# CIBSE GUIDE

# Volume B

## INSTALLATION AND EQUIPMENT DATA

CIBSE


Delta House, 222 Balham High Road, London SW12 9BS

*First published as a loose-leaf volume, 1940*
*Revised and published as a bound volume, 1955*
*Reprinted, 1957*
*2nd edition, 1959*
*Reprinted, 1961*
*3rd edition, 1965*
*4th edition, in three volumes, 1970*
*5th edition, in separate sections, 1974 onwards*
*Reprinted in three volumes, 1986*



LONDON

1986

(Reprinted 1988)

Vol. A ISBN 0 900953 29 2
Vol. B IBSN 0 900953 30 6
Vol. C ISBN 0 900953 31 4
Set ISBN 0 900953 28 4
Typeset by Collins & Wilson Ltd.
Printed and bound by Staples Printers (St Albans) Ltd.

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**The Committee wishes to acknowledge the assistance of the following organisations in the preparation of Section B5**
The Fire Protection Association
The British Fire Protection Systems Association
The Institution of Fire Engineers

# FOREWORD TO THE FIFTH EDITION

The publication of this volume completes the fifth edition of the CIBSE Guide. The editorial policy applied has been based firmly on the tradition of providing guidance information for knowledgeable practitioners rather than a text-book approach from first principles. Every effort has been made to achieve a concise presentation.

The Technical Publications Committee appreciates the large number of advance orders placed for the new Guide and have taken this as an indication of the continued interest of the membership in this familiar and long-standing publication. This volume, the last of the three to be completed, is also the most thoroughly revised.

Time deals relatively gently with the Design Data and Reference Data which comprise the content of Volumes A and C respectively. The updating needs are modest and most of the changes stem from enhanced techniques of calculation. The changes in application practice tend to be much more pronounced. New equipment becomes available and advances in technology are made – all of which need to be considered and, where appropriate, covered in the revision.

Therefore, this volume contains several sections which have been completely rewritten and several more which have been substantially revised and amended. The remainder have been reviewed and updated where necessary.

The numbering of the sections has been left unchanged for consistency with the previous edition and the separate sections. Thus Sections B9 (Lighting) and B17 (Commissioning) are omitted, having been superseded by the wide range of CIBSE lighting publications and the series of Commissioning Codes, respectively. Although it was originally envisaged that this volume would contain an index to all three volumes, practical considerations and the necessity for future revision have meant that an index to this volume only is included. A full index of all CIBSE publications will be contained in the next edition of the OPUS Building Services Design File.

This volume is the result of a huge effort by the Task Groups and individuals concerned. What is more, this effort has been concentrated into a very short time period. On behalf of the Council and the membership of the Institution I wish to record thanks to all the Members, non-Members and technical staff who have played a part in this demanding work.

It is, perhaps, a natural reaction to look ahead to the next revision and to earmark the areas less well covered on this occasion for special attention next time. Plans are already being considered in this respect. Nonetheless, the hope is expressed that this present Volume will serve members well as a working handbook.

Ronald Oughton.

R. J. Oughton

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## Volume B: Installation and Equipment Data

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# CONTENTS

## Volumes A and C

# SECTION B1 HEATING



*This edition of Section B1 first published: 1986*

# SECTION B1 HEATING

## INTRODUCTION

This Section is concerned primarily with space heating but also covers some other aspects of the use of heating in building services.

Information is provided on the heat emission from appliances used for space heating (e.g. radiators, convectors, fan convectors, unit heaters), from other heat emitters (such as embedded panels, heated ceilings and floors) and from electrical heating (both storage and direct). The Section also discusses ways in which the comfort heating requirement for a space may vary with the method of heating, and comments on some design aspects of steam, hot water and other types of heating system.

### Statutory Regulations

Except for some defined types of accommodation, the use of fuel or electricity to heat premises above a temperature of 19°C is prohibited by the Fuel and Electricity (Heating) (Control) Order 1980[1]. The current Order is an amendment to an earlier Regulation which limited the temperature to a maximum of 20°C. Although the 19°C is generally taken to refer to air temperature the Order does not specify this.

### Dry Resultant Temperature

*Convective and Radiant Components*

Section A1 of the *Guide* concludes that for the occupancy of conventional buildings with low air movement, the most acceptable index temperature for comfort conditions is the dry resultant temperature, which is defined as:

$$t_c = \tfrac{1}{2}\, t_{ai} + \tfrac{1}{2}\, t_m \qquad \text{B1.1}$$

where:

$t_c$ = dry resultant temperature .. .. °C
$t_{ai}$ = inside air temperature .. .. °C
$t_m$ = mean surface temperature .. .. °C

For new buildings with the thermal insulation standards required by Parts L2 and L3 of the Building Regulations[2] the differences between air and surface temperatures will be insignificant. Nevertheless it is important to identify situations where these temperatures differ appreciably since this may affect the output required from heating appliances. For some appliances, e.g. fan heater units, the heat output depends only on the difference between entering air temperature and heating medium temperature; with other appliances, e.g. radiant panels, the emission is affected by the temperature of surrounding surfaces.

Tables B1.1 and 2 have been compiled to illustrate this comparison for mainly radiant and mainly convective systems. For general purposes, radiators, natural convectors, skirting heaters, fan convectors and warm air heating systems can be regarded as convective systems. Similarly, heated ceilings, horizontal overhead radiant panels and floor heating systems can be regarded as radiant systems. The tables give the air temperature obtained in different types of buildings when the dry resultant temperature is maintained at 15, 18 and 21°C with an outside air temperature of -1°C. Table B1.1 shows that with convective heating the air temperature is higher than the dry resultant temperature and can be as much as 5 K higher in a poorly insulated space. Table B1.2 shows that with radiant heating, which here refers to appliances with more than 33% of their output

**Table B1.1.** Heating system mainly convective. Air temperature corresponding to dry resultant temperatures.

| Enclosure | | Air temperature, $t_{ai}$, corresponding to the following dry resultant temperature, $t_c$ (for an outdoor temperature, $t_{ao}$, of -1°C) (°C) | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | 15 | | | 18 | | | 21 | | |
| | | with average $U$ values of enclosing surfaces (W/m$^2$K) | | | | | | | | |
| Type | Exposed surfaces | 0·6 | 1·8 | 3·0 | 0·6 | 1·8 | 3·0 | 0·6 | 1·8 | 3·0 |
| Factory 30 m × 30 m × 9 m | Roof and 4 walls | 15·7 | — | — | 18·8 | — | — | 21·9 | — | — |
| | Roof and 2 walls | 15·5 | 16·5 | — | 18·6 | — | — | 21·7 | — | — |
| | Roof and 1 wall | 15·4 | 16·3 | — | 18·5 | 19·5 | — | 21·5 | 22·6 | — |
| | 1 wall | 15·1 | 15·3 | 15·5 | 18·1 | 18·4 | 18·6 | 21·1 | 21·4 | 21·6 |
| Open office 30 m × 30 m × 3 m | Roof and 4 walls | 15·6 | — | — | 18·8 | — | — | 21·8 | — | — |
| | Roof and 2 walls | 15·5 | 16·7 | — | 18·7 | 19·9 | — | 21·7 | — | — |
| | Roof and 1 wall | 15·5 | 16·5 | — | 18·6 | 19·7 | — | 21·6 | 23·0 | — |
| | 1 wall | 15·1 | 15·1 | 15·3 | 18·1 | 18·1 | 18·3 | 21·1 | 21·2 | 21·4 |
| Private office 5 m × 4 m × 3 m | Roof and 2 walls | 15·5 | 16·5 | — | 18·6 | — | — | 21·7 | — | — |
| | Roof and 1 wall | 15·4 | 16·1 | — | 18·4 | 19·2 | — | 21·1 | 22·4 | — |
| | 2 walls | 15·3 | 15·9 | 16·5 | 18·3 | 19·0 | — | 21·3 | 22·2 | — |
| | 1 wall | 15·2 | 15·4 | 15·7 | 18·1 | 18·5 | 18·8 | 21·2 | 21·5 | 21·8 |

**Table B1.2.** Heating system mainly radiant. Air temperature corresponding to dry resultant temperatures (irrespective of average U-value of enclosing surfaces).

| Enclosure | | Air temperature, $t_{ai}$ corresponding to the following dry resultant temperature, $t_c$ (for an outdoor temperature, $t_{ao}$ of -1°C) (°C) | | |
|---|---|---|---|---|
| Type | Ventilation allowance (W/m³K) | 15 | 18 | 21 |
| Factory 30 m × 30 m × 9 m | 0·33 | 13·7 | 16·4 | 19·2 |
| | 0·66 | 13·2 | 15·8 | — |
| Open office 30 m × 30 m × 3 m | 0·33 | 14·3 | 17·2 | 20·1 |
| | 0·66 | 13·8 | 16·5 | 19·3 |
| Private office 5 m × 4 m × 3 m | 0·33 | 14·6 | 17·0 | 20·5 |
| | 0·66 | 14·4 | 17·2 | 20·1 |

in the form of radiation, the air temperature is lower than the dry resultant temperature. In a factory building where the ventilation allowance is 0.66 W/m$^3$K (2 air changes per hour) the air temperature can be as much as 3 K lower than the dry resultant temperature.

Tables B1.1 and 2 also show that in buildings heated by convective systems the required air temperature is unaffected by the rate of ventilation. In buildings heated by radiant systems, however, the air temperature is affected by the ventilation rate but is unaffected by the rate of heat loss through the building fabric.

In circumstances where air temperatures differ markedly from dry resultant temperatures, it may be necessary to alter the sizing of appliances. For instance, if the air temperature requires to be 5°C above the dry resultant temperature, it would be necessary to increase the heating surface area of a convective appliance by approximately 10%, and by a further 5% to allow for increased heat loss to ventilation air. In such a case, however, the first action should be to improve the standard of fabric insulation.

In reasonably well insulated buildings, i.e. those with an average U-value less than 1·0 W/m$^2$K, appliances can be sized using dry resultant temperature as if it were air temperature.

Temperature conditions not covered by the Tables can be estimated by interpolation, or calculated using the method set out in Section A9.

### Allowance for Height of Space

In heat loss calculations a uniform temperature throughout the height of the heated space is assumed, although certain modes of heating cause vertical temperature gradients which lead to increased heat losses, particularly through the roof. These gradients need to be taken into account when sizing appliances. Additions to the calculated heat loss to allow for this are proposed in Table B1.3; however these percentages should not be added to replacement heat to balance that in air mechanically exhausted from process plant.

**Table B1.3.** Allowances for height of heated space.

| Method of heating and type or disposition of heaters | Percent addition for following heights of heated space (m) | | |
|---|---|---|---|
| | 5 | 5 to 10 | > 10 |
| MAINLY RADIANT | | | |
| Warm floor . . . . . . | Nil | Nil | Nil |
| Warm ceiling . . . . . . | Nil | 0 to 5 | * |
| Medium and high temperature downward radiation from high level . . . . . . . . | Nil | Nil | 0 to 5 |
| MAINLY CONVECTIVE | | | |
| Natural warm air convection . . | Nil | 0 to 5 | * |
| Forced warm air | | | |
| Cross flow at low level . . | 0 to 5 | 5 to 15 | 15 to 30 |
| Downward from high level . . | 0 to 5 | 5 to 10 | 10 to 20 |
| Medium and high temperature cross radiation from intermediate level . . . . . . | Nil | 0 to 5 | 5 to 10 |

* Not appropriate to this application.

Attention is also drawn to the means of reducing the effect of temperature stratification discussed below under *Warm Air Heating Systems.*

## CHARACTERISTICS OF HEAT EMITTERS

### The Use of a Margin

Designers will need to decide whether it is necessary to add a margin to the output of heat emitters. During the warm-up cycle with intermittently operated heating systems, emitter output will be higher than design because space temperatures are lower. Also, boost system temperatures may be used to provide an emission margin during warm-up. The need for heat emitter margins to meet extreme weather conditions will depend on the design parameters used in determining heat losses.

In summary, although the addition of a modest margin to heat emitter output would add little to the overall system cost and that a margin on the heat generator or boiler output can only be utilised if the appropriate emitter capacity is available, the decision should be based on careful discrimination rather than using an arbitrary percentage allowance. Section A9 provides the necessary design guidance for the selection of an appropriate margin for systems and the margin for heat emitters should be based on this. In general, for buildings of traditional construction and for the incidence of design weather in normal winters in the UK, an emitter margin in excess of, say, 5% or 10% is unlikely to be justified. However, for well insulated buildings the heat loss reduces in significance relative to the heat stored in, or needed to warm up the structure. For such applications a larger heating system margin is required and the emitter margin provided would need to be considered accordingly.

### Test Conditions

Heat emission data are based on test methods detailed in various British Standards (e.g. BS 3528[3] for radiators and convectors, BS 4856[4] for fan powered heaters) which do not relate exactly to the conditions in the space to be heated. It is, therefore, necessary to check the installation of the emitters and the design conditions in the space against those used for test rating of the appliances.

**Radiators**

BS 3528[3] specifies a standard method of test for radiators and natural convectors, and calls for heat emission to be expressed as :

$$q = ks\ (t_m - t_{ai})^n \quad \text{.. .. .. ..} \qquad \text{B1.2}$$

wher e:

$q$ = heat emission .. .. .. .. W

$k$ = a constant for a given height and design of radiator

$s$ = number of sections or units of length

$t_m$ = mean temperature of heating fluid .. °C

$n$ = an index

Independent reports[5,6,7] confirm that for radiators, the index, $n$, is 1·3.

Emissions are commonly quoted by manufacturers for a temperature difference of 60 K: emissions for other temperature differences can be derived by multiplying by the factors given in Table B1.4. Emissions measured by standard tests such as BS 3528 tend to be lower than those which occur in actual installations, due to the effect of air movement and ventilation, and to architectural features and finishes. The effect of air movement can be considerable, raising the emission by up to 15%. An empirical addition of 10% to the test heat emissions is accepted by some continental authorities to allow for this effect.

For the testing of radiators, BS 3528 specifies top and bottom opposite end connections (TBOE). Measurements[8] have shown that other connection arrangements have a marginal effect on heat output. Taking the heat output for TBOE connections as 1·0, the following values can be expected:

top and bottom same end connections (TBSE) 1·04

bottom opposite end connections (BBOE) 0·85

Long radiators, with a length to height ratio greater than 5, may suffer a reduction in output due to short circuiting in the order of 5 to 10%.

The effects of architectural features on emission are summarised in Table B1.5.

**Table B1.4.** Emission factors $\left(\frac{Dt}{60}\right)^{1\cdot 3}$

| Dt K | 0 | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|---|---|---|
| 20 | 0·24 | 0·25 | 0·27 | 0·28 | 0·30 | 0·32 | 0·34 | 0·36 | 0·37 | 0·38 |
| 30 | 0·41 | 0·43 | 0·44 | 0·46 | 0·48 | 0·49 | 0·51 | 0·53 | 0·55 | 0·57 |
| 40 | 0·59 | 0·61 | 0·63 | 0·65 | 0·67 | 0·69 | 0·71 | 0·73 | 0·75 | 0·77 |
| 50 | 0·79 | 0·81 | 0·83 | 0·85 | 0·87 | 0·90 | 0·92 | 0·94 | 0·96 | 0·98 |
| 60 | BASIS | 1·02 | 1·04 | 1·07 | 1·09 | 1·11 | 1·13 | 1·16 | 1·18 | 1·20 |
| 70 | 1·23 | 1·25 | 1·27 | 1·30 | 1·32 | 1·34 | 1·37 | 1·39 | 1·41 | 1·43 |
| 80 | 1·45 | 1·48 | 1·50 | 1·52 | 1·55 | 1·57 | 1·59 | 1·62 | 1·65 | 1·67 |
| 90 | 1·69 | 1·72 | 1·74 | 1·77 | 1·80 | 1·82 | 1·84 | 1·87 | 1·89 | 1·92 |
| 100 | 1·94 | 1·96 | 1·99 | 2·02 | 2·05 | 2·07 | 2·10 | 2·12 | 2·15 | 2·18 |

**Table B1.5.** Effects of architectural features and surface finish on the heat emission from radiators.

| Parameter | Effect |
|---|---|
| Ordinary paints and enamels | Nil, irrespective of colour. |
| Metallic paints such as aluminium or bronze | Will reduce radiant output by 50% or more and overall emission by between 10 and 25%. Emission is substantially restored by two coats of clear varnish. |
| Open-fronted recess | Will reduce emission by 10%. |
| Encasement with front grille | Will reduce emission by 20% or more, depending on design. |
| Radiator shelf | Will reduce output by up to 10%. |
| Fresh air inlet at rear with baffle at front | May increase emission by up to 15%. This increase should not be taken into account when sizing radiator but should be allowed for in pipe and boiler sizing. A damper should always be fitted. |
| Distance of radiator from wall | The minimum distance from the wall to the nearest part of the radiator of 25 mm is recommended. Below this, emission can be apparently reduced. Above this the emission can increase slightly, e.g. by about 8% at 150 mm. |
| Height of radiator above floor | Little effect above a height of 100 mm. If radiators are mounted at a high level, the emission will depend on the temperature conditions at that level. |

### Natural Convectors

BS 3528[3] details test methods for natural convectors. Since design features will affect the output markedly, it is not possible to give any generalised figures for output, and authenticated data for the individual appliance should be used. The most appropriate value of the index, *n*, for natural convector emitters is 1·40. The water flow rate can have a considerable effect on the water surface film resistance and hence, on the heat emission. It is important to seek information from proprietary manufacturers on this aspect.

### Use of Reflective Backing

Where radiators or natural convectors are installed against outside walls the high losses through the building fabric behind the emitter may be reduced by applying material with a high reflectivity to the inner surface of the wall. Aluminium foil is typically used for this purpose and manufacturers offer products combining the foil with a panel of thermal insulation. The effectiveness of the principle depends on reflectivity which, in practice is likely to deteriorate with age and use. The application of reflective foil behind emitters installed at windows can be much more effective than that installed behind radiators sited elsewhere.

### Fan Coil Heaters

BS 4856[4] gives details of test methods for fan coil units for thermal and air movement performance with and without attached ducting, and for noise levels without attached ducting. The index *n* is unity for these appliances and for unit heaters. The characteristics of natural and forced convective appliances are, therefore, different and this should be taken into account when designing controls for mixed systems.

The output of fan coil units is much more sensitive to air-side deficiencies than to water flow problems. This should be borne in mind when investigating output shortcomings.

Coils for hot water system fan convectors and unit heaters are often fabricated from copper tubes. While copper is anodic to iron in most situations, the presence of minute traces of sulphides or of ammonia can cause copper to become cathodic and can result in local, preferential corrosion, particularly where the metal has been extensively cold-worked or heated for brazing etc. during manufacture. Guidance should be sought from unit manufacturers and reference may be made to Section B7.

### Plane Surfaces

Tables B1.6 and 7 have been prepared using the theoretical data given in Section C3. The radiation and convection emissions are shown separately since there may be significant differences between the mean radiant temperature of the enclosure and the air temperature within it. The convection emission applies to draught-free conditions and appreciable increases will occur if air movement is present. For example, with a local air movement velocity of 0·5 m/s an increase of 35% in convective heat emission could be expected.

### Heat Emission from Water Distribution Pipework

The heat transfer coefficients for various pipe diameters are listed in Section C3. These have been used to produce the unit values for heat emission per metre run of horizontal steel and copper pipes given in Tables B1.8 and 9. As explained in detail in Section C3, heat emission under site conditions may vary by ±10% from these tabulated figures due to air movement, paint finishes etc. These aspects should be taken into consideration in system design and allowances made appropriate to the circumstances. It may be prudent, for instance, to use a value lower than the tabulated figure in estimating heat output available and a higher value than that tabulated in calculating mains losses, as these affect the requirement for heat generation.

#### *Vertical Pipes*

Pipes installed vertically have a heat emission which differs from that of horizontal piping due to the variance in the thickness of the laminar boundary layer of air around the pipe surface. The correction factors quoted in Table B1.10 are for use in conjunction with the data listed in Tables B1.8 and 9.

**Table B1.6.** Heat emission from plane surfaces by radiation.

| Surface tempera-ture/°C | Heat emission/(W/m²) | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Surface emissivity 0·3 | | | | | | | Surface emissivity 0·6 | | | | | | | Surface emissivity 0·9 | | | | | | |
| | Enclosure mean radiant temperature/°C | | | | | | | Enclosure mean radiant temperature/°C | | | | | | | Enclosure mean radiant temperature/°C | | | | | | |
| | 10 | 12·5 | 15 | 17·5 | 20 | 22·5 | 25 | 10 | 12·5 | 15 | 17·5 | 20 | 22·5 | 25 | 10 | 12·5 | 15 | 17·5 | 20 | 22·5 | 25 |
| 20 | 16 | 12 | 8·3 | 4·2 | 0 | -4·3 | -8·8 | 33 | 25 | 17 | 8·4 | 0 | -8·7 | -18 | 49 | 37 | 25 | 13 | 0 | -13 | -26 |
| 30 | 34 | 30 | 26 | 22 | 18 | 14 | 9·2 | 69 | 61 | 53 | 44 | 36 | 27 | 18 | 103 | 91 | 79 | 67 | 54 | 41 | 28 |
| 40 | 54 | 50 | 46 | 42 | 38 | 34 | 29 | 108 | 100 | 92 | 84 | 76 | 67 | 58 | 162 | 151 | 139 | 126 | 114 | 101 | 87 |
| 50 | 76 | 72 | 68 | 64 | 60 | 55 | 51 | 152 | 144 | 136 | 128 | 120 | 111 | 102 | 228 | 216 | 204 | 192 | 179 | 166 | 153 |
| 60 | 100 | 96 | 92 | 88 | 84 | 79 | 75 | 200 | 192 | 184 | 176 | 168 | 159 | 150 | 300 | 288 | 276 | 264 | 251 | 238 | 225 |
| 70 | 126 | 122 | 118 | 114 | 110 | 106 | 101 | 253 | 245 | 237 | 229 | 220 | 211 | 203 | 379 | 367 | 355 | 343 | 330 | 317 | 304 |
| 80 | 155 | 151 | 147 | 143 | 139 | 134 | 130 | 310 | 302 | 294 | 286 | 278 | 269 | 260 | 465 | 453 | 441 | 429 | 416 | 403 | 390 |
| 90 | 186 | 182 | 178 | 174 | 170 | 166 | 161 | 372 | 365 | 357 | 348 | 340 | 331 | 322 | 559 | 547 | 535 | 523 | 510 | 497 | 484 |
| 100 | 220 | 216 | 212 | 208 | 204 | 200 | 195 | 440 | 432 | 424 | 416 | 408 | 399 | 390 | 660 | 649 | 637 | 624 | 612 | 599 | 585 |
| 120 | 297 | 293 | 289 | 285 | 280 | 276 | 272 | 593 | 586 | 577 | 569 | 561 | 552 | 543 | 890 | 878 | 866 | 854 | 841 | 828 | 815 |
| 140 | 386 | 382 | 378 | 374 | 370 | 365 | 361 | 772 | 764 | 756 | 747 | 739 | 730 | 721 | 1160 | 1150 | 1130 | 1120 | 1110 | 1100 | 1080 |
| 160 | 489 | 485 | 481 | 477 | 473 | 468 | 464 | 978 | 970 | 962 | 954 | 945 | 936 | 928 | 1470 | 1450 | 1440 | 1430 | 1420 | 1400 | 1390 |

**Table B1.7.** Heat emission from plane surfaces by free convection.

| Surface tempera-ture/°C | Heat emission/(W/m²) | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Horizontal looking down | | | | | | | Vertical | | | | | | | Horizontal looking up | | | | | | |
| | Air temperature/°C | | | | | | | Air temperature/°C | | | | | | | Air temperature/°C | | | | | | |
| | 10 | 12·5 | 15 | 17·5 | 20 | 22·5 | 25 | 10 | 12·5 | 15 | 17·5 | 20 | 22·5 | 25 | 10 | 12·5 | 15 | 17·5 | 20 | 22·5 | 25 |
| 20 | 11 | 7·9 | 4·8 | 2·0 | 0 | -5·8 | -14 | 30 | 20 | 12 | 4·7 | 0 | -4·7 | -12 | 36 | 25 | 14 | 5·8 | 0 | -2·0 | -4·8 |
| 30 | 27 | 23 | 19 | 15 | 11 | 7·9 | 4·8 | 75 | 63 | 51 | 40 | 30 | 20 | 12 | 91 | 77 | 62 | 49 | 36 | 25 | 14 |
| 40 | 45 | 40 | 36 | 31 | 27 | 23 | 19 | 129 | 115 | 101 | 88 | 75 | 63 | 51 | 157 | 140 | 123 | 107 | 91 | 77 | 62 |
| 50 | 64 | 59 | 54 | 50 | 45 | 40 | 36 | 189 | 174 | 158 | 144 | 129 | 115 | 101 | 230 | 211 | 192 | 174 | 157 | 140 | 123 |
| 60 | 85 | 80 | 75 | 69 | 64 | 59 | 54 | 255 | 238 | 221 | 205 | 189 | 174 | 158 | 309 | 289 | 269 | 249 | 230 | 211 | 192 |
| 70 | 107 | 101 | 96 | 90 | 85 | 80 | 75 | 324 | 307 | 289 | 272 | 255 | 238 | 221 | 394 | 372 | 351 | 330 | 309 | 289 | 269 |
| 80 | 130 | 124 | 118 | 112 | 107 | 101 | 96 | 398 | 379 | 361 | 342 | 324 | 307 | 289 | 484 | 461 | 438 | 416 | 394 | 372 | 351 |
| 90 | 153 | 147 | 141 | 135 | 130 | 124 | 118 | 476 | 456 | 436 | 417 | 398 | 379 | 361 | 578 | 554 | 530 | 507 | 484 | 461 | 438 |
| 100 | 177 | 171 | 165 | 159 | 153 | 147 | 141 | 556 | 536 | 516 | 495 | 476 | 456 | 436 | 675 | 651 | 626 | 602 | 578 | 554 | 530 |
| 120 | 228 | 222 | 215 | 209 | 202 | 196 | 190 | 726 | 705 | 683 | 661 | 640 | 619 | 598 | 882 | 856 | 829 | 803 | 777 | 751 | 726 |
| 140 | 281 | 274 | 267 | 261 | 254 | 248 | 241 | 907 | 884 | 861 | 838 | 816 | 793 | 771 | 1100 | 1070 | 1050 | 1020 | 990 | 963 | 936 |
| 160 | 336 | 329 | 322 | 315 | 308 | 301 | 295 | 1100 | 1070 | 1050 | 1020 | 1000 | 977 | 954 | 1330 | 1300 | 1270 | 1240 | 1220 | 1190 | 1160 |

*Multiple Pipe Installations*

Where pipes are arranged in a horizontal bank, each pipe directly above another at close pitch, the overall heat emission is reduced due to the cumulative interference with the convected output. Table B1.11 lists correction factors for emissions from multiple pipe installations.

*Effect of Proximity of Walls*

Where pipes are installed near to cold external walls, the emission by radiation is likely to increase and that by convection to remain unchanged; with internal walls there may be a reduction in radiation and a slight increase in convection. However, these variations are likely to be appreciably smaller than variations in emission due to air movement, paint finishes etc.

**Heat Emission from Room Surfaces**

Heat liberated at a plane within a structural slab will raise the temperatures of both surfaces by increments proportional to the thermal resistance on either side of the plane. If the structural slab is an intermediate floor, the resulting emissions may both be 'useful'. If it is a floor in contact with earth, a roof, or an external wall, one of the components will generally be waste heat and the thermal resistance on the outer side should be adjusted to minimise the loss.

*Comfort Considerations*

The surface temperatures of the slab must be limited to a level which will not cause discomfort to occupants. For ceilings, the permissible surface temperature is a function of room height, the dimensions of the heated panel and room temperature (see Section A1). For floors on which people will be standing for extended periods, the surface temperature should not exceed 25°C and, for areas subject to foot traffic but not continuous occupancy, the normal design limit is 27°C. Providing the floor finish is suitable, these temperatures may be exceeded in situations such as room perimeters, where foot traffic is unlikely, (34°C) or in bathrooms (32°C) where higher temperatures would be acceptable.

**Table B1.8.** Heat emission from single horizontal steel pipes with a surface emissivity of 0·9 and freely exposed in ambient air at temperatures between 10 and 20°C.

| Nominal pipe size | | Heat emission (W/m run) | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Temperature difference, surface to surroundings/K | | | | | | | | | | | | | | | | | | | |
| (mm) | (in) | 40 | 45 | 50 | 55 | 60 | 65 | 70 | 80 | 90 | 100 | 120 | 140 | 160 | 180 | 200 | 220 | 240 | 260 | 280 | 300 |
| 15 | ½ | 40 | 46 | 53 | 59 | 66 | 73 | 80 | 96 | 110 | 130 | 170 | 200 | 240 | 290 | 340 | 400 | 450 | 520 | 600 | 680 |
| 20 | ¾ | 48 | 56 | 62 | 70 | 78 | 87 | 95 | 110 | 130 | 150 | 200 | 250 | 300 | 350 | 420 | 480 | 560 | 640 | 730 | 830 |
| 25 | 1 | 58 | 68 | 78 | 88 | 98 | 110 | 120 | 140 | 160 | 190 | 240 | 300 | 370 | 430 | 510 | 590 | 680 | 780 | 880 | 1000 |
| 32 | 1¼ | 71 | 82 | 93 | 110 | 120 | 130 | 150 | 170 | 200 | 230 | 300 | 370 | 450 | 540 | 630 | 730 | 840 | 950 | 1100 | 1200 |
| 40 | 1½ | 78 | 92 | 105 | 118 | 130 | 150 | 160 | 190 | 230 | 260 | 330 | 410 | 510 | 600 | 710 | 830 | 950 | 1100 | 1200 | 1400 |
| 50 | 2 | 96 | 112 | 130 | 150 | 170 | 180 | 200 | 240 | 270 | 320 | 410 | 500 | 620 | 740 | 860 | 1000 | 1200 | 1300 | 1500 | 1700 |
| 65 | 2½ | 120 | 140 | 160 | 180 | 200 | 220 | 240 | 290 | 330 | 390 | 500 | 610 | 750 | 900 | 1100 | 1300 | 1400 | 1600 | 1800 | 2000 |
| 80 | 3 | 140 | 160 | 180 | 210 | 230 | 250 | 280 | 330 | 380 | 450 | 580 | 710 | 900 | 1100 | 1300 | 1500 | 1700 | 1900 | 2100 | 2300 |
| 100 | 4 | 170 | 200 | 230 | 260 | 290 | 310 | 350 | 410 | 480 | 550 | 710 | 900 | 1100 | 1300 | 1500 | 1800 | 2100 | 2400 | 2700 | 3100 |
| 125 | 5 | 200 | 240 | 270 | 310 | 350 | 380 | 420 | 490 | 580 | 660 | 850 | 1100 | 1300 | 1600 | 1900 | 2100 | 2500 | 2800 | 3200 | 3700 |
| 150 | 6 | 230 | 280 | 320 | 350 | 400 | 440 | 480 | 570 | 660 | 770 | 1000 | 1200 | 1500 | 1800 | 2100 | 2500 | 2900 | 3300 | 3800 | 4300 |
| 200 | 8 | 290 | 350 | 400 | 460 | 510 | 560 | 610 | 720 | 840 | 970 | 1300 | 1600 | 2000 | 2300 | 2800 | 3200 | 3800 | 4300 | 4900 | 5600 |
| 250 | 10 | 360 | 420 | 490 | 560 | 610 | 700 | 760 | 900 | 1100 | 1200 | 1600 | 2000 | 2400 | 2900 | 3400 | 4000 | 4600 | 5200 | 6000 | 6800 |
| 300 | 12 | 410 | 500 | 580 | 650 | 710 | 800 | 880 | 1100 | 1200 | 1400 | 1800 | 2300 | 2800 | 3400 | 4000 | 4600 | 5300 | 6100 | 7000 | 8000 |

**Table B1.9.** Heat emission from single horizontal copper pipes freely exposed in ambient air at temperatures between 10 and 20°C.

| Pipe size | | Heat emission (W/m run) | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Outside diameter (mm) | Nominal (in) | Temperature difference, surface to surroundings/K | | | | | | | | | | | | | | | | | | | |
| | | Painted pipe $\epsilon \approx 0·9$ | | | | | | | | | | Tarnished pipes $\epsilon \approx 0·5$ | | | | | | | | | |
| | | 40 | 45 | 50 | 55 | 60 | 65 | 70 | 80 | 90 | 100 | 40 | 45 | 50 | 55 | 60 | 65 | 70 | 80 | 90 | 100 |
| 15 | ½ | 28 | 31 | 35 | 40 | 45 | 51 | 58 | 76 | 98 | 130 | 22 | 25 | 29 | 32 | 36 | 40 | 45 | 56 | 70 | 88 |
| 22 | ¾ | 39 | 44 | 40 | 56 | 63 | 71 | 80 | 110 | 130 | 170 | 29 | 33 | 38 | 43 | 49 | 56 | 63 | 81 | 110 | 140 |
| 28 | 1 | 48 | 56 | 64 | 72 | 81 | 91 | 110 | 130 | 170 | 200 | 35 | 41 | 47 | 52 | 59 | 66 | 74 | 95 | 120 | 150 |
| 35 | 1¼ | 58 | 67 | 76 | 87 | 98 | 110 | 122 | 160 | 200 | 250 | 42 | 49 | 56 | 64 | 72 | 82 | 93 | 120 | 160 | 210 |
| 42 | 1½ | 68 | 78 | 90 | 100 | 120 | 130 | 140 | 190 | 240 | 280 | 48 | 56 | 64 | 74 | 83 | 96 | 110 | 150 | 180 | 240 |
| 54 | 2 | 88 | 100 | 110 | 130 | 140 | 160 | 180 | 230 | 310 | 360 | 61 | 70 | 80 | 90 | 110 | 120 | 130 | 170 | 210 | 270 |
| 76 | 3 | 120 | 140 | 160 | 180 | 200 | 230 | 250 | 330 | 420 | 530 | 85 | 100 | 120 | 130 | 150 | 160 | 180 | 230 | 280 | 340 |
| 108 | 4 | 160 | 190 | 210 | 230 | 260 | 290 | 320 | 430 | 570 | 700 | 110 | 130 | 150 | 160 | 180 | 200 | 220 | 270 | 330 | 410 |
| 133 | 5 | 190 | 220 | 250 | 280 | 310 | 350 | 390 | 520 | 650 | 820 | 130 | 150 | 180 | 200 | 220 | 240 | 270 | 330 | 400 | 500 |
| 159 | 6 | 220 | 260 | 290 | 320 | 360 | 410 | 450 | 580 | 730 | 920 | 150 | 180 | 210 | 230 | 260 | 280 | 320 | 390 | 480 | 600 |

**Table B1.10.** Correction factors for heat emission from vertical pipes.

| Pipe size | | Multiply emission from Tables B1.8 and B1.9 by following factors to obtain emission for vertical pipes | | | |
|---|---|---|---|---|---|
| | | Range of temperature difference /K | | | |
| (mm) | (in) | 40 to 45 | 50 to 70 | 80 to 130 | 140 to 260 |
| 15 | ½ | 0·75 | 0·76 | 0·77 | 0·81 |
| 20/22 | ¾ | 0·78 | 0·79 | 0·80 | 0·83 |
| 25/28 | 1 | 0·80 | 0·82 | 0·83 | 0·85 |
| 32/35 | 1¼ | 0·83 | 0·84 | 0·85 | 0·87 |
| 40/42 | 1½ | 0·85 | 0·86 | 0·87 | 0·89 |
| 50/54 | 2 | 0·87 | 0·88 | 0·89 | 0·91 |
| 65 | 2½ | 0·89 | 0·90 | 0·91 | 0·93 |
| 76/80 | 3 | 0·91 | 0·92 | 0·93 | 0·94 |
| 100/108 | 4 | 0·94 | 0·95 | 0·96 | 0·96 |
| 125/133 | 5 | 0·96 | 0·97 | 0·98 | 0·98 |
| 150/159 | 6 | 0·98 | 0·99 | 1·00 | 0·99 |
| 200 | 8 | 1·02 | 1·03 | 1·04 | 1·03 |
| 250 | 10 | 1·04 | 10·5 | 1·06 | 1·05 |
| 300 | 12 | 1·06 | 10·7 | 1·08 | 1·06 |

**Table B1.11.** Correction factors for multiple banks of pipes (horizontal pipes one above another at close pitch).

| Number of pipes in bank | Emission from each pipe as fraction of theoretical single pipe value |
|---|---|
| 2 | 0·95 |
| 4 | 0·85 |
| 6 | 0·75 |
| 8 | 0·65 |

The limiting temperatures given as being appropriate to avoid foot discomfort refer to the occupied period of the building. The initial floor surface temperature may exceed the above values where heat if provided from cables energised by "off-peak" electricity.

*Floors*

The relationship between upward heat emission and floor surface temperature is given by the following equation:

$$q_u = k_u (t_{sf} - t_{ei}) \quad \ldots \quad \text{B1.3}$$

where:

$q_u$ = upward heat emission . . . . . . W/m²
$t_{sf}$ = temperature of exposed floor surface °C
$t_{ei}$ = room environmental temperature . . °C
$k_u$ = coefficient of upward heat emission W/m²K

The coefficient of upward heat emission is a function of the relative values of the floor surface temperature, the air temperature and the mean radiant temperature above the floor, the ventilation rate across the floor surface and, in practical applications, the lateral transmission from the heated panel to adjoining areas of unheated floor. A theoretical value of 9·08 has been proposed[9] for situations where the globe thermometer and air temperatures are approximately equal. For the conditions usually encountered in floor-heating installations, a value of 10 is recommended for calculating the required area of heated surface. In multi-storey buildings, calculations of heated surface should start on the top floor; on lower floors the heat flow downwards from the ceiling above should be subtracted from the space heat loss to arrive at the required floor emission.

Care must be taken where hours of use of adjacent floors are not consistent. Although the heating to a floor not in use could be isolated using automatic controls, neighbouring floors would be affected. Downward emission from the floor above would tend to increase and would be wasted while the floor below would not receive downward emission and may consequently suffer a heat deficiency.

If the ratio of required emission to available floor area exceeds the upward heat emission obtained from substitution of appropriate values of floor surface and environmental temperatures in equation B1.3, consideration should be given to reducing the heat loss from the space or raising the temperature of selected areas of the floor surface, at the perimeter or in other zones with limited foot traffic, to provide increased heat emission.

Floor insulation is necessary and the type used must be loadbearing. A 50 mm thickness of polystyrene is usually satisfactory. The position of the insulating layer may be arranged to maximise the depth of structure available for heat storage for off-peak electric floorwarming as in Fig. B1.1 (a), or to limit the structural heat storage capacity as in Fig. B1.1 (b). Increasing the cover screed thickness also provides more heat storage capacity where this is required, the pracical limit being 100 mm. The thermal insulation properties of carpet or other floor coverings will have the effect of raising the floor temperature and, hence, increasing the heat storage capacity.

Furniture which masks a complete area of floor surface (e.g. cupboards, filing cabinets) will interfere with heat emission. Provided the masked area does not exceed, say, 30% of the exposed floor surface this should not affect design space temperatures. In any case, the extent of masked areas should be brought to the attention of the system manufacturer.

Structural considerations will usually determine the thermal resistance above and below the plane of the embedded heater, as illustrated in Fig. B1.2. Figs. B1.3 to 5 show the relationship, for various pipe or cable spacings, between the variables concerned as represented by the equation:

$$q_u = k'_u \, (t_w - t_{ei}) \qquad \text{.. .. .. ..} \qquad \text{B1.4}$$

where:

- $k'_u$ = coefficient of upward heat emission related to $R_1$ and $R_2$ .. .. .. W/m$^2$K
- $R_1$ = thermal resistance of structure etc. between the plane of the embedded heater and the exposed floor surface m$^2$K/W
- $R_2$ = thermal resistance of structure etc. below the plane of the embedded heater .. .. .. .. .. m$^2$K/W
- $t_w$ = mean temperature of heating element °C

Figs. B1.3 to 5 are drawn for the condition where the environmental temperatures above and below the structure are equal. Where this is not the case, an adjusted value should be used as calculated from:

$$R'_2 = R_2 \left( \frac{t_w - t_{ei}}{t_w - t'_{ei}} \right) \qquad \text{.. .. ..} \qquad \text{B1.5}$$

where:

- $R'_2$ = adjusted value of thermal resistance of structure below the plane of the embedded heater .. .. .. m$^2$K/W
- $t'_{ei}$ = environmental temperature in space below embedded heater .. .. °C

Values of thermal resistance for typical floor and ceiling materials and finishes may be calculated from data given in Section A3.

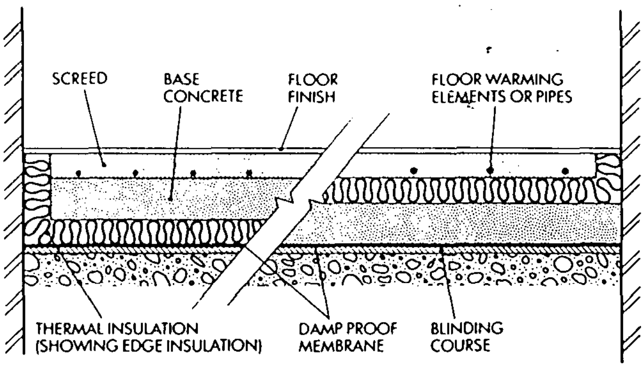


**Fig. B1.1.** Typical ground floor arrangements.

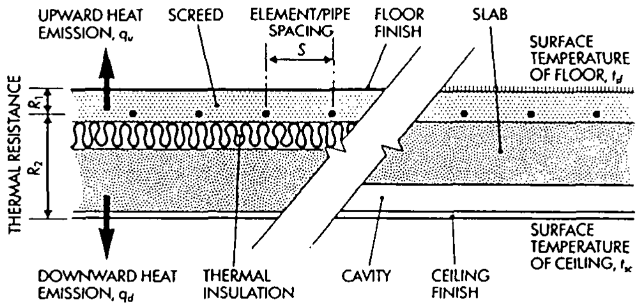


**Fig. B1.2.** Embedded underfloor heating – typical intermediate floor arrangements showing symbol conventions.

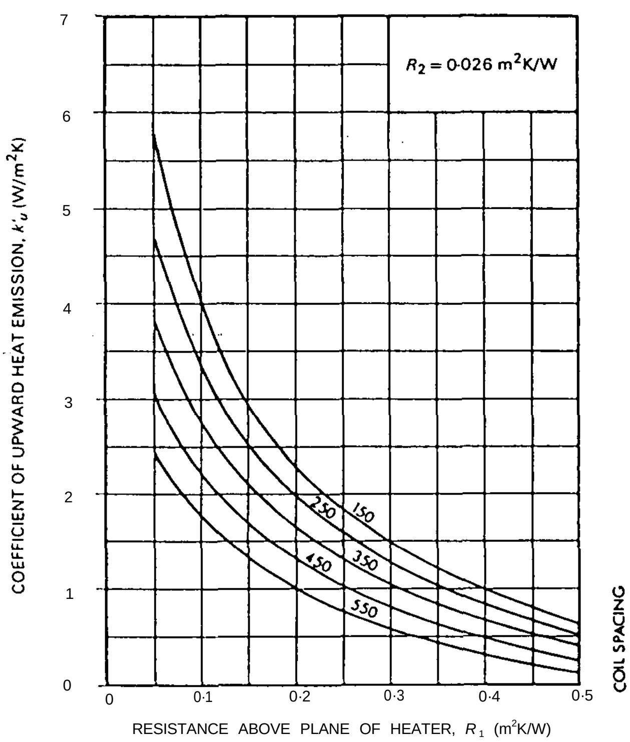


**Fig. B1.3.** Coefficient of upward heat emission.

## *Heat Loss to the Ground*

Losses to earth from heated ground floors may be divided into:

(*a*) Those arising at the vertical edge of the heated floor when the surface of this floor is above ground level (edge losses).

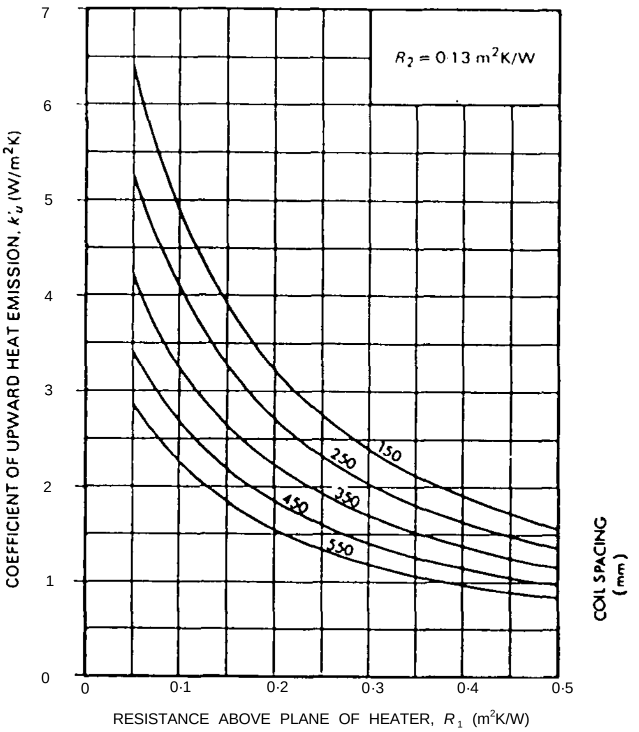


**Fig. B1.4.** Coefficient of upward heat emission.

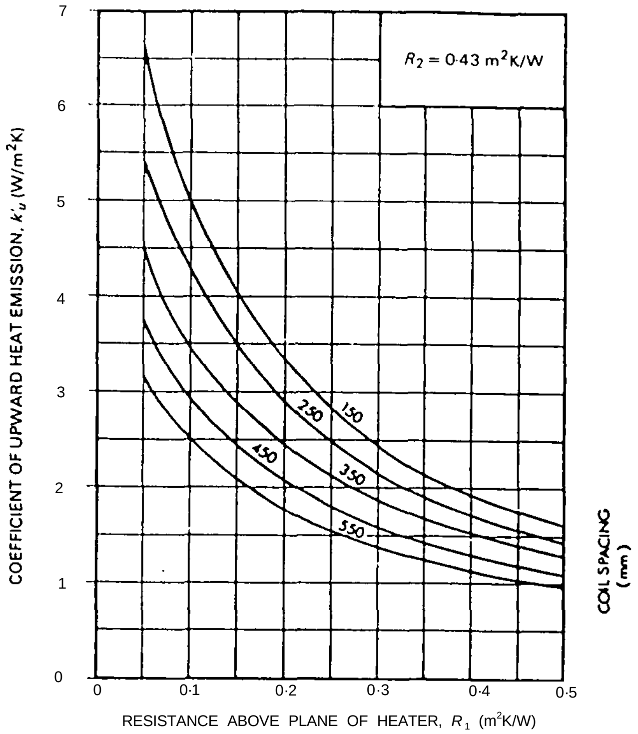


**Fig. B1.5.** Coefficient of upward heat emission.

(*b*) Those arising over the remainder of the heated floor area and travelling downwards and, near the edge, towards the ground surface outside the building (downward and perimeter losses).

Edge loss may be calculated using the following expression:

$$q_e = \frac{Ph\,(t_p + t_{sf} - 2\,t_{ao})}{2(R_w + R_{so})} \quad .. \quad .. \quad .. \quad \text{B1.6}$$

where:

$q_e$ = edge loss.. .. .. .. .. .. W
$P$ = length of exposed perimeter .. .. m
$h$ = height of inside floor surface above ground level .. .. .. .. m
$t_{ao}$ = outside design air temperature .. °C
$t_p$ = heated plane temperature .. .. °C
$R_w$ = thermal resistance of wall structure m²K/W
$R_{so}$ = external surface resistance of wall .. m²K/W

*Note:* (*i*) No inner surface resistance is taken for the wall.
(*ii*) Where edge insulation is used, the additional resistance is included in that of the wall structure.

The heated plane temperature is given by the equation:

$$t_p = R_1\, q_u + t_{sf} \quad .. \quad .. \quad .. \quad .. \quad \text{B1.7}$$

Downward and perimeter losses from a heated floor can be dealt with in terms of a ground heat loss coefficient which allows for the heat stored in the soil below the centre of the floor as well as that concentrated near the edge. Values for $C_g$ in equation B1.8 can be obtained from Figs. B1.6 to 8 for soils of different thermal conductivities. The values are given for floors of infinite length and various widths. These are adjusted for different numbers of exposed edges by means of the correction factor derived from Fig. B1.9.

$$q_d = C_g\,(t_p - t_{ao}) \quad .. \quad .. \quad .. \quad .. \quad \text{B1.8}$$

where:

$q_d$ = average heat flow of total floor area covering both perimeter and central regions .. .. .. .. .. W/m²
$C_g$ = ground heat loss coefficient .. .. W/m²K

The total heat losses for a heated floor on the ground are the sum of the edge and the downward losses, whereas for a suspended intermediate floor they are the sum of the edge loss plus the emission to the space below.

To utilise higher water temperatures than are permissible with directly embedded pipes, floor coils can be enclosed in asbestos-cement sheaths. The resistance of the sheath and air-gap surrounding the pipe must be added to the upward and downward resistances of the structural components. The sheath resistance is given[10] as 0·47 K/W per linear metre of pipe.

Typical design mean water temperatures are 40 to 45°C where heaters are embedded directly in the structure and up to 77°C for pipes or elements installed in asbestos-cement sheathing. If any substantial departure from these temperatures is proposed, detailed consideration should be given to the occupant comfort implications of maximum temperatures of the floor and ceiling, and effects on floor finishes, pattern staining or cracking of plastered finishes, etc.

*Ceilings*

The basic equation for the relation between downward heat emission and exposed ceiling surface temperature is:

$$q_d = k_d \,(t_{sc} - t_{ei}) \quad \ldots \quad \text{B1.9}$$

where:

$q_d$ = downward heat emission .. .. W/m$^2$
$t_{sc}$ = temperature of exposed ceiling surface .. .. .. .. .. °C
$t_{ei}$ = room environmental temperature .. °C
$k_d$ = coefficient of downward heat emission .. .. .. .. .. W/m$^2$ K

The value of the coefficient of downward heat emission varies in a manner similar to that described for upward emission; a significant factor is whether the heated surface occupies the whole, or only a part, of the total ceiling. While a value of 7 may be used for the whole ceiling, a more appropriate figure can be up to 9 for smaller heated areas.

As for floor heating, structural considerations are the dominant factors where the pipe coils or elements are actually embedded as illustrated in Fig. B1.2.

To illustrate the relationship, Fig. B1.10 to 12 have been prepared for various pipe spacings to represent the equation:

$$q_d = k'_d \,(t_w - t_{ei}) \quad \ldots \quad \text{B1.10}$$

where:

$k'_d$ = coefficient of downward heat emission related to $R_1$ and $R_2$ .. .. W/m$^2$K

In instances where the environmental temperatures above and below the structure are not the same, an adjustment should be made using the expression:

$$R'_1 = R_1 \left( \frac{t_w - t'_{ei}}{t_w - t_{ei}} \right) \quad \ldots \quad \text{B1.11}$$

where:

$R'_1$ = adjusted value of thermal resistance of structure above the plane of the embedded heater .. .. .. m$^2$K/W
$t'_{ei}$ = environmental temperature in space above embedded heater .. .. °C

In multi-storey buildings, calculation of heated surface area should start on the ground floor. On upper floors, heat flow upwards from the floor should be subtracted from the room heat loss to arrive at the required ceiling emission.

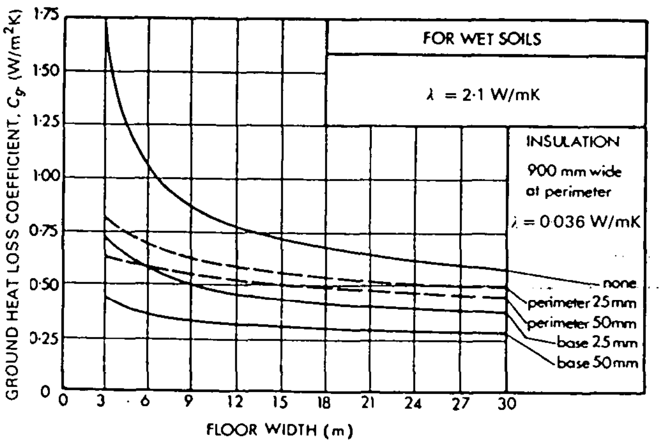


**Fig. B1.6.** Ground heat loss coefficient.

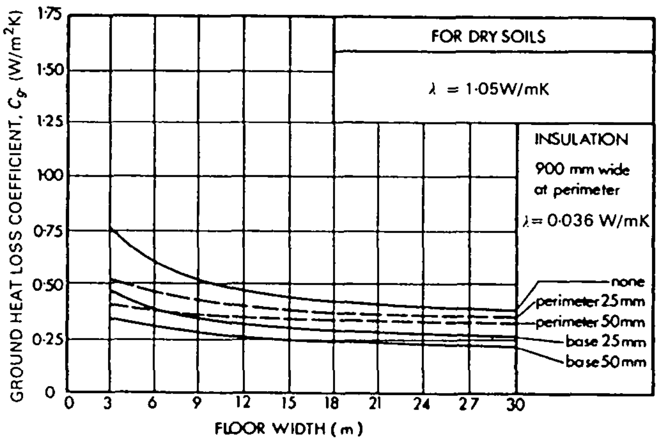


**Fig. B1.8.** Ground heat loss coefficient.

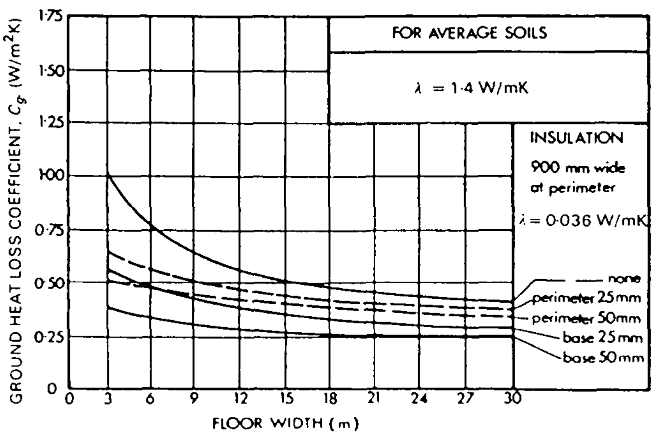


**Fig. B1.7.** Ground heat loss coefficient.

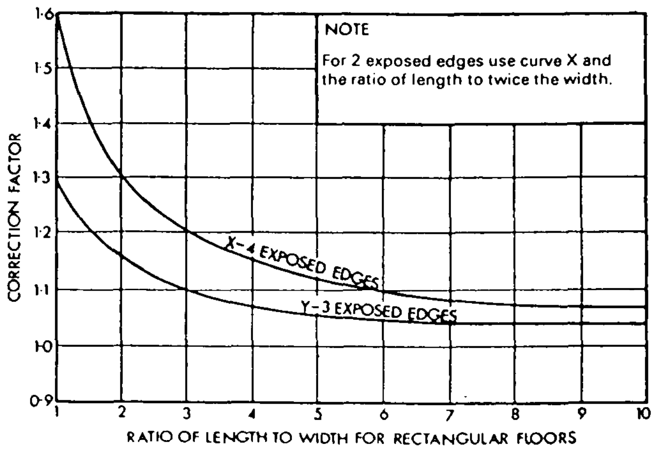


**Fig. B1.9.** Correction factor.

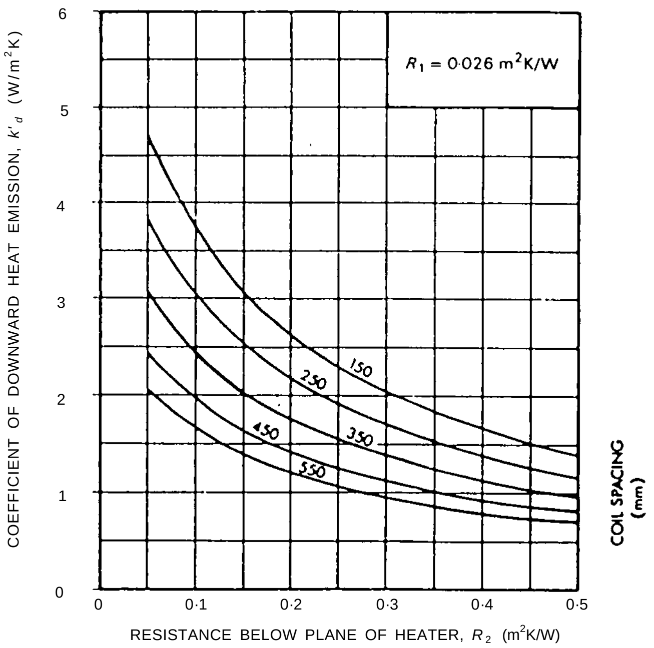


**Fig. B1.10.** Coefficient of downard heat emission.

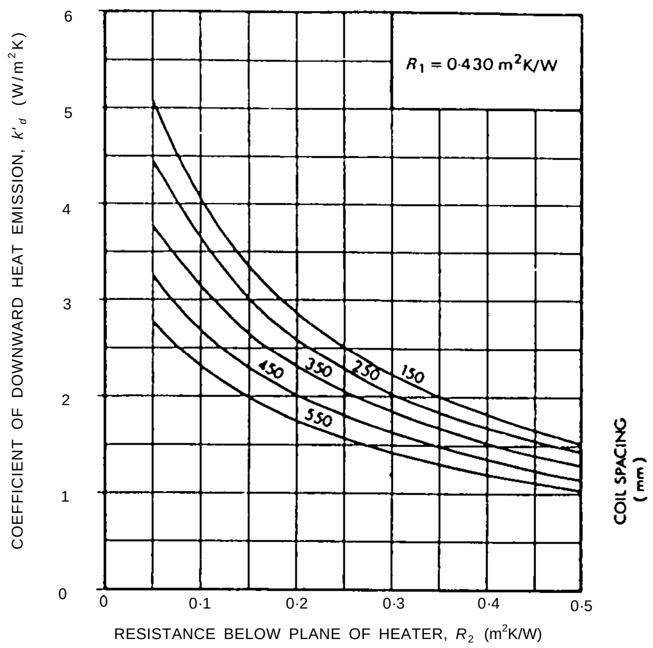


**Fig. B1.12.** Coefficient of downward heat emission.

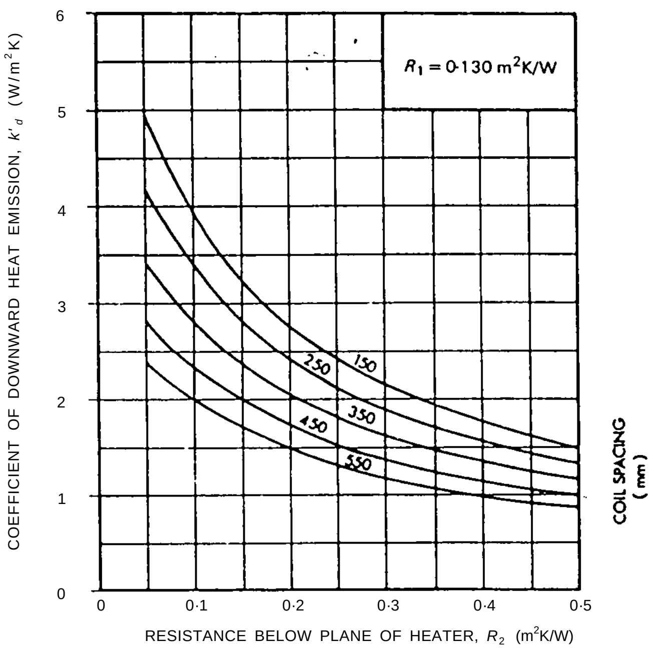


**Fig. B1.11.** Coefficient of downward heat emission.

## HEATING SYSTEMS

### Warm and Hot Water Heating Systems

Warm water or low, medium or high temperature hot water systems are categorised in Table B1.12.

Warm water systems may use heat pumps, fully condensing boilers or similar generators, or reclaimed heat. In many cases, the system design may incorporate an alternative heat generator for stand-by purposes or for extreme weather operation. Under such circumstances, the system may continue to function at warm water temperatures or could operate at more conventional LTHW temperatures.

**Table B1.12.** Design water temperatures for warm and hot water heating systems.

| Category | System designs water temperatures/°C |
|---|---|
| Warm | 40 to 70 |
| LTHW | 70 to 100 |
| MTHW | 100 to 120 |
| HTHW | over 120 |

Note: Account must be taken of the margin necessary between the maximum system operating temperature and saturation temperature at the system operating pressure.

LTHW systems are usually under a pressure of static head only, with an open expansion tank, in which case the design operating temperature should not exceed 83°C. (i.e. 17 K margin)[11]. See also Table B1.21.

Where MTHW systems operating above 110°C are pressurised by means of a head tank, an expansion vessel should be incorporated in the feed and expansion pipe. This vessel should be adequately sized to take the volume of expansion of the whole system so that boiling will not occur in the upper part of the feed pipe. On no account should an open vent be provided for this type of system.

MTHW and HTHW systems require pressurisation such that the saturation temperature at operating pressure at all points in the circuit exceeds the maximum system flow temperature required. Typical values are given in Table B1.21 and these temperatures also apply for systems employing steam space in the boiler. For completely filled systems a minimum margin of 17 K is recommended.

The 17 K margin referred to is based on the use of conventional automatic boiler plant and includes an allowance for tolerances on temperature set-points for the automatic control of heat generation output. A check must be made on actual tolerance used in the design of a control system to ensure that this allowance is adequate.

When selecting the operating pressure, allowance must be made for the effect of static head reduction at the highest point of the system and velocity head reduction at the circulating pump section, to ensure that all parts of the system are above saturation pressure with an adequate anti-flash temperature margin.

Additionally, the margin on the set point of the high temperature cut-out control should be 6 K, except for boilers fired with solid fuel automatic stokers, where the margin should be at least 10 K.

Medium and high temperature systems should be fully pressurised before the operating temperature is achieved and remain fully pressurised until the temperature has dropped to a safe level.

In all systems the heat generator or boiler must be mechanically suitable to withstand the temperature differentials, and the return temperature to the boiler must be kept high enough to minimise corrosion. Automatic controls may be used to achieve this.

*Design Water Flow Temperature*

For low temperature heating systems using natural convective or radiant appliances, the normal design water flow temperature to the system is 83°C (see also Table B1.21). Boost temperatures may be used to accelerate the warm-up rate for operation during extreme weather conditions, provided system pressures offer an adequate anti-flash margin (17 K minimum). Also, reduced temperatures are suitable for mild weather operation using flow temperature modulation. Careful design is needed where forced convective emitters are used on modulated temperature systems because of the changes in heat output characteristics with varying temperatures. Additionally, comfort aspects must be borne in mind, as forced convective emitters operating on modulated temperature systems can deliver airstreams at unacceptably low temperatures.

For MTHW and HTHW systems, heat emitters may be as for LTHW systems, except that for safety reasons, units with accessible surfaces at water temperature would not normally be employed[12, 13] (See also *Warm/Hot Water Underfloor Heating Systems*).

Embedded panel coils may be used in conjunction with a MTHW or HTHW distribution system, with insulating sleeves around the coil piping to reduce the heat flow. Alternatively, the coils can be operated as reduced temperature secondary systems by allowing only a small, carefully controlled proportion of flow temperature water to be mixed with the water circulating in the coils. Design arrangements for reduced temperature secondary systems, sometimes referred to as injection circuits, include fixed provisions for minimum dilution rates. Conventional system balancing devices with three-port automatic modulating valves to regulate mixed water temperatures and, hence, heat output are used. Automatic safety controls must prevent excessive temperatures arising in the coil circuits, as floor fabrics or finishes could be damaged very rapidly.

*Maximum Water Velocity*

The maximum water velocity in pipework systems is limited by noise generation and erosion/corrosion considerations. Noise is caused by the free air present in the water, sudden pressure drops (which, in turn, cause cavitation or the flashing of water into steam), turbulence or a combination of these. Noise will, therefore, be generated at valves and fittings where turbulence and local velocities are high, rather than in straight pipe lengths.

A particular noise problem can arise where branch circuits are close to a pump and where the regulating valve used for flow rate balancing might give rise to considerable pressure differences. Over-sizing regulating valves should be avoided as this will result in poor regulation characteristics; the valve operating in an almost shut position and creating a very high local velocity.

High water velocities can result in erosion or corrosion due to the abrasive action of particles in the water and the breakdown of the protective film which normally forms on the inside surface of the pipe. Erosion can also result from the formation of flash steam and from cavitation caused by turbulence.

Published data on limiting water velocities are inconclusive. Table B1.13 summarises the available information.

**Table B1.13.** Limiting water velocities in pipework.

| Pipe diameter/ (mm) | Steel pipework Non-corrosive water/ (m/s) | Steel pipework Corrosive water/ (m/s) | Copper pipework/ (m/s) |
|---|---|---|---|
| 50 and below | 1·5 | 1 | 1 |
| Above 50 | 3 | 1·5 | 1·5 |
| Large distribution mains with long lengths of straight pipe | 4 | 2 | – |

*Minimum Water Velocity*

Minimum water velocities should be maintained in the upper floors of high rise buildings where air may tend to come out of solution because of reduced pressures. High velocities should be used in down return mains feeding into air separation units located at low level in the system. Table B1.14 can be taken as a guide.

**Table B1.14.** Minimum water velocities.

| Pipe diameter/ (mm) | Minimum water velocity/ (m/s) |
|---|---|
| 50 and below | 0·75 |
| Above 50 | 1·25 |

Water velocities shown in Tables B1.13 and 14 are indicative parameters only; on the one hand to limit noise problems and erosion and, on ther other, to try to ensure air entrainment. Within these parameters the design engineer will need to discriminate on the selection of water velocities in a distribution system based on other considerations. It is particularly necessary to bear in mind the effect of low water velocities on flow measuring components used in balancing flow rates in systems.

*System Temperature Drop*

British practice on LTHW systems uses a typical system temperature drop of 11 K and a maximum system temperature drop of 17 K. Continental practice had tended to use higher drops; up to 40 K.

An advantage of a higher system temperature drop is the reduction in water flow rates. This will result in reduced pipe sizes with savings in capital cost and distribution heat losses, and a reduced pump duty with savings in running costs.

A digadvantage of higher system temperature drops is the need for larger and consequently more expensive heat emitters. However, if it is possible to raise the system flow temperature so that the mean water temperature remains the same, then with certain types of emitter only a small increase in size is required. With large system temperature drops the average water temperature in a radiator tends to fall below the mean of flow and return temperature and, thus, a larger surface is needed. Furthermore, on one-pipe circuits, the progressive reduction in temperature around the circuit may lead to excessively large heat emitters.

Higher system temperature drops can be used with MTHW and HTHW systems since the mean temperature of the heat emitters will be correspondingly higher. Additionally these media are well suited to use for primary distribution systems, conveying heat over long distances.

Precautions should be taken to prevent the danger of injury from contact with hot surfaces. The safe temperature for prolonged contact is relatively low and reference should be made to BS 4086[14] and other sources[12, 13].

*Use of Temperature Limiting Valves on Emitters*

On some group and district heating schemes, outlet limiting valves which permit flow only when the water temperature has dropped to a specified low level are used. This procedure minimises the water quantity to be pumped and permits indicative heat metering by water quantity alone. In such cases, care must be taken to size emitters to suit the available water temperatures. The effect of low water velocities through the emitter must also be taken into consideration, since the heat output of some convective appliances is greatly reduced under such conditions.

*Miscellaneous Components*

Data regarding relief valves, feed and expansion cisterns, etc., are given in Tables B1.15 and 16. Cistern sizes shown in Table B1.16 are based on typical system designs and are approximate only. An estimate of the water content of the particular system should always be made where there is any doubt regarding these typical data, to ensure that the cistern capacity is adequate to contain the expansion volume.

**Table B1.15.** Hot water heating boilers–recommended sizes of relief valves.

| Rated output of boiler/ kW | Minimum clear bore of relief valve/ mm | Equivalent area/ $mm^2$ |
|---|---|---|
| Up to 250 | 20 | 310 |
| 250 to 350 | 25 | 490 |
| 350 to 450 | 32 | 800 |
| 450 to 500 | 40 | 1250 |
| 500 to 750 | 50 | 1960 |
| 750 to 1000 | 65 | 3320 |

*Note:* The above sizes apply to boilers fired with solid fuel. For oil and gas-fired boilers the relief valve should be one size larger.

**Table B1.16.** Approximate sizes of feed and expansion cisterns for low pressure hot water heating systems.

| Boiler or water-heater rating (kW) | Cistern size (litre) | Ball-valve size (mm) | Cold-feed size (mm) | Open-vent size (mm) | Over-flow size (mm) |
|---|---|---|---|---|---|
| 15 | 18 | 15 | 20 | 25 | 25 |
| 22 | 18 | 15 | 20 | 25 | 32 |
| 30 | 36 | 15 | 20 | 25 | 32 |
| 45 | 36 | 15 | 20 | 25 | 32 |
| 60 | 55 | 15 | 20 | 25 | 32 |
| 75 | 68 | 15 | 25 | 32 | 32 |
| 150 | 114 | 15 | 25 | 32 | 32 |
| 225 | 159 | 20 | 32 | 40 | 40 |
| 300 | 191 | 20 | 32 | 40 | 40 |
| 400 | 227 | 20 | 40 | 50 | 40 |
| 450 | 264 | 20 | 40 | 50 | 50 |
| 600 | 318 | 25 | 40 | 50 | 50 |
| 750 | 455 | 25 | 50 | 65 | 50 |
| 900 | 636 | 25 | 50 | 65 | 50 |
| 1200 | 636 | 25 | 50 | 65 | 80 |
| 1500 | 910 | 25 | 50 | 65 | 80 |

*Notes:*
1. Cistern sizes are actual.
2. Cistern sizes are based on radiator-heating systems and are approximate only.
3. The ball-valve sizes apply to installations where an adequate mains water pressure is available at the ball valve.

*Distribution System Design*

The design of pipework distribution systems must allow for the following:

(*a*) Future extensions, where required, by the provision of valved, plugged or capped tee connections.

(*b*) Provision for isolation for maintenance (see Section B16 and BS 759[15]). Where it is necessary to carry out maintenance on a “live” system, valves must be lockable and may need to be installed in tandem.

(*c*) Thermal expansion (see Section B16).

(*d*) Provision for distribution flow rate balancing for initial commissioning or re-balancing to meet changed operational requirements (see Commissioning Code W). Typical provisions for balancing comprise the following:

(*i*) a measuring station - which may be an orifice plate, a venturi, an orifice valve or other proprietary device - provided with a pair of tappings to permit the measurement of upstream and downstream system dynamic pressures.

(*ii*) an associated regulating valve - preferably a double regulating valve or other arrangement which permits the required setting to remain undisturbed by closure.

(*e*) Provision for drainage, including drainage after pre-commission flushing; water circulation during flushing must be in excess of design flow-rates and, in order to discharge the flushing effluent effectively, drainage connections must be full diameter.

(*f*) Removal of air from the system by provision of:

(*i*) air separators, one form of which uses the principle of centrifugal force to separate the heavier constituent (water) from the lighter one (non-condensable gases); best results are achieved by locating the separator at the highest temperature point of the system where air has a greater tendency to come out of solution. The velocity of the medium requires to be above the minimum stated by the manufacturer (usually about 0·25 m/s).

(*ii*) automatic air vents for systems operating at temperatures below atmospheric boiling point.

(*iii*) air bottles with manually operated needle valves to release accumulated air, for systems operating at temperatures in excess of atmospheric boiling point.

(*g*) Provision of test points for sensing temperature and pressure at selected locations.

**Sealed Heating Systems**

Pressurisation of medium and high temperature hot water sealed heating systems referred to above may take the following forms.

*Pressurisation by Expansion of Water*

The simplest form of pressurisation uses the expansion of the water content of the system to create a sufficient pressure in an expansion vessel to provide an anti-flash margin of, say 17°C at the lowest pressure (highest point) of the system. The expansion volume for an atmospheric pressure of 1·013 bar and constant air temperature may be found from the formula:

$$V = \frac{P_1 P_2 E}{1{\cdot}013\,(P_2 - P_1)} \quad \text{B1.12}$$

where:

$V$ = volume of expansion vessel .. .. litre
$P_1$ = static pressure of system .. .. bar
$P_2$ = final pressure in the expansion vessel bar
$E$ = expansion of water contents of the system .. .. .. .. .. litre

$$E = \left(\frac{r_1 - r_2}{r_2}\right) C \quad \text{B1.13}$$

where:

$r_1$ = density of water at filling temperature (see Table B1.17) .. .. .. $kg/m^3$
$r_2$ = density of water at operating temperature .. .. .. .. .. $kg/m^3$
$C$ = system capacity .. .. .. litre

Correcting for the difference in expansion vessel air temperature between filling water temperature and final ambient temperature the formula would be:

$$V = \frac{T_1 P_1 P_2 E}{1{\cdot}013\,(T_1 P_2 - T_2 P_1)} \quad \text{B1.14}$$

where:

$T_1$ = absolute air temperature at filling .. K
$T_2$ = absolute air temperature when working (ambient temperature of location) K

An alternative formula derived by the American Society of Mechanical Engineers within the limits of 70°C and 140°C is:

$$V_t = \frac{(0{\cdot}00074t - 0{\cdot}0335)\ V_s}{\left(\dfrac{P_a}{P_f} - \dfrac{P_a}{P_o}\right)} \quad \text{B1.15}$$

where:

$V_t$ = minimum expansion tank volume .. litre
$V_s$ = system volume .. .. .. .. litre
$t$ = maximum average operating temperature .. .. .. .. °C
$P_a$ = absolute pressure in expansion tank when water first enters .. .. bar
$P_f$ = initial fill or minimum absolute pressure at tank .. .. .. .. bar
$P_o$ = maximum operating absolute pressure (including pumping head at tank) bar

The main disadvantage of a naturally pressurised expansion vessel is the ability of water to absorb air and the consequent risk of oxygen corrosion.

A diaphragm expansion vessel is divided into two compartments by a special membrane or diaphragm of rubber or rubber composition which prevents the water coming into contact with the air. On one side of the diaphragm the vessel is filled with air or nitrogen at the required pressure. The other section of the vessel is connected directly to the water system. A correctly positioned air separator will assist in de-aerating the water in the system. The tank may be sized using equation B1.15 above.

*Pressurisation by Elevated Header Tanks*

Given very careful attention to design, installation and commissioning, MTHW systems may be operated with the necessary system pressure provided by an elevated feed and expansion tank. Where the system operating temperature exceeds 110°C an expansion vessel should be sized to absorb the volume of expansion for the complete system, thus preventing water at operating temperatures entering the feed and expansion tank and causing boiling. On no account should an open vent be provided for this type of system.

*Gas Pressurisation with Spill Tank*

This form consists of a pressure cylinder maintained partly filled with water and partly with gas, usually nitrogen, which is topped up from pressure bottles. Water expansion is usually arranged to discharge from the system through a pressure control valve into a spill tank open to atmosphere or to a closed cylinder lightly pressurised with nitrogen. A pump is provided to take water from the spill tank and return it under pressure to the system as cooling-down results in a pressure drop. The pump operation is regulated by a system pressure sensor.

*Hydraulic Pressurisation with Spill Tank*

In this form the pressure is maintained by a continuously running centrifugal pump. A second pump under the control of a pressure switch is provided to come into operation at a pre-determined pressure differential and as an automatic standby to the duty pump. Surplus water is delivered to or taken from a spill tank or cylinder as described previously.

*Example of Pressure Differential*

| | |
|---|---|
| Assume system flow temperature of 120°C | |
| Allow 17 K anti-flash margin–137°C Corresponding absolute pressure | 3·4 bar |
| Assume static absolute pressure on system | 2·0 bar |
| Minimum absolute pressure at cylinder | 5·4 bar |
| Allow operating differential on pressure cylinder say– | 0·5 bar |
| Minimum operating absolute pressure of system | 5·9 bar |

*Example of Water Expansion*

| | | |
|---|---|---|
| Assume water capacity of system | | 200 000 litre |
| Assume ambient temperature of 10°C | | |
| Assume system maximum flow temperature of 120°C | | |
| Assume system minimum return temperature of 65°C | | |
| Increase in volume from 10°C to 65°C | | |
| $200\,000 \dfrac{(999{\cdot}7 - 980{\cdot}5)}{980{\cdot}5}$ | = | 3916 litre |
| Increase in volume from 65°C to 120°C | | |
| $200\,000 \dfrac{(980{\cdot}5 - 943{\cdot}1)}{943{\cdot}1}$ | = | 7931 litre |
| Total increase in volume | = | 11847 litre |

**Table B1.17.** Density of water.

| Temperature (°C) | Density (kg/m$^3$) | Absolute pressure* (bar) |
|---|---|---|
| 0 | 1000 | 1·013 |
| 20 | 998 | 1·013 |
| 40 | 992 | 1·013 |
| 60 | 983 | 1·013 |
| 80 | 972 | 1·013 |
| 100 | 958 | 1·013 |
| 120 | 943 | 1·985 |
| 140 | 926 | 3·614 |
| 160 | 907 | 6·180 |
| 180 | 887 | 10·03 |
| 200 | 865 | 15·55 |
| 220 | 840 | 23·20 |
| 240 | 814 | 33·48 |
| 260 | 784 | 46·94 |
| 280 | 751 | 64·19 |

*For temperatures of 100°C and above, the pressure given are saturation pressures. They are *not* safe working pressures.

*Maintenance Practices for Water Heating Systems*

A common practice in many hot water heating installations is to drain the complete system during summer months. This practice, involving a complete change of raw water every year, is to be deprecated. It introduces additional hardness salts and oxygen to the system resulting in very significant increases in scaling and corrosion. Where it is necessary to drain the boiler or heat generator or other parts of the system for inspection or maintenance purposes, isolating valves or other arrangements should be used to ensure that the section drained is kept to a minimum.

**Steam Heating Systems**

These are designed to use the latent heat of steam at the heat emitter. Control of heat output is generally by variation of the steam saturation pressure within the emitter. For heating applications with emitters in occupied areas low absolute pressures may be necessary in order to reduce the saturation temperature to safe levels[12, 13, 14].

The presence of non-condensable gases in steam systems (e.g. air and $CO_2$) will reduce the partial pressure of the steam, and hence its temperature, thus affecting the output of the appliance. A further adverse effect is the presence of a non-condensable gas at the inside surface of a heat emitter. This impedes condensation and, hence, heat output. It is therefore, imperative that suitable means are provided to prevent formation of $CO_2$ and to evacuate all gases from the system.

To minimise the storage volume it may be acceptable in particular situations to allow the system volume increase of water between ambient and minimum return water temperatures to go to waste, and thus, to provide storage volume sufficient only to accommodate the system volume increase of water between minimum return water temperature and maximum flow water temperature. Generally, this would only be acceptable for applications where continuous operation is required so that the water loss incurred would be occasional only.

*Safety Devices*

In addition to safety valves and normal operational automatic controls, closed heating systems require the following safety arrangements:

(*a*) high temperature cut-out

(*b*) low pressure cut-out

(*c*) high pressure cut-out.

Cut-outs must be arranged to de-energise the heat generator and re-setting must be manual.

Superheat, which must be dissipated before condensation occurs, can be used to reduce condensation in the distribution mains.

On-off control of steam systems can result in the formation of a partial vacuum, leading to condensate locking or back feeding, and infiltration of air which subsequently reduces the heat transfer.

When using modulating valves for steam, heat emitter output must be based on the steam pressure downstream of the valve, which often has a high pressure drop across it, even when fully open. (See Section B11).

Steam traps must be sized to cope with the maximum rate of condensation (which may be on start-up) but must perform effectively over the whole operational range, minimising the escape of live steam. (See Section B16).

Partial water-logging of heater batteries can lead to early failure due to differential thermal expansion. Steam trap selection should take account of this.

Where high temperatures are required (e.g. for process work) and lower temperatures for space heating, it is desirable to use flash steam recovery from the high temperature condensate to feed into the low temperature system, augmented as required by reduced pressure live steam.

**High Temperature Thermal Fluid Systems**

Where high operating temperatures are required, high temperature thermal fluid systems may be used instead of pressurised water or steam systems. These systems operate at atmospheric pressure using non-toxic media such as petroleum oil for temperatures up to 300°C or synthetic chemical mixtures where temperatures in excess of this are required (up to 400°C).

Some advantages and disadvantages of thermal fluid or heat transfer oil systems are listed below:

*Advantages*

No corrosion problems.

Statutory inspections of boilers/pressure vessels not required.

No scale deposits.

No need for frost protection of system.

Cost of heat exchangers/heat emitters less, as only atmospheric pressures involved.

Better energy efficiency than steam systems.

Operating temperature can be increased subsequent to design without increasing operating pressure.

*Disadvantages*

Medium more expensive than water (but no treatment costs).

Medium is flammable under certain conditions.

Heat transfer coefficient is inferior to that of water.

Care necessary in commissioning and in heat-up rates due to viscosity changes in medium.

Circulating pump necessary (not required for steam systems).

Air must be excluded from the system.

In the event of leakage the medium presents more problems than water.

**Warm Air Heating Systems**

These may be provided with electric or indirect oil or gas fired heaters or with a hot water heater or steam battery supplied from a central source[16, 17]. Because the radiant heat output of warm-air systems is negligible, the space air temperature will generally need to be higher for equivalent comfort standards than for a system with some radiant output; this will increase energy use, and legislative standards for limiting space temperatures should be considered. Attention is drawn to the vertical temperature gradient with convective systems and, when used for cellular accommodation, the likelihood of some spaces being overheated due to the difficulty of controlling such systems on a room by room basis.

Warm air systems incorporate many of the elements and characteristics of ventilation systems. For commercial and industrial applications reference should be made to Sections B2 and B3.

With the advent of natural gas, direct fired warm air systems are used where the heat and products of combustion, diluted by fresh air introduced into the system, are distributed to the heated spaces. In designing such installations account must be taken of the requirements of the Building Regulations 1985[2], Part J and of the Regional Gas Authority. Care must also be taken in design and application to ensure that the moisture in the products of combustion will not create condensation problems. Direct fired systems are more suited to large, single space low occupancy applications such as warehouses and hangers and should not be used to serve sleeping accommodation[18, 19].

*Reducing the Effect of Temperature Stratification*

As with all convective systems, warm air heating installations produce large temperature gradients in the spaces they serve. This results in the inefficient use of heat and high heat losses from roofs and upper wall areas. To improve the energy efficiency of warm air systems, pendant type punkah fans or similar devices may be installed at roof level in the heated space. During operational hours of the heating system these fans operate either continuously or under the control of a roof level thermostat and return the stratified warm air down to occupied levels.

The energy effectiveness of these fans should be assessed, taking into account the cost of the electricity used to operate the units. The following factors should also be borne in mind:

(*a*) the necessary mounting height of fans to minimise draughts

(*b*) the effect of the spacing of fans and the distance of the impellor from the roof soffit

(*c*) any risk to occupants from stroboscopic effects of blade movements

(*d*) the availability of multi or variable speed units.

Punkah fans may also be operated during summer months to provide air movement and offer a measure of convective cooling for occupants.

*High Temperature High Velocity Warm Air Heating Systems*

These systems, best suited to heating large, single spaces, may use indirect heating by gas or oil or direct gas heating. Relatively small volumes of air are distributed at high temperature (up to 235°C) and high velocity (30 to 42·5 m/s from the heater unit) through a system of well-insulated conventional ductwork. Air outlets are in the form of truncated conical nozzles discharging from the primary ductwork system into purpose designed diffuser ducts. The high velocity discharge induces large volumes of secondary air to boost the outlet volume and reduces the outlet temperature delivered to the space, thereby reducing stratification. Most of the ductwork thermal expansion is absorbed by allowing free movement and long, drop-rod hangers are used for this purpose. Light, flexible, axial-bellows with very low thrust loads can also be used where free expansion movement is not possible. System design and installation is generally handled as a package-deal by specialist manufacturers.

*Domestic Warm Air Heating Systems*[20]

In domestic warm air installations, air velocities should be limited to the values given in Table B1.18 to minimise noise problems. A measure of sound attenuation may be achieved by the configuration of the ductwork distribution system, but many domestic installation designs are based on very short duct runs and attenuation in such cases would be minimal.

**Table B1.18.** Air velocities for acceptable noise levels.

| Location | Maximum velocity/ m/s |
|---|---|
| Main supply/extract ducts | 3·2 |
| Branch supply/extract ducts | 2·7 |
| Face velocities: | |
| outlet grille, living rooms | 2·2 |
| outlet grille, bedrooms | 1·8 |

Supply air grilles should normally be sited at low level to provide good air diffusion and to minimise pattern staining of ceilings. Wherever possible, supply air grilles should be positioned to counteract the major heat losses of the spaces served. Since duct heat losses would not normally contribute to the occupied space heating requirement, ducts should be well insulated to give a heat loss no greater than 50 W/m length. Additionally the designer must give careful consideration to the influence of domestic warm air heating installations on the movement of smoke in the event of fire and to the need to prevent smells from the kitchen etc. from spreading throughout the dwelling.

### Warm/Hot Water Underfloor Heating Systems

*Tube Materials*

While copper, some stainless steels and black mild steel tubes with a bonded plastic coating would all meet the requirements for pipe coils embedded in a structure, modern systems use plastic tubing almost exclusively.

The plastics used are:

(*a*) Cross-linked vulcanised polythene (limiting fluid content temperature 70°C). Not suitable for thermo-welding.

(*b*) Polypropylene (limiting fluid content temperature 60°C). Tends to become brittle and difficult to work at temperatures around 0°C.

(*c*) Polybutylene (limiting fluid content temperature 90°C).

Plastics manufacturers may claim tubes to be capable of withstanding higher temperatures, but the above limiting values are consistent with a long service life of, say, 50 years.

Tubes are joined by welding (or brazing for copper) and under no circumstances should mechanical joints be embedded in the structure. Mild steel tubes are butt welded and bandaged with proprietary material to protect the joint. A socket weld is used for most small-bore plastics tubing, the process and its temperature control being automatic.

Tubes are laid in a continuous rectangular spiral format with flow and return side-by-side to even out temperature differences across the area. A maximum length of around 200 m is generally used to limit pumping pressures. Pipe coils must be fully pressure tested prior to embedding in the structure.

*Proprietary Systems*

Many systems now comprise a plastic base plate bonded to an expanded plastic thermal insulation slab. Before installing these base plates, the structural finish should be swept clean and a polythene sheet laid overall. The base plate is provided with a repetitive pattern of moulded protrusions or heads, designed for snap-in fixing of the plastic piping. The plates are used in conjunction with fixing clips and edge insulation to make up the format for a complete underfloor heating installation for any shape of space.

The thermal insulation slab also provides some sound insulation benefits by giving a measure of isolation of the structural elements.

### Electric Floorwarming

Electric floorwarming is a storage heating system which uses the mass of the floor structure to retain heat from an overnight charge and disperse it during the following day.

The cables may either be embedded directly in a concrete screed, or in tubes so that renewal is possible. In the former case, multi-stranded cables insulated with cross-linked polythene are commonly used, with screened and armoured cables being available for more demanding situations. Withdrawable systems use silicone rubber or glass fibre insulated cable.

The use of floor insulation is discussed earlier, see Figs. B1.1 and 2.

In common with other electric storage systems the design should be based on the 24 hour requirements of the building. To preserve comfort the maximum mean 24 hour floor surface temperature should not exceed 25°C for floors on which people will be standing for long periods. This imposes a limit on the maximum heat output and it will be found that well insulated buildings are necessary for storage floorwarming to be fully effective. It also follows that mechanical ventilation should have its own heat supply.

*Design*

The design of an installation involves four principle considerations:

(*a*) Calculating the heat requirements for the space, making due allowances for incidental heat gains.

(*b*) Determining the heat flow upwards and downwards from the cables, and the edge losses (see *Heat Emission from Room Surfaces*).

(*c*) Calculating the required installed load, taking account of the available hours of charge.

(*d*) Examining the swing in space temperature resulting from the variation of heat emission as the floor slab cools, and ensuring that this is acceptable.

Item (*d*) requires a complex calculation involving the thermal properties and mass of elements of the floor construction and finishes, the admittances of surfaces of the heated space and the ventilation rate. A calculation method is proposed by Danter[21].

*Operation and Control*

The off-peak period generally available is of 7 hours duration (8 or 8½ hours in Scotland). While floorwarming designs are commonly based on this charge period, it should not prevent more flexible operation. A seasonal average day charge usage of 10% for instance, usually regarded as an acceptable economic limit, can be achieved with as much as 33% day charge usage in design day weather, which occurs infrequently. Similarly, using a maximum demand tariff with an off-peak feature, it may be possible to expand the night operation during design weather without affecting demand charges, if these are determined by the daytime load.

Where it forms the sole source of heat, electric floorwarming can be controlled by room thermostat, weather sensitive charge controller or in some instances, a floor thermostat. Where supplementary heating is used, a room thermostat may be inappropriate since the air temperature is a function of the secondary heat source. In a large building, the installation should be divided into appropriate control zones such that, for instance, areas with different heat gains or orientations are controlled according to their individual requirements.

## STORAGE HEATING SYSTEMS

In storage heating systems sufficient energy must be generated and stored during the operational period of the heat generator to provide heating for the remainder of the period of demand. Thus, when sizing a thermal storage system it is necessary to consider the aggregate heating energy requirements for the entire occupational period for a design day. This includes the decay in space temperature which may occur when the system is off.

To calculate the 24 hour heating requirement it is necessary to consider the thermal weight of the building. A thermally heavyweight building absorbs a large amount of heat into its fabric and dissipates it slowly; hence the mean internal temperature remains high. Conversely a thermally lightweight building absorbs less heat and its internal temperature falls rapidly, resulting in a lower 24 hour mean internal temperature.

System losses must also be considered. The standing loss is energy lost from the surface of a water storage vessel or outer casing of a solid core storage unit, and this loss is directly related to the level of insulation of the vessel. Secondary losses, associated with the distribution system, must also be taken into account in sizing the storage vessel.

### Method of Sizing

The sizing procedure and data below are suggested by the Electricity Council. Attention is also drawn to the procedures given in Sections A5 and A9. The method below applies to water thermal storage and electric solid core storage systems which incorporate full control of the heat output from the storage unit.

The required charge acceptance for the operating period of the heat generator on a design day is given by:

$$E = 24(Q_u + Q_i)\,I + H_p(Q_v - Q_i) - G + L_s \qquad \text{B1.16}$$

where:

| | | |
|---|---|---|
| $E$ | = charge acceptance during heat generator operating period on a design day .. .. .. .. .. | kWh |
| $Q_u$ | = design fabric heat loss .. .. | kW |
| $Q_i$ | = background infiltration heat loss .. | kW |
| $I$ | = intermittency factor | |
| $H_P$ | = occupied period .. .. .. | h |
| $Q_v$ | = design ventilation heat loss .. .. | kW |
| $G$ | = useful design day heat gain .. .. | kWh |
| $L_s$ | = 24 hour system distribution heat loss | kWh |

The intermittency factor, $I$, is a function of the daily heating and preheating period, and the thermal response factor for the building:

$$I = \frac{Cf_r}{Cf_r + B - C} \qquad \text{B1.17}$$

where:

$$B = (24 - H_h)(Q_u + Q_i) + H_h(Q_u + Q_v) \quad .. \qquad \text{B1.18}$$

$$C = H_h P(Q_u + Q_i) + H_p(Q_u + Q_v) \quad .. \quad .. \qquad \text{B1.19}$$

$$f_r = \frac{Y'}{U'} \quad \frac{\Sigma(AY) + \frac{1}{3}NV}{\Sigma(AU) + \frac{1}{3}NV} \quad .. \quad .. \quad .. \qquad \text{B1.20}$$

where:

| | | | |
|---|---|---|---|
| $H_h$ | = | preheat period .. .. .. .. | h |
| $H_p$ | = | occupancy period .. .. .. | h |
| $P$ | = | overload ratio | |
| $f_r$ | = | building thermal response factor | |
| $Y'$ | = | thermal admittance and ventilation coefficient .. .. .. .. | W/K |
| $U'$ | = | thermal transmittance and ventilation coefficient .. .. .. .. | W/K |
| $N$ | = | mean 24 hour natural ventilation rate | $h^{-1}$ |
| $V$ | = | room volume .. .. .. .. | $m^3$ |
| $A$ | = | area of surface bounding the heated space .. .. .. .. .. | $m^2$ |
| $Y$ | = | thermal admittance of the surface .. | $W/m^2K$ |
| $U$ | = | thermal transmittance of the walls etc. bounding the heated space .. | $W/m^2K$ |

Table B1.19 suggests preheating times for a range of building thermal response factors and overload ratios, and Table B1.20 gives typical values of $f_r$

**Water Storage Heating Systems**

Having calculated the net heating energy requirement and system losses, the total vessel water capacity and an appropriate temperature drop can be determined. The temperature drop will be the difference between the minimum and maximum storage temperatures: the greater this temperature difference the greater will be the heat storing capacity of a given water volume. Maximum operating water temperatures are given in Table B1.21.

Table B1.21 lists the data required for the design of storage vessels. Absolute operating pressures and typical related anti-flash margins will be seen to differ. For high working temperatures an anti-flash margin of 17 K is recommended in accordance with HSE Guidance Note PM5[11]. At lower storage temperatures an anti-flash margin of 10 K is acceptable. The reasons for this are twofold:

(*a*) Storage systems have a high thermal (water) content relative to the heat generator output, especially where immersion heaters are used. Hence any residential heat in the heat generator is quickly dissipated in the large water mass, minimising the water temperature overswing.

(*b*) Between the temperatures of 100 and 105°C the rise in saturated steam pressure is relatively low compared with the rise in saturated steam pressure relative to a 5 K rise at higher pressures.

Anti-flash margins must be maintained both with and without circulating pumps in operation. Depending on the location of a pump, the pump pressure could have the effect of increasing or decreasing system static pressure.

In storage applications it is important that the calibration of thermostat sensors is accurate, and that they are located in a representative position where high temperatures will be sensed readily.

*Thermal Insulation of the Storage Vessel*

Because of the high temperatures at which water is stored, the vessel must be well insulated. If standing losses from the vessel are high, then additional storage capacity and thus a larger tank will be required.

The annual standing energy loss per unit area of tank surface is illustrated in Fig. B1.13 for insulation with a thermal conductivity of 0·048 W/m K for a range of thicknesses and storage temperatures.

The optimum thickness of insulation for the storage vessel can be found from Fig. B1.13 by determining the cost of standing loss energy over the required pay back period, and comparing this with the cost of insulation. The optimum insulation thickness occurs where the sum of these two costs is a minimum.

*Storage Expansion Volume*

The expansion volumes which need to be accommodated in a feed and expansion tank or a separate expansion vessel for a range of maximum storage temperatures, are shown in Table B1.21. Volumes are given in litres per 1000 litres of storage capacity for a cold start where

**Table B1.20.** Typical values of $f_r$

| Building thermal characteristics | Thermal response factor, $f_r$ |
|---|---|
| Heavyweight | 6·5 |
| Mediumweight | 4·0 |
| Lightweight | 2·5 |

**Table B1.19.** Recommended preheat times for storage heating systems.

| Thermal response factor $f_r$ | Hours of occupancy, $H_p$ | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 6 | | | 8 | | | 10 | | | 12 | | | 16 | | |
| | Overload ratio $P$ | | | Overload ratio $P$ | | | Overload ratio $P$ | | | Overload ratio $P$ | | | Overload ratio $P$ | | |
| | 1·2 | 1·5 | 2·0 | 1·2 | 1·5 | 2·0 | 1·2 | 1·5 | 2·0 | 1·2 | 1·5 | 2·0 | 1·2 | 1·5 | 2·0 |
| 2·5 | 5·4 | 4·0 | 2·8 | 5·0 | 3·7 | 2·5 | 4·6 | 3·3 | 2·2 | 4·0 | 3·0 | 2·0 | 2·9 | 2·2 | 1·5 |
| 3 | 5·8 | 4·4 | 3·0 | 5·4 | 4·1 | 2·8 | 5·0 | 3·7 | 2·5 | 4·4 | 3·3 | 2·2 | 3·3 | 2·4 | 1·7 |
| 4 | 6·1 | 4·7 | 3·2 | 5·7 | 4·4 | 3·0 | 5·2 | 3·9 | 2·7 | 4·6 | 3·5 | 2·4 | 3·5 | 2·6 | 1·9 |
| 6 | 6·6 | 5·2 | 3·5 | 6·2 | 4·9 | 3·4 | 5·6 | 4·3 | 3·1 | 5·0 | 3·9 | 2·8 | 3·9 | 3·0 | 2·2 |
| 8 | 7·0 | 5·6 | 3·8 | 6·6 | 5·2 | 3·6 | 5·9 | 4·6 | 3·3 | 5·3 | 4·1 | 3·0 | 4·2 | 3·2 | 2·4 |
| 10 | 7·1 | 5·7 | 3·9 | 6·7 | 5·3 | 3·7 | 6·0 | 4·7 | 3·4 | 5·4 | 4·2 | 3·1 | 4·2 | 3·3 | 2·5 |

**Table B1.21.** Temperatures, pressures and storage volumes for water storage heating systems.

| Water temperature | | Anti-flash margin/ K | Absolute saturation pressure/ * bar | Nominal absolute operating presure/ † bar | | Approximate height of water column/ m | Water density/ * kg/m$^3$ | Minimum storage volume for 100 kW heat content/ * (litre/100kW) | Expansion volume of storage water/ (litre/1000 litre stored) | |
|---|---|---|---|---|---|---|---|---|---|---|
| Storage/flow °C | Drop/ K | | | | | | | | Cold start | Normal operation |
| 90 | 50 | 10 17 | 0·701 | 1·013 | 1·295 | 13·2 | 965·3 | 1773 | 35·6 | 27·9 |
| 95 | 55 | 10 17 | 0·845 | 1·209 | 1·532 | 15·6 | 961·9 | 1615 | 39·3 | 31·5 |
| 100 | 60 | 10 17 | 1·013 | 1·433 | 1·805 | 18·4 | 958·3 | 1484 | 43·2 | 35·4 |
| 105 | 65 | 10 17 | 1·209 | 1·691 | 2·114 | 21·6 | 954·8 | 1373 | 47·1 | 39·2 |
| 110 | 70 | – 17 | 1·433 | – | 2·468 | 25·2 | 951·0 | 1278 | 51·2 | 43·3 |
| 120 | 80 | – 17 | 1·985 | – | 3·318 | 33·8 | 943·1 | 1123 | 60·0 | 52·0 |
| 130 | 90 | – 17 | 2·701 | – | 4·390 | 44·8 | 934·8 | 1002 | 69·4 | 61·4 |
| 140 | 100 | – 17 | 3·614 | – | 5·724 | 58·4 | 926·1 | 906 | 79·5 | 71·4 |
| 150 | 110 | – 17 | 4·760 | – | 7·364 | 75·1 | 916·9 | 826 | 90·3 | 82·2 |
| 160 | 120 | – 17 | 6·181 | – | 9·356 | 95·4 | 907·4 | 760 | 100·7 | 93·4 |
| 170 | 130 | – 17 | 7·920 | – | 11·750 | 119·8 | 897·3 | 705 | 114·1 | 105·7 |

Notes:
System return temperature = 40°C
Cold start vessel water temperature = 10°C
Normal operation water temperature = 40°C

* These data are based on the actual water storage temperature
† These pressures allow for the anti-flash storage temperature

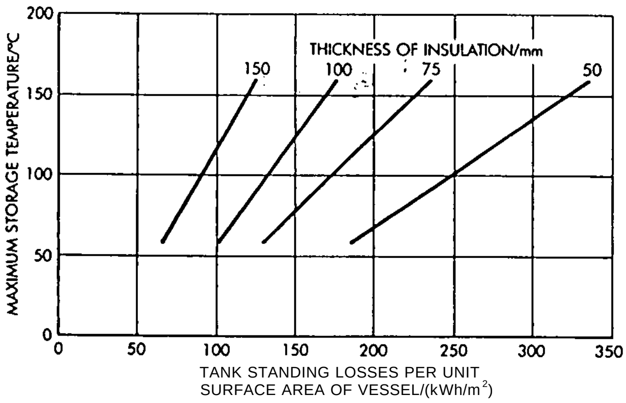


**Fig. B1.13.** Optimum thickness of insulation.

initial water temperature is assumed to be 10°C, and for normal operation where the initial temperature will be 40°C. This expansion volume is for the storage vessel only; for the total expansion the water content of the primary and secondary circuits must also be considered.

*Principle of Operation*

When heating in the building is called for, high-temperature water is drawn from the top of the storage vessel and replaced by return water at the bottom. A proportion of the return water is mixed with the high-temperature water through a thermostatic mixing valve which controls the temperature of the water flowing to the system (see Figs. B1.14 and 15 on the following pages). This can be varied in accordance with outside temperature conditions or maintained at a suitable constant temperature to serve air heater batteries etc.

The process continues throughout the day, the level of cool water at the bottom of the storage vessel gradually rising but not mixing with the hot. At the end of the heating period in design weather, the hot water store would be practically exhausted.

*Automatic Control of Water Storage Systems*

This is dealt with below for electric water storage systems but the principles apply generally.

## ELECTRIC STORAGE HEATING

Electric storage systems operate on a tariff in which electricity is charged at a lower rate during an off-peak period at night. The night rate periods and the off-peak tariff arrangements vary slightly depending on location but are generally 7 hours within the period midnight to 08.00.

In buildings which operate intermittently, the 24 hour building energy requirement is calculated on the basis of a mean daily internal temperature that will reflect the internal temperature depressions occurring during unoccupied periods.

The thermal storage media in common use are elevated temperature water systems, and high density refractory materials. The electric heating charge in water thermal storage systems is by immersion heaters, either within the storage tank, in separate flow boilers or, in very large installations, by means of electrode boilers. Solid core systems are heated by direct electric resistance elements.

Heat is discharged from water thermal storage vesels by conventional LTHW systems. Solid core storage units can be arranged to discharge heat directly by air, or by LTHW. Thermal storage in its simplest form, as a room storage heater, heats the space directly by surface radiation and convection.

*Storage Materials and Thermal Insulation*

Whatever the method of heat dissipation, the quantity of heat stored is normally a function of the specific heat and mass of the storage material and the temperature differential during heating and cooling. With certain storage materials this may be augmented by the latent heat of fusion or the heat of hydration.

Suitable storage materials are refractories capable of withstanding temperatures of up to 900°C and various alloys of iron usable up to 750°C. Densities of the refractories are in the range 1920 to 2800 kg/m$^3$ with specific heat capacities of 465 to 605 J/kg. The metals have densities of about 8000 kg/m$^3$ and specific heat capacities of 233 to 372 J/kg.

Thermal insulation is used to minimise uncontrolled surface heat losses. Most materials have higher thermal conductivities at temperatures of 800°C than at 200°C, the notable exception being an opacified silica aerogel which has a $\lambda$ -value of 0·036 W/mK at 800°C.

*Preheat*

Where electric thermal storage systems are designed to operate in intermittently heated buildings, it is often advantageous due to tariff structures, to provide the bulk of the preheat energy as direct heating from the immersion heaters, flow boilers etc. Preheating of the building will therefore take place during the same period in which the thermal store is being charged. More preheat energy introduced at this time leads to less energy needed to be stored, and hence a smaller thermal storage can be specified. Optimum start controllers can be employed to reduce preheat periods during milder weather.

**Types of Electric Heat Generators**

Electric heaters may be one of two types according to the size of the installation and the electricity supply available:

(*a*) Immersion heaters, consisting of resistance elements inside metallic tubes or blades and suitable for low voltages up to 250 V or, when in balanced arrangement, up to 500 V.

The heaters are usually contained in a tubular flow-boiler external to the storage vessel and with an independent circulating pump. In small installations the heaters are sometimes set in the storage vessel near the base, though this may lead to servicing difficulties. For immersion heater applications total installed load is rarely more than 300 kW per vessel.

(*b*) Electrode heaters, which may be connected to a three-phase a.c. supply of medium voltage (400 to 600 V) or high voltage (600 to 11,000 V), are suitable for installations up to 5,000 kW or more. Electrode heaters make use of the resistance of the water, the current passing through the water from one electrode to the other. The load is varied by increasing or decreasing the length of path which the current has to take, by interposing non-conducting shields between or around the electrodes.

Electrode heaters usually take the form of separate packaged units, since the load-regulating mechanism is complex and the electrode shields must be accessible for inspection and maintenance without involving water loss. Additionally, the heat release at the current path is such that brisk circulation of water is necessary and circulating impellers for this purpose are often built into larger heaters. One heater may be used in conjunction with a number of storage cylinders to provide the capacity required, a separate pump or pumps being arranged to circulate water between the heater and the cylinders during the charging period. The volume flow rate for the circulating pump is usually based on a temperature rise of between 8 and 11 K at heater rated output. As the temperature in an electrode heater rises the resistivity of the water decreases and the load correspondingly tends to increase. In such cases the shield positioning mechanism is often controlled automatically to maintain a constant pre-selected load.

*Element Sizing*

Energy delivered to the building via the storage vessel or from direct preheating will be provided within the off-peak period. The element must be sized such that there is sufficient capacity to achieve the preheating of the building as well as charging the vessel for the normal occupation period heating. During the charge period, additional heating capacity must be available to offset the primary and secondary pipework heat losses, together with tank standing loss.

*Automatic Control of Electric Water Thermal Storage Systems*

For the arrangement shown in Fig. B1.14, where immersion heaters are installed in the vessel directly, the immersion heaters will charge the vessel under the control of a sensor at $C_1$. The sensor is located above the plane of the immersion heaters and remote from the rising convection currents. Turbulence within the tank must be minimised during charging and preheat periods in order to promote stratification in the vessel. To achieve this, the flow from the tank should be designed with a low velocity exit to preserve the stratification set up by the sparge pipe.

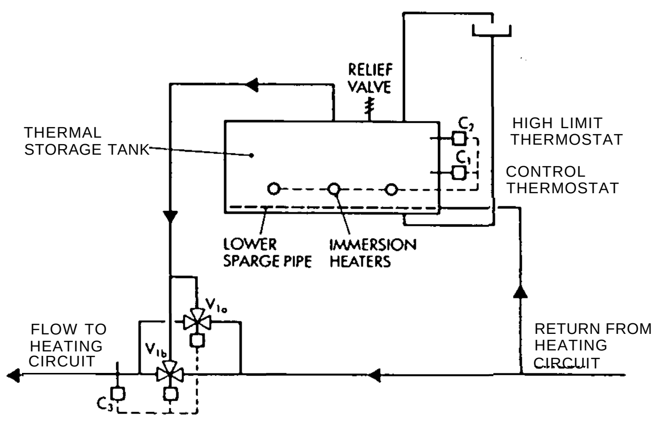


**Fig. B1.14.** Arrangement of thermal storage tank heated by immersion heaters.

The safety controls for a flow boiler as shown in Fig. B1.15 or for an electrode boiler comprise a flow switch and a high limit thermostat. Heat output from the boiler is controlled by a sensor at $C_1$. During the charging and preheat period, the primary pump circulates water from a sparge pipe at the bottom of the tank, through the boilers and returns the heated water through a sparge pipe positioned along the centre of the tank. The return sparge pipe is located to maintain any stratification that may exist from the previous day's charge.

The flow rate of the primary pump should be selected so that the water temperature rise through the boiler is no more than 10 K. This ensures that the water temperature in the vessel at the end of the charge period is as near uniform as practically possible which gives maximum utilisation of the storage capacity.

Heat input to the heating system is controlled by a thermostat, $C_3$, which regulates the heat injection valve arrangement to maintain a constant water flow temperature. On a design winter day the tank would discharge all its heat to the heating system and, towards the later stages of operation, the full heating flow will be diverted through the tank. The heat injection valve arrangement must, therefore, be capable of stable control over a wide range of injection flow rates as the storage temperature decays from its design maximum to its minimum level. Improved control stability is achieved by using two three-port valves, $V_{1a}$ and $V_{1b}$, in parallel as shown in Figs. B1.14 and 15. These are controlled in sequence from the sensor, $C_3$. The smaller valve is selected to regulate low injection rates up to about 25% of the system flow rate, when the larger valve would take over. Sensor $C_3$ should be located to sense the water temperature with a minimum time-lag after injection so that the regulation of the three-port valve is stable and hunting does not occur.

**Solid Core Storage Equipment**

Controlled output solid core storage heaters are available as proprietary units with up to 1800 MJ of storage capacity or in banks up to the required capacity. They are sized using the same procedure as for hot water thermal storage. Figs. B1.16 and 17 show the two basic types of unit. Fig. B1.16, known as electricaire provides warm air directly and Fig. B1.17, known as a dry core boiler incorporates an integral heat exchanger for the provision of LTHW. In both cases air is used as the primary medium for extracting heat from the core.

The electricaire unit can be used for space heating either as a free standing unit discharging directly into the space or as a remote unit, using distribution ductwork. Electricaire can also be used for supplying heat in plenum heating or ventilation systems. The dry core boiler serves a conventional LTHW heating system.

Storage fan heaters operate on the same principle as electricaire units and are sized on the same basis. They are usually designed specially for application in individual rooms, but stub ducts may be used for extension to adjacent rooms.

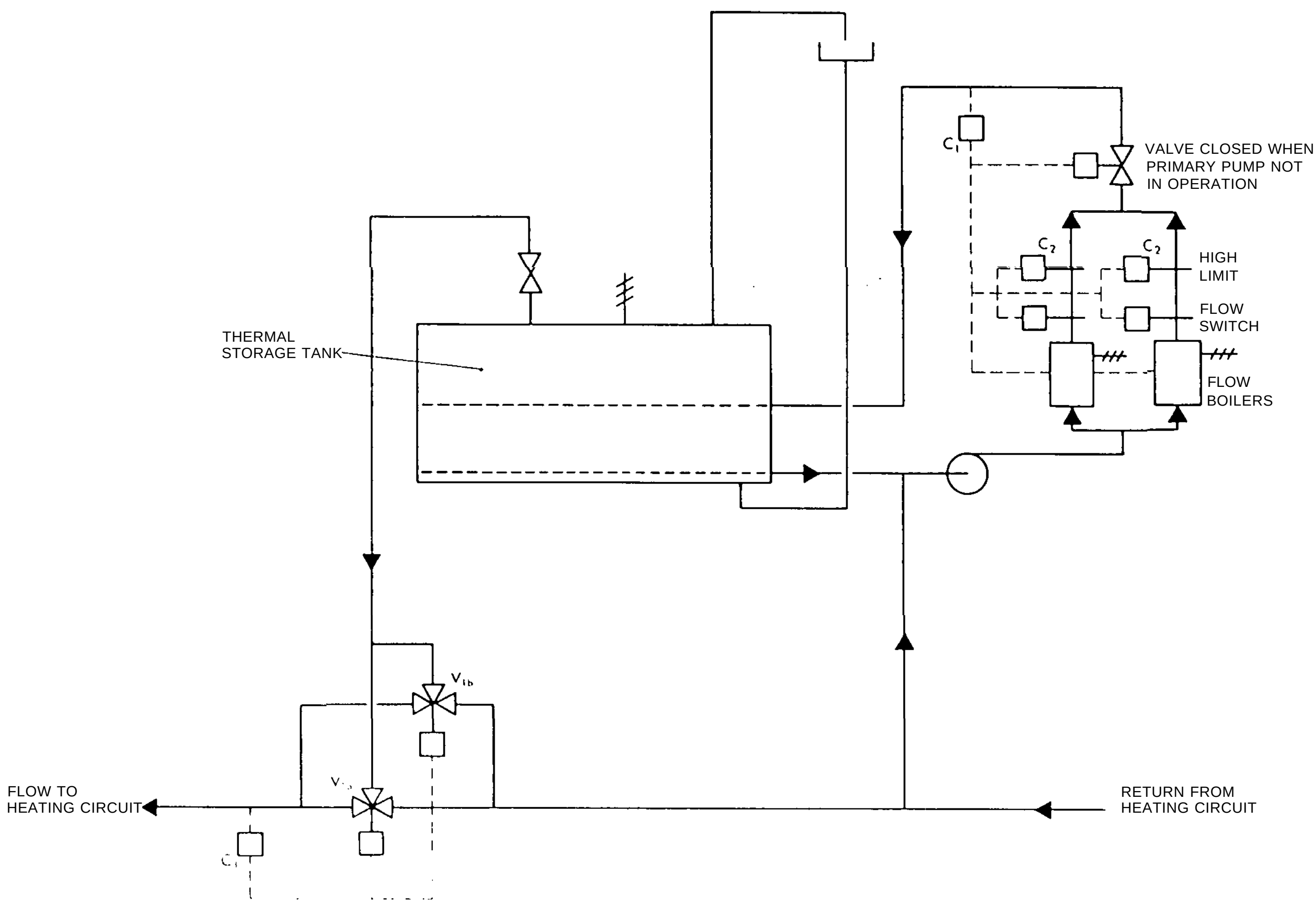


**Fig. B1.15.** Arrangement of thermal storage tank heated by flow boilers.

*Room Storage Heaters*

Room storage heaters are designed to fully charge during the various off-peak periods. The maximum surface temperature of free-standing storage heaters is limited to 85 K above ambient (see BS 3456: Section 2.26: 1973[22]). Surface heat output by radiation and natural convection is approximately 1 kW/m$^2$ for these temperatures, and the rate of convective heat output is usually increased by providing an air space within the outer case, and one or more grille-type outlets. Storage heaters are insulated heavily at the base to minimise the surface temperature at floor level.

Most modern appliances incorporate a damper. The core is arranged to leave an air channel through the centre which is closed by the damper. The damper is opened by a bimetallic thermostat control to give a boost in output, if required, near the end of the day.

**Method of Sizing of Storage Heaters**

The following sizing procedure has been suggested by the Electricity Council and is an extension to that given under *Storage Heating Systems.*

Where there is an extended occupation period it may be necessary to supplement the storage heaters with direct electric heaters. The time at which supplementary heating may be required is given in Table B1.22.

Where the occupation period ends before the time given in Table B1.22, an intermittency factor for storage heater systems has been found from field trial results. The intermittency factor is given by:

**Table B1.22.** Times when storage heaters may need to be supplemented by direct electric heaters.

| Building thermal characteristics | Time when supplementary heating may be required |
|---|---|
| Heavy weight | 21·00 |
| Medium weight | 20·00 |
| Light weight | 19·00 |

$$I = 1{\cdot}1016 - \frac{1}{f_r}\left(0{\cdot}24 + \frac{\overline{G}}{Q_u + Q_v}\right) \quad .. \quad \text{B1.21}$$

where:

$I$ = intermittency factor for storage heater systems

$f_r$ = building thermal response factor

$\overline{G}$ = useful design day heat gain averaged over 24 hours .. .. .. .. kW

$Q_u$ = design fabric heat loss .. .. kW

$Q_v$ = design infiltration/ventilation heat loss .. .. .. .. .. kW

The intermittency factor calculated in equation B1.21 above should then be used in equation B1.16 in order to find the required storage heater charge acceptance, *E.*

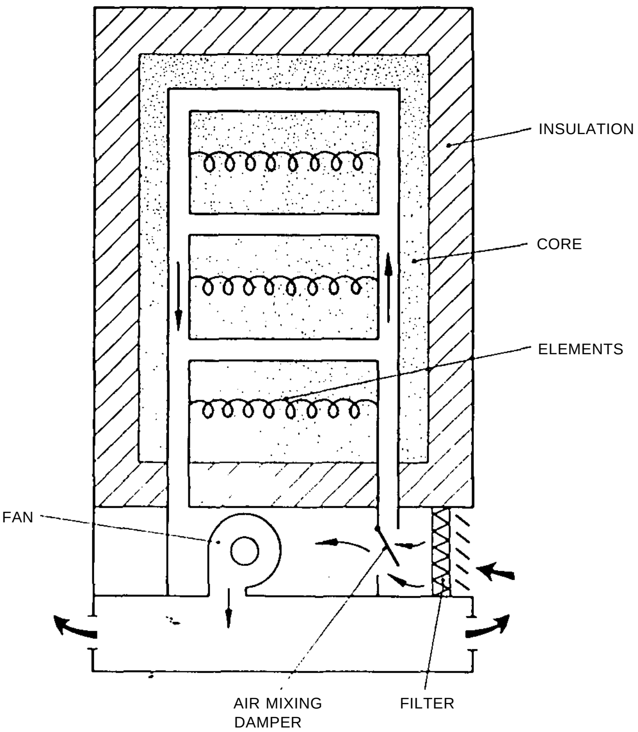


**Fig. B1.16.** Electricaire unit.

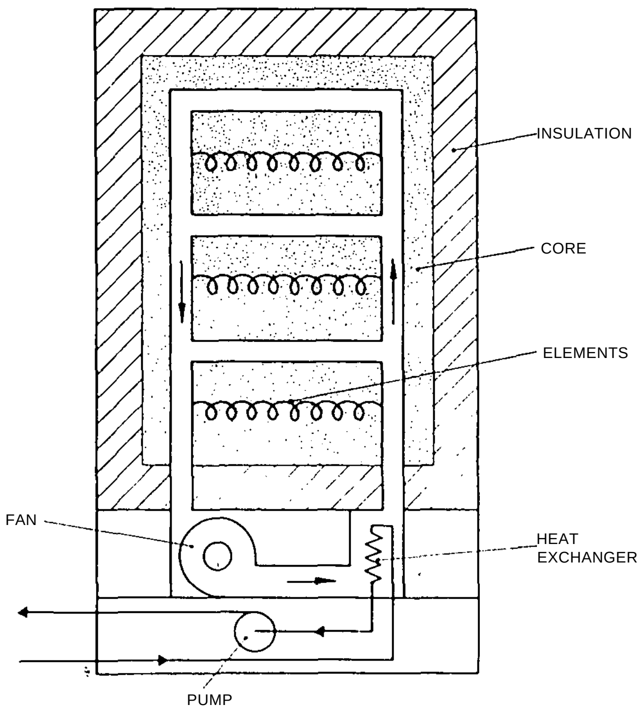


**Fig. B1.17.** Dry core boiler.

### Storage Heater Controls

Storage heating equipment requires some form of input controller to regulate the amount of charge taken, and a protection device to prevent the core from being overheated. The main types are as follows:

*Core Temperature Control*

The simplest and most common form of input control is a manually adjustable core thermostat. The sensing element of the thermostat may be placed in contact with the core, or outside the insulation, where it is termed an inferential thermostat.

*Weather Sensitive Time Control*

This type of charge control regulates the duration of charge as a function of external temperature, by delaying the onset of the charge until some time after the start of the off-peak period. Charging continues until the end of the off-peak period.

*Room Thermostat Control*

A recent development is the use of a room thermostat for the control of storage heaters. Each heater operates under the control of an air thermostat placed in the same room. The thermostat used should have a close differential (less than 0·5 K), be without an accelerator and have a minimal self heating effect.

## HEATING EQUIPMENT – ATTRIBUTES AND APPLICATIONS

### Water System Heating Equipment

The range of heat emitters may be divided into three generic groups:

(*a*) radiant

(*b*) natural convective

(*c*) forced convective.

Table B1.23 lists the principal types of appliance in each group, together with descriptive notes. Typical emission ranges are quoted for each type over its normal span of working temperatures. These are intended as a guide only and manufacturers' catalogues should be consulted for detailed performance values.

**Table. B1.23.** Characteristics of water system heating equipment.

| Type | Description | Advantages | Disadvantages | Emission range |
|---|---|---|---|---|
| **Radiant** | | | | |
| Radiant panel | Consists of steel tube or cast-iron waterways attached to a radiating surface. Back may be insulated to reduce rear admission or may be left open to give added convective emission. Particularly useful for spot heating and for areas having high ventilation rates (e.g. loading bays), the radiant component giving a degree of comfort in relatively low ambient air temperatures. | No moving parts, hence little maintenance required; may be mounted at considerable height or, in low temperature applications, set flush into building structure. | Slow response to control; must be mounted high enough to avoid local high intensities of radiation, e.g. on to head. | 350 $W/m^2$ to 15 $kW/m^2$ of which up to 60% may be radiant. |
| Radiant strip | Consists of one or more pipes attached to an emissive radiant surface. Is normally assembled in long runs to maintain high water flow rates. The back may be insulated to reduce rear emission. When using steam adequate trapping is essential together with good grading to ensure that tubes are not flooded. Multiple tube types should be fed in parallel to avoid problems due to differential expansion. Hanger lengths should be sufficient to allow for expansion without lifting the ends of the strip. Heating media may be steam, hot water or hot oil. | No moving parts, hence little maintenance required; may be mounted at considerable height or, in low temperature applications, set flush into building structure. | Slow response to control; must be mounted high enough to avoid local high intensities of radiation, e.g. on to head. | 150 W/m to 5 kW/m of which radiant emission may be up 10 65% of total. |
| **Natural convective** | | | | |
| Radiators | Despite their name, 70% of the emission from these devices is convective. Three basic types are available: column, panel and high output, the last incorporating convective attachments to increase emission. Panel radiators offer the least projection from the wall but emission is higher from column and high output units.<br><br>In application they should be set below windows to offset the major source of heat loss and minimise cold downdraughts. | Cheap to install; little maintenance required. | Fairly slow response to control. With steel panel radiators there is a risk of corrosive attack in areas having aggressive water, which may be accentuated by copper swarf left in the radiator. This leads to rapid failure unless a suitable inhibitor is used. Not suitable for high temperature water or steam | 450 to 750 $W/m^2$. |

**Table. B1.23.** Characteristics of water system heating equipment–*continued*.

| Type | Description | Advantages | Disadvantages | Emission range |
|---|---|---|---|---|
| **Natural convective (continued)** Natural convectors | Compact units with high emissions. Often fitted with damper to reduce output when full emission not required, usually to about 30% of full output. Heat exchangers normally finned tube. Units may be built into wall of building. | May be used on high temperature hot water or low pressure steam without casing temperature becoming dangerously high; fairly rapid response to control. | Take up more floor space than radiators. Likelihood of fairly high temperature gradients when using high temperature heating media. | 200 W to 20 kW |
| Continuous convectors | Comprise single or double finned tube high output emitters in factory-made sill height sheet metal casings or builders' work enclosures which may be designed to fit wall to wall. Can be fitted with local output damper control, which reduces the emission to approximately 30% of the full output. They should be placed at the point of maximum heat loss, usually under windows. The wall behind the unit should be well insulated.<br><br>For long-run applications, distribution of flow water must be provided to modular sections of the unit to ensure that input remains reasonably consistent over full length of unit. For builders' work casings inlet and outlet apertures must have the free-area requirements stipulated by the manufacturer. | Take up relatively little space; give even distribution of heat in room. May be used with medium temperature hot water or low pressure steam without casing temperatures becoming dangerously high. Return pipework may be concealed within casing. | May produce large temperature gradients on high temperature heating media if poorly sited. | 500 W/m to 4 kW/m |
| Skirting heating | Finned tube emitters in a single or double skirting height sheet metal casing, usually with provision for a return pipe within the casing. Applications and distribution of flow water similar to continuous convectors. | May be used on water or low pressure steam. Give low temperature gradients in the room. All pipe work concealed. | Relatively low output per m of wall. More work involved when installing in existing building as existing skirting has to be removed. | 300 W/m to 1·3 kW/m |
| **Forced convective** Fan convectors | These units give a high heat output for volume of space occupied by the unit, together with the ability to distribute the heat over a considerable area using directional grilles.<br><br>May be used to bring in heated fresh air for room ventilation.<br><br>Leaving air temperatures should be above 35°C to avoid cold draughts. Where mixed systems of radiators and fan convectors are installed it is advisable to supply fan assisted units on a separate constant temperature circuit to avoid the above problems. To minimise stratification, leaving air temperatures above 50°C should be avoided.<br><br>Must not be used on single pipe systems. Care must be taken at design stage to avoid unacceptable noise levels.<br><br>Control by speed variation or on/off regulation of fan. | Rapid response to control, by individual thermostat. By use of variable speed motors rapid warm up available on intermittent systems; filtered fresh air inlet facility. | Electric supply required to each individual unit. | 2 to 25 kW |

**Table. B1.23.** Characteristics of water system heating equipment–*continued*.

| Type | Description | Advantages | Disadvantages | Emission range |
|---|---|---|---|---|
| **Forced convective (continued))** | | | | |
| Unit heaters | A unit with a large propeller or centrifugal fan to give high air volume and wide throws. Louvres direct the air flow in the direction required. May be ceiling mounted, discharging vertically or horizontally or floor mounted. Can be used with fresh air supply to ventilate buildings. Large units may be mounted at a considerable height above the floor to clear travelling cranes etc. May be used with steam or not water but care should be taken to restrict leaving air temperatures, usually 40 to 55°C to avoid reduction of downward throw and large temperature gradients in the building. The air flow from the units should be directed towards the points of maximum heat loss.<br>Control as for fan convectors. | Rapid response to control by individual thermostat; by use of multi-speed motors rapid warm up available on intermittent systems; filtered fresh air inlet facility. | Electric supply required for each individual unit. | 3 to 300 kW |

## Electric Heating Equipment

Where electric heating equipment is installed within the space to be heated, the total electrical input is converted into useful heat. There are two categories of electric heating equipment, direct acting and storage heating.

The two types of electric heating can be used independently, or to complement one another to meet particular heating requirements. Table B1.24 gives a brief description of the different types of electric heating.

**Table B1.24.** Electric heating equipment.

| Type | Description | Advantages | Disadvantages | Emission range |
|---|---|---|---|---|
| **Radiant** | | | | |
| High intensity radiant heaters | Consists of high temperature elements mounted in front of polished reflector.<br>Element can be silica or metal sheathed wire (up to 900°C) or Quartz lamp (up to 2200°C). | Fast response time. Little regular maintenance required. May be mounted at considerable height.<br>Quartz lamp heaters have improved beam accuracy allowing higher mounting heights. Especially suitable for spaces with high air movement. | Must be mounted sufficiently high to avoid local high intensities of radiation e.g. on to head. | 0·5 to 6 kW per heater |
| Low temperature radiant panels | Consists of low temperature elements (300°C and below) mounted behind a radiating surface.<br>Thermal insulation behind the elements minimises heat loss. Very low temperature elements (40°C) used in ceiling heating applications. | No regular maintenance required. Set flush into building structure and unobtrusive. | Slow response time. | Up to 200 $W/m^2$ |
| **Natural convective** | | | | |
| Storage heater | Consists of a thermal storage medium which is heated during off-peak electricity periods. A casing and thermal insulation around the medium enables the heat to be gradually released throughout the day. Manual or automatic damper control allows 20% of heat output to be controlled. | No regular maintenance required. | Not intermittent. Limited control of heat charging and output. | Storage element sizes 1·4 to 3·4 kW |
| Panel heaters, convectors or skirting heaters | Consists of a heating element within a steel casing with air grilles allowing the natural convection of air across the element. Generally controlled by an integral room thermostat. | No regular maintenance required. Cheap to install. Suitable for low heat loss applications. East response time for intermittent operation. | High surface temperatures. High temperature gradients if poorly sited. | 0·5 to 3 kW output |

**Table B1.24.** Direct electric heating equipment–*continued*.

| Type | Description | Advantages | Disadvantages | Emission range |
|---|---|---|---|---|
| **Forced convectivc** | | | | |
| Storage fan heaters/ electricaire | These storage heaters have increased core thermal insulation and contain a fan which distributes warm air to the space to be heated. The fan is usually controlled by a room thermostat. Storage fan heaters are single room units but can heat an adjacent room with a stub duct. Electricaire units are larger and can be ducted to serve several areas. Up to 80% controllable heat. | Uses off-peak electricity. Suitable for intermittent operation. | Heavyweight. | Fan storage heaters: 3 to 6 kW<br>Electricaire 6 to 15 kW<br>Industrial models up to 100 kW |
| Fan convectors | These wall mounted or free standing units incorporate a fan which forces air over the heating elements into the space. High output rate relative to size. Can incorporate integral room thermostat control. | Low maintenance. Fast response. Accurate temperature control. Suitable for highly intermittent heating applications. Low surface temperatures. Fan only operation for summer use. | | 2 to 3 kW |
| Down flow fan convector | These units are forced air convectors mounted to direct the heated air downwards. High air flow and heat output. Often used to provide a hot air curtain over entrance doorways. | Suitable for localised heating. | Can be noisy. | 3 to 18 kW |

## Gas and Oil-fired Heating Equipment

Where gas or oil appliances are used for heating and installed within the heated space, between 70 and 90% of the total energy content of the fuel input will be converted into useful heat. Table B1.25 gives particulars of some gas-fired equipment types and Table B1.26 gives similar details for some oil-fired heaters. The first three types of equipment detailed in Table B1.25 and the first two in Table B1.26 are usually used for local warming of individuals rather than to provide a particular temperature throughout the space.

**Table B1.25.** Direct gas-fired heating equipment.

| Type of heater | Usual rating/ (kW) | Surface temperature/ (°C) | Flue system | Notes |
|---|---|---|---|---|
| Radiant convector gas fires | 4·4 to 7·3 | 1100 at radiant tips | Conventional | Wall mounted or at low level |
| Overhead radiant heaters | 3·1 to 41 | 850 to 900 | Flueless* | High level or ceiling mounted |
| Overhead tubular radiant heaters | 10 to 15 | 315 mean | Conventional or fan-assisted flue or flueless* | High level or ceiling mounted |
| Convector heaters | 2·5 to 16·7 | – | Conventional or balanced flue or flueless* | Wall mounted or at low level |
| Fan convectors | 1·4 to 3·7 | – | Balanced flue or fan-assisted flue | Wall mounted or at low level |
| Make-up air heaters† | 49 to 250 | – | Flueless* | Mounted at high level and fan-assisted |
| Unit air heaters† | 17 to 350 | – | Flued or flueless* | Mounted at high level and fan-assisted |

* The use of flueless appliances should be discouraged since they discharge much moisture into the heated space (see Section A10).

† The installation of flueless appliances in excess of 44 kW is not permitted by the Building Regulations (1985) and application should be made to the Local Authority for the necessary waiver where installations of this size are contemplated.

**Table B1.26.** Direct oil-fired heating equipment.

| Type of heater | Usual rating/ (kW) | Surface temperature/ (°C) | Flue system | Notes |
|---|---|---|---|---|
| Radiant paraffin heaters | 1·5 | Less than 121† | Flueless* | Mounted at low or floor level |
| Flued radiant convector heaters | 8·5 to 10·9 | Less than 121‡ | Conventional | Mounted at low or floor level |
| Convective paraffin heaters | 1 to 4 | Less than 121† | Flueless* | Mounted at low or floor level |
| Air heaters | 8·5 to 13·5 | Less than 121‡ | Conventional | Mounted at low or floor level |
| Blown air heaters | 10·7 to 16·7 | Less than 121‡ | Conventional | Mounted at low or floor level |
| Warm air heaters | 50 to 450 | Maximum 60§ | Flued | Floor mounted or overhead |
| Make-up air heaters | 90 to 450 | Maximum 60§ | Flued | Can be flueless.<br>Floor or overhead mounted. |

* The use of flueless appliances should be discouraged since they discharge much moisture into the heated space (see Section A10).

† See BS 3300 ‡ See BS 799 § See BS 4256

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# SECTION B2 VENTILATION AND AIR CONDITIONING (REQUIREMENTS)




*This edition of Section B2 first published: 1986*

# SECTION B2 VENTILATION AND AIR CONDITIONING (REQUIREMENTS)

### Introduction

This Section examines the requirements and applications of ventilation and air conditioning within buildings. Ventilation is treated both generally and with reference to particular applications. Air conditioning is discussed primarily in terms of special purpose and process requirements since Volume A covers comfort air conditioning in detail. Section B3: Ventilation and Air Conditioning (Systems, Equipment and Control), should be consulted for information on system design and components.

## VENTILATION REQUIREMENTS

Ventilation requirement is defined here as the amount of outdoor air ('fresh air') needed to be supplied to a space to meet criteria associated with the use of that space. It is not necessarily the total quantity of air supplied because in many systems the room air supply is a mixture of outdoor and recirculated air.

The provision of a fresh air supply is necesary for one or more of the following purposes:

(*a*) human respiration

(*b*) dilution and removal of airborne contaminants produced within the space. These may include body odours, tobacco smoke, water vapour, and emanations from machinery or processes

(*c*) combustion appliances

(*d*) thermal comfort

(*e*) smoke clearance in the event of a fire

### Human Respiration

The rate of ventilation required for the supply of oxygen for breathing is far outweighed by the requirements for the dilution of exhaled carbon dioxide ($CO_2$). The maximum allowable concentration of $CO_2$ for 8 hour exposures for healthy adults is 0·5% by volume (5000 ppm)[1]. In the USA, one half of this limit (0·25%) has been taken[2] as appropriate for general building environments. To prevent these limits being exceeded, minimum ventilation rates for various activity levels are given in Table B2.1.

**Table B2.1.** Ventilation rates required to limit $CO_2$ concentration.

| Activity | Minimum ventilation requirement/(litre/s per person) | |
|---|---|---|
| | 0·5% $CO_2$ limit | 0·25% $CO_2$ limit |
| Seated quietly | 0·8 | 1·8 |
| Light work | 1·3-2·6 | 2·8-5·6 |
| Moderate work | 2·6-3·9 | — |
| Heavy work | 3·9-5·3 | — |
| Very heavy work | 5·3-6·4 | — |

However, for most applications involving human occupancy, it is the dilution of body odours that is the most critical factor in determining ventilation requirements.

### Body Odour Dilution

Until recently, the studies on body odour acceptability by Yaglou[3] were commonly used as the basis for ventilation standards. Yaglou concluded that the ventilation requirement varied with the space available per person. However, recent research[4,5,6] has shown that this is not so and that the requirement can simply be expressed as an air flow rate per person.

The ventilation rate required depends on whether the criterion is acceptability to the occupants or acceptability to visitors entering the occupied space. In studies in auditoria[7], it has been found that the occupants themselves were insensitive to changes in ventilation over the range 5 to 15 litre/s per person although there were always nearly 10% of the occupants dissatisfied with the odour level.

Similarly it has been shown that an outdoor flow rate of 7 to 8 litre/s per person was required to restrict the level of body odour so that no more than 20% of the entrants to the occupied space were dissatisfied. The sensitivity was such that halving the ventilation rate increased the proportion dissatisfied to 30% while more than 3 times the ventilation rate was required before the proportion decreased to 10%.

In the absence of further information it is recommended that 8 litre/s per person should be taken as the minimum ventilation rate to control body odour levels in rooms with sedentary occupants.

There is evidently a relationship between $CO_2$ concentration and body odour intensity in occupied rooms. Thus for intermittent or varying occupancy, the control of ventilation rates by $CO_2$ concentration monitoring can be effective in matching the rate of air supply to the changing requirements.

### Tobacco Smoke

There are no definitive criteria for the required dilution of tobacco smoke. Uncertainties relate particularly to the respirable particulate component. Evidence suggests that particle removal by filtration is necessary to avoid excessively high ventilation rates.

Smoking produces an undesirable odour, particularly to non-smokers. A recent study has shown that filtration of the smoke particles did not alleviate the odour nuisance. As is the case for body odour, recent research has superseded that of Yaglou and the indications are that much higher ventilation rates than previously are now required to avoid dissatisfaction of more than 20% of visitors to a room occupied by cigarette smokers. Ventilation rates for smokers of 4 or 5 times that required for non-smokers have been suggested[4] although, allowing for the fact that a minority of the occupants may be smokers, the overall ventilation rate may be only twice that needed for non-smoking situations (see Table B2.2).

**Table B2.2.** Recommended outdoor air supply rates for sedentary occupants.

| Condition | Recommended outdoor air supply rate (litre/s per person) |
|---|---|
| with no smoking | 8 |
| with some smoking | 16 |
| with heavy smoking | 24 |
| with very heavy smoking | 32 |

### Condensation

Condensation in buildings and their structures is discussed in Section A10 and the BRE Digest 110[8]. Ventilation is one of the primary methods for controlling the moisture content of indoor air to prevent or at least minimise condensation.

The nomogram presented in BS 5925[9] may be used to directly estimate the ventilation rate required to prevent condensation on inner surfaces of walls and windows of various U-values exposed to known outdoor conditions.

### Other Airborne Contaminants

Ventilation may be used to dilute and remove other airborne contaminants released in a space, and which would otherwise rise to unacceptable concentrations. If dilution is the main basis of control then the ventilation system should be designed to produce good mixing of the incoming air and the contaminant within the space. In situations where the contaminant release is from a fixed source, then it is preferable to arrange the extract location as close the source as possible so that direct removal is achieved.

The Health and Safety Executive (HSE) produce guidance on the limits to which exposure to hazardous airborne substances should be controlled in work places. This guidance is in the form of occupational exposure limits for long term (8 hour) and short term (10 minute) exposures. These occupational exposure limits are published by HSE (Guidance Note EH40[1]) for a large number of substances and are updated annually. While these concentration limits must not be exceeded, it is recommended that exposure should be kept as low as is reasonably practical.

For situations where exposure may be longer than 8 hours a day, or where more susceptible members of the general population, such as the elderly, the young and those prone to ill-health, are involved values lower than the occupational exposure limits should be applied. Little information exists on the appropriate limits for this general application, but it has been suggested[10] that one fifth of the occupational exposure limit may be an acceptable standard.

For a single contaminant under steady conditions, equation B2.1 may be applied to determine the flow of outdoor air which, with good mixing, would maintain the contaminant concentration at a specified level.

$$Q = q\left(\frac{1 - C_e}{C_e - C_o}\right) \qquad \text{B2.1}$$

where:

$Q$ = flow rate of outdoor air .. .. litre/s
$q$ = release rate of contaminant within the space .. .. .. .. litre/s
$C_e$ = limiting contaminant concentration within space .. .. .. litre/litre
$C_o$ = concentration of the contaminant in the outdoor air .. .. .. litre/litre

A more comprehensive analysis of the relationship between contaminant concentration and ventilation rate is given in BS 5925[9].

There is no generally accepted approach for the derivation of exposure limits for mixtures of contaminants. In such cases it is recommended that specialist occupational hygienists or toxicoligists should be consulted.

### Combustion Appliances

Care must be taken to ensure that adequate fresh air is supplied to meet the requirements for combustion in fuel burning appliances. Compliance with the regulations governing the ventilation of such appliances must be maintained. Details of these requirements are given in relevant British Standards e.g. BS 5376[11], BS 5410[12] and BS 5440[13], reference should also be made to Section B13.

### Thermal Comfort

Ventilation systems should be designed to avoid discomfort caused by local draughts particularly in winter. The principles of air distribution outlined in Section Al should be followed.

In the summer months, high ventilation rates can be used to control indoor air temperatures which otherwise would rise above comfort levels because of heat gains within the space. The ventilation rate required to prevent excessive indoor temperature can be deduced from the relationships given in Section A8. It is probable that the summertime ventilation rates will need to be at least 10 air changes per hour in the UK.

### Smoke Clearance

Ventilation for the removal of smoke in the event of a fire is a specialised subject and is covered in Section B5.

## NATURAL VENTILATION

Natural ventilation is dependent on the naturally occurring agencies of wind and difference in air temperature between the inside and outside of a building. As these motivating influences are variable, it is almost impossible to maintain consistent flow rates and thereby ensure that optimum ventilation will be achieved at all times. These influences cause movement of outdoor air into and out of the building through any available openings or cracks. Whereas in the past minimum ventilation rates could often be provided by advantageous leakage (infiltration), the trend towards better sealed buildings means that more attention has to be given to the provision of suitable ventilation openings to achieve the necessary air flow rates. However, because of the complexities and vagaries of the natural forces, it is difficult to determine the required size and distribution of ventilation openings to ensure that the supply of fresh air will neither fall too low, with consequent indoor air quality or condensation problems, nor rise too high, resulting in draughts and energy wastage.

With some simplifying assumptions, it is possible to predict the likely infiltration or ventilation rates of a building provided sufficient information is known about the pressure distributions and air leakage characteristics. Calculation methods and the data required for them are presented in Section A4 and other sources[9,14]. More sophisticated prediction techniques[15] using computer programs are being developed and validated.

Although natural ventilation can be provided through inlet and outlet openings both on the same external face of a room, more advantage of the wind effect can be obtained with openings on opposite sides of a building. This is particularly relevant in summer when much higher ventilation rates are required to assist in dissipating heat. However, there needs to be a free flow path from one side to the other so care should be taken in the layout or extent of any internal partitioning. It is generally considered that such cross flow ventilation is able to give reasonable air distribution for a distance up to 6 m inwards from the external façade.

## MECHANICAL VENTILATION

Mechanical ventilation systems rely on fans to produce air flow. These systems vary in complexity from a single wall-mounted fan to ducted air distribution from and to centrally located fans with the possible addition of filtration, acoustic attenuation, heating, cooling, humidification and heat recovery equipment.

There are 3 basic types:

Supply
Extract
Balanced

### Supply Ventilation

In a supply system, the air is delivered by a fan to the treated space and allowed to exhaust through purpose provided or other openings. The supply air can be treated as required. The slight positive pressure (relative to outside) established within the space helps to prevent inward leakage of air and so this type of system is advantageous when the extraneous entrance of outside air or air from other parts of the building is to be avoided. With suitable ducting systems, the supply air can be distributed throughout the building to give uniform ventilation or to match individual air flow rates to those areas requiring different ventilation levels.

### Extract Ventilation

In extract ventilation, fan power is applied to exhaust air from within the room or building to outside. Replacement air enters through any available gaps and purpose provided openings. This type of system is commonly used for the removal of hot or polluted air, for example in kitchens, bathrooms and toilets, and in other situations where the uncontrolled escape of contaminated air from the room is to be prevented. Unless special precautions are taken, the incoming air may cause problems such as local draughts. The risk of noise being tranmitted through the openings for the make-up air should also be considered.

A particular application of extract ventilation is the use of local exhaust openings or hoods adjacent to the source of contamination for its direct removal. Design information relating to local exhaust ventilation is given in Section B3 and a general description of the application of this type of ventilation in industry follows in the section below.

### Balanced Ventilation

The combination of supply and extract systems is known as balanced ventilation, by which control of both incoming and exhaust air can be achieved. Additionally, the particular advantages of a supply system can be obtained by having a slightly greater supply flow capability than extract. Alternatively, the extract flow could be made slightly higher than the supply flow.

It should be noted that mechanical ventilation systems are not totally immune from the effects of external climate (see Section A4), particularly the pressure variations caused by wind. These effects should be considered in relation to the required stability of flow through the system and to the location on the façade of the building of the system inlets and outlets. For most systems it is necessary to pay particular attention to avoid exhaust air re-entering the building (see Section B3).

## INDUSTRIAL VENTILATION

Ventilation applied in an industrial context is usually used to remove airborne contaminants arising from processes or machines. Where contaminant sources are weak and of low toxicity, and they are either scattered or mobile, satisfactory ambient conditions can be achieved by dilution. However, it is usually more appropriate to remove the contaminant at or close to its source by means of local exhaust, e.g. vehicle exhaust removal systems in garages.

Industrial contaminant sources often require large extract air flow rates to ensure that the released pollution is effectively captured and conveyed away by the extract system. In such cases, particular attention should be paid to ensure that adequate replacement or 'make-up' air is provided without discomfort to the occupants and without disturbance to the industrial process. It may be necessary to directly heat the incoming air in winter or, if that would impose an unacceptably high energy consumption, to duct the outdoor air directly to the source location.

For certain processes, such as paint spraying, filtration of the incoming air may be necessary. Similarly, it may be necessary to remove the contamination from the exhaust air before it is discharged to the outside air. Special industrial air cleaning devices are available for this purpose (see Section B3).

### Choice Between Natural and Mechanical Ventilation

The basic factors that affect the choice between natural and mechanical ventilation are:

quantity of air required
quality of air required
consistency of control required
isolation required from external environment

*Quantity*

The ventilation flow rates that can be generated by natural ventilation are limited by the exposure of the building to the available thermal and wind forces. For high flow rates mechanical ventilation will almost certainly be required.

*Quality*

Natural ventilation provides very little opportunity for treatment of the incoming air. If the air is required to be heated, cooled, filtered or otherwise treated, mechanical ventilation will need to be used.

*Consistency of control*

Mechanical ventilation systems can be designed to provide constant or variable flow rates distributed as required throughout the building. Consistency and precise control are not possible with natural ventilation because of the variability of climatic conditions.

*Isolation*

When a building is located in a noisy environment, it is often impractical to provide adequate natural ventilation without excessive sound transmission through the openings. In such circumstances, mechanical ventilation systems with appropriate acoustic treatment can successfully be used.

Mechanical ventilation can also be designed to control room pressures to prevent ingress or egress of contaminants. Such isolation is used, for example to minimise the spread of airborne bacteria in hospitals and dust into clean rooms.

## REQUIREMENTS FOR SPECIFIC PURPOSES

For many types of application it is possible to give guidance on the conditions required to provide suitable environments. In most of the examples that follow, indoor temperature is the key parameter. In some cases humidity is also important; in others ventilation rates are recommended with, where appropriate, the quality of filtration required. Table B2.3 summarises the ventilation requirements for a wide range of building types. For many types of building, statutory requirements apply and checks should be made to ensure compliance with the relevant regulations. For industrial applications of a specialist nature, expert guidance should always be sought from appropriate professional authorities.

### Animal Farming[16, 17]

There are two main groups of farm animals; those such as milking-cows, breeding-pigs and sheep (hardy stock) that do not require any great control of conditions in their housing, and those such as pig farrowing, fattening houses, veal calf houses, laying and broiler poultry houses which require such control of environmental conditions that the highest possible productivity is obtained at the lowest food and management costs.

Hardy stock only require housing to protect them from extremes of weather, ventilation being provided by low level and ridge open ventilators with protection against direct and through draughts. However, care must be taken to ensure adequate ventilation in high density enclosed houses where forced ventilation will be necessary. Humidity is not found to be a problem.

For animals requiring close control of conditions, mechanical ventilation is essential, provided by supply and/or extract fans depending on the requirement for positive or negative pressures within the houses. Winter recirculation can be used to conserve heat. Safeguards must be provided against fan failure or livestock will be seriously affected during hot weather. Adequate ventilation will also restrict high humidities. Table B2.4 gives optimum air temperatures and ventilation rates.

### Animal Rooms[18, 19, 20]

The environmental conditions and degree of control required for animal rooms depends on the species and the intended use of the facilities. Tables B2.5 and B2.6 show the conditions required for various animals and for different applications.

For precise experimental work, close control of temperature (±1K) and relative humidity (±10%) may be required at different conditions within the overall operational stage. Uniformity of the environment throughout the space is also important and in some cases the direction of air movement needs to be controlled to minimise, for example, the pollution in the spaces through which the laboratory operatives move.

Requirements may also include standby equipment and/or safety features that are automatically initiated in the event of a failure of the main system.

### Car Parks and Garages[21, 22, 23, 24]

Because of the dangerous nature of the accessories to the repair and storage of motor vehicles and the risk of pollution from waste gases and products, the heating, ventilation, fire protection and safety etc. of functional structures is carefully regulated. This is particularly true in the case of, for example, underground garages. The following notes give some guidance, but local authorities should be consulted at an early stage on the applicability of fire, planning, building and office regulations.

*Heating*

High level radiant heating or flexibly ducted hot-air systems will be preferred in ancillary workshops because of the high air change rates and safety requirements. General car parking areas can remain unheated. The overriding condition is that the equipment should be safe in the presence of inflammable vapours, e.g. motors for fan units should be either bifurcated or centrifugal with the motor and drive out of the air stream.

*Ventilation*

Free air flow across above-ground car parks is to be encouraged, and for natural ventilation, openings in outside walls should have an aggregate area equal to at least 5% of the floor area at each level, and at least half of that area should be in opposite sides. For mechanical ventilation, between 6 and 10 air changes per hour are necessary, according to the type of building. Extract points should be placed so as to eliminate pockets of stale air where dangerous fumes could collect. Extract should be from both high and low level, with special attention given to low points and drains. An automatic CO detection system should be used to initiate increased ventilation and raise an alarm when the CO concentration rises above 100 ppm.

**Table B2.3.** Ventilation requirements for a range of building types.

| Room or building type | Recommended fresh air supply rate | Recommended total air supply rate air changes/hour unless otherwise stated | Comments |
|---|---|---|---|
| Animal rooms | see Table B2.4 | based on heating/ cooling requirements | |
| Boiler houses and engine rooms | as required for combustion plus ventilation | 15-30 | |
| Banking halls | as required for occupants (see Table B2.2) | 6 | |
| Bathrooms (dwellings) | 6 litre/s (mechanical extract) | no recirculation | possibly intermittent |
| Boardrooms, conference rooms | as required for occupants (with allowance for smoking) | 6-10 | |
| Canteens | as required for occupants | 8-12 | |
| Cinemas | as required for occupants | 6-10 | |
| Dining halls, restaurants | as required for occupants | 10-15 | |
| Garages– | | | |
| public parking | 6-10 air changes/h | no recirculation | |
| repair shops | as required to dilute vehicle exhaust | no recirculation | |
| Kitchens, hotel and industrial | as required for appliances | not less than 17·5 litre/m$^2$ of floor space, nor less than 20 air changes/h | |
| Laboratories | as required by equipment, processes and occupants | 4-15 | |
| Laundries | as required for heat/ moisture removal | 10-15 | |
| Lavatories and toilets | not less than 3 air changes/h or 6 litre/s per pan (extract), preferably at least 5 air changes/h | no recirculation | |
| Libraries, museums, galleries | as required for occupants | 3-4 | good air circulation needed around shelves etc. |
| Offices | as required for occupants | 4-6 | |
| Schools | at least up to 8.3 litre/s per person | – | |
| Shops, department stores, supermarkets | as required for occupants | based on heating/ cooling requirements | |
| Sports halls-general | as required for competitors and spectators | based on heating/ cooling requirements | some sports require an environment with low air movement |
| Swimming pools– | | | |
| pool hall | 10-15 litre/m$^2$ of water surface and wetted surround | no recirculation | extract preferably slightly greater than supply |
| changing area | 6-10 air changes/h | no recirculation | extract from clothes storage area |
| Theatres | as required for occupants | 6-10 | |

*Note:* The above recommendations do not take account of all statutory requirements, and reference should be made to national and local regulations.

Standby fans connected to a secure power supply should be fitted.

Where many vehicle engines are likely to be running simultaneously, e.g. at car park entrances and exits and in vehicle repair centres, it is important to relate the ventilation rate to the rate of pollution generation in the exhaust gases. Generally, carbon monoxide is the most critical pollutant although some of the other exhaust gas constituents, e.g. oxides of nitrogen and hydrocarbons, may need to be taken into account or at least checked. Table B2.7 gives some indication of the composition of vehicle exhaust from correctly-tuned diesel, petrol and liquid petroleum gas (LPG) engines. Concentrations several times higher may result from badly tuned or cold running engines.

In the absence of more specific information, an approximate assessment of the degree of pollution in a confined space can be made assuming the rate of exhaust discharge is 1·2 litre/s per brake horse power.

The limiting concentration of exhaust pollutants are included in the Health and Safety Executive Occupational

**Table B2.4.** Temperatures and ventilation rates suitable for housed livestock.

| Animal species | Optimum temperature range/°C | Ventilation rate winter/ (litre/s per kg of body weight) | Ventilation rate summer/ (litre/s per kg of body weight) |
|---|---|---|---|
| Adult cattle | 0 to 20 | 0·5 | 0·20 to 0·38 |
| Calves | 10 to 15 | 0·10 | 0·26 to 0·53 |
| Pigs | 5 to 25 | 0·10 | 0·26 to 0·53 |
| Piglets: at birth | 35 | 0·08 | 0·08 |
| after 2 days | 28 to 33 | 0·06 | up to 0·06 |
| Fattening pigs | 11 to 22 | 0·10 | 0·26 to 0·53 |
| Laying poultry | 20 to 25 | 0·4 | 1·5 to 2·6 |
| Broiler chickens | 15 to 25 | 0·2 | 0·8 to 1·3 |

Exposure Limits[1]. Equation B2.1 may be used to determine the necessary ventilation rate. In the case of stationary vehicles, ventilation and energy needs can be substantially reduced by the direct extraction of their exhaust fumes. Specially designed systems are available for this purpose.

## Computer Rooms[25, 26]

Under operational conditions, computer equipment is susceptible to the temperature, humidity and cleanliness of the surrounding environment. In all cases an accurate design brief for each project is required from the manufacturer of the computer equipment, with careful assessment and consideration given to temperature and humidity limitations.

Air conditioning will normally be required to control this environment to ensure satisfactory operation of the equipment and provide comfortable conditions for the operators. It is usual for the air conditioning to be extended to cover all ancillary areas, e.g. printer rooms, tape library, stores etc.

### *Temperature and Humidity*

For the computer equipment itself, steady conditions of temperature and humidity, within limits, are generally more important than the actual values of the temperature and humidity. In practice, it will normally be the comfort of the occupants which is the criterion, and room conditions may therefore be based on the information given in Section A1, but normally will be within the range 20 to 25°C and 40 to 60% relative humidity.

Independent heating should be provided in all critical areas, such as the computer room, the tape store and the paper store, to maintain a minimum temperature of 18°C under most adverse winter conditions during computer shut-down. In temperate climates this requirement may need to be coupled with an overriding humidistat designed to increase the space temperature if the relative humidity exceeds 80%.

Close control of temperature and humidity will not normally be achieved unless all doors to the air conditioned area are provided with air locks and the building structure is airtight and vapour sealed (including the false ceiling space) to reduce the infiltration of unconditioned air and the migration of water vapour.

### *Filtration*

Filtering of the air supply should be provided to the standard specified by the computer manufacturer, as this will vary with the type of equipment to be installed. As a minimum standard a filter of 95% efficiency based on tests to BS 6540 Part 1[26] should be used. In many cases, the filtration standards will be high and it should be borne in mind that considerable quantities of dust are brought into the suite by the operators unless suitable precautions are taken.

Humidifying equipment should be of a type which does not produce dust.

### *Air Distribution*

Air supply is normally through the ceiling or floor. Supplying air at low level and extraction over the computer equipment has the advantage that the heat released upwards from the equipment can more easily be removed without it affecting the occupied areas. High level supply may be through diffusers or a ventilated ceiling.

Extract points located above the heat source can be used effectively to reduce the room temperature differential or the room total load if ducted directly to atmosphere.

**Table B2.5.** Animal room environmental design conditions.

| Animal | Surface area ($m^2$) | Average metabolic rate at 21°C* (W) | Number of animals per 10 $m^2$ of floor area | Typical animal room gain ($W/m^2$) | Receommended temperature range (°C) | Relative humidity % |
|---|---|---|---|---|---|---|
| Mice | 0·01 | 0·5 | 2000 | 100 | 21-23 | 40-70 |
| Rats (60 days) | 0·031 | 1·5 | 485 | 73 | 21-23 | 40-60 |
| Guinea pigs (60 days) | 0·07 | 3·0 | 400 | 120 | 17-20 | 40-70 |
| Chicken (4 weeks) | 0·04 | 2·4 | 230 | 55 | 21-23 | 40-60 |
| Chicken (24 weeks) | 0·21 | 12·0 | 100 | 120 | 16-19 | 40-60 |
| Rabbits (adult) | 0·20 | 11·0 | 32 | 35 | 16-19 | 40-60 |
| Cats | 0·20 | 8·0 | 16 | 13 | 18-21 | 40-60 |
| Dogs (male) | 0·65 | 26·0 | 5 | 13 | 12-18 | 40-70 |
| Dogs (female) | 0·58 | 22·0 | 5 | 11 | 12-18 | 40-70 |

*Notes:*
*Based on resting metabolism. Handbook of Biological Data, 1956.
1. Assume 35-40% as latent gain.
2. Figures should be used as a guide only and will vary depending on conditions.
3. Animal numbers per $m^2$ based on figures for an average experimental holding room.
4. Resting metabolic rates should be used for animals in confined spaces, not basal rates given in Table A7.3.

**Table B2.6.** Animal room categories and environmental data for experimental units.

| Animal room type | Environmental condition | | | | Filtration[1] | | | Air pressure | |
|---|---|---|---|---|---|---|---|---|---|
| | Air change rate ($h^{-1}$) | Outdoor air (%) | Temperature (°C) | Humidity (%) | Supply | Extract | Terminal | Corridor | Room |
| Breeding/holding | 8–12 | 50 | 20 | No control | Roughing | Nil | Nil | slightly positive | slightly negative |
| Holding (exp) | 12–18 | 100 | Variable | regulation between 40–60 | Medium efficiency | Nil | Nil | slightly positive | slightly negative |
| Laboratory (exp) | 8–12 | 100 | 20 | maintain between 40–60 | Medium efficiency | Nil | Nil | slightly positive | slightly negative |
| Autopsy/operating | 12–15 | 100 | Variable | regulation between 40–60 | High efficiency | Nil | Supply UHE Extract HE | slightly positive | more positive |
| Specified pathogen free | 12–18 | 100 | Variable | maintain between 40–60 | High efficiency | Nil | Supply UHE Extract HE | positive to outside | positive to corridor |
| Radioactive labs. | 12–15 | 100 | 21 | maintain between 40–60 | Medium efficiency | Ultra high efficiency | Supply nil Extract UHE | positive to outside | negative to corridor |
| Infected animals | 12–15 | 100 | 21 | maintain between 40–60 | High efficiency | Ultra high efficiency | Supply HE Extract UHE | positive to outside | negative to corridor |
| Cage washing | 15–20 | 100 | 10–15 | No control | Roughing | Nil | Nil | slightly positive | slightly negative |

*Note:*
[1] J. E. Firman: Filter data, *Journal of the Institute of Animal Technicians:* Dec. 1966[19].

The outdoor air flow rate requirement for the generation of a positive pressure in the room will probably outweigh that necessary for the occupants as these are usually few in number. An outdoor air quantity of between 10 and 15% of the supply air flow rate is normally taken.

*Maintenance Requirements*

Consideration should be given to the operating and maintenance requirements of the installation. Temperature and humidity recording/alarm devices may be necessary together with other operational alarms. Short and long term maintenance must be covered in the design brief with provision for standby motors, plant and electrical supply, etc. where appropriate.

## Dark Rooms

Small dark rooms for occasional use or for purely developing processes may often be ventilated naturally with a suitable light trap, although consideration should be given to providing mechanical extract using an air change rate of 6 to 8 per hour.

For general-purpose dark rooms, however, the air change rate should be ascertained from consideration of the heat gain from the enlarger, lights, etc. plus the occupants, on the basis of a temperature rise of 5 to 6°C. In industrial and commercial dark rooms which have machine processing, the machines will very often have their own extract ducting, the air supply being drawn from the room itself. It will usually be necessary to provide a warmed and filtered mechanical inlet in such cases. In special cases, involving extensive washing processes, the humidity gain may be significant and require consideration.

**Table B2.7.** Main contaminants in vehicle exhaust.

| Engine | Carbon monoxide/ ppm | Oxides of nitrogen/ ppm | Hydrocarbons/ ppm | Aldehydes/ ppm |
|---|---|---|---|---|
| Petrol | 20 000-50 000 | 600-4 000 | 10 000 | 40 |
| Diesel | 200-4 000 | 200-2 000 | 300 | 20 |
| LPG | 10 000 -20 000 | 700-800 | 600 | – |

## Dwellings

For information on the required provisions for ventilation of habitable rooms reference should be made to the Building Regulations[27] (for England and Wales) and The Building Standards (Scotland) Regulations[28].

The provisions that are deemed to meet the Building Regulations are given in Approved Document F1[29]. In habitable rooms, kitchens and bathrooms natural ventilation may be provided by at least one opening with an area of at least one-twentieth of the floor area of the room. Alternatively mechanical ventilation may be provided to produce one air-change per hour in habitable rooms and three air-changes per hour in kitchens and bathrooms.

Slightly different requirements apply to Scotland[28].

### Factories[30]

The environmental conditions required in factories differ widely depending mainly on the manufacturing processes, although in some cases the needs of the operatives may predominate. It is essential that the requirements for the particular process in question are clearly specified so that the design and selection of systems and their controls can be matched to the application.

In some applications air cleanliness is the most critical factor. This is so in electronic assembly areas, for example, and for these and other minute component assembly facilities special clean rooms with very highly efficient filtration may be required (see Section B3).

For most applications temperature control is important but the degree of control necessary depends on the sensivity of the process or product.

Data on typical requirements for some applications are given in Tables B2.8–10, but these should only be used in the absence of more specific information.

**Table B2.8.** Environmental conditions in pharmaceutical production areas.

| Process | Temperature (°C) | Relative humidity (%) | Filtration | |
|---|---|---|---|---|
| | | | Efficiency (%) | Particle size (mm) |
| Tabletting and ointments | 21–24 | 35–40 | 95 | 5 |
| Sugar coating | 21 | 20 | 85 | 5 |
| Hard capsule | 24 | 15–20 | 95 | 5 |
| Ampoule filling (sterile) | 21–24 | 15–20 | 99 | 1 |
| Clean/sterile rooms | 21–24 | 40–50 | 99 | 1 |
| Packing deliquescent products | 21–24 | 20–40 | 85 | 5 |

**Table B2.9.** Typical environmental conditions for printing processes.

| Process | Temperature (°C) | Relative humidity (%) |
|---|---|---|
| Multicolour offset lithography . . | 24 | 43–47 |
| Other sheet fed printing . . . . | 24–26·5 | 45–50 |
| Newspaper and other wet printing | 21–24 | 50–52 |
| Stock room for colour lithography | 22·5–26·5 | 48–55 |
| Other paper storage . . . . | 21–26·5 | 45–50 |
| Binding, cutting, folding, gluing . . | 21–26·5 | 45–50 |
| Roll storage . . . . . . | 22·5–26·5 | 50 |

**Table B2.10.** Typical environmental conditions for tobacco processes.

| Process | Temperature (°C) | Relative humidity (%) |
|---|---|---|
| Packing . . . . . . | 21 | 57 |
| Cigarette making . . . . | 21 | 67 |
| Tobacco stores . . . . | 23·5 | 62 |
| Blending . . . . . . | 22 | 75 |
| Cutting . . . . . . | 22 | 75 |

### Food processing

Food processing covers cooking and/or dehydration, bottling, canning and packing. The processes of food production normally require mechanical ventilation and sometimes air conditioning. The designer should take into account the heat dissipation based on the energy used in the production process and should make an approximate heat balance for the calculation of air quantities. The ambient air temperature in a food production area should not be allowed to rise much more than 6°C above external summer design conditions (i.e. 30 to 35°C). Where it is not possible to limit the temperature rise to about 6°C, consideration should be given to some degree of cooling of the incoming supply in limited areas. The air change rates in cooking areas may well be in excess of 20/hour.

In addition to local ventilation, general ventilation will be necessary. It is preferable to supply air over working areas and extract over cooking equipment or other high heat dissipation areas, but care must be taken to avoid local excess cooling of the processes (see ‘Kitchens’).

In sweet and chocolate manufacture, a fairly closely controlled temperature will be necessary and local cooling is an essential part of the manufacturing process.

Ductwork should be run in a manner that will enable cleaning to be carried out at reasonable intervals. Drains may be necessary in some cooling processes, as may fire dampers and grease filters (see ‘Kitchens’).

The mechanical ventilation of many special food manufacturing processes will need very detailed consideration, and plant may need to be designed to meet individual requirements. Consultation with food production managers should be maintained during this stage.

### Horticulture[31, 32]

Environmental conditions in greenhouses must be favourable to plant growth. This involves heating during cold weather and the limitation of high temperatures due to solar gains in the hot weather. In some cases, carbon dioxide enrichment and humidity restriction will also be required. The internal design temperature should be in the order of 16°C when the external temperature is –7 to –10°C.

Greenhouse crops require ventilation to limit the rise in air temperature, provide carbon dioxide for photosynthesis, and restrict the rise in humidity due to transpiration. Automatic ventilators, controlled by an air thermostat, can be opened at a pre-determined temperature (approximately 24°C). Rates of ventilation of the order of 30 to 50 litre/s per $m^2$ of greenhouse floor area are desirable, which is equal to 45 to 60 air changes per hour for conventional houses. Low level ventilators may be required in addition to the ridge ventilators to increase the stack effect during still conditions.

Propeller extract fans (side wall mounted) with ventilation duties to the rates given, have the advantage of positive air movement through crops, thus promoting growth. Inlet air should have a velocity not exceeding 1 m/s and be diverted with an upward component, thus preventing cooler air being drawn directly on to the crops. A combination of automatic ventilators and fans will allow for failures of either system.

A complete mechanical ventilation system, using PVC ductwork with air supply discharge holes, can be used for winter heating with heated recirculated air, and summer cooling with 100% fresh outdoor air. Fans giving a constant 10 to 20 air changes per hour can be supplemented by automatic ventilators or extract fans during hot weather. This type of system has the advantage of even, closely controlled temperatures, with positive air movement throughout the year, although the initial outlay is likely to be high.

Other aspects worthy of consideration are: automatic solar shading equipment, automatic day and night temperature and lighting sequencing, evaporative cooling pad air inlet and exhaust fan system, air purification, plant cooling by evaporation using overhead spraying, and earth heating plant propagation beds.

### Hospitals and Medical Buildings[33]

Reference should be made to design guidance published by the Department of Health and Social Security.

### Kitchens[34, 35, 36]

The principal objective of the engineer designing services for kitchens, as for any occupied space, should be to enable the occupants to pursue their working activities in comfort. Often it is not possible to achieve normal ‘comfort’ conditions in kitchens because of the extremely high expenditure required to counteract the heat released from appliances. Under these circumstances, care should be taken to ensure that ‘acceptable’ working conditions are not exceeded.

*Air Flow Rate*

Generally air is extracted at a constant rate from cooking and subsidiary areas, with replacement air supplied from a separate system and additional make-up from adjoining areas. The extract air flow rate is based on allocating nominal quantities to various types and sizes of kitchen appliances according to Table B2.11.

For operation in winter, the ventilation may be reduced to two thirds of the tabulated values.

When the cooking equipment details are not available, an approximate extract rate (in litre/s) can be calculated by multiplying the number of meals served in one hour by 10 to 15. If the hood sizes are given, use a minimum velocity of 0·35 m/s through the hood opening.

In all cases the extract rate should not be less than 17·5 litre/s per $m^2$ of floor area, nor less than 20 to 30 air changes per hour. It is not unusual to find 120 air changes per hour where basement kitchens are restricted to the minimum height of 2·5m.

In larger kitchens, areas are subdivided to form wash-ups, preparation, pastry, stores etc. These areas require a minimum of 10 air changes/h to create a feeling of comfort (see also Replacement Air). In serveries care must be taken to avoid premature cooling of food by excessive air movement. Radiant heat from hot plates in serveries or salamanders in grilling areas will cause considerable discomfort, and in such cases, spot cooling can be effective using punkah type or other supply grilles giving individual manual adjustment for volume and direction.

**Table B2.11.** Nominal extract rates for kitchen appliances.

| Equipment | Air extraction rates (litre/s) | |
|---|---|---|
| | Unit | Per $m^2$ net area of appliance |
| Roasting and grilling | | |
| Ranges (unit type) approximately 1 m square .. .. .. | 300 | 300 |
| Pastry ovens .. .. .. | 300 | 300 |
| Fish fryers .. .. .. | 450 | 600 |
| Grills .. .. .. .. | 250–300 | 450 |
| Steak grills .. .. .. | 450 | 900 |
| Salamanders (special grills) .. | 450 | 900 |
| Steaming and vapour producing | | |
| Boiling pans (140–180 litre) .. | 300 | 600 |
| Steamers .. .. .. | 300 | 600 |
| Sinks (sterilizing) .. .. | 250 | 600 |
| Bains-marie .. .. .. | 200 | 300 |
| Tea sets .. .. .. .. | 150–250 | 300 |

*Replacement Air*

The method of replacing the air removed by the kitchen extract system will depend entirely on the areas immediately adjacent to the kitchen and the necessity of maintaining the kitchen under negative pressure. If the kitchen is in a sealed area (adjoining areas–non-dining) by virtue of the architectural design, or situated in a basement area (see also Local Authority Regulations), a separate kitchen supply system should be used to introduce 80 to 85% of the air extracted, the difference being the infiltration keeping the kitchen under negative pressure. It should be noted that in basement areas containing kitchens and restaurants, the supply plant to the restaurant areas should be sufficient to offset the down-draught from the street level in addition to supplying air to the kitchens.

When non-air conditioned, properly ventilated restaurants adjoin the kitchens, the majority of the air may be drawn from the dining area. If the restaurant is air conditioned, air may be drawn from it at the maximum rate of 7 litre/s per head. The difference between the extract and replacement air (in both cases) should be provided by a separate kitchen supply system. Outdoor air supplied directly to the kitchen in this way can be used effectively to create spot cooling and reduce the discomfort of occupants working for long periods under conditions of high radiant heat and vapour saturation levels.

When adjoining restaurants are used for replacement air, permanent grilles sized on 1·0 to 1·5 m/s through the free area should be provided in cases where serving hatches are small or likely to be shut for long periods. Velocities through hatches should not exceed 0·25 m/s.

*Hoods and Grease Filters*

It is essential to place hoods above all the foregoing equipment, and good practice to supply hoods for tea making equipment and boilers, to localise the escape of cooking odours and convected heat, and also to protect decor. Hoods are normally of the valance or island type provided with an undrained gutter and constructed for easy cleaning, using stainless steel, anodised aluminium, glass, or painted galvanised steel according to architectural design.

*Local Authority Regulations*

Local regulations relating to fire precautions, basement kitchens, and smell nuisance to adjoining property must be observed. The latter usually implies terminating the kitchen extract in a building at high level.

**Laboratories**[37, 38]

Laboratories have specific but differing requirements depending upon the type of work or process undertaken. For this reason it is not recommended that any design work is completed solely on the basis of the information given under this section but that this is amplified by a concise brief from the establishment requiring the facility.

*General Ventilation and Air Conditioning*

Ventilated and air conditioned laboratories should be designed individually, taking the following factors into account:

(*1*) Type of laboratory: teaching, research, analytical, workshop.

(*2*) Room environmental conditions.

(*3*) Room location and orientation.

(*4*) Special rooms, e.g. chromatography with high and low level extraction.

(*5*) Cross-contamination and separation.

(*6*) Positive or negative room pressure.

(*7*) Room equipment (maximum/minimum variation of heat load).

(*8*) Limitations imposed by laboratory processes.

(*9*) Degree of specific extraction: fume cupboards, bench ventilation, equipment hoods, etc.

(*10*) Diversity in use of laboratories.

(*11*) Plant running period.

(*12*) Plant stand-by facilities.

(*13*) Fire precautions and explosion hazards.

(*14*) Laboratory partition flexibility.

In this country, mechanical ventilation can be effectively designed to maintain the required room conditions. The replacement air provided for the process extract should be used for this purpose. Normally, for this, 100% outdoor air is recommended with a good standard of filtration (90% efficiency down to 5µm particles). The make-up air should normally enter the building at a temperature of 18°C and be supplied into the corridors, ceiling voids, or distributed directly into the laboratory if positive air movement is required, leaving a secondary heating system to deal with fabric and nominal infiltration heat losses. The air change rate within rooms is governed by the need for extraction of air and removal of heat, and as a guide should fall between 6 and 15 air changes per hour. It should not normally be less than 4 or 8 air changes per hour for perimeter and core areas respectively.

Most laboratories are designed to be under negative pressure with respect to the corridor to stop cross-contamination between laboratories.

*Bench Ventilation*

For bench-top processes involving high heat releases or emissions of toxic fumes, local bench ventilation may be used. Simple extract canopies over or at the back of the bench may be suitable but more effective contaminant control can be achieved by a combination of supply and extract known as the ‘push-pull’ system. Under this system fresh air is introduced across the bench to direct the contaminant into a suitably sized canopy.

*Fume Cupboards*

Extraction from fume cupboards should be an integral part of the overall design and is subject to the following factors:

(*1*) Face velocity across the open sash (see Table B2.12).

(*2*) Fume cupboard utilisation.

(*3*) Aerodynamic design of cupboard and its location.

(*4*) Ventilated or air conditioned laboratory.

(*5*) Local or central control.

(*6*) Interconnection of extract ductwork.

(*7*) Need to discharge exhaust air to avoid re-entry.

(*8*) Fire precautions.

**Table B2.12.** Air velocities through sash openings.

| Fume cupboard use | Velocity (m/s) |
|---|---|
| Teaching .. .. .. .. | 0·2 –0·3 |
| Research .. .. .. .. | 0·25–0·5 |
| Analytical .. .. .. .. | 0·3 –0·6 |
| Highly corrosive and toxic .. .. | 0·5 –0·75 |
| *Radioactive .. .. .. .. | 0·5 –2·0 |

*Notes:*
* Radioactive grade determines minimum velocity.
1. Velocity with sash fully open (normally between 0·75-1 m).
2. Higher velocities may be reduced with aerodynamic openings.
3. For walk-in fume cupboards use a diversity on sash opening.

The basic fume cupboard should draw the total quantity of extract air from the laboratory over the sash opening, with a velocity recommended in Table B2.12.

Information on the extract flow rate required to produce the specified velocity should be obtained from the fume cupboard manufacturer.

Care must be taken to ensure that the air movement in the vicinity of a fume cupboard does not adversely affect its performance.

When the flow rate required for fume cupboards is much greater than that otherwise needed in the room where they are situated, energy savings can be achieved by supplying make-up air directly at the faces of the cupboards. However, it must be confirmed that it can be done without causing problems to the laboratory personnel or processes.

*Specialised Laboratories*

For those laboratories handling highly dangerous materials, specialist advice must be sought lo determine the appropriate environmental requirements and safety features.

**Lavatories**

The ventilation of sanitary accommodation is covered by regulations and by-laws, and reference must be made to the relevant statutory requirements, including Borough Council By-Laws, the 1985 Building Regulations[27] and the Building Standards (Scotland) Regulations 1981[28].

The Borough Council By-Laws are often open to interpretation by Public Health Officers as to the methods permissible to achieve adequate ventilation, and therefore it is advisable to obtain the opinion of the Officer before final approval is sought.

*Ventilation Rates*

Using mechanical ventilation, a minimum extract rate of 3 air changes per hour or 6 litre/s per WC pan or wash-hand basin, whichever is the greater, is satisfactory to meet the statutory requirements. In the case of public lavatory facilities, it is good practice to increase the extract rate to within the range 5 to 10 air changes per hour, otherwise this air would be displaced by the building's general extract system. For dwellings, the BRE[39] recommends an extract rate of 6 litre/s from a WC or a bathroom without a WC pan and an extract rate of 12 litre/s from a bathroom with a WC pan.

Mechanical ventilation can be operated intermittently on individual systems provided it continues to run for at least 15 or 20 minutes after the use of the room or space stops.

For sanitary rooms with direct access to outdoor air, natural ventilation requirements are specified as at least one opening with an area of at least $^1/_{20}$th of the floor area of the room or $0{\cdot}1 m^2$, whichever is the greater.

*Air Replacement in Ventilated Lobbies*

Replacement air for lavatories in public buildings can be supplied by using common or separate toilet and lobby supply plants, or made up by an air supply totally via the lobby. The air passes into the lavatory normally through a louvred door or one with a non-vision grille. In any case, the replacement air should be introduced at, or close to the comfort temperature, and at a velocity not exceeding 1·5 m/s.

The need for a ventilated lobby should be checked with the by-laws. Some by-laws pertaining to mechanical ventilation of toilets within a fully air conditioned building permit the dispensing with of a ventilated lobby area.

The practice of extracting slightly more air than is supplied is usually adopted, thus preventing egress of odours from the toilet area. If incinerators requiring room air for combustion are fitted in female toilets, care should be taken to avoid negative pressures that would cause malfunction of the burner.

*Stand-by Facilities*

Duplicate fans and/or motors are usually required for WC extract systems. Where centrifugal fans are installed, changeover dampers will be required, designed so that the air pressure within the system assists in holding them in position. Automatic changeover facilities may be required in addition to the above.

*Cross-talk*

The transmission of noise along ductwork can be a problem especially if toilets are located back to back. Branches should be designed offset and attenuators installed if necessary.

**Museums, Libraries and Galleries[40, 41, 42]**

Deterioration of books and paper may result from acidic atmospheric contaminants (particularly $SO_2$), direct sunlight and very dry or very moist environments. For the preservation of such materials as paper, parchment, leather and wood, the ambient relative humidity should be kept constant and within 50 to 60%. Slightly drier atmospheres may be more suitable for metals.

For most materials, normal room temperatures are satisfactory and again consistency is more important than the actual value. However, temperatures less than 16°C or greater than 29°C should normally be avoided.

Fresh air supply rates will normally be determined by the requirements of the occupants. In rooms with partitions or shelving, care should be taken to provide good air circulation.

High efficiency filtration and gaseous contaminant removal may be required, particularly in city locations.

**Schools[43]**

In working areas the desirable temperature during the heating season varies with expected clothing levels and activities as follows:

| | |
|---|---|
| lightly clad and inactive | 21°C |
| normal clothing and activity level | 18°C |
| vigorous activity | 15°C |

The variation about the mean room temperature should not exceed 2°C. In summertime the recommended room temperature is 22°C, although increases up to 28°C may be tolerated. Working areas should be capable of being ventilated up to at least 8·3 litre/s of fresh air per person.

**Sports Centres[44, 45, 46, 47, 48]**

In sports centres, both the competitors and the spectators need to be considered. Multi-purpose halls will require variable conditions ranging from 5 to 22°C. Generally in the UK there is no justification for refrigeration in the majority of sports centres other than may be required for restaurants, meeting rooms etc. For some sports, such as badminton and table tennis, particular attention must be given to the avoidance of draughts, as air velocities in excess of 0·1 m/s are considered unacceptable.

*Swimming Pools*

Mechanical services for swimming pools should be designed to prevent condensation on the inner surface of the structure, to counteract downdraughts from large windows and to maintain a satsifactory level of temperature, relative humidity and airborne pollution control. Achievement of the first two criteria will be facilitated by good thermal insulation and double-glazed windows.

Relative humidity may be within the range of 40 to 90% but should preferably be at about 60%.

The ambient air temperature in the pool hall should be 1°C or so above pool temperature, which should be as follows:

| | |
|---|---|
| pools for competitive swimming | 23-25°C |
| recreation and training pools | 25-28°C |
| learner pools | 28-30°C |
| diving pools | 28°C |

Background heating can be provided by underfloor or wall-embedded heaters at the pool surround. Exposed convectors or radiant panels may be used but suitable measures need to be taken to prevent corrosion.

Mechanical ventilation should be provided in the hall supplied at a rate of 10 to 15 litre/s per $m^2$ of water surface and wetted surround. Supply and extract flow rates should be balanced, or preferably be such as to produce a marginally lower pressure than that outdoors and in the adjoining accommodation. This slight depression will inhibit the migration of moisture and odour. Although bathers out of the water will be susceptible to draughts, air movement at the pool surface must be sufficient to prevent the accumulation of gases released from the chemically treated water. In winter, the ventilation plant should be run 24 hours a day but the flow rate can be substantially reduced during unoccupied periods because of reduced moisture release from the pool.

Warmed air to maintain a temperature of 24°C should be provided for the changing rooms, and preferably supplied at low level to assist floor drying if no provision is made for under-floor heating. The latter is a suitable means of providing background heating.

Permanent extraction from the clothes storage area should be balanced by an air supply at a rate of between 6 and 10 air changes per hour. A separate system with duplicate fans should be provided for the toilet extract system.

*Squash Courts*

For normal recreational use, squash courts need only be heated to 10°C with a ventilation rate of about 4 air changes per hour. However, spectator areas should be heated to 18°C.

**Standards Rooms**

It is usual for standards rooms to be designed to meet the same conditions as those maintained for the manufacturing processes, and reference should be made to the appropriate section. In practice, the environmental conditions within standards rooms may well be more exacting, in order to:

*(a)* sample and test equipment over varying environmental conditions for set time periods.

*(b)* sample and test equipment manufactured in various sections of the factory maintained at differing conditions.

**Television Studios**

The primary requirement is to provide a comfortable environment within the constraints imposed by the production of television programmes. The multiplicity of studio arrangements, changing sets, differing degrees of occupancy, and the personal discomfort caused by heavy and restrictive costumes and make-up, require a flexible type of air conditioning plant.

Within the occupied zone near floor level environmental conditions should be 21°C ±1°C, rising to 23°C ±1°C at peak loads, and humidities between 30 and 60% RH. Air speeds should be in the order of 0·2 m/s, but not higher than 0·3 m/s to avoid visual disturbance of hair, clothing and scenery drapes and adverse microphone effects.

Minimum fresh air should be calculated on the maximum anticipated occupancy. Contemporary television studios may accommodate an audience of as many as 400 persons, together with production staff numbering 100 or more.

Lighting presents a special problem, usually constituting by far the largest component of the cooling load. Invariably the total energy input to the lights is all reduced to low-grade heat within the studio and dissipated directly to the air, and may amount to as much as 400 W/$m^2$ of floor area. Generally a 50% diversity factor is appropriate but this heat gain may be concentrated on only two thirds of the total floor area. Also high-level lighting grids are a feature of studios and some consideration should be given to the comfort of people at high level.

The building usually has some thermal storage effect provided the studio is not in continuous peak production, but internal acoustic treatment usually confines this effect to not more than 10% of the peak load. Heavy construction and substantial acoustic treatment usually limit transmission gains or losses but this should be carefully checked when building designs have been completed.

The design and number of the air conditioning plants employed depend upon the type and method of air distribution applied. Broadly, the alternatives are a variable volume air supply, where the quantity of cool air introduced is adjusted by dampers fitted in take-offs from a ring main at high level and directed down into the working areas by nozzles, or a fixed volume reheat or recool system where the total air quantity remains constant.

Sound attenuation of plant and system is particularly important, and it is usual for the performance specification to call for a background noise level not exceeding NR 25, with particular attention given to the attenuation of airborne noise and isolation of mechanical vibration. Anti-vibration gear and duct acoustic protection are usually considered essential.

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# SECTION B3 VENTILATION AND AIR CONDITIONING (SYSTEMS, EQUIPMENT AND CONTROL)




*This edition of Section B3 first published 1986*

# SECTION B3 VENTILATION AND AIR CONDITIONING (SYSTEMS, EQUIPMENT AND CONTROL)

## INTRODUCTION

The aim of this Section is to provide a source of information on current practice in the design of ventilation and air conditioning systems. The characteristics of the component parts of the systems and alternative arrangements for these components along with various control strategies are outlined.

The information contained provides an overview of the options available and, together with the design criteria and applications requirements given in Sections A1, *Environmental Criteria for Design*, and B2, *Ventilation and Air Conditioning (Requirements)*, provides the basic information required for the design of mechanical ventilation and air conditioning systems.

This Section refers to many other sources of information, *see References and Bibliography*, and these should be consulted where possible.

## ROOM AIR DIFFUSION

The effectiveness of all ventilation and air conditioning systems depends on the method by which supply air is introduced to, and vitiated air removed from the space. The condition of the supply air, the air currents which are induced, the noise emitted and the appearance and position of the openings are all factors which directly influence the occupants perception of the effectiveness of the system.

The usual results of poor air terminal selection and/or positioning are draughts, stagnation, poor air purity, large temperature gradients and unwanted noise.

The terminal type and layout may be affected by architectural or structural considerations but, conversely, particular room air diffusion requirements may influence building design and/or structure (e.g. floor supply).

Air diffusion is the main interface between the system and the occupants. If the air diffusion is not well designed the system will fail, no matter how accurately building loads were modelled and how carefully the plant and equipment were selected.

The parameters which influence the quality of the air at a particular point in a room can be summarised as follows:

(*a*) air supply velocity

(*b*) temperature differential between room and supply air

(*c*) purity of supply air

(*d*) position of air supply terminals

(*e*) room shape and surface geometry, including projections

(*f*) position, size and shape of all sources and sinks for heat and contaminants

(*g*) temperature of heat sources and sinks

(*h*) rates of evolution and sorption of contaminants

(*i*) other factors influencing air movement (e.g. movement of occupants, machinery, infiltration through openings).

### Criteria for Design

The response of the occupants will be mainly determined by the following:

(*a*) velocity of air adjacent to uncovered or lightly covered skin (e.g. neck and ankles)

(*b*) temperature of the airstream in relation to the temperature of “still” air adjacent to other parts of the body

(*c*) level of activity taking place

(*d*) occupants’ clothing

(*e*) purity of air in the breathing zone

(*f*) individual susceptibility and acclimatisation.

The above are discussed in Section A1.

These factors interact with other parameters which influence perceptions of comfort such as the temperatures of surrounding surfaces, air humidity, ion content, noise and glare to produce a cumulative subjective reaction to the environment[1].

ISO 7730[1] recommends that for moderate thermal environments with light, mainly sedentary activity during cooling, the mean air velocity should be less than 0·25 m/s, whilst in winter it should be less than 0.15 m/s. No minimum velocity is suggested since, although stagnant zones could result in temperature gradients between ankle and neck greater than the 3 K recommended, it is likely that sufficient air movement will be generated by other means[2].

An occupied zone can be defined as a region, the outer limits of which are described by an envelope 1·8 m from the floor and 0·15 m from walls. However, in the case of low level supply terminals, the occupied zone is any region where the occupants are likely to linger for significant periods. In the case of desk terminals mixing occurs over the desk surface and, for seatback terminals, in the region above and between the seats.

An assessment[1] of Predicted Percentage Dissatisfied (PPD) for a wide range of activity levels, clothing, body temperatures and velocities shows that, even at low activity levels, velocities as high as 1·0 m/s can be acceptable in offsetting high temperatures. This technique has been applied to the concept of 'spot' cooling in some industrial applications[3] whereby heat stress in the workers is avoided by keeping the local conditions below an agreed threshold limit value for wet-bulb globe temperature.

**Ventilation effectiveness[4, 5, 6]**

Due to local convection currents and the uneven distribution and mixing of contaminants within a space, there will be uneven distribution of both temperature and contaminant concentration. If heat and oxygen transfer can occur within the occupied zone the condition of the air above this zone is usually unimportant. However, conventional air conditioning systems utilise dilution ventilation whereby mixing occurs outside the occupied zone and, under ideal conditions, all the air in the space is at the same temperature and of the same quality.

Ventilation effectiveness may be defined in two ways[4], as follows:

effectiveness for contaminant control:

$$E_c = \frac{C_e}{C_o} \quad \text{B3.1}$$

where:

$E_c$ = effectiveness for contaminant control ..
$C_e$ = concentration of contaminant in exhaust .. .. .. .. .. ppm
$C_o$ = concentration of contaminant in occupied zone .. .. .. .. ppm

effectiveness for heat exchange:

$$E_t = \frac{t_e - t_s}{\bar{t} - t_s} \quad \text{B3.2}$$

where:

$E_t$ = effectiveness for heat exchange.. ..
$t_e$ = exhaust air temperature .. .. °C
$t_s$ = supply air temperature .. .. °C
$\bar{t}$ = average temperature in occupied zone °C

High effectiveness result from local removal of heat and contaminants with a corresponding reduction in energy consumption at the central plant.

**Air Terminal Categories[7]**

Air can be supplied to a space in a number of ways, the principal division being between diffusers and perpendicular jets. Airflow patterns for both types of terminal are strongly dependent on the presence or absence of the Coanda effect, see below.

Diffusers may be radial, part-radial or linear and normally utilise the Coanda effect and/or "swirl" to reduce the risk of excessive room air movement.

A perpendicular jet is formed by discharging air through grilles, louvres, nozzles or any opening which allows perpendicular flow: direction and spread adjustment can be provided using blades and/or swivel adjustment.

Supply air terminal devices can be incorporated into any room surface, e.g. ceiling (flat or sculptured), floor, wall (high or low level), desk top, seat back or under seats. Table B3.1 summarises the types of air terminal devices and provides information on typical face velocities (based on any local control dampers being fully open) and noise levels.

**Air Terminal Phenomena**

Many studies of jets and their effect on room air movement have been undertaken[8-17]. Fig. B3.1[18] shows the predicted air flow patterns for various types and positions of air terminal device.

It should be noted that these patterns are based on stylised terminals. For predictions of air movement appropriate to specific air terminals, the manufacturers' data must be consulted. For non-standard situations it may be necessary to model room air movement using a mock-up. In many cases it will be necessary to allow for on-site adjustment of airflow pattern, either during commissioning or during operation by the occupant (e.g. desk mounted terminals).

**Air Diffusion Terminology**

ISO 3258[19] gives definitions and standard terminology used in connection with air movement. Some of the more important parameters are outlined below.

*Throw*

A free jet having a given momentum on discharge will establish velocity profiles, known as isovels, the shape of which depend on the geometry of the terminal, the temperature of the jet and any other disturbing influences. The velocity decays with increasing distance from the terminal. Throw is defined as the distance from the terminal (measured perpendicular or parallel to the face of the air terminal device depending on predominant direction of flow) to the 0·5 m/s isovel.

Normally lower velocities are required for air entering the occupied zone, typically 0·25 m/s for cooling, 0.15 m/s for heating. Reference should be made to manufacturers' literature for throw data and recommended mounting distances from solid surfaces and neighbouring terminals.

Maximum throw for an air terminal device depends on the characteristics of the device, the mounting height and the influence of neighbouring surfaces.

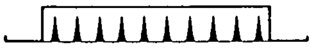
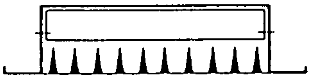
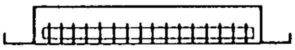
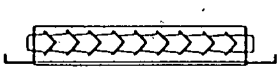
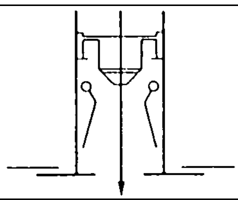
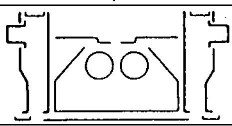
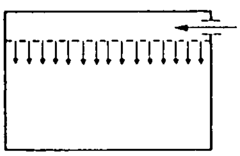
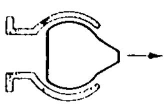
*Spread*

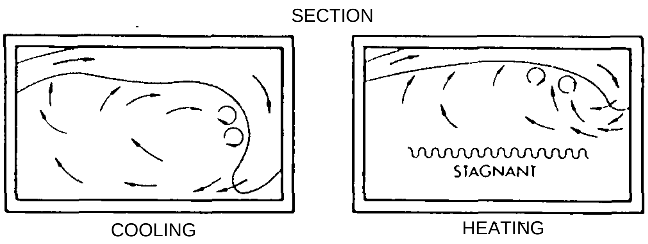
The spread of a horizontal jet is defined as the width of the 0·5 m/s isovel. Note that most manufacturers give the width of the 0·25 m/s isovel, which is generally of more use to the designer.

*Drop*

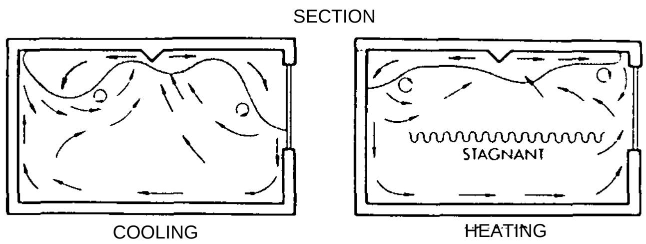
The drop is defined as the vertical distance from the centre-line of the terminal to the bottom edge of the 0·25 m/s isovel.

**Table B3.1.** Types of air handling unit.

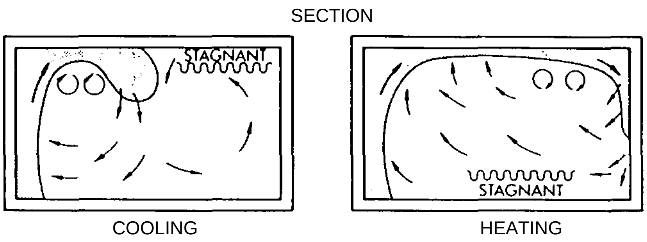
| Type | Diagram | Application and location | Core velocity / (m/s) | | Description and remarks |
|---|---|---|---|---|---|
| | | | Quiet | Commercially quiet | |
| 1. Perforated or stamped lattice |  | Supply<br>Extract<br>Transfer<br><br>Ceiling<br>Sidewall<br>Floor | Up to 4 | Up to 6 | Simple form of grille with small free area. Alternatively can be used as supply diffuser with high air entrainment allowing large quantities to be diffused. For low level "laminar flow" panels to give displacement ventilation, velocity of 0·25 m/s is used. |
| 2. Aerofoil blades — one row adjustable |  | Supply<br>Extract<br><br>Ceiling<br>Sidewall<br>Desk top | 7 | 10 | Frequently used grille with large free area. Directional control in one plane only for supply applications. |
| 3. Aerofoil blades — two rows adjustable |  | Supply<br><br>Sidewall | 7 | 10 | As 2 but with directional control in two planes. |
| 4. Fixed blade |  | Supply<br>Extract<br><br>Ceiling<br>Sidewall<br>Floor<br>Seat back | 6 | 9 | Robust grille with limited free area. Some directional control possible using profiled blades. |
| 5. Non-vision |  | Extract<br>Transfer<br><br>Sidewall | 7 | 10 | Low free area. Designed to prevent through vision. |
| 6. Egg crate |  | Extract<br><br>Ceiling<br>Sidewall | 7 | 10 | Generally largest free area grille available. |
| 7. Fixed geometry diffusers |  | Supply<br>Extract<br><br>Ceiling<br>Floor<br>Desk top | 7 | 10 | Radial discharge diffusers offer good air entrainment allowing diffusion of large air quantities. Square or rectangular diffusers can provide 1, 2 or 3 way diffusion. Angled blades or slots can be used to apply twisting motion to supply air. |
| 8. Adjustable diffusers |  | Supply<br><br>Ceiling | 4 | 6 | As 7 but offers horizontal or vertical discharge. Can be thermostatically controlled. |
| 9. Slot and linear diffusers |  | Supply<br>Extract<br><br>Ceiling<br>Sidewall<br>Desk top<br>Under window | 6 | 9 | Offers vertical or horizontal discharge; single or multiple slots. Care must be taken with design of plenum box. Desk top units may incorporate induction of room air. |
| 10. Air handling luminaires |  | Supply<br>Extract<br><br>Ceiling | 7 | 10 | As 9 but single slot only. Normally used in conjunction with extract through luminaire. |
| 11. Ventilated ceiling |  | Supply<br>or<br>Extract | | | Void above ceiling is pressurised to introduce air at low velocity through many single holes or through porous panels. Air entrainment is restricted and natural air currents may affect room air diffusion. |
| 12. Nozzles, drum and punkah louvres |  | Supply<br><br>Ceiling<br>Sidewall<br>Under window<br>Seat back | | | Adjustable type shown can be rotating drum or swivelling ball with or without nozzle jet for long throws and personal air supply/spot cooling. For high-induction applications fixed multiple nozzles are used Velocities depend on throw, noise and induction requirements. |



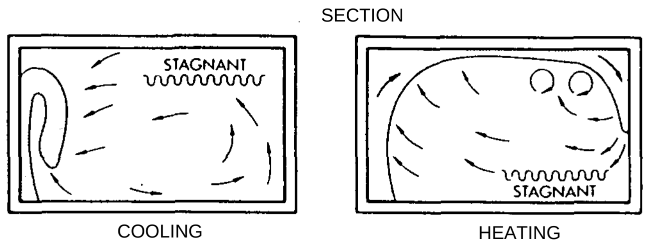

(a) High sidewall outlet



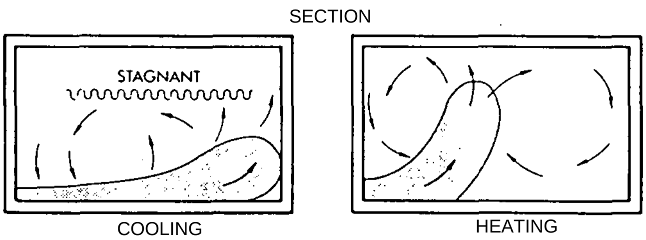

(b) Ceiling outlet



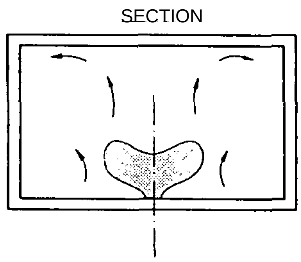

(c) Outlet in or near floor, non-spreading jet



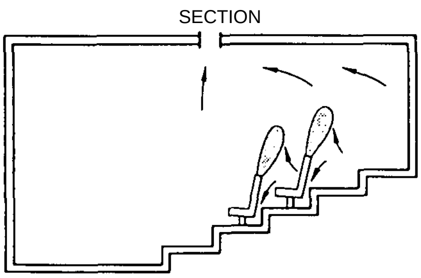

(d) Outlet in or near floor, spreading jet



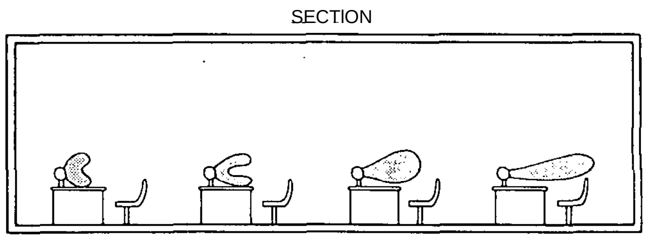

(e) Outlet near floor, horizontal discharge




(f) Outlet in floor, spreading vertical jet with twist




(g) Outlet in seat back, non-spreading vertical jet




(h) Personal adjustable desk outlet

**Fig. B3.1.** Predicted air flow patterns for typical air diffusers.

**Entrainment, Mixing and Boundaries**

Frictional forces cause a momentum transfer to take place between the jet and adjacent room air which draws the room air in the same direction as the jet. The jet expands with distance from the terminal as it entrains adjacent room air. Hence kinetic energy is expended in creating turbulence which transfers thermal energy and assists the dilution of contaminants. This process of diffusion may be enhanced by the introduction of a rapidly expanding jet and still further by imparting a swirling motion to the jet.

A jet which is constrained by the walls of a room, such as a full width slot, will entrain less room air and expand more slowly than a free conical jet[7, 18].

**Effect of Temperature Differential**

Fig. B3.1 shows that a jet which is nor influenced by the proximity of a solid surface follows a path which is a function both of velocity and temperature: a warm jet tends to rise until it attaches itself to a horizontal surface, whilst a cold jet falls. Care must be taken to ensure that this does not lead to unacceptable temperature gradients in the occupied zone during heating and excessive air velocities during cooling. The terminal must be mounted such that the 0·25 m/s isovel does not enter the occupied zone.

The difference in temperature between supply and return air may be greater than that between the supply air and the room air, particulary with low level supply designed to encourage high level stratification. This temperature difference is related to sensible heat gain and supply air mass flow, as follows:

$$q_s = \dot{m}\ C_{ph}\ \Delta t \qquad \text{B3.3}$$

where:

$q_s$ = total sensible heat gain .. .. kW
$\dot{m}$ = rate of mass flow of supply air .. kg/s
$C_{ph}$ = humid specific heat of air .. .. kJ/kg K
$\Delta t$ = room air to supply air temperature differential .. .. .. .. K

Therefore, the mass flow rate, and hence the cost of air handling, will depend upon the value of temperature difference chosen by the designer. This decision will also be influenced by the evaporator temperature and the level of control of humidity.

For example, a displacement system with low level input can supply air at 18°C with a temperature difference of about 10 K. This can be achieved with high evaporator temperatures and correspondingly low compressor power. However, high level humidity control will suffer unless supply air is over-cooled and reheated, normally an undesirable combination at peak load. Alternatively, a permanent bypass around the cooling coil can be provided and, if motorised dampers are incorporated at the coil face and in the bypass, part load control supply temperature can be achieved by damper modulation (see System Data Sheets, *Single Zone Systems*).

For comfort applications air change rates are unlikely to exceed 10 per hour corresponding to a cooling temperature differential range of 8 to 12 K. A free horizontal jet from a rectangular grille is likely to create down-draughts if providing more than 8 air changes per hour with a cooling temperature differential greater than 8 K.

A maximum cooling temperature differential of 10 K can be applied when either:

(*a*) the presence of Coanda effect (see below) is assured

(*b*) for a free jet, mixing of supply air with room air outside the occupied zone can be assured without promoting discomfort.

Table B3.2 gives general guidance on the maximum air change rates that can be achieved using various air terminal devices supplying air with a cooling temperature differential of 10 K.

**Table B3.2.** Typical maximum air change rates for air terminal devices.

| Device | Air change rate/h$^{-1}$ |
|---|---|
| Sidewall grilles | 8 |
| Linear grilles | 10 |
| Slot and linear diffusers | 15 |
| Rectangular diffusers | 15 |
| Perforated diffusers | 15 |
| Circular diffusers | 20 |

If sufficient mixing cannot be guaranteed between terminal and occupants (e.g. with low level supply), then the minimum supply temperature of 18°C applies with a temperature differential in the occupied zone of 4 to 5 K. However, the cooling temperature differential is ultimately determined by the maximum exhaust air temperature[20], see Table B3.3.

**Table B3.3.** Typical cooling temperature differentials for various applications.

| Application | Maximum temperature difference / K |
|---|---|
| High ceiling<br>large heat gains/low level input | 12 |
| Low ceiling<br>air handling luminaires/low level input | 10 |
| Low ceiling<br>downward discharge | 5 |

The larger temperature differential indicated for high ceilings is possible due to the smaller influence of ceiling temperature on the mean radiant temperature experienced by the occupants.

Downward discharge is generally only satisfactory for very high air change rates and hence small temperature differentials, and where room convection is not significant (see below).

**Effect of Room Convection Currents**

High level supply jets have to overcome the buoyancy forces in the room air generated by heat emitters, solar gain, occupants etc., whereas low level input cultivates these forces to assist the supply jet. For this reason a low level supply system is most satisfactory for applications with high room gains and high ceilings. For low ceilings, the radiant heating effect of the ceiling itself may be significant. This may also be a problem where the ceiling void is used as an exhaust air plenum, carrying air heated by air handling luminaires.

Free descending jets are not recommended for normal use since the low velocity approaching the occupied zone would cause instability. This could result in localised high velocities due to deflection by convective forces elsewhere in the room, see Fig. B3.2. An exception to this general guidance is the case of laminar downflow clean rooms[21, 22] where an even velocity, across the full area, of 0·4 m/s should be maintained from ceiling to floor. However, even in these circumstances, sources of extremely buoyant upflow should be avoided.

**Coanda Effect**

When a jet is discharged from a terminal device adjacent and parallel to an unobstructed flat surface, the jet entrains air from one side only resulting in deflection of the axis of the jet towards the surface. This phenomenon, known as the Coanda effect, is due to frictional losses between the jet and the surface.

The effect diminishes with distance form the terminal as increasing volumes of air are entrained from the room-side of the jet, resulting in a reduction of jet velocity. However, Coanda effect is maintained despite

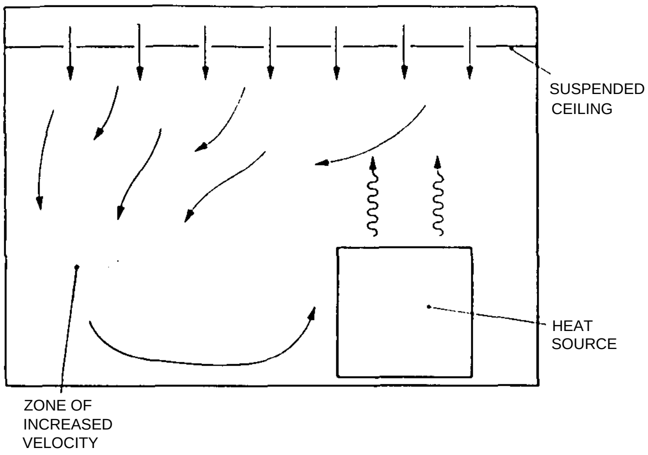



**Fig. B3.2.** Air flow pattern produced by unstable vertically descending air jets.

temperature differences between the jet and the room air and it is a critical factor influencing the selection and positioning of supply air terminals, particularly for rooms with low ceilings which have little space above the occupied zone in which mixing can occur.

If the Coanda effect is not present the maximum throw for any terminal is reduced by approximately 33%.

The main factors which influence whether Coanda Effect occurs are:

(*a*) distance between terminal and surface

(*b*) width of jet exposed to surface

(*c*) velocity of jet

(*d*) projections and other disturbing influences.

The importance of these influences for sidewall terminals with various aspect ratios, velocities and temperature differences are discussed elsewhere[15]. The most important factor is temperature difference, i.e. buoyancy effects. For the usual range of temperature differences for cooling of 8 to 12 K the opening should be within 300 mm of the surface to guarantee attraction. For systems designed to make use of the Coanda effect, provision should be made for on-site adjustment of the jet.

When a jet adheres to a surface dust particles will be deposited on the surface leading to staining, hence supply air cleanliness is of paramount importance (see *Equipment: Air Cleaners and Filtration*). Cleanliness of the exhaust air is difficult to control and some staining of surfaces near to exhaust openings is inevitable.

**Effect of Projections and Junctions**

Techniques exist[13] for predicting the influence of projections, such as downstand beams and surface mounted luminaires, on a jet flowing across an otherwise smooth surface. An obstruction may cause the jet to separate completely from the surface, hence destroying the Coanda effect, or it may separate and rejoin some distance downstream of the obstruction.

The critical distances at which these phenomena are likely to occur depend on the depth and shape of the obstruction and the size of the supply opening. The influence of supply air to room air temperature differential is small but depends on the extent to which mixing has occured before the jet meets the obstruction.

Fig. B3.3 shows the effect of a horizontal surface on a jet rising close to a vertical surface. The Coanda effect is maintained after the change in direction provided that the velocity is adequate, particularly in the case of cooling jets, and that the temperature differential between supply and room air is not too large. A design procedure for selecting optimum supply velocities and temperatures is given elsewhere[16].

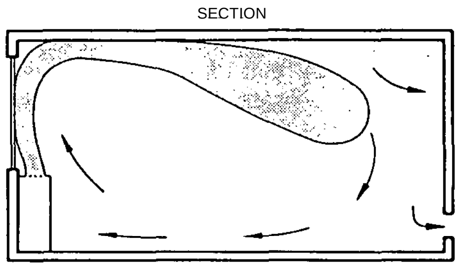


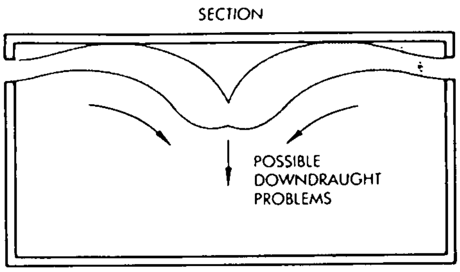

**Fig. B3.3.** Vertical cooling jet discharging under a window.

**Interaction Between Jets**

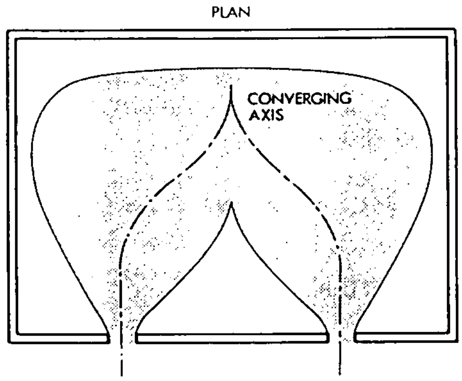
Fig. B3.4(a) shows possible room air velocity patterns for two jets directed towards each other along a 3 m high ceiling. The individual velocities of the two airstreams must not be greater than 0·25 m/s at the boundary otherwise discomfort due to excessive downdraughts may occur.

The envelopes of two converging jets may also interfere with each other, combining to form a single wider jet with a maximum velocity at the new axis between the two jets, see Fig. B3.4(b). A similar phenomenon occurs with two jets moving in tandem, see Fig. B3.4(c). The downstream jet entrains and accelerates the decaying upstream jet and forms a wider jet with an axis further from the neighbouring surface. The cumulative effect of a series of single-way jets can result in a deep jet which intrudes into the occupied zone resulting in unacceptably high room velocities.

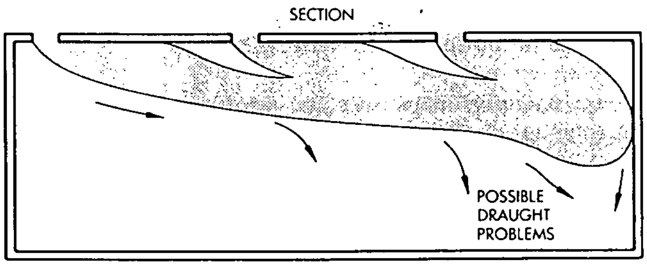
Fig. B3.5 shows examples of possible layouts for ceiling diffusers. The main problems likely to be encountered are those described above. Downdraughts may be encountered in areas marked “X” and this problem may be eliminated by avoiding terminals with excessive throw, particularly in large spaces where stagnation between terminals is unlikely to occur. The layout shown in Fig. B3.5(c) may cause convergence problems with long rooms.




(a) Opposing jets

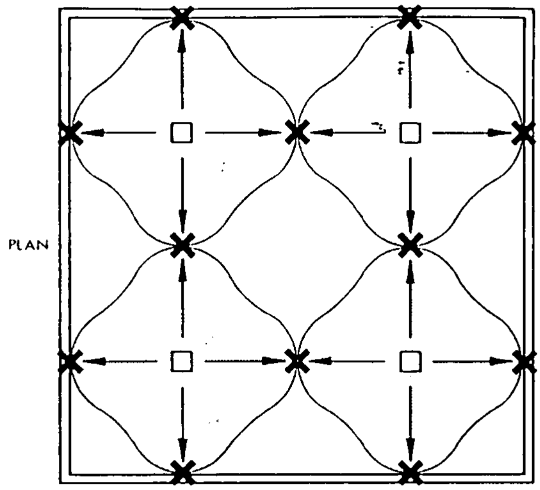



(b) Converging jets

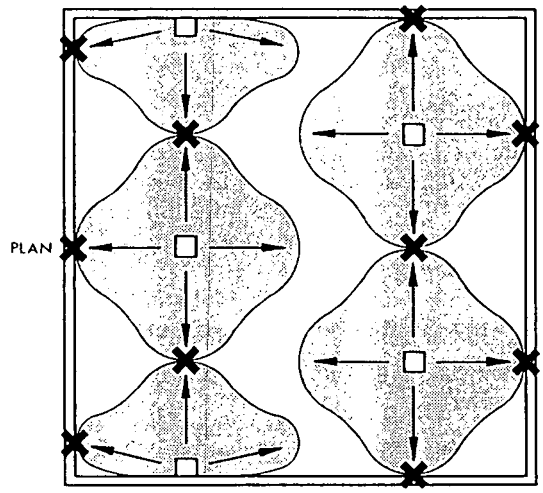



(c) Three jets in series

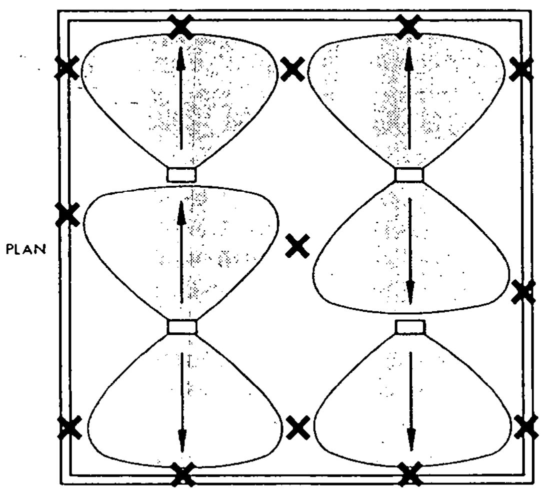
**Fig. B3.4.** Interactions between jets.




(a) Four-way ceiling diffusers, symmetrical layout




(b) Four-way ceiling diffusers, off-set layout




(c) One- and two-way celing diffusers, contra-flow layout

**Fig. B3.5.** Supply terminal layouts for open-plan spaces.

For sidewall applications, the spacing of diffusers should be in accordance with manufacturers' recommendations. However, in the absence of such information Table B3.4 gives typical data. For a terminal mounted close to a wall the spacing should be halved to give the minimum distance from centreline to wall.

**Table B3.4.** Spacing of ceiling diffusers – typical data.

| Deflection / degree | Spacing / m | |
|---|---|---|
| 0 | 0·20 $L_v$ | 0·33 $L_v$ |
| 22·5 | 0·25 $L_v$ | 0·50 $L_v$ |
| 45 | 0·30 $L_v$ | 1·0 $L_v$ |

*Note:*

$L_v$ = throw in metres where axial velocity has decayed to 0·25 m/s

$L_v$ = throw in metres where axial velocity has decayed to 0·5 m/s

Table B3.5[20] indicates typical turndown limits for various types of fixed area air terminal device.

**Table B3.5.** Turndown limits for various types of fixed air terminal device.

| Type of outlet | Maximum turndown / % |
|---|---|
| Ceiling mounted | |
| not using Coanda effect | 50 |
| using Coanda effect | 40 |
| Floor-mounted outlets | |
| perforated plate and fixed bar grille | 60 |
| free jet outlets | 50 |
| outlets with swirl | 40 |
| Desk outlets | |
| linear type | 50 |
| ball type | 50 |
| outlets with swirl | 40 |

### Exhaust Terminals

The zone of localised high velocities associated with exhaust openings is very close to the opening. Therefore the position of the opening has little influence upon the airflow pattern in the space (see *Industrial Ventilation and Local Exhaust).*

Exhaust terminals may be sited to advantage as follows:

(*a*) in a stagnant zone where supply jet influence is limited

(*b*) close to a source of unwanted heat and/or contamination, e.g. above a luminaire

(*c*) close to an excessively cold source to increase its surface temperature and thereby reduce radiant losses and cold draughts

(*d*) at a point of local low pressure, e.g. the centre of a ceiling diffuser.

The following positions should be avoided:

(*a*) within the zone of influence of a supply air terminal, because this allows conditioned air to pass directly into the exhaust without having first exchanged heat with its surroundings resulting in very low ventilation efficiency

(*b*) close to a door or aperture which is frequently opened, since this leads to the exhaust handling air from outside the conditioned space

(*c*) in a position which causes contaminated room air to be drawn through the occupants' breathing zone.

### Duct and Plenum Design

Air terminal devices will only perform as intended if the approach velocity profile is even. If the duct connections and/or volume flow regulators create eddies at the terminal, the following problems may arise:

(*a*) unpredictable throw, spread and drop

(*b*) breakdown of Coanda effect

(*c*) high noise levels

(*d*) balancing is difficult or impossible.

Design procedures for duct and plenum connections to various types of air terminal are given elsewhere[7, 23].

If the ceiling is to be used as an exhaust plenum it is important to create a uniform negative pressure throughout the whole ceiling void to ensure even exhaust through all terminals. This is particularly important when exhaust is via air handling luminaires, the performance of which varies with air flow rate.

Ceiling voids should be made as large as possible and, if obstructed by luminaires, ductwork etc., exhaust stub ducts should be provided to ensure even exhaust over the full ceiling area.

## NATURAL VENTILATION

Openings in the building fabric can be designed and positioned to provide sufficient outside air for:

(*a*) carbon dioxide and odour dilution

(*b*) prevention of condensation

(*c*) limiting summertime overheating.

These openings constitute the ventilation system and must be designed to achieve the desired performance.

Ideally, a building should be of air-tight construction with adjustable openings that:

(*a*) allow ingress of the exact quantity of outside air required for the above under any outside conditions, without causing discomfort from draughts

(*b*) enable larger quantities of outside air to enter to limit the internal temperature rise in summer under light wind conditions.

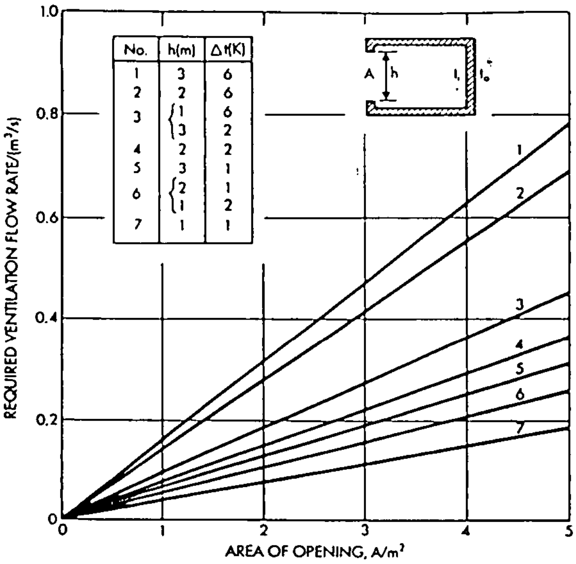
In practice, all structures are subject to some degree of uncontrolled air leakage, referred to as infiltration.

Ventilation for the dispersal of smoke during a fire may require larger openings than those provided for natural ventilation (see Section BS, *Fire Safety Engineering*).

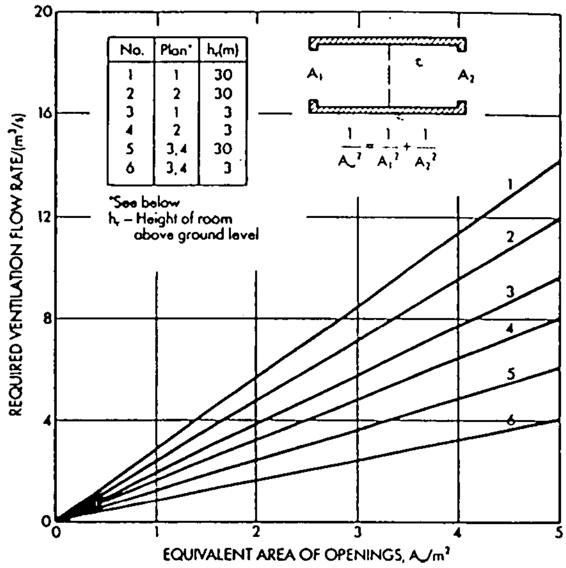
### Summer Ventilation

Usually, the sizes of openings will be dictated by the ventilation rate required to control the rise in internal temperatures in summer and Section A8, *Summertime Temperatures in Buildings,* gives a method for predicting summertime temperatures. Section A4, *Air Infiltration and Natural Ventilation* gives methods of assessing the rates of natural ventilation under a range of conditions.

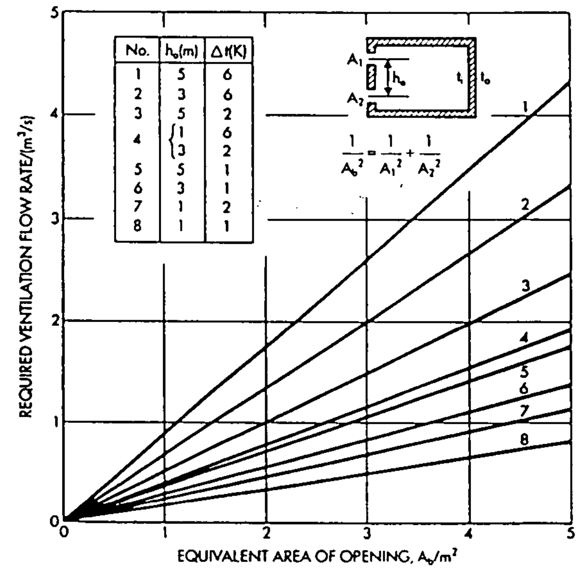
Fig. B3.6 has been constructed using such methods[24] to assist in the sizing of openings to achieve the required summer ventilation rate. Figs. B3.6(a) and (b) assume no wind and show the ventilation rates due to stack effect alone for various differences between room and



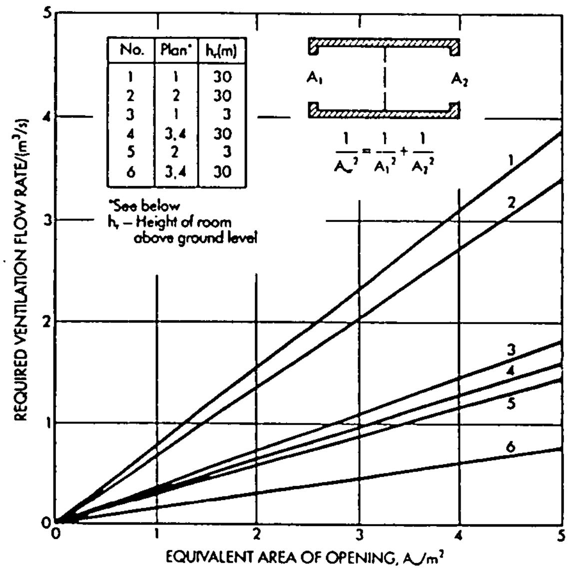

(a) Single opening on one side, stack effect only



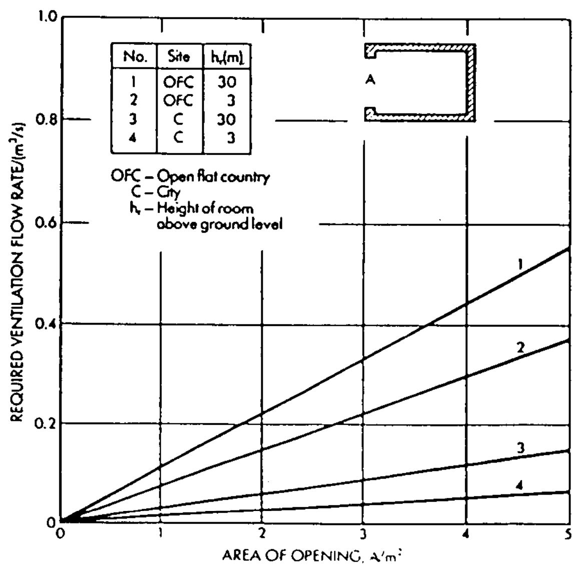

(d) Single opening on two sides, wind only; exposed location in open, flat country



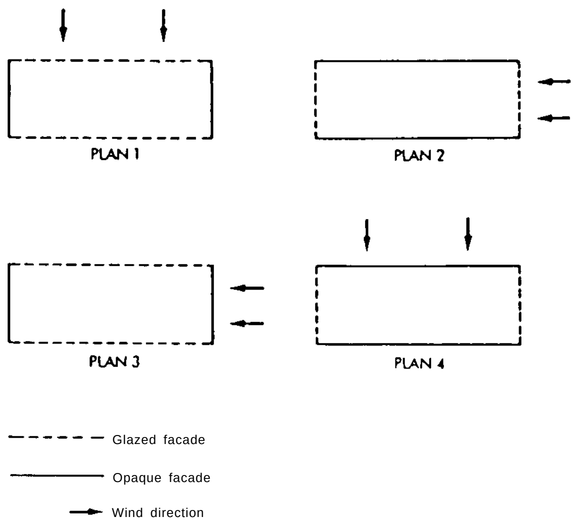

(b) High level outlet and low level inlet on one or two sides, stack effect only




(e) Single opening on two sides, wind only: sheltered location in urban area




(c) Single opening on one side, wind only




(f) Plan of rectangular tower block for (d) and (e)

**Fig. B3.6.** Summer ventilation by open window lights.

extract temperature. They should be used for extremely sheltered rooms, e.g. courtyards and light-wells. Figs. B3.6(c) to (c) assume stack effect is insignificant in comparison with a light wind which is deemed to be exceeded for 80% of the year.

Exposure, terrain, height above ground level, building shape, and wall/wind orientation all influence the rate of airflow into the building. Figs. B3.6(c) to (e) should be used for any window that has significant exposure to wind. In the case of graphs for winds at right angles to openings a minimum flow pressure coefficient of 0·2 has been assumed.

It is assumed that, when open, the airflow through the ventilation opening is not impeded by the moveable parts of the device.

All openings for summer use should be located so as to provide air movement over the occupants to minimise discomfort due to high temperatures. However, to avoid disturbing papers on desks etc., local velocities should not be greater than 0.8 m/s.

**Winter Ventilation**

In winter the required opening is much smaller and the flow is usually characterised by the equation appropriate to a crack in the external fabric. Section A4 gives a method of calculating the rate of infiltration through cracks and this should be compared with the minimum ventilation rates recommended in Section B2, *Ventilation and Air Conditioning (Requirements).* The infiltration rate should be calculated under the same wind conditions used for determining maximum heat loss, since the adjustable portion can be opened if necessary under low wind conditions.

BS 6375: Part 1[25] specifies four grades of crack leakage and the method contained therein can be used to optimise between winter crack air flow against summer opening crack length. In extreme winter climates the highest grade (i.e. smallest area) of window crack should be specified with separate provision for winter ventilation by means of adjustable ventilators. All wintertime ventilation openings should be located to avoid subjecting the occupants to unheated draughts and, if possible, the incoming air should be mixed with a warm airstream on entry into the occupied space.

*Sliding Windows*

Sliding windows provide good summer ventilation but need very carefull weather-stripping to minimise leakage.

*Vertical Sliding (Sash) Windows*

Sash windows provide good control, particularly of stack effect since opening size can be adjusted at both top and bottom of the window. However, crack leakage is difficult to control.

*Side Hung Windows*

The combination of top hung winter ventilators and side hung summer windows (with effective weather-stripping) provides good all-round performance. The top hung winter ventilator can also provide a secure opening for summer ventilation which, in combination with the side hung opening, will enhance stack effect.

*Horizontal or Vertical Centre Pivot Windows*

These provide good directional control of summer ventilation but need carefully designed weather-stripping to achieve high grades of crack leakage.

*Louvred Ventilators*

Whilst providing good control over summer ventilation adjustable louvres usually present the greatest crack length for a given opening. However, conventional hinged louvres are usually difficult to seal when closed.

*Air Bricks*

Air bricks incorporate no provision for control of infiltration rate and automatic ventilators, which provide nominally constant infiltration under variable wind velocities, should be considered as an alternative.

*Roof Ventilators*

In combination with low level openings in the fabric roof ventilators can be used to take advantage of summer stack effect, particularly for tall spaces. However, they must be specified to have low crack leakage or wind induced draughts will cause discomfort in winter.

Rooftop ventilators generally fall into two categories ridge and circular. The ridge type is less obstrusive but their efficiency is impaired by variations in wind direction whereas circular stack outlets are not affected if positioned correctly. For maximum effect the outlet should be on the ridge of a pitched roof and the cap should project sufficiently above the ridge to minimise the influence of turbulence arising from wind blowing up the slope of the roof. Natural ventilation openings should never be installed on the slope of a roof nor should they be located in high pressure areas of the building environment, nor where downdraughts are likely to occur.

*Fixed Lights*

Fixed lights may give crack leakage rates between zero[2] and 1 $m^3$/h per meter length of visible perimeter of glass depending on the gasket material. Therefore crack leakage from fixed lights should not be relied upon to provide winter ventilation.

*Automatic Control*

Temperature sensitive thyristors may be incorporated into roof ventilators or top-hung ventilators to provide automatic control over summer ventilation. The practice of providing automatic control over minimum winter ventilation rates is not widespread but should be considered.

*Airflow Between Rooms*

The algorithms for air flow due to stack effect, wind, partition resistance and room networks are contained in Section A4. Various computer programs are also available for the calculation of airflow rates between rooms.

*Structural Leakage*

Care should be taken during construction to avoid infiltration paths through the structure and guidance in constructional detailing is given elsewhere[26]. Current ‘wet trades’ construction methods exhibit infiltration rates through walls and window surrounds that are likely to be in excess of the window crack leakage rates as calculated using Section A4.

## SYSTEMS: GENERAL NOTES

In this Section, air conditioning is differentiated from mechanical ventilation by virtue of the incorporation, or otherwise, of mechanical refrigeration; that is, many of the systems described below would be regarded as mechanical ventilation systems on omission of the cooling coil.

An air conditioning system need not provide humidification of the supply air since there will be many instances where this facility is unnecessary in meeting the comfort needs of the occupants. Control over a rise in humidity needs to be provided only insofar as comfort or process requirements demand and to limit the risk of microbiological growth (i.e. to maintain room percentage saturation below 70%).

For the purposes of this Section, the following categories of air conditioning are identified:

(a) comfort air conditioning

(b) close control air conditioning.

For comfort air conditioning, it is usually satisfactory to supply air with a sufficiently low moisture content to cater for maximum latent gain and limit room percentage saturation by overriding either the humidity sensor or the temperature sensor in the air leaving the cooling coil, as appropriate.

Close control air conditioning is difficult to achieve with multiple zone systems since each zone requires a dehumidifying cooling coil, reheater and humidifier to give total control of humidity.

Cooling systems with low level input and high level extract (see *Room Air Diffusion*) require higher supply temperatures for summer cooling and can sometimes dispense with the need for mechanical refrigeration by some combination of the following:

(*a*) drawing outside air from a shaded north facing aspect

(*b*) drawing outside air from a point clear of the "heat island" at ground level

(*c*) drawing outside air through a buried earth-cooled duct

(*d*) supplying the cooling coil with indirectly and evaporatively cooled water.

In the last case, the potential hazards of microbiological contamination must be considered.

If mechanical refrigeration is not provided, humidity control will be difficult since little dehumidification is available from the above, largely passive, sources of cooling. However, with low level input, moisture from the occupants will not mix thoroughly but be carried to a high level with the upward moment of the air.

If heat gains are moderate, it may be possible to use all-air systems without cooling to limit the rise in material summertime temperatures, in which case larger air change rates would be required than for air conditioning. Again, it may be possible to limit the rise in inside temperature by drawing air into the central plant at a lower temperature than the outside dry-bulb temperature.

When used as make-up air for industrial local exhaust systems, appropriate zoning and local control can be used to cater for the intermittency and/or diversity of the industrial process taking place.

### System Control

Automatic controls are dealt with in Section B11, *Automatic Controls* and, in greater detail, in the CIBSE Applications Manual, *Automatic Controls and their Implications for Systems Design*[27]. The following notes discuss some aspects of control peculiar to air conditioning and mechanical ventilation systems.

Fig. B3.7 summarises the various control options for single zone applications. Control options for full fresh air systems are similar to those for recirculation systems but must include provision for frost protection upstream of the filters. In most recirculation applications it will be worthwhile incorporating motorised dampers (sequenced with the coils) so that outside air, when available at an appropriate condition, may be used to achieve the desired room conditions with minimal load on the central plant.

It may also be worth incorporating a means of holding the mixing dampers on full fresh air, cycling to minimum fresh air when outside air enthalpy ($h_o$) is greater than room enthalpy[27] ($h_r$), see Appendix 1, Fig. B3.68.

It should be borne in mind that the more complex the control scheme, the greater the capital cost and the greater the chances of control malfunction. In particular, humidity sensing is prone to inaccuracy and drift.

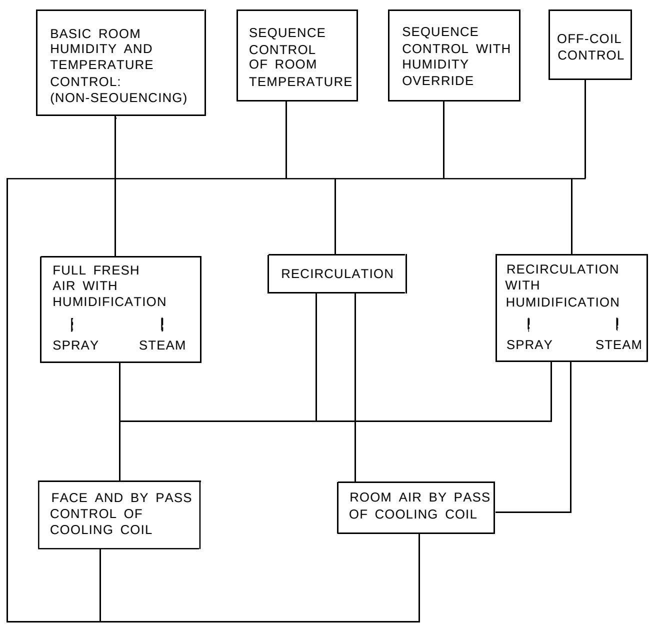



**Fig. B3.7.** Control techniques for single zone systems.

High humidity can be controlled by one of the following methods:

(*a*) Closed loop control by means of a humidity sensor located in the space or exhaust air duct which monitors changes in latent gain subject to time lags (associated with the room, the control system and the plant) and the inherent inaccuracy of the humidity sensors.

(*b*) Open loop control using a sensor to measure the temperature of the air leaving the cooling coil. (This method of control is often known as “dew-point” control but will be referred to here by the more correct term of “off-coil” control.)

In the latter case, the supply air moisture content is controlled by the dry-bulb temperature sensor. This gives accurate humidity control providing the cooling coil is efficient and variation in room humidity is predictable (a humidity sensor can be incorporated to override the cooling coil operation if the occupancy increases above the usual level). Also, simultaneous cooling/dehumidification and reheating will occur for much of the cooling season. With a system serving externally influenced spaces the off-coil sensor set point may be reset when the moisture content of the incoming air falls below that required to deal with latent gains. “Dew-point control” is dealt with more fully elsewhere[27].

Humidity sensors can be used to limit humidity rise by:

(*a*) controlling the output of a cooling coil by proportional control (with integral action if required)

(*b*) overriding the action of a temperature sensor controlling some combination of heating coil, cooling coil, mixing dampers and/or extract air heat recovery device

(*c*) resetting the set point of an off-coil sensor

(*d*) overriding control over the reheater so that the sequencing room temperature sensor calls for further cooling and hence dryer air is supplied

(*e*) overriding control over the reheater in a variable air volume zonal reheat terminal or the mixing dampers in a dual duct terminal so that the zonal temperature sensor calls for a larger volume of dry air to be supplied to that zone.

If control of high humidities is not required, the limits of the proportional band of a sequence controller can be the winter and summer design room conditions. Otherwise, different conditions for summer and winter can only be achieved by using integral action to remove offset (sequential control will normally require a wide proportional band, particularly if mixing dampers are included), and by resetting the set point of the room temperature sensor in response to an outside temperature sensor.

Humidity sensors can be used to control low humidity by:

(*a*) providing step or on/off control of a steam humidifier (see Fig. B3.10)

(*b*) providing proportional control of a preheater and/or mixing dampers to provide appropriate on-conditions to a water spray-coil humidifier, with the spray pump running continuously (see Fig. B3.12)

(*c*) switching on a spray washer pump or spinning-disc humidifier and provide appropriate on-conditions by proportional control over the preheater and/or mixing dampers.

If off-coil sensors are not employed, a low limit sensor may be required to bring-in the heating coil if the supply air temperature falls below the minimum design value. This is necessary where room or return air temperature sensors are likely to be slow to respond to low supply conditions.

Full fresh air and variable air volume systems are particularly susceptible to frosting of filter and water coils. Other systems may also suffer during damper sequencing from a room sensor (with inherent time lags) under high gain conditions in winter. Stratification through the mixing box may also be a problem (see *Equipment: Mixing Boxes*). In these cases, electric or water-fed serpentined coils should be provided, switched at 4 to 5°C from a downstream thermostat.

Other controls, not indicated on the following systems schematics, may be required to deal with early morning boost, heat recovery and variable occupancies[27] (see also Section B11).

Simultaneous heating and cooling can be avoided by bypassing the cooling coil with either outside, mixed or room air. This relies on accurate damper positioning for control over room conditions and may produce elevated room humidities.

**Position of Fans**

The systems schematics which follow (see *Systems Data Sheets*) mainly indicate a “draw-through” arrangement for the supply air handling plant, with separate extract fan, see Fig. B3.8(a). Alternative arrangements include “blow-through” and combined supply/extract fan. The former is the normal configuration for dual duct systems.

*“Blow-through” Central Plant*

The main advantages of positioning the fan upstream of the cooling coil are that:

(*a*) a lower supply air moisture content can be achieved at a particular apparatus dew-point and chilled water temperature (see Fig. B3.8(c))

(*b*) the cooling coil condensate drain will be under positive pressure which reduces the chances of drawing airborne contaminants from the drainage system or plant room into the system.

The main disadvantages are that:

(*a*) since the cooling coil is under positive pressure there is a greater risk of condensate leakage through the casing

(*b*) an additional plenum or transition piece is required at fan discharge to reduce the air velocity to an appropriate value at coil face and ensure even distribution over face area.

Soiling of the fan may be reduced by locating the filter upstream of the fan (see Fig. B3.8(a)).

*Combined Supply/extract Fan*

A single fan can both draw air through the extract system and blow air through the supply distribution system, providing that a balance can be achieved between extract and intake pressure losses using an appropriate combination of fixed resistance and damper in the intake.

In most cases free cooling from full fresh air will be required. Therefore, means must be provided for varying the proportion of return air to outside air at the mixing box by damper modulation. Some means of pressure relief will be required in the building or system and Fig. B3.8(c) shows a relief damper controlled from a room pressure sensor, $P_R$. For extract systems having low resistance this damper could be replaced with simple weighted pressure relief flaps. (See also *Equipment: Mixing Boxes*).

## Zoning

Loads on an air conditioning plant are rarely constant due to changes in solar gain, occupancy, use of lights etc.

If the loads throughout a building vary together (i.e. are in phase), or the variations are not large enough to cause the internal conditions to drift outside the acceptable limits, single zone control can be adopted. However, if different areas experience load changes which are out of phase, supply air must be provided at a rate or condition appropriate to each zone.

Most deep plan buildings require division into perimeter and internal zones. The depth of perimeter zones mainly depends on the penetration of sunlight and daylight which is determined by orientation, external shading, shape and size of windows, characteristics of the glass and the type and pattern of use of blinds.

For a typical multiple zone application the following should be noted:

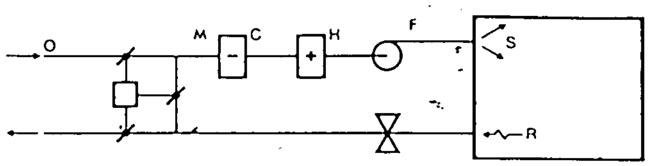
(*a*) For a constant volume flow rate to be maintained to each zone the system must be capable of supplying air at various temperatures at any one time; this may involve simultaneous heating and cooling of supply air.

(*b*) All rooms with similar solar gain patterns can be zoned together provided that other variables are in phase. However, the number and position of the zonal sensors will be important. Corner rooms pose further problems.

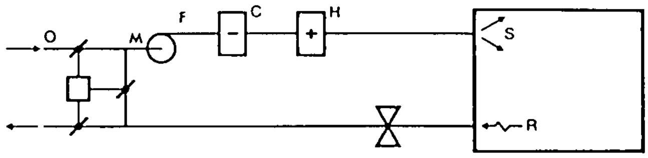
(c) North facing rooms experience less variation and can be grouped with internal zones for cooling provided that heating is dealt with by other means.

(d) Gains through poorly insulated roofs are similar to gains on south facing surfaces but if adequately insulated, they may be treated as intermediate floors.

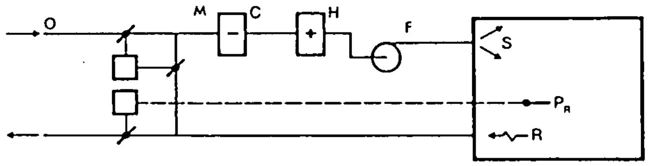
The success of an air conditioning system depends largely on wise zoning and careful positioning of sensors in relation to the sources of heat gains.




(a) "Draw-through" arrangement




(b) "Blow-through" arrangement




(c) Combined supply/extract fan

**Fig. B3.8.** Supply air handling point – alternative arrangements for position of fans.

## SYSTEMS DATA SHEETS

### Nomenclature

For the purposes of the system schematics, load charts and plant operation charts discussed later, the following nomenclature is adopted.

*System Schematics*

Sensors are identified as follows:

| | |
|---|---|
| E | enthalpy |
| H | humidity |
| P | pressure |
| Q | volume flow rate |
| T | dry bulb temperature |
| V | air velocity |

Sensors are differentiated by the following subscripts:

| | |
|---|---|
| C | air leaving cooling coil |
| H | air leaving heater battery |
| LL | low limit |
| M | mixed air |
| O | outside air |
| R | room air |
| S | supply air |
| W | water |

For multiple zone systems the orientation of the zone (NSEW) is indicated by an additional subscript. Humidity sensors are identified as "high" or "low" by the subscript H or L, respectively.

Single capital letters are used to relate locations within the systems to corresponding points on the psychrometric charts.

*Load and Plant Operation Charts*

The following symbols are used:

| | | |
|---|---|---|
| $A$ | = area of fabric element | $m^2$ |
| $U$ | = thermal transmittance coefficient | $W/m^2$ K |
| $g$ | = moisture content | kg/kg |
| $h$ | = enthalpy of air | kJ/kg |
| $t$ | = dry-bulb temperature | °C |
| $t_{dp}$ | = dew-point temperature equivalent to given moisture content | °C |
| $t'$ | = wet-bulb temperature | °C |
| $D_t$ | = difference between inside air and outside air temperatures | K |
| MAX LOAD: | estimated maximum sensible heat load for zone having orientation indicated | kW |
| MIN LOAD: | estimated minimum sensible heat load during normal usage | kW |

Temperatures and/or humidities are differentiated by the subscripts given for sensors, plus the following:

| | |
|---|---|
| Smax | warmest supply air to deal with MIN LOAD |
| Smin | coolest supply air to deal with MAX LOAD |
| ADP | apparatus dew-point temperature |
| MST | mean surface temperature |
| n % | room percentage saturation |

**Load and Plant Operating Charts**

Many air conditioning systems fail to give satisfactory service because their operation at conditions other than design extremes has not been considered.

Although various computer programs now exist[28] which enable overall energy consumption to be modelled, these may not be available to all designers. Therefore, it is useful to have an easily constructed visual representation of:

(*a*) load variation with outside temperature (load chart)

(*b*) the possible variations in temperatures and humidities throughout the system (plant operation chart).

A procedure for constructing load and plant operation charts, applied to an example building having a particular load profile, is given in Appendix 1. Load and plant operation charts for various systems, based on the example building used in the Appendix, form part of the following Systems Data Sheets.

On analysis[29], see Appendix 1, three main strategies emerge from the plant operation chart:

(*a*) a resetting schedule for the off-coil sensor ($T_C$)

(*b*) the likely mixing damper operation

(*c*) the likely boiler and refrigeration plant operation.

These strategies are shown at the foot of each plant operation chart. However, it should be noted that these are based on the specific load profile appropriate to the example building discussed in Appendix 1.

The Systems Data Sheets enable the designer to identify the dominant characteristics of the most common types of mechanical ventilation and air conditioning systems. The control schematics and plant operation charts are based on a building with perimeter zones only (see Appendix 1). Buildings with significant areas in which external influences have minimal effect would be treated differently and the resetting schedules given for the off-coil sensor may not be appropriate.

Most of the systems described can utilise either dilution or displacement ventilation for room air diffusion and for removal of unwanted heat and contaminants.

The ability of supply air to offset sensible heat gains depends upon the volume flow rate and the temperature difference between the supply air and the room air. Most of the systems discussed below operate by controlling the supply air temperature and hence, the temperature difference. This includes all-air systems of the terminal reheat, dual duct and hot deck/cold deck types and all of the air/water and unitary types. Variable air volume (VAV) systems generally control the flow rate at the terminals, with the occasional use of variable supply temperature.

**Control Methods**

The Systems Data Sheets indicate the method of automatic control over the outputs from the heating and cooling coils and the position of the mixing damper in outline only. The locations and types of sensing element (temperature, humidity, enthalpy, air velocity, volume flow rate and pressure) are shown and related to the items of equipment, the output of which is being controlled.

The plant operation charts shown for all-air multiple zone systems are based on the following control assumptions:

(*a*) an off-coil sensor ($T_C$) is used to control coils and dampers in sequence, see Fig. B3.9

(*b*) enthalpy comparators are used to cycle mixing dampers between full and minimum fresh air positions when outside and room enthalpies are equal.

No indication is given regarding the method of control signal processing, i.e. pneumatic, electric or electronic. An outline of these systems is contained in Section B11 and a more detailed comparison is given elsewhere[27].

**Types of System**

(*a*) *All-air systems:* employ central plant and duct distribution to treat and move all the air supplied to the conditioned space, fine tuning of supply temperature or volume occuring at the terminals.

(*b*) *Air-water systems:* usually employ central plant to provide fresh air only, terminals being used to mix recirculated air with primary air and to provide fine tuning of room temperature.

(*c*) *Unitary systems:* small scale versions of single zone systems within packaged units.

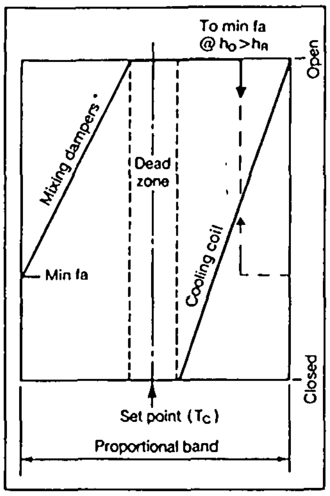



**Fig. B3.9.** Sequential control of mixing dampers and cooling coil.

Although systems which employ air as the main heat transfer medium between central plant and the space require large and costly air distribution systems they do offer the potential for “free cooling” from this air when outside conditions are appropriate. With air/water systems it is possible to utilise “free cooling” only indirectly, by passing cooling water from the coil in the terminal unit through the primary air cooling coil. However, this may be worthwhile only if the primary coil is sprayed.

It is difficult to achieve a good standard of filtration through small room units due to lack of space and poor fan pressure development. Induction units cannot normally filter recirculated room air due to the very high nozzle pressures required.

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## SYSTEMS DATA SHEET 1

### Single Zone Systems: Full Fresh Air

See Figs. B3.10 to 16. Refer to *Systems: General Notes.*

*Room Temperature and Humidity Control*

A temperature sensor $T_R$ controls the reheater (see Fig. B3.10), and a humidity sensor $H_{RH}$ provides proportional plus integral control of the cooling coil duty and hence supply air moisture content. Any overcooling required to limit humidity rise is detected by the room temperature sensor which operates the reheater as necessary.

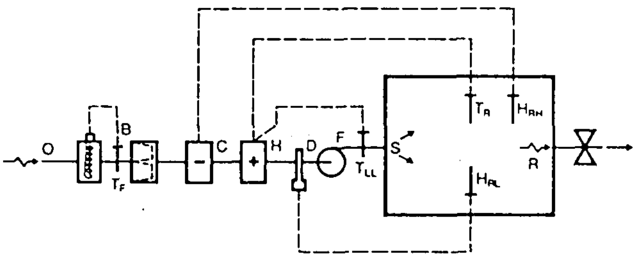



**Fig. B3.10.** System schematic: full fresh air with steam humidification; room temperature and humidity control.

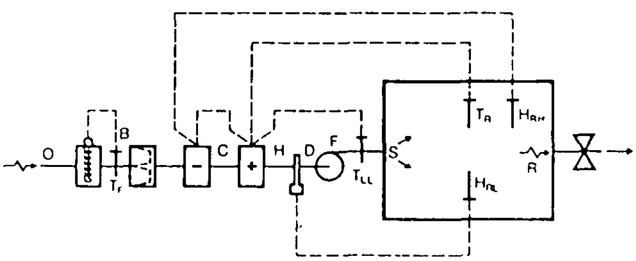



**Fig. B3.11.** System schematic full fresh air with steam humidification; sequence control with humidity override.

*Sequence Control with Humidity Override*

For the arrangements shown in Figs. B3.11 and 12 the temperature sensor $T_R$ controls the cooling coil and reheater in sequence within its proportional band (see Fig. B3.13). Psychrometric analysis under part load conditions will indicate whether room humidities are likely to rise above design limits: in which case a high humidity sensor $H_{RH}$ will bring in the cooling coil out of sequence and $T_R$ will call for simultaneous reheat to deal with overcooling.

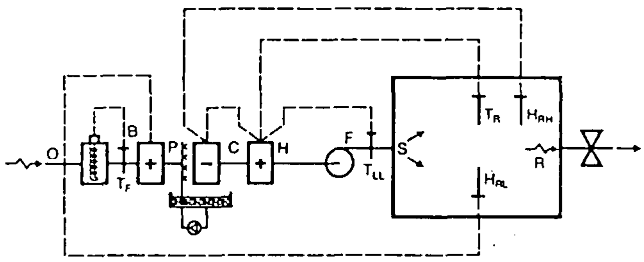



**Fig. B3.12.** System schematic: full fresh air with spray cooling coil; sequence control with humidity override.

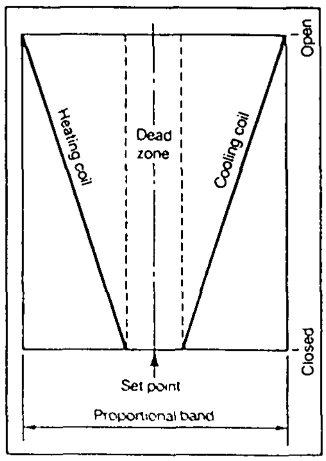



**Fig. B3.13.** Sequential control of heating and cooling coils.

*Sequence Control with Face-and-bypass Dampers*

In the scheme shown in Fig. B3.14, $T_R$ positions the face-and-bypass dampers in sequence with the heating coil to provide an appropriate supply condition rather than controlling cooling directly via the cooling coil.

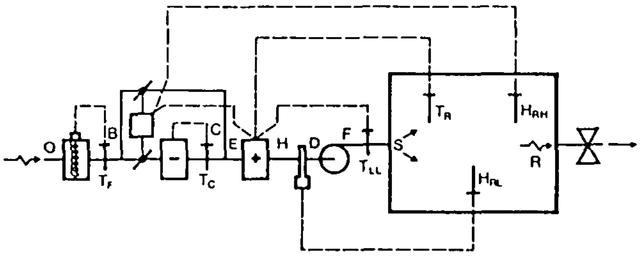


**Fig. B3.14.** System schematic: full fresh air with steam humidification, sequence control with face-and-bypass dampers.

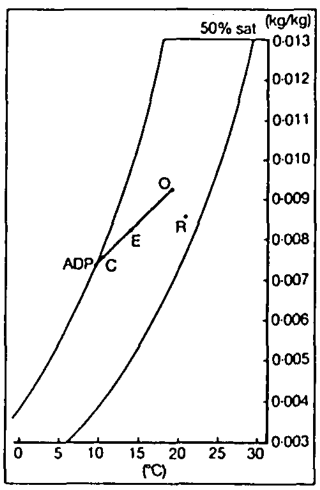


**Fig. B3.15.** Psychrometric chart: full fresh air with steam humidification; sequence control with face-and-bypass dampers.

When combined with appropriate cooling media temperatures (see Fig. B3.15) this method provides adequate humidity control without wasteful reheat. However, room humidities will rise, particularly at low sensible heat loads but protection against high humidities can be provided by using a humidity sensor $H_{RH}$ to override damper control, the reheater being brought in to deal with resultant overcooling.

The cooling coil can be installed without a control device, provided that chilled water temperatures are maintained at an appropriate level.

*Off-coil Control*

Fig. B3.16 shows an arrangement in which the off-coil dry-bulb temperature sensor $T_C$ controls the cooling coil and preheater in sequence to achieve its set point adjusted against the outside temperature sensor $T_O$ if appropriate. The room temperature sensor $T_R$ controls the output of the reheater to achieve the desired room temperature.

The preheater is optional, but if not present a low limit sensor should be provided to bring in the reheater to prevent cold draughts on start-up and during wide load variations.

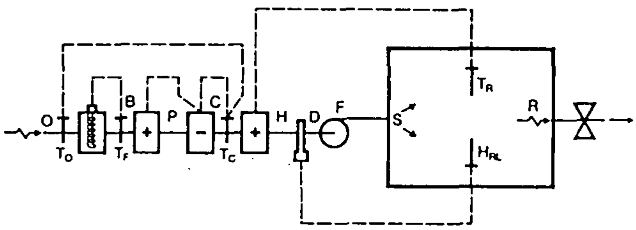


**Fig. B3.16.** System schematic: full fresh air with steam humidification; off-coil control.

## SYSTEMS DATA SHEET 2

### Single Zone Systems: Recirculation

See Figs. B3.17 to 26. Refer to *Systems: General Notes.*

*Sequence Control with Humidity Override*

In the arrangement shown in Fig. B3.17 the temperature sensor $T_R$ controls the cooling coil, mixing dampers and reheater in sequence within its proportional band (see Fig. B3.18). Other than the additional control over the mixing dampers which can be provided, control is similar to that for a full fresh air system.

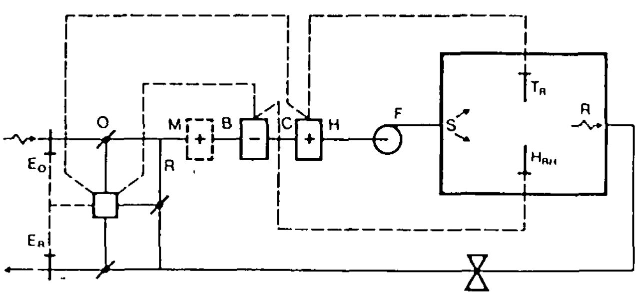
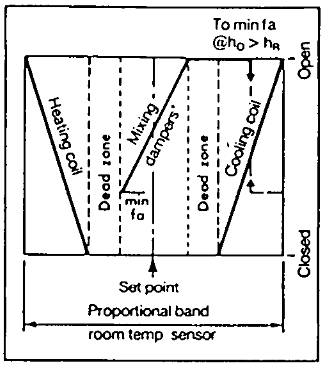


**Fig. B3.17.** System schematic: recirculation; sequence control with humidity override.




For a full fresh air system with heat recovery, the mixing dampers are replaced by the heat recovery device in the above sequence.

**Fig. B3.18.** Sequential control for recirculation system.

Referring to the plant operation chart, Fig. B3.19, the lines A to F and G to L represent the likely ranges of dry-bulb and dew-point temperatures for the air entering the plant. The shaded area represents the likely range of supply conditions. This should be compared with the plant operation chart for off-coil control, Fig. B3.25.

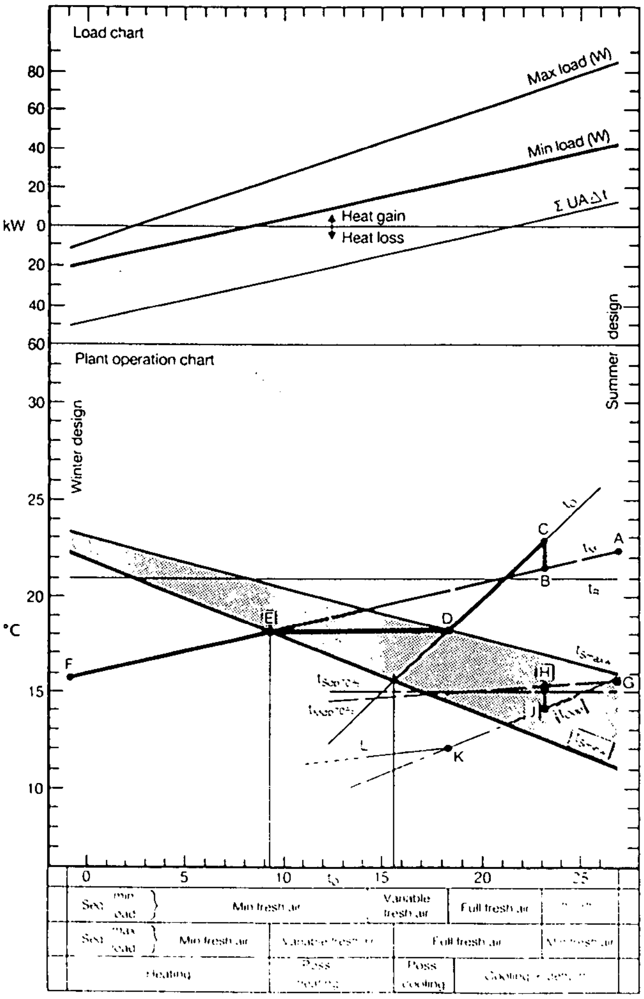



**Fig. B3.19.** Load/plant operation chart: recirculation, sequence control with humidity override.

*Sequence Control with Room-air Bypass*

In the scheme shown in Fig. B3.20, instead of directly controlling flow of the cooling medium through the cooling coil, an appropriate supply condition is provided by positioning the bypass and recirculation dampers in sequence with the heating coil in response to an appropriate signal from $T_R$. This gives closer control of room humidity than face-and-bypass dampers because controlled room air only is bypassed around the cooling coil.

A part-load analysis of mass flow and temperature balance can be used to determine the on- and off-coil conditions for the cooling coil and hence the resultant room percentage saturation (see Figs. B3.21(a) and (b)). Control is otherwise similar to face-and-bypass control.

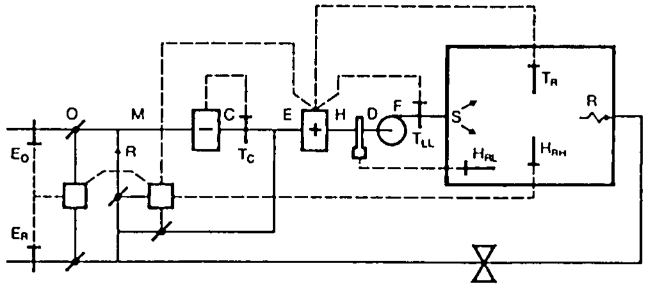
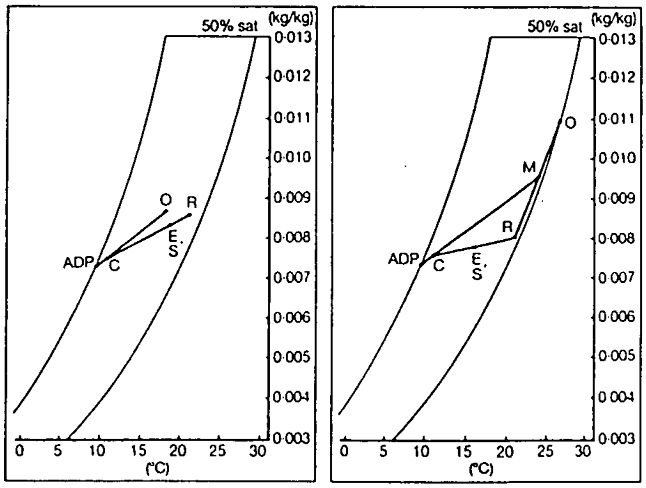



**Fig. B3.20.** System schematic: recirculation with steam humidification; sequence control with room air bypass.




(a) $t_o$ = 18°C (b) $t_o$ = 27°C

**Fig. B3.21.** Psychrometric chart: recirculation with steam humidification; sequence control with room air bypass.

*Off-coil Control*

In the arrangement shown in Fig. B3.22, the off-coil dry-bulb temperature sensor R controls the cooling coil and mixing dampers in sequence within its proportional band (Fig. B3.9).

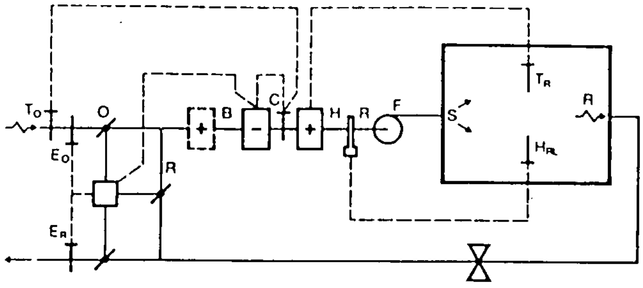


**Fig. B3.22.** System schematic: recirculation; off-coil control.

Further control over mixing damper position may be provided (see *Systems: General Notes*), otherwise control is similar to that for the full fresh air system.

Fig. B3.23 shows the plant operation chart for the south facing zone which indicates that significant amounts of reheat will only be required to deal with part load operation. (See Fig. B3.24 for typical psychrometric cycles.) This should be compared with Fig. B3.25 which shows the plant operation chart for a west facing zone and the resulting off-coil sensor resetting schedule for a room humidity range of 45 to 55% saturation. This indicates substantial simultaneous cooling and reheating.

The preheater (shown dotted) is incorporated into the sequence only if large fresh air rates promote high mixing ratios, and hence low winter temperatures, through the air handling plant. Alternatively, a low limit sensor could be used to bring in the heating coil as necessary (see *Systems: General Notes*). If adiabatic humidification is employed to deal with the associated low winter moisture contents, a preheater may be necessary to heat mixed air.

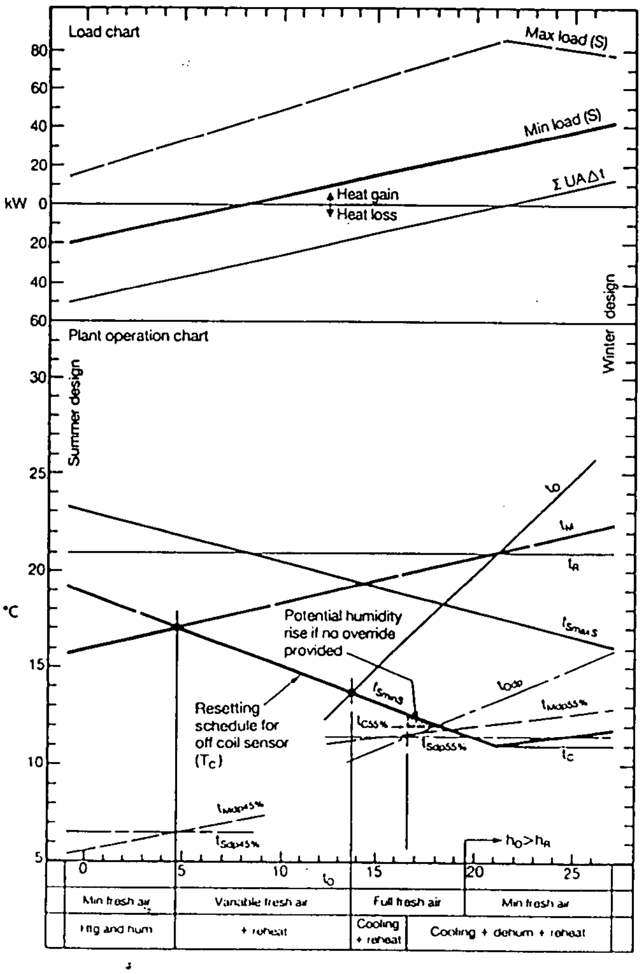
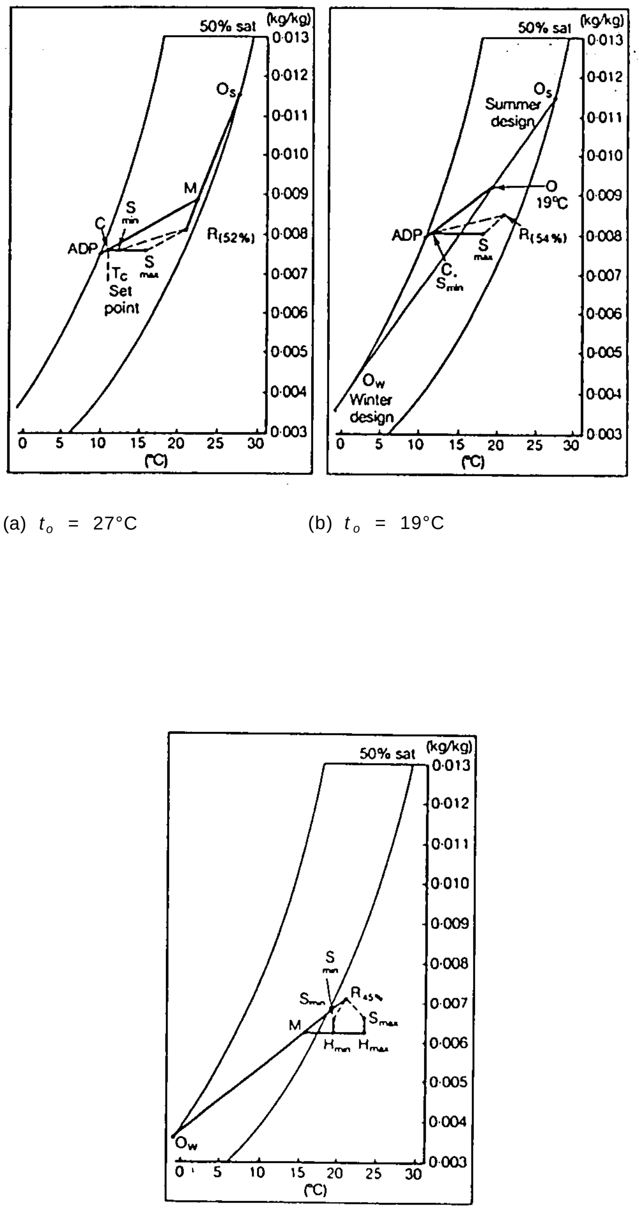


**Fig. B3.23.** Load/plant operation chart: recirculation, south facing; off-coil control.



(a) $t_o$ = 27°C (b) $t_o$ = 19°C

(c) $t_o$ = –1°C

**Fig. B3.24.** Psychrometric chart: recirculation, south facing, off-coil control.

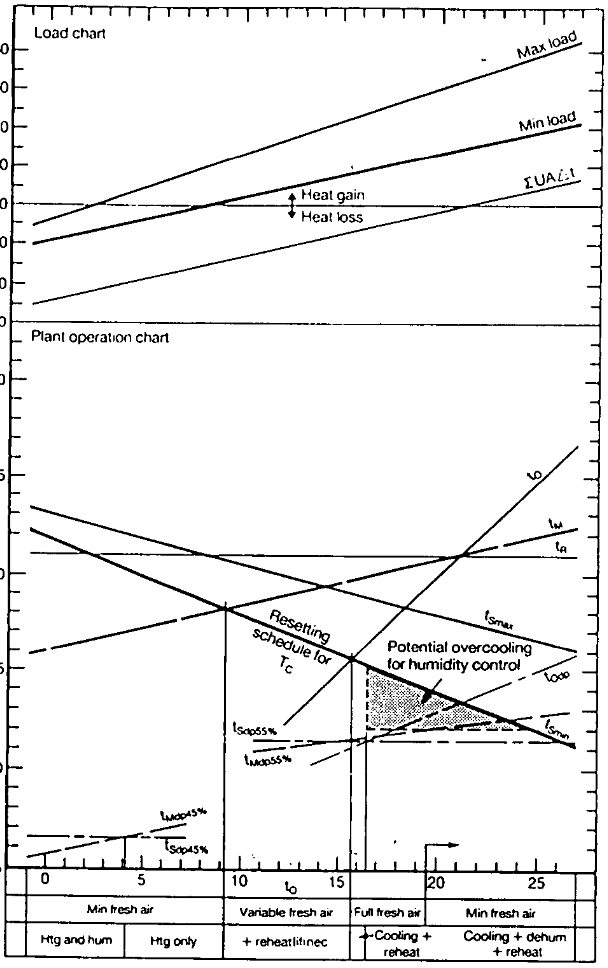
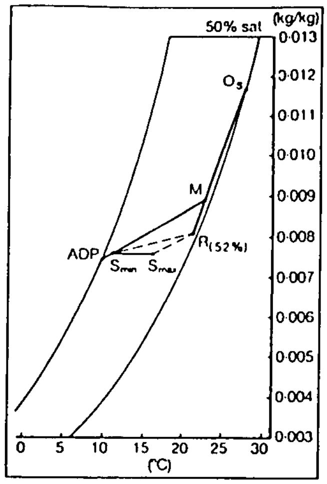
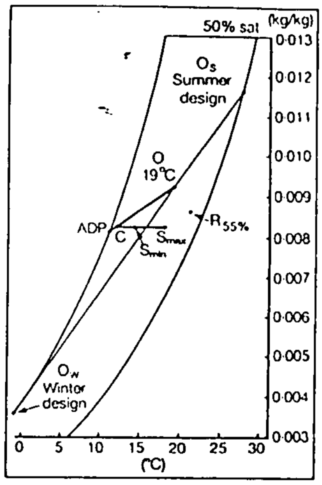


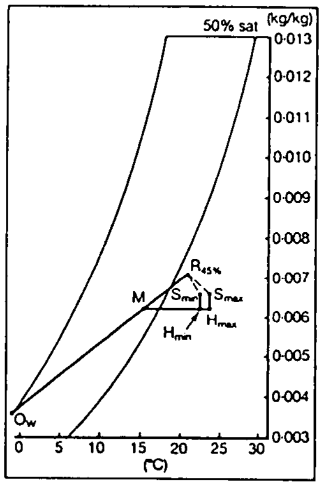

Fig. B3.25. Load/plant operation chart: recirculation, west facing; off-coil control.




(a) $t_o$ = 27°C




(b) $t_o$ = 19°C




(c) $t_o$ = –1°C

Fig. B3.26. Psychrometric chart: recirculation, west facing; off-coil control.

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## SYSTEMS DATA SHEET 3

### Multiple Zone Systems: All-air, Terminal Reheat

See Figs. B3.27 to 29. Refer to *Systems: General Notes.*

*General*

Air is treated centrally and distributed at a common temperature and moisture content, see Fig. B3.27. The temperature is sufficiently low to deal with the greatest sensible heat gain (or lowest net loss) and the moisture content is at a level which will satisfy the zone having the lowest sensible heat ratio and provide adequate fresh air to the zone having the highest mixing ratio of 1ocal fresh air to supply air.

Figs. B3.29(a) and (b) show that the resultant percentage saturations will be different for each zone ($R_F$ and $R_W$).

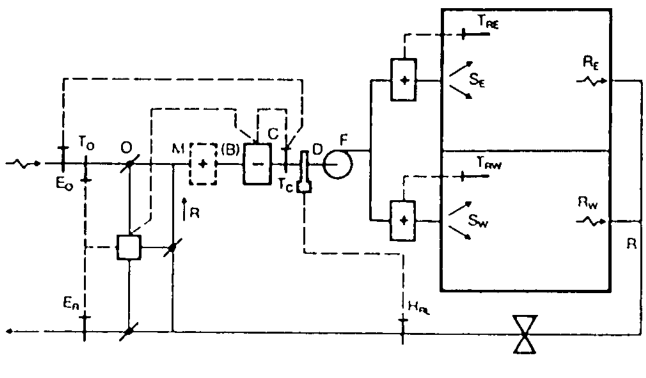



Fig. B3.27 System schematic: terminal reheat.

Therefore, extract air condition R is the result of mixing the two airstreams at their respective mass flow rates and moisture contents.

For any zone which experiences overcooling by the centrally treated air, the room temperature sensor $T_R$ brings in the respective zonal reheater.

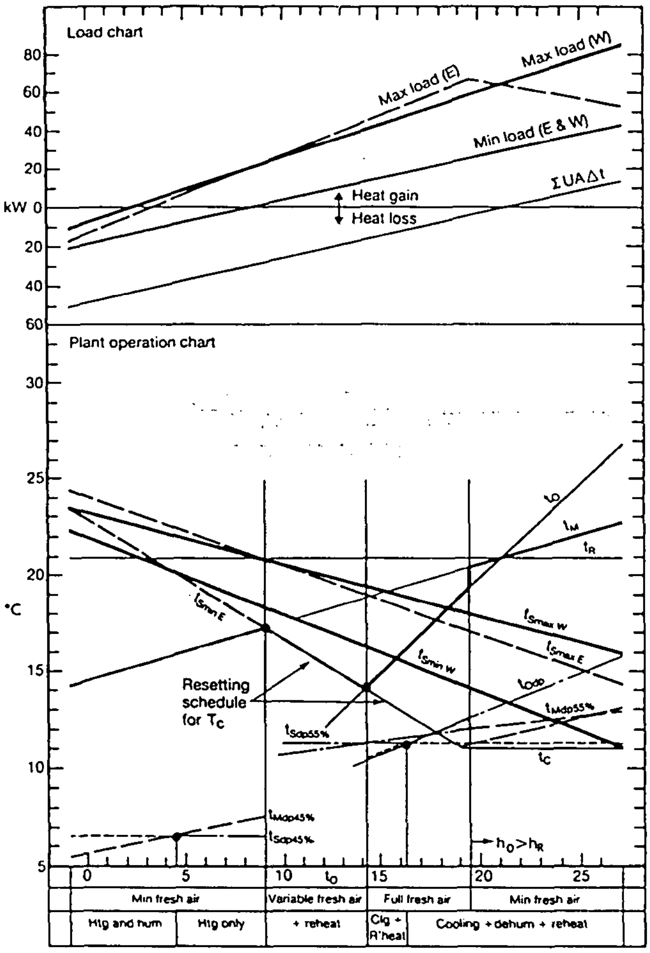
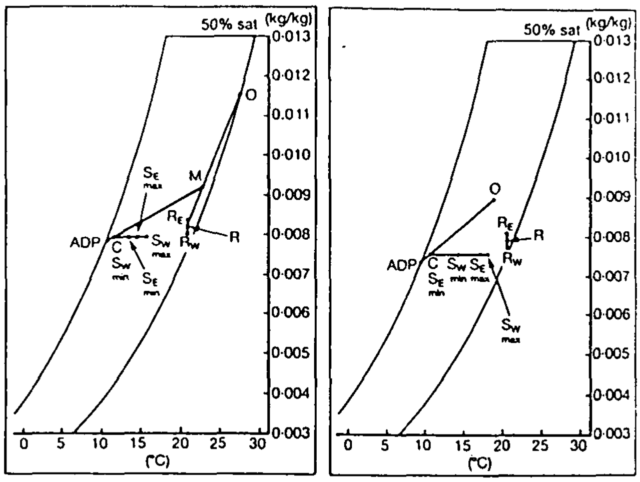



**Fig. B3.28.** Load/plant operation chart: terminal reheat.




(a) $t_o$ = 27°C (b) $t_o$ = 19°C

**Fig. B3.29.** Psychrometric chart: terminal reheat.

The condition of the distribution air can be varied with outside temperature (see Appendix 1) when the system is serving perimeter zones only (see Fig. B3.28). Internal zones are likely to experience peak cooling load even at low external temperatures, hence the air leaving the central plant must be kept at the minimum design condition.

Serving perimeter and internal zones from one plant can prove wasteful of energy unless humidity control necessitates low supply temperatures or it is possible to achieve low supply temperatures by utilising sources of "free cooling".

In order to reduce unnecessary reheat, control signals from the reheater control actuators can be analysed centrally, the resetting schedule for the off-coil sensor being based on the zone requiring the lowest supply air temperature, i.e. minimum reheat requirement.

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## SYSTEMS DATA SHEET 4

### Multiple Zone Systems: All-air, Variable Air Volume

See Figs. B3.30 to 35. Refer to *Systems: General Notes.*

*General*

The control of VAV systems is dealt with in detail in Section B11 and elsewhere[27].

*VAV: Constant Temperature*

If the volume of air supplied to a space is varied the following consequences result:

(*a*) its ability to offset sensible gains is reduced

(*b*) its ability lo offsite latent gains is reduced

(*c*) if mixing ratio remains constant, its ability to dilute odours, carbon dioxide etc. is reduced

(*d*) unless special air terminal devices are utilised, its ability to create room air movement is reduced.

The volume of supply air is normally varied in relation to room air temperature (sensors $T_{RF}$ and $T_{RW}$ in Fig. B3.30) and will respond only to changes in sensible gain. see Fig. B3.31. Hence, unless the main load variations are caused by occupancy changes, unacceptable humidity rise and depletion of fresh air can result. The effect on room air movement will depend largely on the turndown efficiency of the terminal device, see *Room Air Diffusion.*

Generally, humidity rises on turndown can be kept within acceptable limits provided that a cooling differential of about 8 to 12 K is used, see Fig. B3.32.

Fresh air rates on turndown can be maintained by means of on inlet velocity sensor to control the position of the mixing dampers.

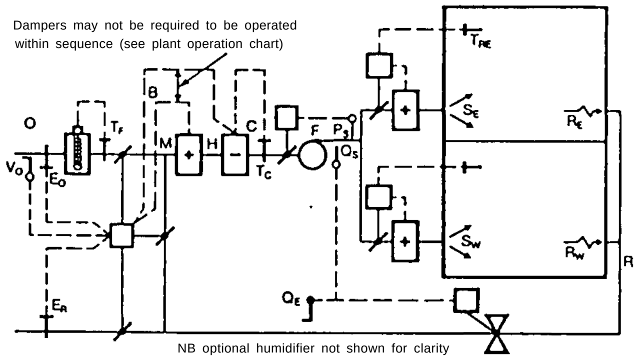


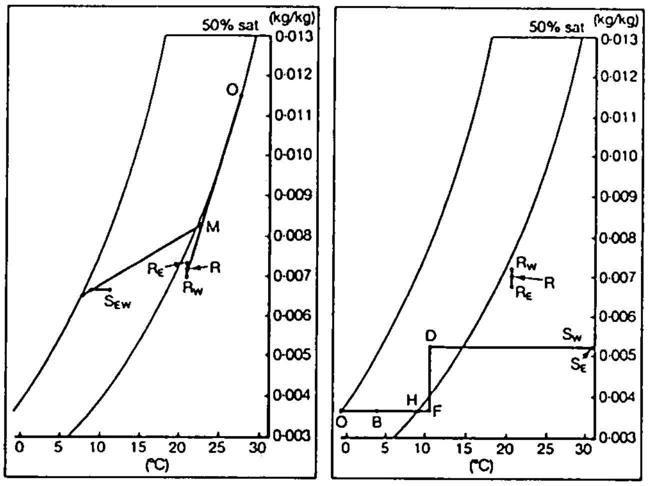

**Fig. B3.30.** System schematic: VAV with terminal reheat.

The efficiency with turndown depends on:

(*a*) the position selected for sensing flow changes

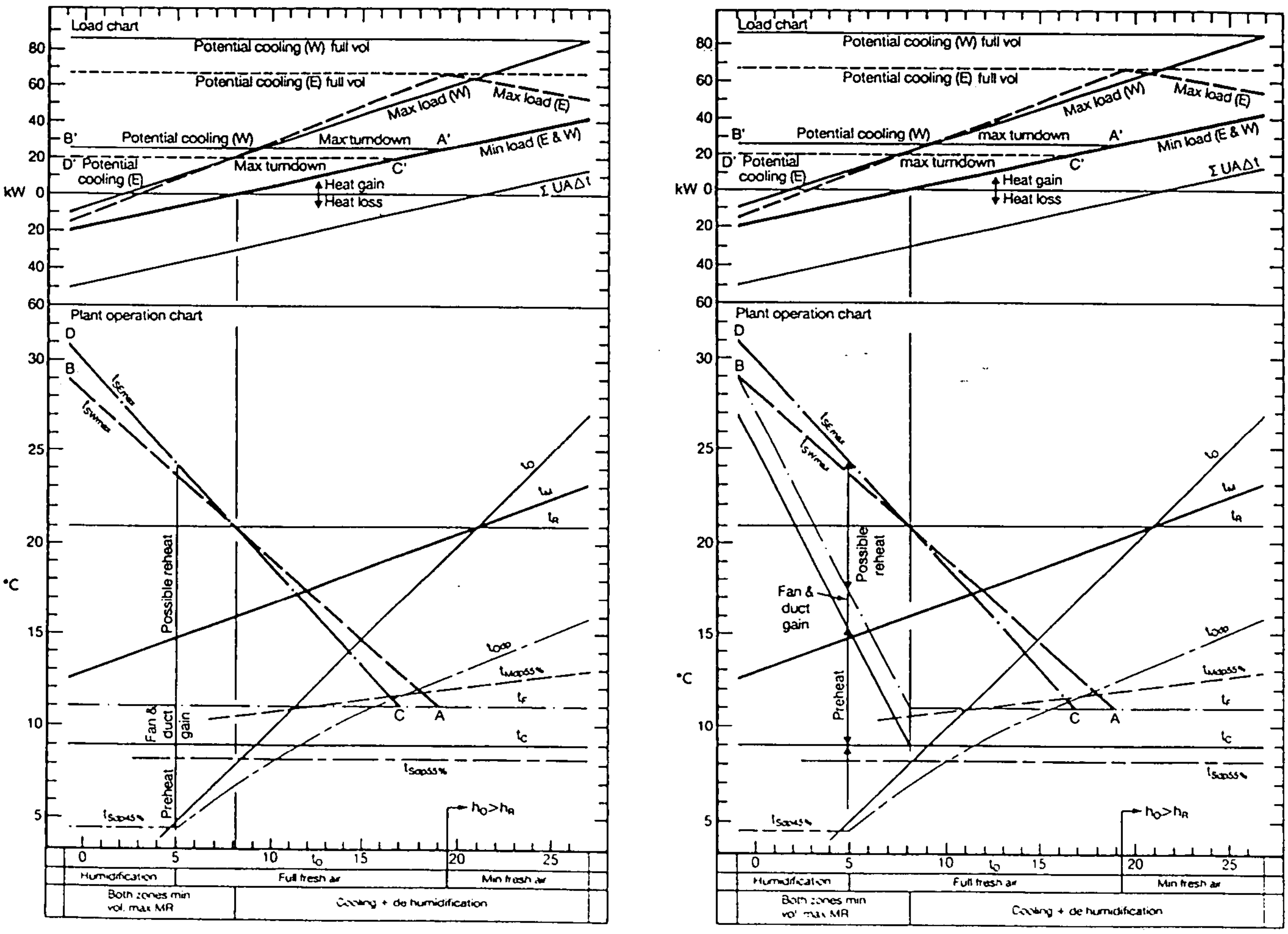
(*b*) the mechanism used for reducing total flow rate

(*c*) the mechanism by which flow dependant signals are converted to movement at the actuator.




(a) $t_o$ = 27°C (b) $t_o$ = −1°C (full turndown)

**Fig. B3.32.** Psychrometric chart: VAV with terminal reheat.




(a) Without resetting (b) With resetting at turndown

**Fig. B3.31.** Load/plant operation charts: VAV with terminal reheat.

If the supply fan duty is to be modulated from a static pressure sensor in the supply ductwork, the sensor must be in a position which gives a reasonable indication of total flow requirements. Medium to high duct velocities are needed to improve sensor sensitivity to flow changes[27,30,31].

The extract system must repond to changes in the supply flow rates to avoid over-pressurisation of the building. This may be dealt with at two levels:

(*a*) *Individual control zones:* if zones are separated by solid partitions any imbalance in supply flow rates between zones must produce corresponding changes in extract flow rates. Thus the extract duct for each zone will contain a damper controlled to follow changes in supply volume. In the case of a multi-storey open plan building, this may be necessary on a floor by floor basis.

(*b*) *Choice of fan characteristics:* the supply and extract fans will usually be of different types to cope with dissimilar system pressure requirements. Hence, their characteristics will differ accordingly.

For perimeter zones where minimum loads fall below the potential cooling at full turndown, some means of heating will be necessary to avoid overcooling.

If a step change in load from net cooling to net heating occurs in all zones simultaneously, a changeover coil in the central plant may be used. This system then provides a constant temperature heated supply or a constant temperature cooled supply.

A computer controlled changeover system is available for small applications where zonal load changes during spring and autumn do not result in prolonged periods of simultaneous heat gain and loss. Following analysis of the sensor information, either cooled or heated air will be provided from central plant depending on whether the majority of zones require cooling or heating. The system will cycle between heating and cooling as necessary and the thermal inertia of the building limits hunting.

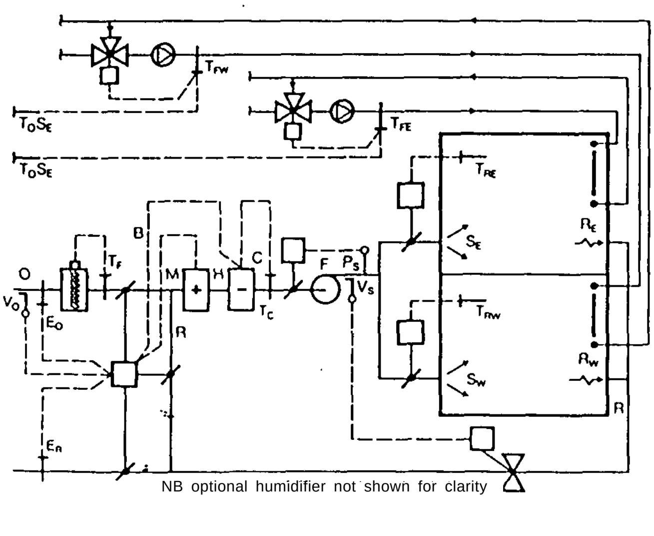



**Fig. B3.33.** System schematic: VAV with perimeter heating.

Alternatively, it may be possible to reset the set point of the off-coil sensor in the manner of a variable temperature system. This has the advantage of expanding the range of loads which the system can accommodate and eliminating some of the disadvantages of turndown. However, fan running costs increase because of the reduced turndown over the whole year.

*VAV: Terminal Reheat*

In Fig. B3.31, the lines AB and CD represent the supply temperatures required to deal with the minimum loads on the west and east zones respectively. The regions bounded by these lines and the line representing the temperature at the reheaters ($t_F$) indicate the maximum likely reheat required for each zone. The reheat coil and VAV dampers are controlled in sequence from a room temperature sensor.

*VAV: Perimeter Heating*

If significant perimeter downdraught is likely, under-window heating may be desirable, see Fig. B3.33. The output of the heating system must be controlled in such a way as to prevent the heat appearing as a cooling load. Therefore, water temperature must be scheduled against outside air temperature and solar compensated for different orientations, if appropriate. The resetting schedule is shown in Fig. B3.34 and is based on providing sufficient heating to deal with greatest potential cooling at maximum turndown (A′ B′ in Fig. B3.31). An extension of this principle is to utilise a VAV system for internal zones and a variable temperature air conditioning system to deal with perimeter loads.

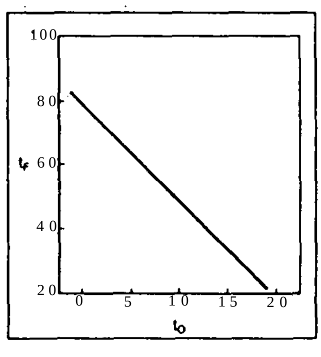



**Fig. B3.34.** Resetting schedule for flow temperature sensors against outside air temperature – VAV with perimeter heating.

*Induction VAV*

(*a*) *Air terminal induction:* a separate constant volume primary air duct or system is used to encourage constant throw from supply air terminals. A separate source of primary air can be used to provide a constant fresh air supply, scheduled against outside air temperature as appropriate. Primary air is discharged at a constant volume through induction nozzles or slots at the variable volume supply outlet which may be in the form of a sidewall grille, ceiling diffuser or induction nozzle.

(*b*) *Ceiling plenum induction:* a unit which mixes cooled air from central plant with air from the ceiling void which has been heated through exhaust luminaires. This results in better control and improved fan economy whilst room air movement is greater than that obtained from conventional throttling devices.

*Fan-assisted VAV*

In principle, this system is similar to the ceiling plenum induction system but uses a fan within each terminal to enhance room air movement on turndown and blend warm air from the ceiling void with that from the central plant.

There are two arrangements in common use whereby the fan and VAV damper are connected either in parallel (see Fig. B3.3S(e)) or in series (Fig. B3.35(f)). The parallel arrangement requires the fan and damper to be controlled in sequence, the fan being brought in only on full turndown. With the series configuration the fan runs continuously, thus maintaining constant room air movement (and noise generation) with varying proportions of air drawn from the ceiling void. A reheat coil can be incorporated into the device if insufficient heat is available from the luminaires.

*Dual Duct VAV*

The cold duct functions in the same manner as the basic VAV system providing the facility, at full volume flow, to deal with maximum cooling load for each zone. The hot duct connection is kept closed until the cooling VAV damper reaches its minimum setting. Any further reduction in cooling loads is dealt with by opening the hot duct damper. Hot duct temperature may be programmed against outside air temperature as appropriate. See also System Data Sheet 5, *Dual Duct Systems.*

*VAV Devices*

The device for varying airflow to a control zone may either form a part of the air distribution system and serve a number of conventional air terminal devices or it may be the air terminal device itself. In the latter case it may incorporate some means of maintaining reasonably constant throw. Fig. B3.35 shows examples of these devices.

Flow rate may be modulated by:

(*a*) throttling by dampers

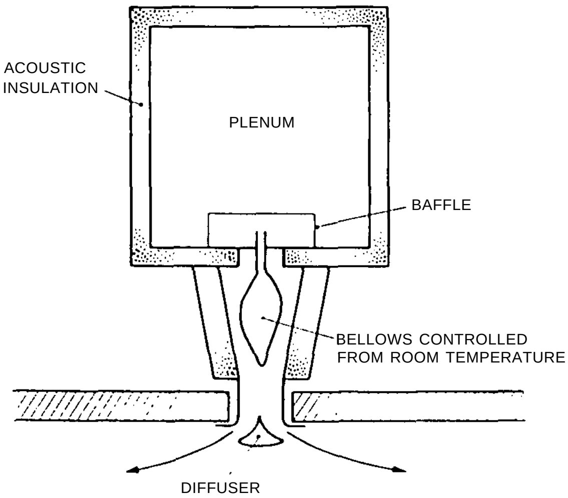
(*b*) throttling by variable area

(*c*) mechanical bypass by diverting supply air back to the air handler (constant volume fan).

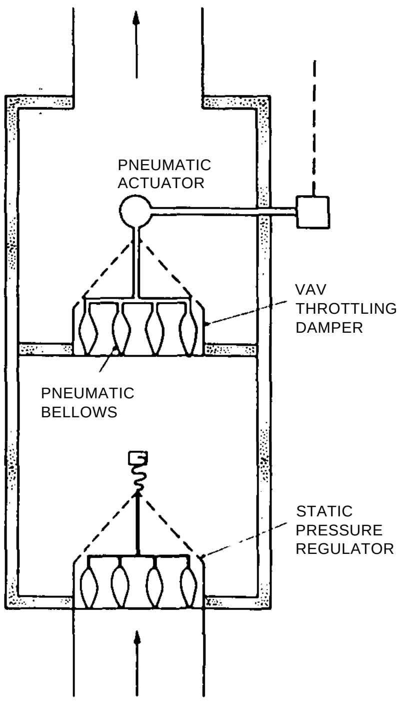
Control of the device can be achieved by the system being operated, utilising the pressure available in the supply duct, or by the use of an external power source, either electric or pneumatic.

The device may also incorporate some means of maintaining system balance under varying flow conditions, normally by automatic damper adjustment from a static pressure sensor. Alternatively, this function may be fulfilled by a separate damper box.

VAV devices may be actuated mechanically, by means of a spring loaded regulator which closes as pressure increases, or pressure-actuated using the changes in branch pressure to position a throttling damper. Both types increase fan pressure requirement by 100 to 200 Pa.

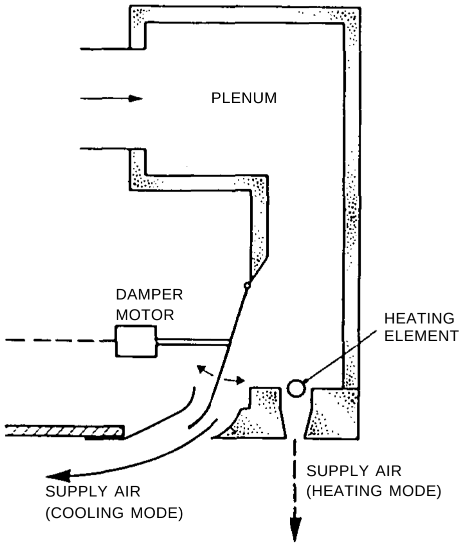



(a) Throttling air terminal device (variable velocity)

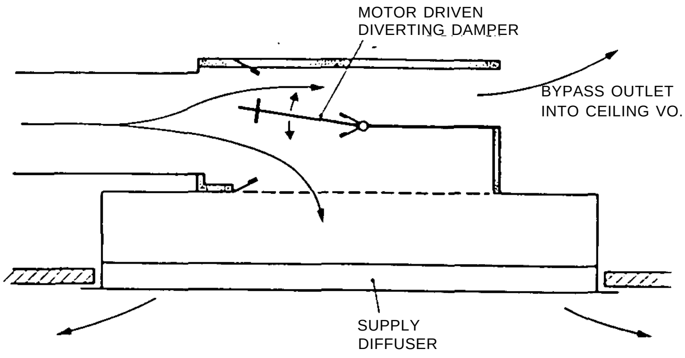



(b) Throttling control unit with static pressure regulator (mechanical)

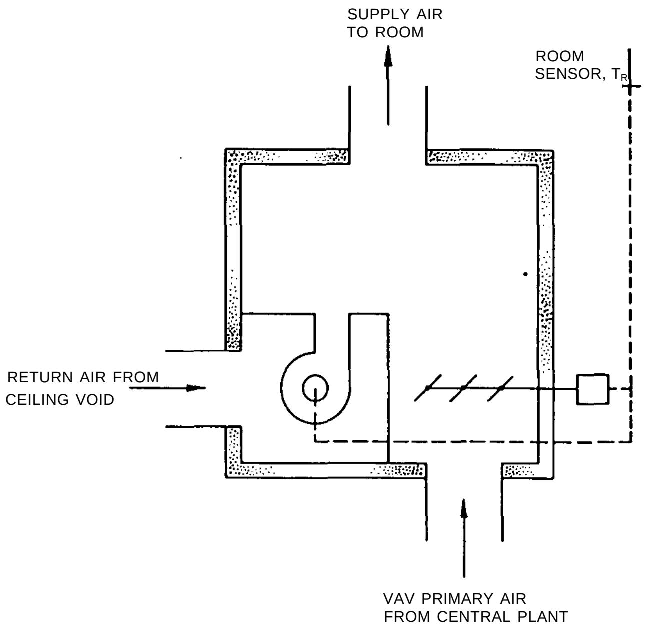
**Fig. B3.35.** Variable air volume devices (*continued overleaf*).



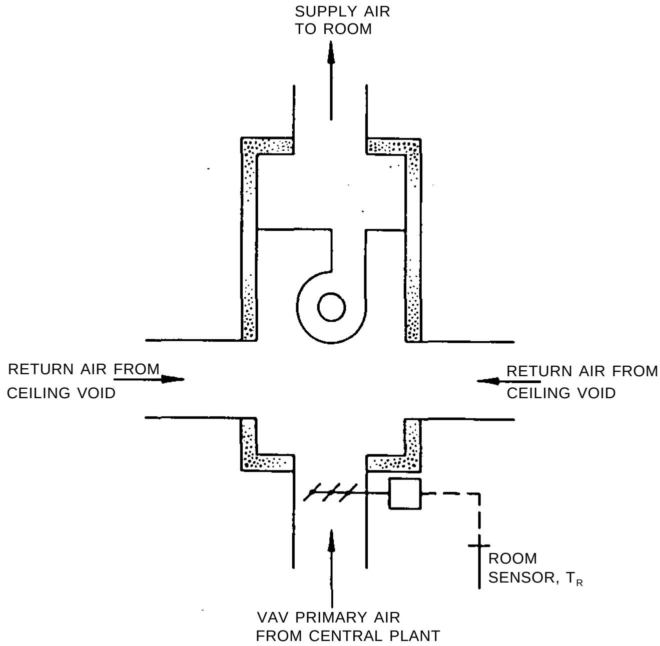

(c) Variable area air terminal device with reheat



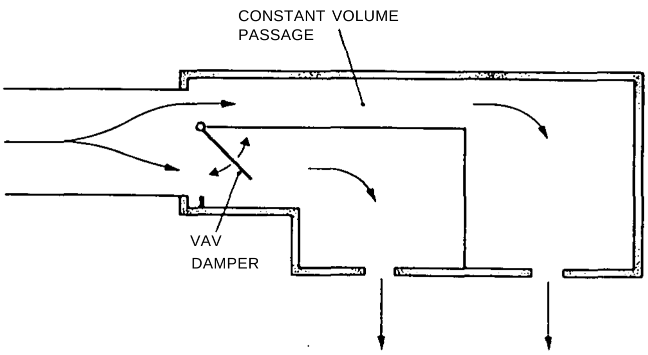

(d) Mechanical bypass air terminal device (slave diffusers possible)



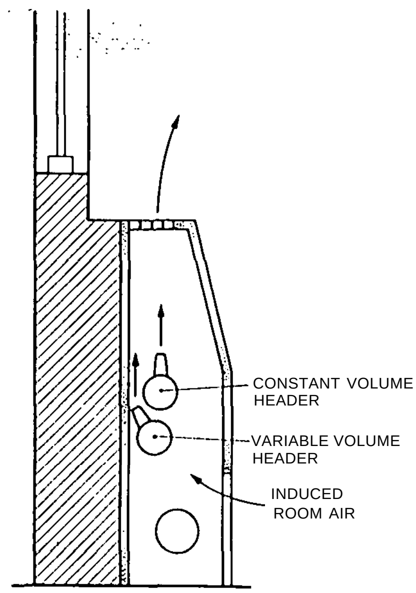

(e) Fan assisted control unit – parallel arrangement




(f) Fan assisted control unit – series arrangement




(g) Induction air terminal device using slots




(h) Induction air terminal device using nozzles – located under window

**Fig. B3.35**–*continued.* Variable air volume devices.

## SYSTEMS DATA SHEET 5

### Multiple Zone Systems: All-air, Dual Duct

See Figs. B3.36 to 40. Refer to *Systems: General Notes.*

*Basic System*

This system employs a two duct distribution system circulating separately cooled and heated air to zonal mixing boxes. Zonal temperature sensors ensure that air from the hot and cold ducts are mixed in appropriate proportions to deal with the prevailing load. The cold duct temperature is maintained constant if internal gains are constant or scheduled with outside temperature if perimeter loads only are to be dealt with. However, allowance must be made for the reduction in latent cooling due to mixing at part load.

The hot duct temperature must be scheduled to provide air at an appropriate temperature to deal with the minimum load when blended in equal proportions with air from the cold duct.

This system combines rapid response time with the ability to deal with heating and cooling loads simultaneously. Room air movement is constant and wet services above ceilings are avoided.

Central plant and distribution systems tend to be larger and more costly than other systems, despite the practice of sizing ductwork for high velocities.

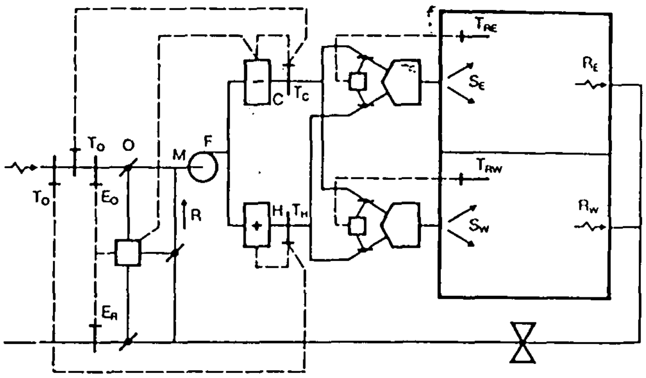
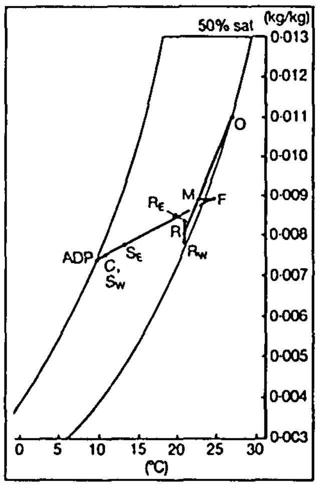


**Fig. B3.36.** Systems schematic: dual duct.




$t_o$ = 27°C

**Fig. B3.38.** Psychrometric chart: dual duct, hot deck/cold deck.

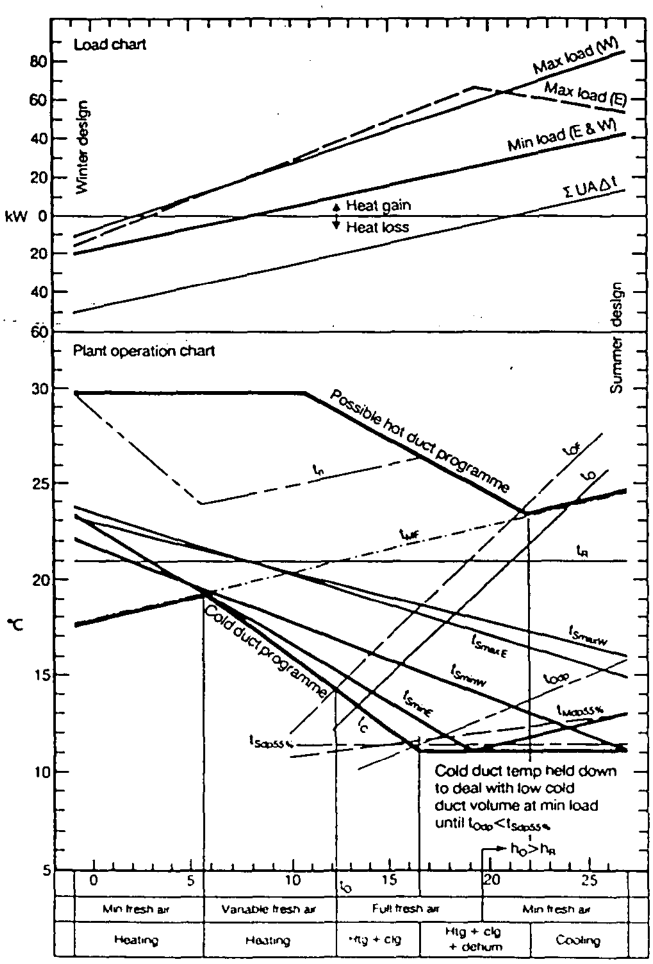
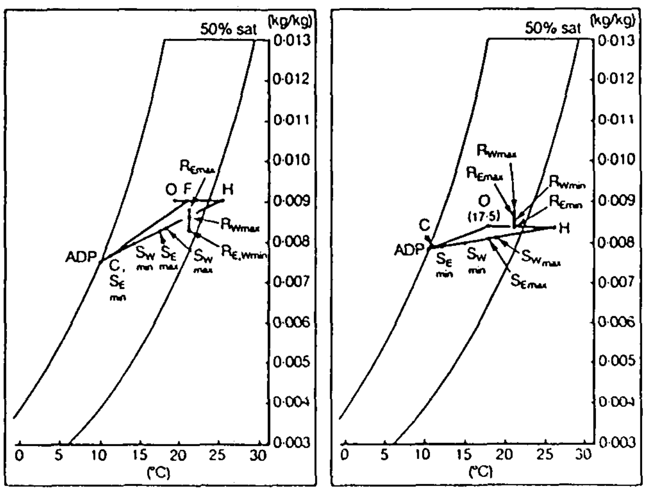



**Fig. B3.37.** Load/plant operation chart: dual duct.




(a) $t_o$ = 19°C (b) $t_o$ = 17·5°C

**Fig. B3.39.** Psychrometric chart: dual duct, hot deck/cold deck.

*Alternative Arrangements*

Dual duct systems can incorporate additional features to deal with specific requirements:

(*a*) *Fresh air preheat:* a preheater can be incorporated into the fresh air intake to deal with minimum fresh air quantities in winter. This avoids the possibility of freezing of the cooling coil due to stratification of fresh and return air through the mixing box and fan.

(*b*) *Fresh air dehumidification:* if the outside air is likely to be very humid at part load, a separate dehumidifying coil can be located in the fresh air inlet to avoid using very low temperatures at the main cooling coil.

(*c*) *Dual duct reheat:* the cooling coil is located within the central plant so that all the air is cooled and dehumidified, some being reheated in the hot duct, this providing better humidity control.

(*d*) *Dual duct/dual fan:* the provision of separate fans for the hot and cold ducts enables the hot duct to handle air recirculated through air handling luminaires. This assists with winter heating and increases cold duct volume and hence availability of dry air in summer. Sufficient fresh air must be assured for zones drawing minimum quantity from cold duct. A bypass between hot and cold ducts will ensure that fans handle constant volumes.

(*e*) *Dual duct VAV:* alternative arrangements incorporate single or dual supply fans, either with all fans being variable volume or with variable volume cold duct and constant volume hot duct. A cooling coil may also be incorporated into the constant volume system and hence provide the facility to serve some zones with constant volume variable temperature air, some with variable volume constant cooling, and others with a mixture[32]. See also Systems Data Sheet 4, *Dual Duct VAV.*

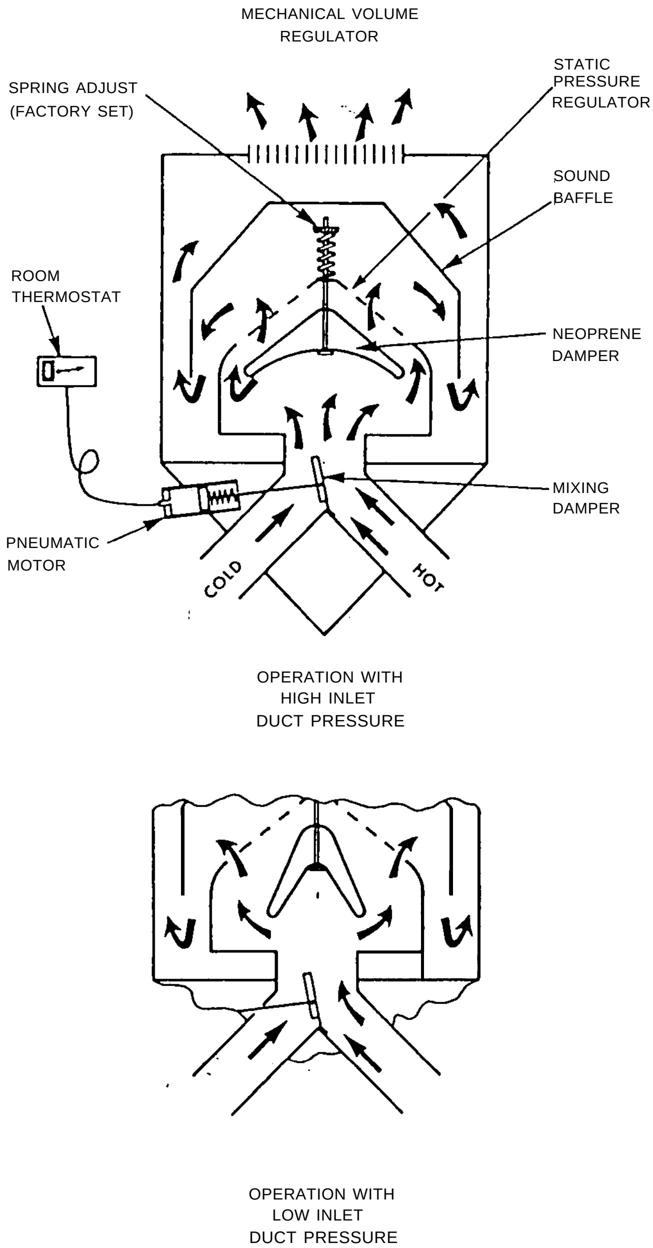



**Fig. B3.40.** Constant volume dual duct unit with integral static pressure regulator and air terminal device.

*Mixing Devices*

There are many types of mixing box using various methods of operation. Devices are available both in constant volume form and with sequenced cold duct VAV and mixing.

Basic functions usually performed include:

(*a*) mixing air from hot and cold ducts in appropriate proportions to match room load under the dictates of a room air temperature sensor

(*b*) mixing air thoroughly to avoid stratification

(*c*) attenuating noise generated at mixing dampers

(*d*) maintaining constant supply volume against variations in duct pressure (self balancing).

Fig. B3.40 shows one type of mixing device. Such devices may be individually controlled or several may be slaved from one master device as with VAV systems.

Leakage will always occur through “closed” dampers. Leakage rates vary from 0·03 to 0·07 of full flow rate for small, well-made devices up to 0·1 to 0·2 for larger and site-assembled units. This leakage represents an additional load on the cooling system under peak load conditions.

*Air Volume Stability*

Although the total volume flow handled by the fan remains constant each duct handles a variable volume, consequently the same problems of static pressure fluctuation occur as in VAV systems and require similar remedies at the terminals. Furthermore, with mixing devices operating under part load there is a risk of cross flow between the two ducts if significant imbalance exists between inlet pressures.

The following methods can be used to maintain system balance:

(*a*) Change in duct static pressure resets the set points of the sensors controlling hot and cold duct temperatures, hence maintaining constant flow rate in each duct; an unusual solution.

(*b*) Static pressure sensors in each duct cause the operation of dampers at the inlet to both hot and cold ducts; suitable for small systems only.

(*c*) Employ mixing devices with integral factory-set constant volume regulators; the most common solution. Most types are capable of maintaining a preset volume to within ±5% despite fluctuations of duct static pressure between 250 and 2000 Pa, if necessary. Factory-set volumes need to be checked after installation.

The two main types of static pressure regulator are:

*Mechanical:* a spring loaded regulator in the mixed airstream closes as the pressure increases, the mixing dampers operating as a single unit direct from a room sensor.

*Pressure actuated:* a room sensor operates the hot duct damper whilst the cold duct damper responds to resultant changes in flow sensed by a static pressure differential sensor across an integral resistance.

*Dual Duct VAV Terminals*

These operate in the same manner as VAV terminals down to minimum volume with the room sensor operating the cold duct VAV regulator and mixing dampers in sequence. Static pressure regulation requirements are as described above. The cold duct fan should be regulated under the dictates of a static pressure sensor, in a similar manner to that of a conventional VAV system.

*Stratification*

This can occur if there is inadequate mixing after the terminal and is a particular problem if a multiple outlet mixing device is installed with its outlets stacked vertically.

*Noise*

Noise regeneration at the unit is normally reduced by suitable lining materials and internal baffles. Larger terminals may require separate attenuation.

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**SYSTEMS DATA SHEET 6**

**Multiple Zone Systems: All-air, Hot Deck/Cold Deck**

See Figs. B3.41 and 42. Refer to *Systems: General Notes.*

*General*

In principle, the system shown in Fig. B3.41 is similar to the basic dual duct system, the major difference being that zonal mixing occurs at the discharge from the central air handling plant. Hence each zone requires a separate supply from the central plant. This reduces the problems of branch imbalance on damper turndown, hence low velocity distribution is possible, giving reduced fan running costs. However, problems can occur with interaction between separately controlled zones having very different volume flow requirements.

This arrangement is best suited to applications involving a small number of zones and where plant can be located centrally. It may also be successfully applied to noise sensitive spaces. Packaged "multizone" air handling units capable of serving a maximum of twelve zones are available (see Fig. B3.42) whilst site-constructed coil/damper arrangements may have as many zonal branches as can be physically incorporated.

Damper quality is an important factor in ensuring saisfactory part load control and economy of operation. A maximum leakage of 5% when closed should be specified. Precise control action is required in the transmission of the signal from room sensor through control system, actuators and damper linkages.

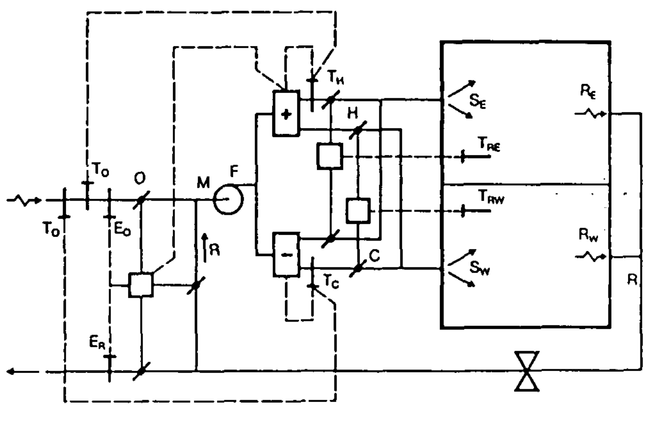
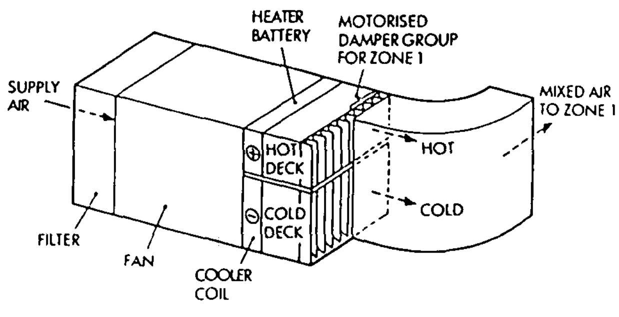



**Fig. B3.41.** System schematic: Multizone hot deck/cold deck.




Motorised mixing dampers operate in opposition, thermostatically controlled for each zone

**Fig. B3.42.** Typical packaged multizone arrangement.

## SYSTEMS DATA SHEET 7

### Multiple Zone Systems: Air/Water, Induction

See Figs. B3.43 to 48. Refer to *Systems: General Notes* and Systems Data Sheet 1, *Single Zone Systems: Full Fresh Air.*

*General*

Induction units use the power in a high velocity primary jet to induce room air to flow over a coil and hence promote adequate air circulation within the conditioned space.

Primary air plant and distribution ductwork are much smaller than for an all-air system since air/water units usually handle only the fresh air requirement of the occupants. However, the units may intrude on available floor space.

Induction unit coils are rarely used to dehumidify room air due to the inconvenience of condensate disposal. Therefore, all latent loads must be dealt with by primary air. Secondary water temperatures must be well above the maximum likely dew-point temperature of the room air, hence it is common practice to pump the return water from the primary cooling coil via a constant temperature mixing circuit (see Fig. B3.46). "Free cooling" may also be available if the refrigeration plant is bypassed when outside temperatures are low, i.e. water circulated through the primary coil is chilled by outside air before distribution to the room coils (see Fig. B3.47).

A number of alternative piping and control arrangements may be adopted, but the primary air system shown in Fig. B3.43 may be used for all options. In essence, this system is a single zone, full fresh air system with off-coil control of heating and cooling coils, including a steam humidifier, if required. Recirculation occurs at the room unit so that outside air only is filtered. Primary air volume must be adequate to:

(*a*) meet fresh air requirements of occupants

(*b*) provide adequate induction of room air to generate satisfactory room air movement

(*c*) provide sufficient sensible cooling at the zone without generating unacceptable levels of noise

(*d*) deal with dehumidification load at achievable chilled water temperature

(*e*) provide winter humidification, if necessary.

The various types of induction system are characterised by the method adopted for providing winter heating, as follows.

*Two-pipe Changeover*

Room coils can be supplied with either chilled or heated water via a common water circuit connected to boilers and refrigeration plant via three-port changeover valves. This method is appropriate only where the summer/winter transition is easily distinguishable.

*Two-pipe Non-changeover*

The distribution pipework carries chilled water only, each unit incorporating a coil which sensibly cools the room air. The output of each unit is varied by the action of a two- or three-port valve under the dictates of a local room thermostat (usually direct acting). Manual damper control over coil bypass is also possible. All heating is carried out in the primary plant, hence this system is only suitable where maximum heat loss can be accommodated without excessive supply air temperatures, usually limited to a maximum of 45°C.

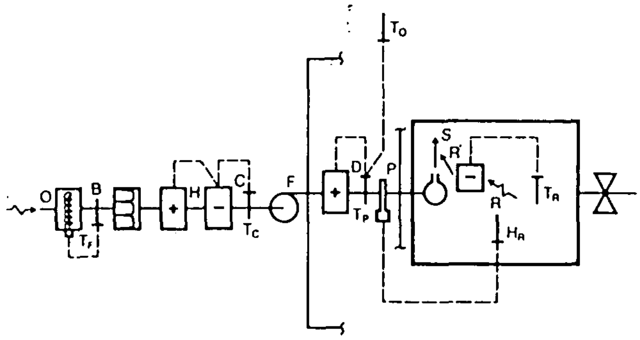



**Fig. B3.43.** System schematic: induction system, two-pipe non-changeover.

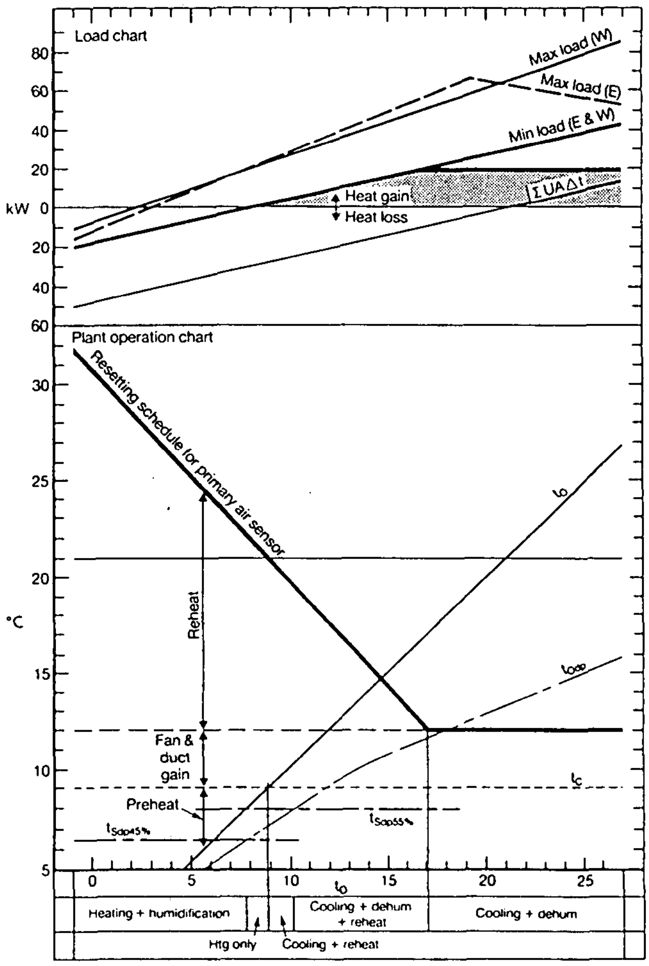


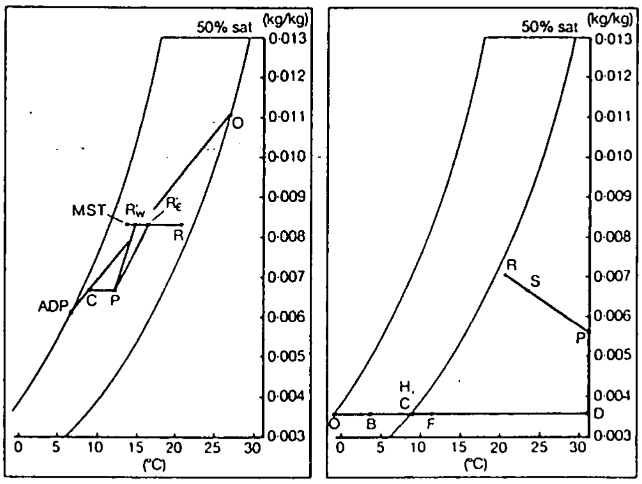

**Fig. B3.44.** Load/plant operating chart: induction system, two-pipe non-changeover.

The primary air temperature is scheduled to meet changes in load where these cannot be met by the induction coils (see Fig. B3.44). The shaded area of the load chart represents the portion of the load which can be met by room coils providing sensible cooling only. Primary air is scheduled to follow the MIN LOAD line, any additional gains being dealt with by the room coils. Alternatively, solar compensated reheat coils may be incorporated into primary air branches serving each orientation.

*Four-pipe*[27]

Each induction unit incorporates separate heating and cooling coils, operated in sequence by two- or three-port valves under the dictates of a room sensor. Four-port valves make efficient use of space, but transmission of heat from hot to chilled water may be a problem. Primary air must be distributed at a condition to meet humidity requirements when mixed with room air, but needs to provide only that portion of the heating load which cannot be dealt with by the room coils. Zonal reheating is unnecessary. This system offers greater flexibility than two-pipe systems but is more costly to install.

Where hot water is available for circulation through room coils, natural convective heating can be used for background heating with zero occupancy and the primary plant off. This may assist with early morning pre-heating, which will be costly in terms of energy consumption unless some provision for recirculation is made.




(a) $t_o$ = 27°C (b) $t_o$ = –1°C

**Fig. B3.45.** Psychrometric chart: induction system, two-pipe non-changeover.

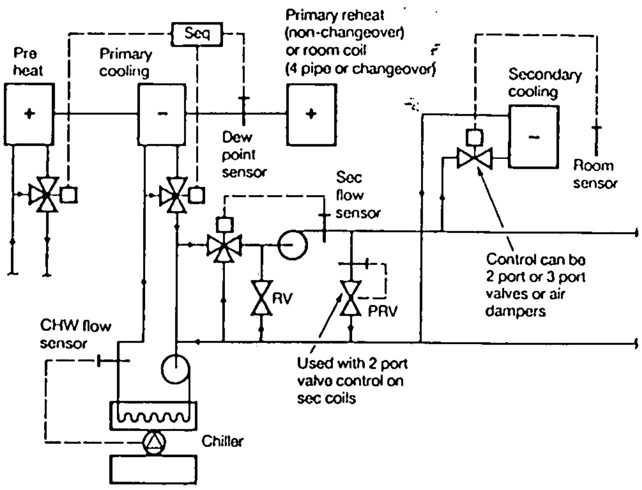



**Fig. B3.46.** Induction system: water control (dry room coils).

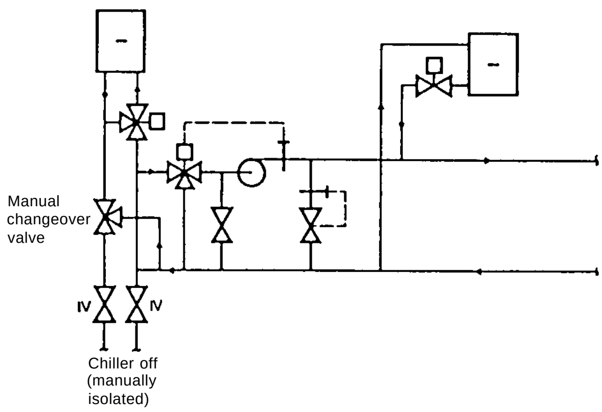


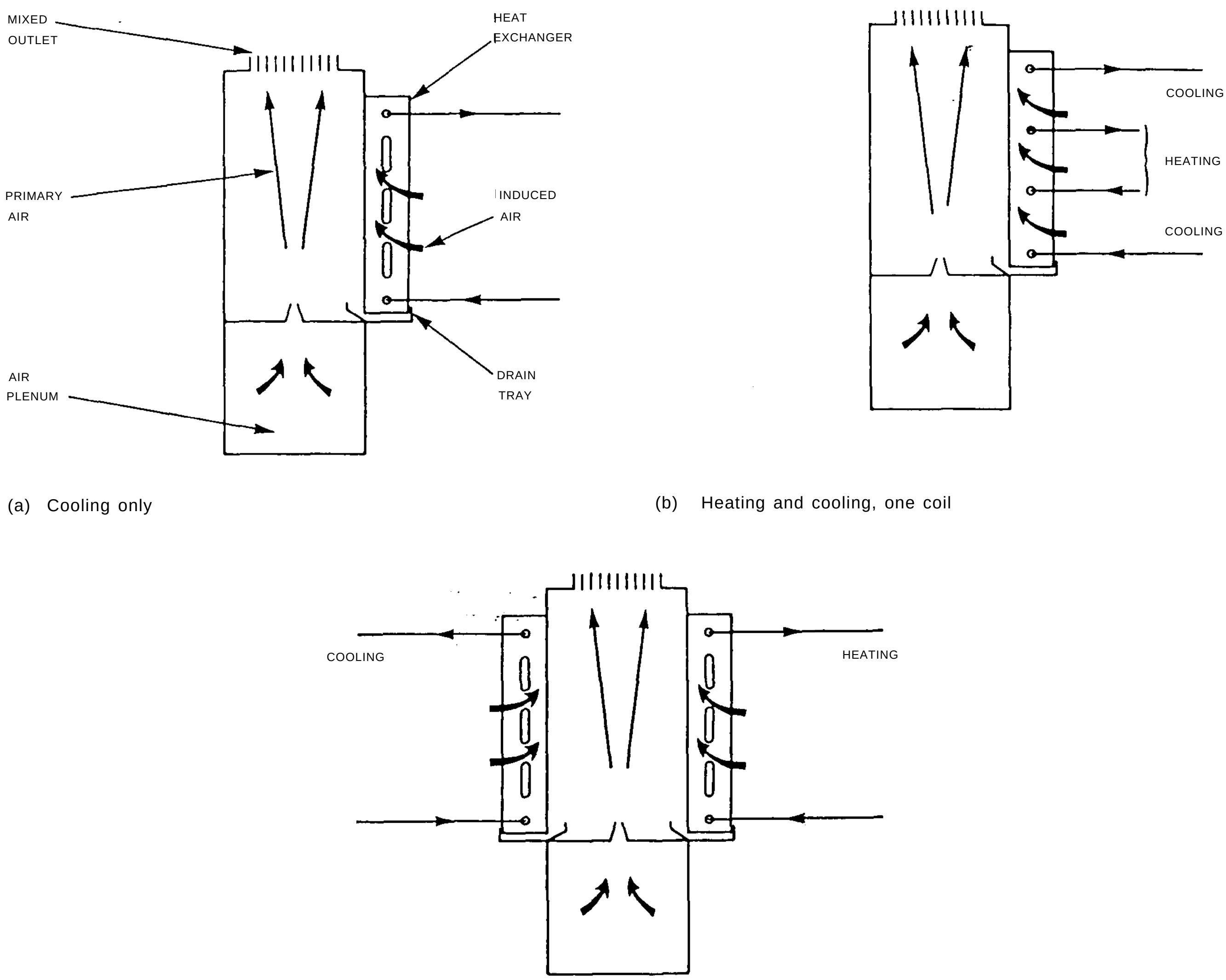

**Fig. B3.47.** Utilisation of "free cooling" in primary chilled water.

*Induction Units*

Units may be fitted with a single heat exchanger supplied with either heating or cooling medium as required (see Fig. B3.48(a)). Alternatively, separate circuits may be incorporated into a single coil (Fig. B3.48(b)) or separate coils may be provided (Fig. B3.48(c)).

These units are normally sited under windows and are particularly suited to perimeter zones in multi-room, multi-floor buildings with fairly evenly distributed occupancy. The non-changeover system cannot deal with zonal depths greater than 6 m. For internal zones, induction units can be installed horizontally above false ceilings.

Induction units operate with nozzle pressures between 125 and 750 Pa, typically 500 Pa, and volume flow rates of air up to 0·1 $m^3$/s per metre length of unit.




(a) Cooling only

(b) Heating and cooling, one coil

(c) Heating and cooling, separate coils

**Fig. B3.48.** Induction units: alternative coil arrangements.

## SYSTEMS DATA SHEET 8

### Multiple Zone Systems: Air/Water, Fan Coils

See Figs. B3.49 and 50. Refer to *Systems: General Notes* and Systems Data Sheet 1, *Single Zone Systems: Full Fresh Air.*

*Fan Coils*

Ventilation air can be introduced in conjunction with fan coil units in the following ways:

(*a*) Primary air can be distributed to inlet plenums where mixing with room air occurs.

(*b*) Fresh air may be introduced into the space independently via conventional air terminal devices, mixing with fan coil discharge air outside the occupied zone.

(*c*) Fresh air may be drawn through an outside wall by the room unit itself (see Systems Data Sheet 10: *Air/Water, Fan Coil Units*).

The options available for piping and control for (*a*) are similar to those for induction systems. The main differences being:

(*a*) Primary air does not entrain room air, hence primary fan power is considerably lower and noise from induction nozzles is reduced.

(*b*) Maintenance is required to individual fan motors.

(*c*) Individual control of units by the occupants is possible by switching fans without upsetting system balance.

(*d*) Local filtration of recirculation air is possible within fan coil units.

(*e*) Fan coils provide the facility for early morning pre-heat with primary plant held off.

A system with separate ventilation air distribution designed to satisfy humidity requirements cannot use

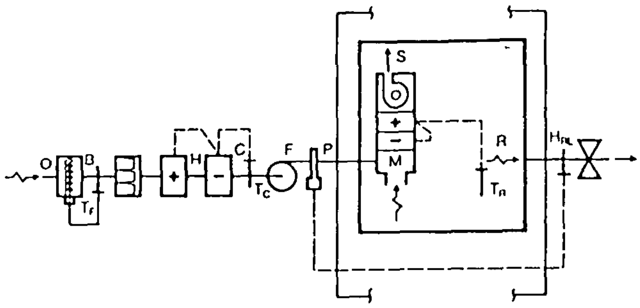



**Fig. B3.49.** System schematic: fan coil system, four-pipe.

this air as the only means of off-setting heat losses unless these losses are very small. If not, excessively high supply temperatures may lead to stratification at the ceiling and poor mixing with fan coil discharge air. A four-pipe system is required in these circumstances.

Control of unit output can be similar to that for induction units: automatic control by cycling of fans is crude and leads to wide swings in temperature and humidity. Furthermore, the change in noise level due to fan switching can be disturbing. Water-side control is preferred.

*Fan Coil Units*

A fan coil unit is a packaged assembly comprising coil(s), condensate collection tray, circulating fan and filter, all

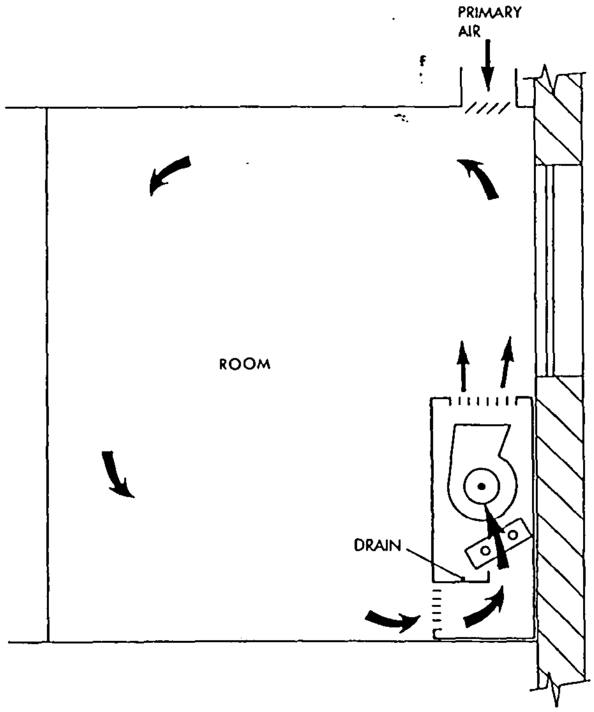



**Fig. B3.50.** Fan coil unit with separate primary air.

contained in a single housing. They are best suited to applications having an intermittent load but where access for maintenance is not a problem, e.g. hotels, shops, restaurants etc.

## SYSTEMS DATA SHEET 9

**Multiple Zone Systems: Air/Water, Reversible Heat Pumps**

See Figs. B3.51 and 52. Refer to *Systems: General Notes.*

*General*

Each unit incorporates a reversible refrigeration machine which comprises a refrigerant/room air coil, a refrigerant/ water heat exchanger, a compressor, an expansion device and a reversing valve. When the controlling thermostat calls for heating, the air coil becomes the condenser, drawing heat from the water circuit, upgraded by the compressor. When there is a net cooling load the air coil becomes an evaporator, rejecting heat to the water circuit via the water-side condenser.

Therefore these units have the ability to operate as heat pumps or as direct expansion coolers. Hence simultaneous heating and cooling may be provided to different zones when required (typically spring and autumn in temperate climates) without recourse to boiler operation. They art not appropriate for hot and humid climates where high cooling water temperatures are inevitable.

The basic system comprises a number of units linked to a two-pipe closed water circuit maintained at near constant temperature by controlled rejection or injection of heat. This temperature, typically 27°C, depends on

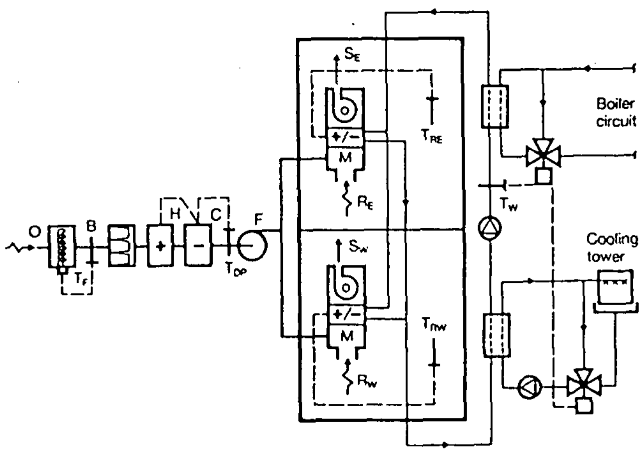



**Fig. B3.51.** System schematic: reversible heat pump.

the method of heat rejection and the external conditions but should be lower in summer than winter.

Dirty water must be prevented from entering the unit coils since they are sensitive to small changes in water flow and temperature. Therefore balancing is critical and reverse return piping is essential (see Fig. B3.51).

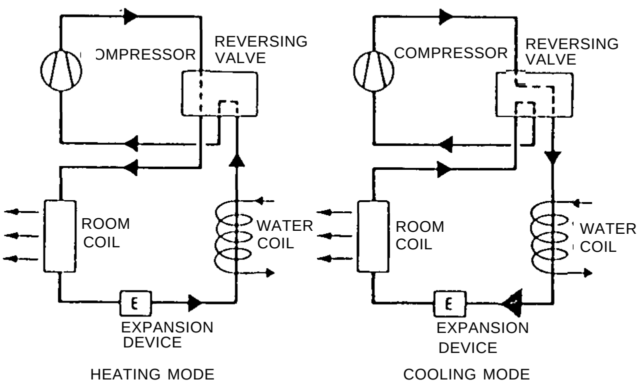



**Fig. B3.52.** Operation of reversing valve.

Fresh air may be distributed to the units from central plant or drawn into each unit locally (see Systems Data Sheet 10: *Air/Water, Various*). Central plant normally distributes air at a constant temperature, typically 10°C, so that "free" heating may be utilised when available during heat pump mode.

Control is normally by a return air thermostat sequencing the compressor and reversing valve. This gives only crude temperature control and swings in relative humidity from 25 to 75% may result, although primary air can be used to limit summer rise.

Compressor noise can be distracting, particularly during the control cycle. If several units are used in combination, simultaneous start-up should be avoided by the use of random start relays or a sequential timer.

## SYSTEMS DATA SHEET 10

### Multiple Zone System: Air/Water, Various

See Fig. B3.53. Refer to *Systems: General Notes* and Systems Data Sheet 1, *Single Zone Systems: Full Fresh Air.*

*Chilled Ceiling*[30]

Chilled ceilings comprise a suspended ceiling with chilled water pipework attached to its upper surface. The amount of cooling per unit surface area is limited by the minimum allowable chilled water temperature (which must be well above the maximum dew-point of the room air if surface condensation is to be avoided) and the heat transfer characteristics of the ceiling. They are often used to deal with the varying component of zonal heat gain (e.g. solar, occupants etc.) whilst the constant components (lighting, structural gains etc.) are dealt with by ventilation air. The ventilation air must also be capable of controlling room humidity.

Each control zone should have a separate coil with chilled water flow controlled by a local room sensor. Response will be slow, depending on the ceiling area. Ventilation air temperature should be scheduled against outside air for zones where perimeter transmission is significant.

The incorporation of some heated pipework into the ceiling above windows will deal with some of the perimeter losses and provide a useful radiant component for winter heating. The heating loops should be controlled in sequence with the neighbouring cooling loops.

A low ceiling temperature reduces the mean radiant temperature experienced by the occupants and hence, higher air temperature can be tolerated (typically 2 K above normal design values).

Stratification of warm air at high level will not occur and air handling luminaires may affect ceiling performance, therefore the ceiling void should not be used as a return air plenum[30].

*Unitary Systems*[30,33]

In this context, unitary systems are differentiated from other systems in that fresh air is drawn into each zone rather than distributed from a central source. With small room units this results in lower filtration efficiencies and the likelihood of external noise and pollution entering the room. Positive wind pressure may produce an increase in ventilation load, whilst negative pressure may reduce ventilation suction.

These systems can easily be incorporated into existing buildings where space is restricted. Units must be located close to an external wall and fresh air intakes may be considered unsightly, whether mounted in the wall or the window. Units may be freestanding, incorporating supply and extract terminals, or concealed. They may supply air direct into the space or, for larger units, via a ducted air distribution system (see Systems Data Sheet 1, *Single Zone Systems*).

*Fan Coil Units*

See also Systems Data Sheet 8, *Multiple Zone Systems: Air/Water, Fan Coils.*

Zonal units must be capable of dealing with fresh air and dehumidification load, hence coils will run wet for some of the time and provision must be made for condensate removal. Humidification, if required, would need to be achieved by separate means (e.g. by injection of vapour direct to the room).

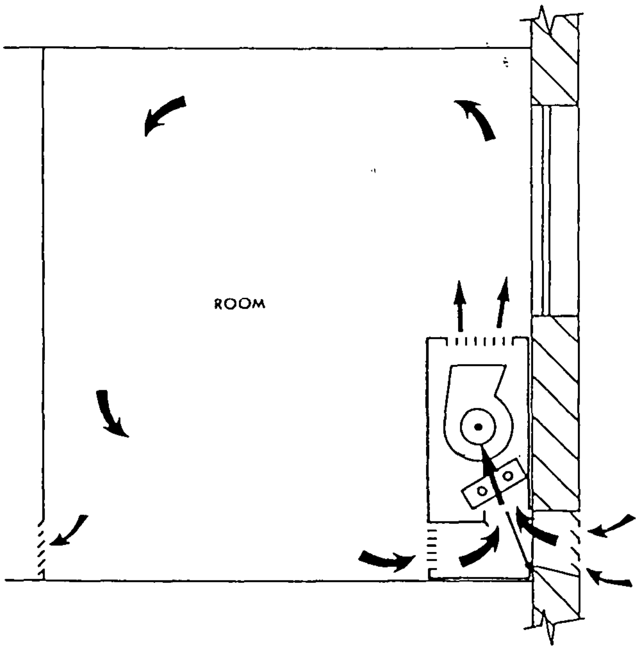



**Fig. B3.53.** Fan coil unit – unitary type.

Control should not be achieved by switching the fan motor since fresh air quantities must be maintained for constant occupancy. A local sensor can be used to control water flow valves to modulate controlling heating and cooling coil outputs in sequence.

It may be possible to incorporate full fresh air facility on each unit to provide "free cooling" when appropriate. However, there may be difficulties in dealing with the corresponding extract air quantities. Depending on the airtightness of building, it may be necessary to provide two stages of extract to cater for minimum and full fresh air quantities respectively. Similarly, it may be possible to cycle to full recirculation for early morning pre-heating. Difficulties may be experienced in incorporating motorised dampers into standard fan coil units.

*Reversible Heat Pumps*

See also Systems Data Sheet 9, *Multiple Zone Systems: Air/Water, Reversible Heat Pumps.*

The room coil duty must be adequate to deal with fresh air load in addition to room load for both cooling and heating modes. Normally, the fresh air quantity is limited to a fixed value, typically 20 to 25% of supply.

*Room Air Conditioners*

Also known as window units and through-wall air conditioners, these are packaged units incorporating a room air side evaporator (direct expansion cooling coil), an outside air cooled condenser, a compressor and an expansion device. Winter heating is often by electric coil, although some manufacturers offer a low pressure hot water coil as an option.

In their basic forms these units offer the crudest form of air conditioning, control normally being through compressor switching, with consequent swings in room temperature, humidity and noise level. Their main advantage is that they are self-contained and require only an appropriate electricity supply and an outside wall in which to be mounted.

Manufacturers' literature needs careful interpretation and corrections to ratings will normally be required to account for UK design conditions.

*Split Systems* [30]

A room air conditioner, or small air handling unit, incorporating a direct expansion cooling coil, can be connected to a remote air or water cooled condensing unit via low pressure vapour and high pressure liquid refrigerant lines. Care must be taken in design to ensure oil entrainment in the refrigerant lines.

With smaller units, control can be achieved by switching the compressor (see *Room Air Conditioners,* above) whilst larger direct expansion coils may incorporate refrigerant flow control or hot-gas bypass, possibly with multi-stage loading and unloading of reciprocating compressors.

## INDUSTRIAL VENTILATION AND LOCAL EXHAUST

Ideally industrial ventilation systems should limit the exposure of workers to airborne contaminants to zero, or as near zero as is practicable and, as a minimum, to below the most recently published occupational exposure limits[34]. These are updated annually and it is essential that current information is used, along with prevailing regulations etc.

If extract rates are too low, short term or long term damage to health will occur or, at least, serious discomfort will be experienced. If too much air is handled, fan and ductwork costs (both capital and running) are excessive, incoming air treatment costs are high, draughts may be difficult and expensive to prevent and the process may be affected by overcooling or costly increased chemical evaporation rates.

The most effective method of preventing a contaminant from entering the breathing zone of a worker is to isolate the process by total enclosure. This solution may be appropriate for automated processes and is essential where highly toxic substances are involved. Normally, some degree of access to the process will be required and it is desirable to limit this access to the minimum necessary for a particular purpose, e.g. access to a low emission chemical process within a fume cupboard via a sliding door, to components to be welded together or to surfaces to be spray-painted. In all cases the contaminant must be drawn away from the breathing zone of the worker.

### Exhaust Hood Suction Dynamics

The velocity of the air induced by suction at an exhaust hood decreases rapidly with distance from the opening. In theory the velocity at a given distance from an opening can be predicted from an equation of the form:

$$V_x = \frac{Q}{bx^n + A} \qquad \text{B3.4}$$

where:

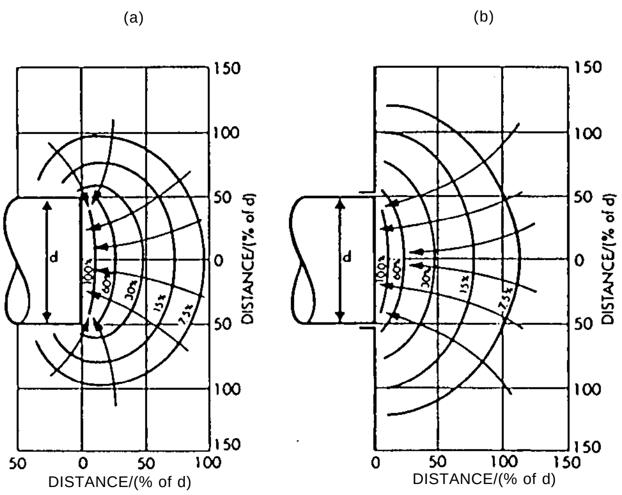
$V_x$ = air velocity at distance $x$ .. .. m/s
$Q$ = volume flow rate of air .. .. $m^3/s$
$x$ = distance from opening .. .. m

$A$, $b$ and $n$ are constants depending on the geometry of the opening and the flow characteristics. Normally, values are obtained experimentally.

Fig. B3.54 shows solutions of this equation for circular openings having unflanged and flanged edges. Note the improvement in performance when the suction is focussed by the flange. The efficiency of capture can be further improved by side screens which also reduce the influence of cross draughts, the ultimate extension of this principle being to enclose the process completely.

Velocities are given as percentages of velocity at opening, $V_o$. Distances from opening are given as percentages of diameter, $d$.

Table B3.6 gives solutions to the above equation for various types of opening and the appropriate equations for air volume flow rates through overhead canopies for both cold and hot processes[35]. Appropriate control velocities and convective heat transfer rates are given in Tables B3.7 and 8 respectively.




(a) Sharp-edged opening
(b) Flanged opening.

**Fig. B3.54.** Isovels for circular opening.

**Table B3.7.** Control velocities for hoods.

| Condition | Example | Control velocity (m/s) |
|---|---|---|
| Released with practically no velocity into quiet air | Evaporation from tanks; degreasing, etc. | 0·25–0·5 |
| Released at low velocity into moderately still air | Spray booths; intermittent container filling; low speed conveyor transfers; welding; plating. | 0·5–1·0 |
| Active generation into zone of rapid air motion | Spray painting in shallow booths; conveyor loading. | 1·0–2·5 |
| Released at high initial velocity into zone of very rapid air motion | Grinding; abrasive blasting. | 2·5–10 |

*Note:*
The higher values apply if: small hoods handling low volumes are used; hoods are subject to draughts; airborne contaminant is hazardous; hoods are in frequent use.

**Table B3.8.** Convective heat transfer rates for horizontal surfaces[35].

| Surface temperature /°C | Rate of heat transfer /($W/m^2$) |
|---|---|
| 100 | 580 |
| 200 | 1700 |
| 300 | 3060 |
| 400 | 4600 |
| 500 | 6600 |

The momentum of the air induced by suction at an opening must be sufficient at the part of the process most remote from the opening to overcome a combination of the following forces:

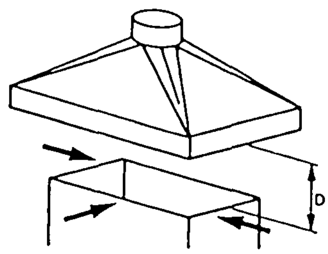
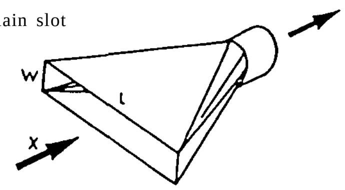
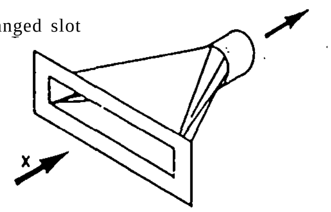
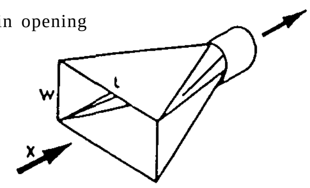
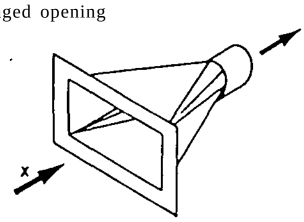
(*a*) *Gravitational:* due to the density of the air/contaminant mixture in relation to the surrounding air.

(*b*) *Frictional:* to overcome drag on the mixture due to the neighbouring bulk of room air.

(*c*) *Dynamic:* due to the initial momentum of the contaminant on release from source and/or disturbing forces due to movement of room air, e.g. cross draughts.

Gravitational and dynamic forces may be used to assist capture. Heavy dust particles having some momentum should be directed into an opening close to the source and, ideally, should be collected and removed from the exhaust without further transport. Transporting large particles through a duet requires very high velocities.

**Table B3.6.** Air volume flow rate equations for hoods and canopies[35].

| Type of opening | Equation | Notes |
|---|---|---|
| Canopy  | Cold source<br>$Q = 1{\cdot}4\ PDv$ | If D exceeds 0·3 B use equation for hot source.<br><br>Canopy should overhang tank by 0·4 D on each side. |
| | Hot source, exposed horizontal surface<br>$Q = 0.038\ A_s \sqrt[3]{h\ D} + 0{\cdot}5\ (A - A_s)$ | Q is progressively under-estimated as D increases above 1 m.<br><br>Canopy should overhang tank by 0·4 D on each side. |
| | Hot source, exposed sides and top<br>$Q = 0.038\ A_s \sqrt[3]{\frac{h\ A_t\ D}{A_s}} + 0.5\ (A - A_s)$ | Q is progressively under-estimated estimated as D increases above 1 m.<br><br>Canopy should overhang tank by 0·4 D on each side. |
| Plain slot  | $Q = Lv(4X\sqrt{\frac{X}{W}} + W)$ | Aspect ratio R should be not less than 10. |
| Flanged slot  | $Q = 0.75\ Lv\ (4X\sqrt{\frac{X}{W}} + W)$ | Aspect ratio R should be not less than 10.<br><br>If X is less than 0·75 W use equation for plain slot. |
| Plain opening  | $Q = v\ (10\ \sqrt{R}\ X^2 + A)$ | Aspect ratio R not to exceed 5. May be used for greater aspect ratios with loss of accuracy. |
| Flanged opening  | $Q = 0{\cdot}75\ v\ (10\ \sqrt{R}\ X^2 + A)$ | Aspect ratio R not to exceed 5. May be used for greater aspect ratios with loss of accuracy.<br><br>If X is less than 0·75 W use equation for plain opening. |

*Symbols:*

A = area of hood/opening . . . . $m^2$
$A_s$ = horizontal surface area of source . . . . . . . . $m^2$
$A_t$ = total exposed heated surface area of source . . . . . . $m^2$
B = breadth of source . . . . m
D = height above source . . . . m
L = length of hood/opening . . m
P = perimeter of source . . . . m
$Q$ = volume flow rate . . . . $m^3/s$
R = aspect ratio (L/W)
W = width of hood/opening . . m
X = distance from source . . . . m
$h$ = rate of convective heat transfer $W/m^2$
$v$ = control velocity . . . . . . m/s

If emitted into a workspace with low momentum, the concentration of contaminant immediately adjacent to its source will be high but normally complete mixing with workspace air will occur within a short distance from the source. An obstructed bouyant plume from a hot source will entrain and mix with room air thus expanding the plume but, if an opening can be used to contain the plume, induction may prove sufficient to avoid the need for additional fan-induced forces.

The basic forms of hoods and canopies are shown in Table B3.6, and Table B3.9 indicates the effects of adjacent surfaces. However, specific processes may require other hood arrangements not shown in the table. The ACGIH Handbook[36] gives a wide range of empirically-based design data sheets for many common industrial processes and should always be consulted before proceeding with the design of a local exhaust system.

The size, aspect ratio, position and number of openings used depends on:

(*a*) size and nature of source (opening must overhang source if possible)

(*b*) dynamics and rate of evolution of contaminant

(*c*) access requirements and position of operator

(*d*) prevailing room air currents (side baffles should be provided if possible).

**Table B3.9.** Effect of side walls and adjacent surfaces[35].

| Type of opening | Baffle | Effect |
|---|---|---|
| Canopy, cold source | Side walls | Reduces effective perimeter, hence flow rate $Q$ is reduced. |
| Canopy, hot source | Side walls | Reduces cross draughts but minimal effect on flow rate $Q$. |
| Plain slot | Long side on flat surface | $Q = \ell v(X\sqrt{\frac{2X}{W}} + W)$ |
| Plain opening | Long side on flat surface | For $R \leq 2$; $Q = v(5\sqrt{\frac{2}{R}}X^2 + A)$ For $2 < R \leq 5$; $Q = v(5\sqrt{\frac{R}{2}}X^2 + A)$ |
| Flanged slot or opening | Long side (not flanged) on flat surface | For $X > 0{\cdot}75W$ calculate flow rate $Q$ for plain arrangement and multiply by 0·75 |

## Overhead Canopies

Overhead canopies are only appropriate for hot processes which cannot be kept covered, and must not be used if the operator is likely to lean over the process or if strong cross draughts are likely to occur. Baffle plates can be incorporated into larger hoods to ensure an even velocity across the opening, whilst very large hoods should be sectioned, each section having its own offtake.

## Lateral Exhaust

For processes in which the emission momentum is small or tends to carry the pollutant horizontally away from the source, horizontal slots or hoods at the edge of a work surface or tank may be used. Slots may be arranged one above the other (see Fig. B3.55) or facing each other along opposite long edges, depending on the vertical distance of the source above the rim of the tank. If the most remote part of the source is less than 0·5 m from the slot, a single exhaust slot along the longer edge is adequate, otherwise two slots, on opposite sides of the source are required.

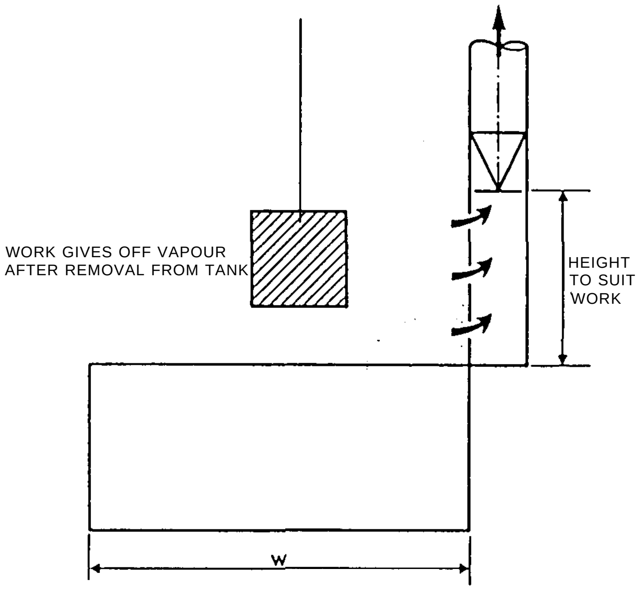



**Fig. B3.55.** Open-surface tank with drying facility: single side exhaust.

## Push-pull Hoods

For sources larger than 1 m across, a push-pull hood arrangement should be used (see Fig. B3.56) whereby a slot or row of nozzles is used to blow air across the source. Design data, for the hood illustrated in Fig. B3.57, are given below[36].

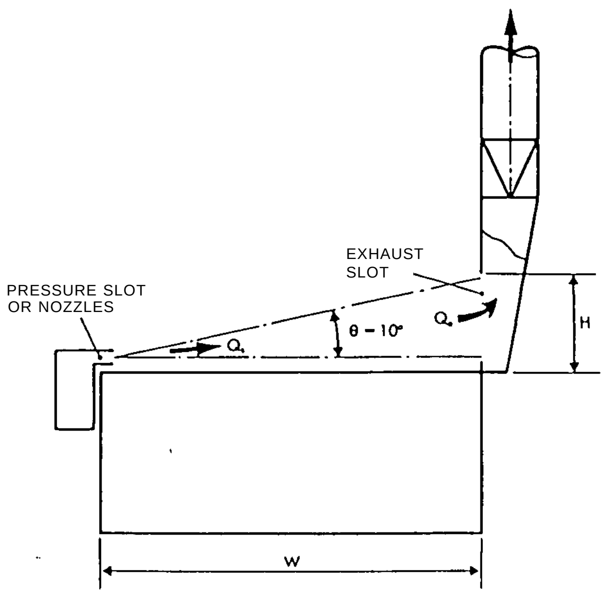



**Fig. B3.56.** Push-pull hood.

Exhaust air quantity:

$$Q_e = (0.5 \text{ to } 0.75) \times A \quad \text{B3.5}$$

where:

$Q_e$ = exhaust air flow rate .. .. .. $m^3/s$
$A$ = area of open surface .. .. .. $m^2$

The value of the numerical factor depends on the temperature of the liquid, presence of cross draughts, agitation of liquid, etc.

Supply air quantity:

$$Q_s = \frac{Q_c}{W \times E} \quad \text{B3.6}$$

where:

$Q_s$ = supply air flow rate .. .. .. $m^3/s$
$W$ = throw length .. .. .. .. m
$E$ = entrainment factor (see Table B3.10)

Height of exhaust opening:

$$H = 0.18\, W \quad \text{B3.7}$$

where:

$H$ = height of exhaust opening .. .. m
$W$ = throw length .. .. .. .. m

Width of supply opening:

Size for a supply velocity of 5 to 10 m/s.

The input air volume is usually about 10% of the exhaust volume and the input air should be tempered to avoid frost damage. The source must not be placed in the input air path since this could result in deflection of the contaminant into the workspace. If necessary, baffles or screens should be used to deflect cross draughts.

**System Design**

Individual exhaust hoods can be either discharged separately to outside via individual fans or connected via a multi-branch system to central fan(s), depending upon:

(*a*) compatibility of substances evolved by different processes: exhausts should be separate if doubts exist

(*b*) access to outside wall: multiple roof penetrations may not be acceptable

(*c*) aesthetics of multiple discharges

(*d*) potential for air cleaning and recirculation of heat recovery from exhaust (see *Equipment: Heat Recovery Devices.*)

(*e*) balancing: multi-branch dust handling systems must be self-balancing, obstructions within ductwork could create blockages

**Table B3.10.** Entrainment factors for push-pull hoods.

| Throw length /m | Entrainment factor |
|---|---|
| 0 – 2·5 | 6·6 |
| 2·5 – 5·0 | 4·6 |
| 5·0 – 7·5 | 3·3 |
| > 7·5 | 2·3 |

(*f*) process usage pattern: ventilation may need to be isolated when a process is not in use and operation of an isolating damper may upset system balance unless a variable volume fan is used. (The VAV fan would be controlled from a system pressure sensor, which could become blocked if dust is transported within the duct).

If make-up air requirements are small they can be drawn from outside or surrounding areas via cracks or openings in the fabric. However, negative pressure must not be allowed to develop to a level at which swing doors are held open or cold draughts are produced in occupied spaces near doors, windows etc. Careful positioning of perimeter heating will minimise discomfort by warming the incoming air.

It is preferable to supply the make-up air via a handling system which cleans, heats (in winter) and, exceptionally, cools and dehumidifies the air, as appropriate.

Large volumes of make-up air may be required, which have considerable implications for energy consumption. Therefore, consideration must be given to:

(*a*) supply of tempered make-up air direct to the process (e.g. by push-pull system)

(*b*) partial recirculation of exhaust air after removal of contaminant using high efficiency air cleaning[36] (see Table B3.11)

(*c*) recovery of heat from exhaust to incoming make-up air but avoiding transfer of contaminants (see *Equipment: Heat Recovery Devices*).

Make-up air must be supplied into the space in such a way as to avoid causing draughts across the process which would affect the efficiency of capture.

## EQUIPMENT

### Ventilation Air Intake and Discharge Points

Each intake and discharge point should be protected from the weather by louvres, cowl or similar device. Any space behind or under louvres or cowls should be "tanked" and drained if there is the possibility of penetration by and accumulation of rain or snow which could stagnate and give rise to unpleasant odours within the building.

Intake points should be situated away from cooling towers, boiler flues, vents from oil storage tanks, fume cupboards and other discharges of contaminated air, vapours and gases, and places where vehicle exhaust may be drawn in. Discharges from other installations, existing or planned, should also be considered.

If the air discharge point is located adjacent to the intake then "short-circuiting" may occur. With systems using recirculated air this is relatively unimportant provided that the discharge is not contaminated. The more remote the intake from the discharge point the less the risk of short-circuiting. If the intake and discharge points are located on the same aspect of a building, short-circuiting is usually more likely to occur than if they are positioned on different aspects. However, wind pressure effects will have a greater effect on the volume of air handled by the system in the latter case.

The position and design of openings must also account for wind forces which may affect fan performance, particularly where fan pressures are low due to low velocity distribution or the absence of obstructions to air flow. The influence of wind pressure can be reduced by:

(*a*) positioning openings within a zone of minimal pressure fluctuation

(*b*) providing balanced openings which face in two or more opposite directions or an omnidirectional roof-mounted cowl.

Birdscreens should be used to prevent entry by birds or other large objects.

Toxic and hazardous exhaust must not be discharged in a manner which will result in environmental pollution and the local authority Environmental Health Officer should be consulted to ensure that proposed discharges will be acceptable. EEC Directive 80/779/EEC[37] gives mandatory air quality standards for smoke and sulphur dioxide, see also Section B2. A vertical discharge stack, capable of imparting a high efflux velocity to the exhaust, may be required. If so, provision must be made for handling rainwater and avoiding corrosion.

In complex arrangements it may be necessary to use a wind tunnel model to predict the effect of wind on discharge performance or pollution dispersal, although prediction may be possible using analytical technique[38, 39].

**Mixing Boxes**

A mixing box is a plenum in which recirculated and fresh air are mixed before entering an air handling unit. It may be part of the ductwork installation, a builder's work chamber or a standard module attached to packaged plant.

Mixing boxes must be designed to provide sufficient mixing so that freezing outside air does not stratify below warm recirculation air on entering the filters. If in doubt, a frost coil at the air intake should be provided.

Mixing dampers may be manual or motorised. Manual dampers are used to fix the minimum fresh air quantity whereas motorised dampers may be cycled from full recirculation for early morning pre-heating to full fresh air to exploit "free cooling" if available.

To improve the rangeability of a motorised control damper the face velocity should be increased to 5 or 6 m/s by adjusting the duct size or by blanking off an appropriate area of the duct at the damper. Damper quality is critical: play in linkages and pivots should be minimal and leakage on shut-off should be less than 0·02.

**Air Cleaners and Filtration**

*Nature of Airborne Contaminants*

Atmospheric dust is a complex mixture of solid particulate matter, comprising dusts, smokes and fumes, and non-particulate vapours and gases. A sample of atmospheric dust may contain minute quantities of soot and smoke, minerals such as rock, metal or sand, organic material such as grain, flour, wool, hair, lint and plant fibres and, perhaps, mould spores, bacteria and pollen. Particles are not generally called dust unless they are smaller than 80 $\mu$m.

Smokes are suspensions of fine particles produced by the incomplete combustion of organic substances such as coal and wood or by the release into the atmosphere of a wide variety of chemical compounds in a finely divided state. Smoke particles vary considerably in size from about 0.3 $\mu$m downwards. Fumes are solid particles, predominantly smaller than 1.0 $\mu$m, formed by the condensation of vapours.

Non-particulate contaminants consist of vapours condensable at normal pressures and temperatures, and gases, of which the most damaging to plants and buildings is sulphur dioxide. Carbon monoxide and various oxides of nitrogen are also present in minute quantities.

There is a wide variation in atmospheric solids between rural, suburban and industrial areas, as shown in Table B3.11. Table B3.12 gives the analysis of a sample of atmospheric dust in terms of total number of particles and total mass for each range of particle sizes. The figures may be considered typical for average urban and suburban conditions but wide variations may be encountered in particular cases.

*Definitions*

The following definitions are commonly used in describing the properties of air cleaners:

*Rating*
The rating of an air cleaner is the air flow for which it is designed, expressed in cubic metres (or litres) per second.

**Table B3.11.** Typical amounts of solids in the atmosphere for various localities.

| Locality | Total mass of solids (mg/m$^3$) |
|---|---|
| Rural and Suburban | 0·05– 0·5 |
| Metropolitan | 0·1 – 1·0 |
| Industrial | 0·2 – 5·0 |
| Factories or work rooms | 0·5 – 10·0 |

**Table B3.12.** Analysis of typical atmospheric dust in relation to particle size.

| Range of particle size: diameter ($\mu$m) | Amount of solid (%) in terms of | |
|---|---|---|
| | Number of particles | Total mass of particles |
| 30–10 | 0·005% | 28% |
| 10– 5 | 0·175% | 52% |
| 5– 3 | 0·250% | 11% |
| 3– 1 | 1·100% | 6% |
| 1– 0·5 | 6·970% | 2% |
| Below 0·5 | 91·500% | 1% |

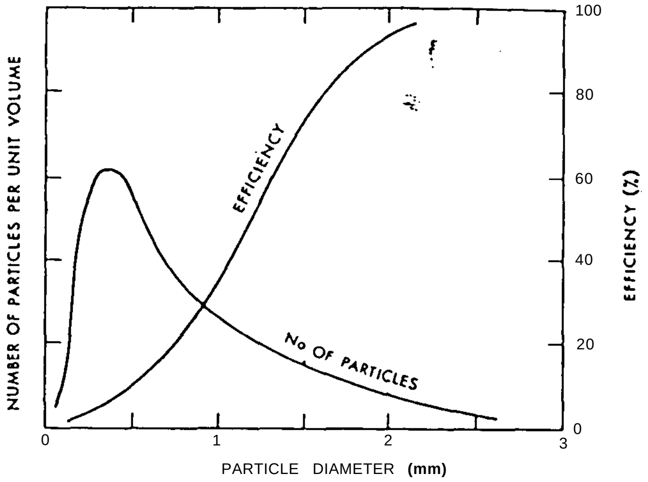



**Fig. B3.57.** Filter efficiency related to particle size.

*Face Velocity*
The face velocity is the average velocity of the air (expressed in metres per second) entering the effective face area of the cleaner. It is determined by dividing the volumetric air flow by the area in square metres of the duct connection to the cleaner.

*Resistance*
The resistance of an air cleaner is the difference between the static pressure upstream and that downstream, usually measured in Pa. Rating tables produced by most manufacturers specify both "clean" and "dirty" resistances for a cleaner when operated at standard rating.

*Efficiency*
The efficiency of an air cleaner is a measure of its ability to remove dust from the air. Expressed in terms of the dust concentration upstream and downstream of the filter:

$$\eta = 100\left(\frac{C_1 - C_2}{C_1}\right) \quad .. \quad .. \quad .. \quad B3.8$$

where:

$\eta$ = filter efficiency .. .. .. %
$C_1$ = upstream concentration
$C_2$ = downstream concentration

The concentration may be expressed in terms of the number of particles, staining power (surface area) or mass of dust.

The efficiency may be expected to depend on particle size. Most filters will remove the largest particles, while allowing some smaller ones to penetrate. If in determining efficiency the total number, area or mass of particles per unit volume of a heterogeneous cloud is used the efficiency may be referred to as *total* or *overall efficiency.* It is this overall efficiency which is usually measured and quoted.

However, if consideration is restricted to dust of a particular size, or small range of sizes, the *grade efficiency* is obtained. This is usually presented as a graph of efficiency against particle size, see Fig. B3.57. Grade efficiency is a true characteristic of a filter, whereas the overall efficiency is not, since the latter depends on the size distribution of the test dust. Given the grade efficiency and the size distribution the overall efficiency can be computed. Furthermore, the grade efficiency is the same whichever criterion (number, staining power or mass of dust) is adopted. This is not the case for the overall efficiency, which is clearly size-dependent.

*Penetration*
The penetration, a term normally used only in connection with "absolute" filters is defined as:

$$P = 100 - \eta \quad .. \quad .. \quad .. \quad .. \quad B3.9$$

where:

$P$ = penetration .. .. .. .. %

*Dust Holding Capacity*
Is defined as the mass of dust which an air cleaner can retain at rated capacity during a resistance rise from its initial "clean" resistance to some arbitrary maximum value. This definition does not apply to automatic self-cleaning filters.

*Arrestance*
A measure of the ability of an air cleaning device to remove injected dust. Calculated on the basis of mass and expressed as a percentage:

$$A = 100\left(\frac{W_1 - W_2}{W_1}\right) \quad .. \quad .. \quad .. \quad B3.10$$

where:

$A$ = arrestance .. .. .. .. %
$W_1$ = mass of dust fed to the device .. kg
$W_2$ = mass of dust passing the device .. kg

*Test Methods*
In non-industrial air conditioning and ventilation the designer is most concerned with the staining power of the dust in the atmosphere, hence a visual measure of efficiency, using atmospheric dust (BS 6540[40]) or sodium chloride (BS 3928[41]) is the most appropriate. These are known as discolouration tests.

However, it is necessary to assess the mass of atmospheric dust that the fillers can remove from the incoming air in order to determine maintenance intervals. This cannot be done using atmospheric dust due to the difficulty of determining input mass, hence a synthetic dust (BS 6540) is used which is similar in composition and concentration to atmospheric dust. The efficiency obtained is known as the efficiency or arrestance.

*Tests for Filters for General Purposes*

A method of comparative testing of air filters for general ventilation purposes is given in BS 6540: Part 1. A more rapid method for determining dust spot efficiency will be given in BS 6540: Part 2*.

BS 6540: Part 1 is based on Eurovent 4/5[42] which in turn is based on ASHRAE Standard 52-76[43]. These standards differ in some respects of detail.

These tests are intended for filters for use in air systems handling more than 0·236 $m^3/s$ and atmospheric dust spot efficiencies no greater than 98%. For higher efficiencies the sodium flame test given in BS 3928 is appropriate.

(*a*) *Atmospheric dust spot efficiency:* otherwise known as the blackness test, this test involves sampling upstream and downstream air quality by drawing sample quantities over target filters and comparing changes in capacity with time. Downstream sampling is continuous, upstream sampling is varied in accordance with expected efficiency on a trial and error basis.

Although this method has been used extensively for on-site testing, BS 6540 specifies test apparatus which enables reasonable reproducibility despite variations in atmospheric composition.

(*b*) *Synthetic dust weight arrestance:* this gravimetric test uses a synthetic dust comprised of carbon, sand and lint in controlled proportions similar to those found in a typical atmosphere.

A known mass of dust is injected into test apparatus upstream of the filter and the dust passing the filter is collected in a more efficient final filter. The increase in mass of the final filter is used to calculated arrestance.

(*c*) *Dust holding capacity:* the above tests can be continued in cycles to achieve a picture of changes in efficiency and arrestance with increasing dust loading until the rated maximum pressure loss or minimum arrestance has been reached. The dust holding capacity can be determined from the total mass of synthetic dust held by the filter.

*Test for High Efficiency Filters*

BS 3928 describes a test method for high efficiency filters not covered by BS 6540; i.e. filters having a penetration less than 2%. The test involves generation of an aerosol of sodium chloride containing particles ranging in size from 0.02 to 2µm.

* Unpublished at time of writing (Sept. 1986)

The amount of particulate matter passing through the filter is determined by sampling both upstream and downstream of the filter and passing each sample through a flame photometer to determine the concentrations of sodium chloride particles captured.

BS 3928 is based on Eurovent 4/4[44], and results achieved under both standards should be comparable.

*On-site Testing*

The efficiency of a filter installation depends not only on the filter efficiency but on the security of the seal between the filter and the air system. This is particularly vital in high efficiency particulate air filter (HEPA) installations, hence penetration must be established immediately prior to use, and at regular intervals throughout the working life of the system[21, 22].

Tests which have been used to determine on-site penetration include:

(*a*) *Di-octyl-phthalate (DOP) test:* DOP is an oily liquid with a high boiling point. Normally, DOP vapour is generated at a concentration of 80 $mg/m^3$ and the downstream concentration is determined using a light scattering photometer via a probe which scans the entire downstream face of the filter installation.

(*b*) *Sodium flame:* a portable version of BS 3928 apparatus which utilises a salt-stick thermal generator to produce an aerosol and an oxy-propane flame and portable photometer for penetration assessment.

*Dust Collector Efficiency*

The efficiency of a dust collector is frequently expressed as grade efficiency for a given range of particle sizes, depending on the composition of the dust to be collected.

*Gas and Vapour Removal*

Most manufacturers quote efficiencies for removal of a wide range of gases and vapours, based on upstream and downstream concentrations. Adsorption filters are also rated in terms of the mass of gas/vapour which can be adsorbed before saturation of the adsorbent.

*Filter Life*

The life of a filter depends on:

(*a*) concentration and nature of entering contaminants

(*b*) filter efficiency

(*c*) dust holding capacity corresponding to a rise in pressure loss between clean and dirty conditions

(*d*) face velocity at the filter.

The following equation[37] gives approximate maintenance intervals:

$$N = \frac{D \times 10^6}{3600\ v\ c\ (h\ /100)} \quad \text{B3.11}$$

where:

| | | | |
|---|---|---|---|
| $N$ | = | duration of use .. .. .. .. | h |
| $D$ | = | dust holding capacity .. .. .. | g/m$^2$ |
| $v$ | = | face velocity .. .. .. .. | m/s |
| $c$ | = | entering pollutant concentration | $\mu$g/m$^3$ |
| $\eta$ | = | filter efficiency .. .. .. .. | % |

It is essential that these parameters are established for airborne contaminants of the same nature and size distribution as those prevailing at the site of the installation. The local conditions may be determined by consultation with the local authority Environmental Health Officer or the Warren Spring Laboratory (Department of Industry). Alternatively, a local survey may be undertaken. Some filter manufacturers provide prediction data for hours of use for different localities. Tables B3.11 and 12 give typical data on the amount and nature of solids in the atmosphere.

If the filter represents a significant proportion of the total pressure loss of the system and there is not provision for automatic fan duty adjustment (e.g. a VAV system), the rise in pressure loss due to filter soiling should not exceed 20% of the total system loss with clean filter.

*Filter Application and Selection*

A forthcoming British Standard* will give guidance on the selection and comparison of air filters. This document may adopt a system[45] by which filters are graded in efficiency terms from grade 2 for coarse pre-filters to grade 9 for near-HEPA filters with efficiency to BS 3928 of 95%. True HEPA filters are excluded from this grading system.

* Title and number unavailable at time of writing (Sept. 1986)

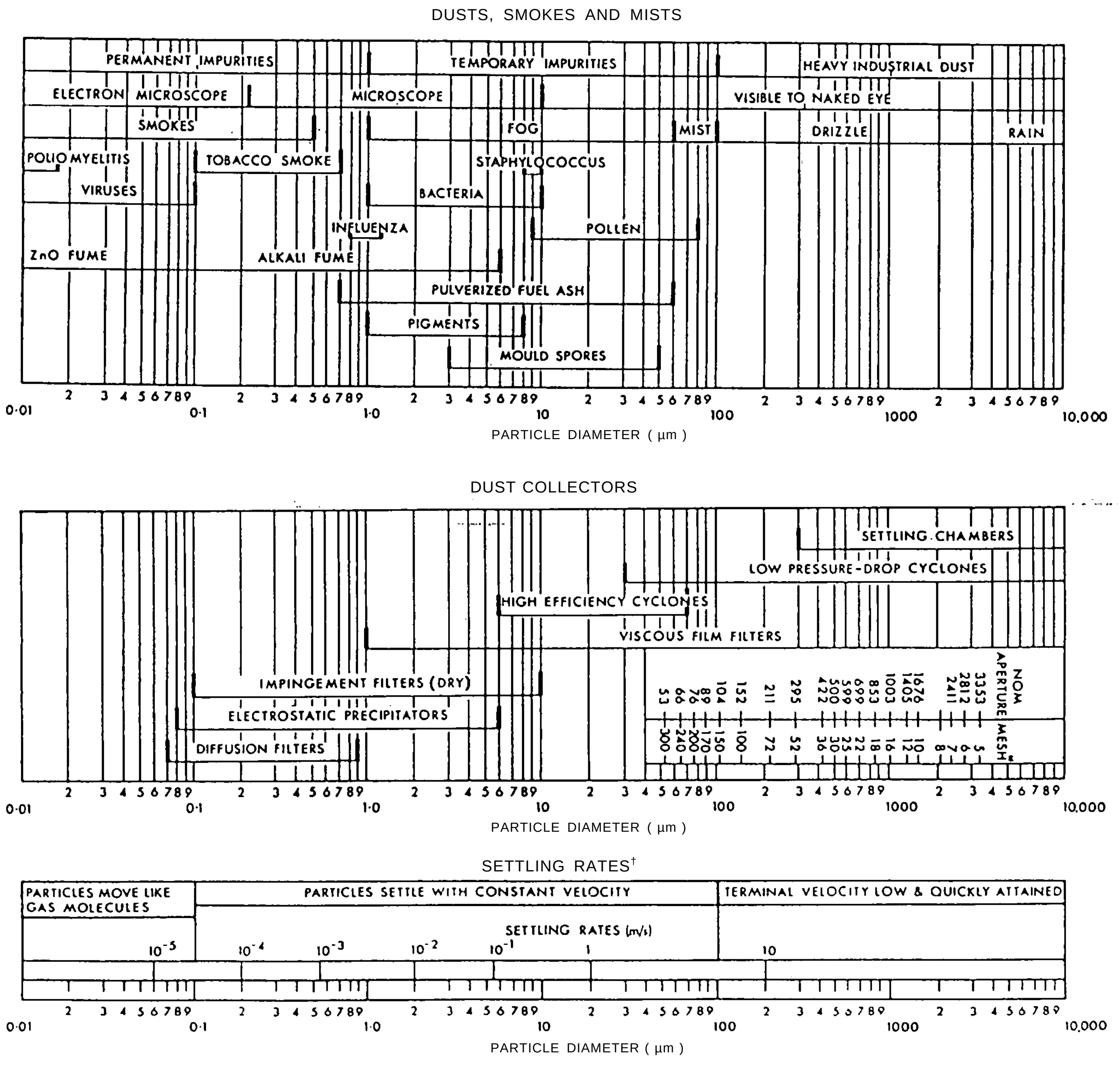



**Fig. B3.58.** Dusts, smokes and mists.

**Table B3.13.** Classification of air cleaning devices

| Type and remarks | Method of cleaning | Face velocity (m/s) | Resistance at face velocity Pa | | Dust holding capacity | Relative efficiency (%) | | Relative cost |
|---|---|---|---|---|---|---|---|---|
| | | | Initial | Final | | BS 3928 Sodium flame | BS 6540 Synthetic dust | |
| VISCOUS IMPINGEMENT | | | | | | | | |
| (a) Panel or unit Thickness ranges 12 to 100 mm; small or intermediate air volumes; good for particles > 10 $\mu$m diameter; efficiency decreases with dust loadings; used as after-cleaners. | Permanent (washable) or disposable | 1·5 to 2·5 | 20 to 60 depending on thickness | 100 to 150 | High, can be critical | 10 | >85 | Low |
| (b) Moving curtain Will handle heavy dust loads; intermediate or large air loads; used as precleaners etc. | Continuous or intermittent, can be automatic | 2 to 2·5 | 30 to 60 | 100 to 125 operating | Self-cleaning by immersion | 10 | >85 | Medium |
| DRY | | | | | | | | |
| (a) Panel, bag, cartridge or unit with fabric or fibrous medium Small or intermediate air volumes. | Usually disposable | 1·25 to 2·5 | 25 to 185 depending on efficiency | 125 to 250 | Generally not as high as viscous impingement. Can be critical | 30 to 80 depending on filter type, medium and face velocity | 96 to 100 | Low to high depending on efficiency |
| (b) Moving curtain Intermediate or large air volumes. | Continuous or intermittent, can be automatic or disposable | 2·5 | 30 to 60 | 100 to 175 operating | Self-cleaning | Can be selected over a wide range | | Medium to high depending on efficiency |
| (c) Absolute or diffusion (HEPA) Prefilter necessary; small air volumes; particles down to 0·01 µm diameter. | Disposable | Up to 2·5 | Up to 250 | Up to 625 | Low | Over 99·9 | 100 | High |
| ELECTROSTATIC | | | | | | | | |
| (a) Charged plate Prefilter desirable; after-filter used to collect agglomerates; power-pack and safety precautions necessary (up to 12 kV); particles down to 0·01 µm diameter; intermediate to large air volumes. | Washable or wipable; can be automatic | 1·5 to 2·5 | Negligible. Resistance (40 to 60) added to improve uniformity of air distribution | | Can be critical | — | Not suitable over 5 µm diameter | High; low maintenance costs |
| (b) Charged medium See charged plate. | Disposable | 1·25 | 25 | 125 | High | 55 to 65 | Not suitable over 5 $\mu$m diameter | High; low maintenance costs |
| ADSORPTION UNITS Should be protected from dust, oil and grease; used for odour removal* | Can be reactivated | Low | Low. Can be selected. Constant | | Medium adsorbs up to half its own weight of many organic substances | 95 dependent on gas to be removed | | High |
| MECHANICAL COLLECTORS Not suitable for particles > 10 $\mu$m in diameter. | To be emptied | According to design | 50 to 100 Constant Some also act as air movers | | High | | | High; low maintenance costs |

WET COLLECTORS
Sometimes air washers used for humidification or dehumidification purposes also act as air cleaning devices. These include capillary air washers, wet filters, absorption spray chambers, etc., for which manufacturers' data should be consulted.

* Odours can also be removed by combustion, masking or liquid absorption devices.

*Filter Installation*

The overall efficiency for the filter installation must be not less than that specified for the filter. Ideally, final filters should be downstream of the fan and under positive pressure to reduce the risk of dust entering the system downstream of the filter. Installation efficiency will suffer if filters are not firmly sealed onto their seatings. This may be difficult if side withdrawal is intended. A means of monitoring pressure loss across the filter should be provided, with warning light and alarm as appropriate.

*Classification of Air Cleaning Devices*

Table B3.13 presents a broad classification of air cleaners and Fig. B3.58 illustrates the various characteristics of dusts, mists, etc. together with other relevant data.

**Air Heater Batteries**

A heater battery comprises one or more rows of finned tube, connected to headers and mounted within a sheet steel casing having flanged ends. Tubes in an individual row are usually connected in parallel but sometimes, for water only, may be series connected as a serpentine coil in a single row. Tubes may be horizontal or vertical except for the cases of serpentine coils, which always have horizontal tubes, or steam batteries which always have vertical tubes. Tube rows are usually connected in parallel.

*Materials*

Tubes should be of solid drawn copper expanded into collars formed on the copper or aluminium fins. Tube wall thickness should be not less than 0·7 mm for LTHW or 0·9 mm for HTHW or steam. Aluminium fins are usually acceptable, except in corrosive atmospheres, and should be not less than 0·4 mm. If copper fins are used they should be of not less than 0·3 mm. Fins should not be spaced more closely than 330 per metre.

Provision should be made in the tube arrangement, by bowing or otherwise, to take up movement due to thermal expansion. Casings and flanges should be of adequate gauge in mild steel, painted with a rust resisting primer. Alternatively, the casings may be in galvanised mild steel with flanges painted in rust resistant primer. Occasionally both casings and flanges may be galvanised after manufacture.

*Test Pressure*

Batteries should be tested with water at 2·1 MPa or 1·5 times the working pressure, whichever is the greater.

*Heating Medium*

This is usually LTHW, MTHW, HTHW or dry steam. Where steam is used for preheat coils, handling 100% outdoor air, so called “non-freeze” heater batteries should be selected. These coils have co-axial steam and condensate tubes which prevent condensate build-up, and consequent risk of freezing, in the lower part of the battery.

**Air Cooler Coils**

A cooler coil consists of one or more rows of horizontal finned tube connected to headers and mounted within a sheet steel casing having flanged ends. Tubes in individual rows are connected in parallel and rows are usually connected in series, although sometimes they may be interlaced. Piping connections must be made so that the coldest air flows over the coldest row, thus approximating contraflow heat exchange. Condensate drain trays through the depth of the coil are essential and these must be fitted at vertical intervals of not more than 1 metre to facilitate proper drainage from the fins. Each such condensate collection tray must be drained using not less than a 22 mm connection. Eliminator plates are necessary if face velocities exceed 2·25 m/s. Cooler coils should normally be located on the low pressure side of the supply fan to avoid condensate leakage through the casing.

*Materials*

Tubes should be of solid drawn copper, electro-tinned and expanded into collars formed in aluminium. Alternatively, solid copper tubes should be expanded into collars formed in copper fins, the whole assembly then being electro-tinned. Tube wall thicknesses should be to suit the test pressure but not less than 0·7 mm. Aluminium fins should be not less than 0·4 mm and copper fins not less than 0·3 mm. Fins should not be spaced more closely than 330 per metre. Facings should be of an adequate gauge of mild steel, welded or with black mild steel angle flanges, the whole assembly being hot-dipped galvanised after manufacture. A suitable alternative, corrosion-resistant construction may be used. Condensate collection trays should be of not less than 2 mm black mild steel, galvanised after manufacture and then coated on the inside with bitumenised paint. Suitable alternative corrosion-resistant materials may be used.

Return bends should be housed within removable covers, allowing sufficient space for the bends to be lagged and vapour-sealed. Alternatively, particularly where a cooler coil is mounted on the high pressure side of a supply fan, return bends should be provided with air-tight galvanised steel covers, with adequate provision for condensate drainage back to the main sump.

*Sprayed Cooler Coils*

These are generally similar to unsprayed coils, except that eliminator plates must always be fitted; the main sump tank is deeper and provides a reservoir of water for the spray pump. An array of standpipes and spray nozzles is fitted on the upstream side of the coil, the main sump of 3·2 mm black mild steel, galvanised after manufacture and coated internally with bitumenised paint. Aluminium fins must not be used.

*Test Pressure*

Cooler coils should be tested with water at 2·1 MPa or 1·5 times the working pressure, whichever is the greater.

*Cooling Medium*

This is visually chilled water or, occasionally, chilled brine. Where the latter is used the reaction of the brine with the piping and pumping materials must be considered and suitable steps taken to prevent corrosion.

*Direct Cooling Coils*

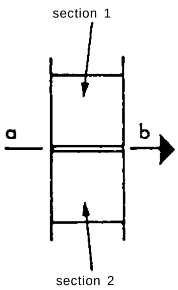
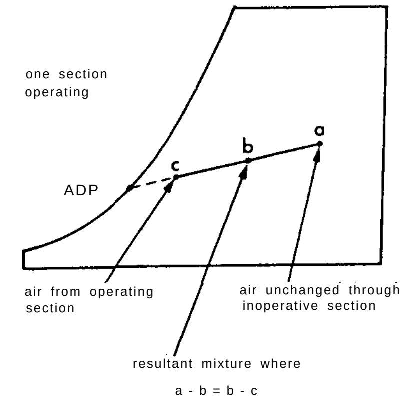
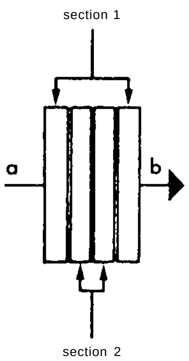
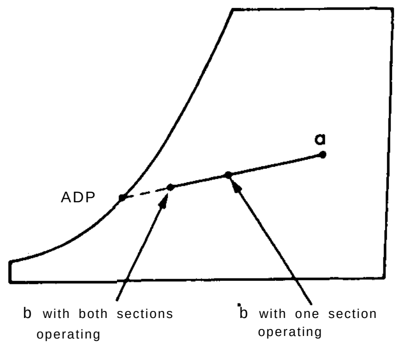
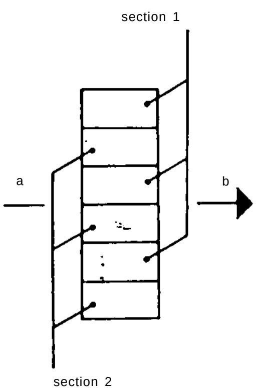
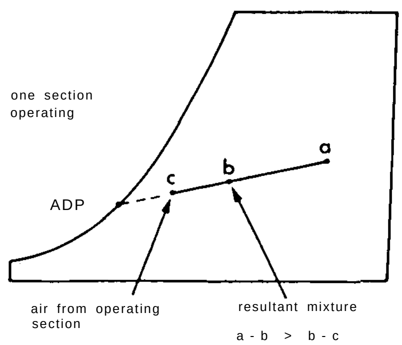
When the coil is a refrigerant evaporator, additional care must be taken with its design and control because of interaction with the refrigeration system. The normal vapour-compression refrigeration system using an oil-miscible refrigerant and thermostatic expansion valve has limiting rangeability features.

To promote adequate oil return to the compressor and prevent oil fouling the coil, an oil entrainment velocity of about 4 to 5 m/s is necessary at the coil outlet and in the suction line. To limit pressure drop between the coil and compressor, the suction line velocity should not be more than 10 to 12 m/s. Therefore rangeability due to oil disentrainment will be between 2:1 and 3:1. A thermostatic expansion valve is often incapable of stable operation at less than a third of its rated capacity, so this may impose a further rangeability limit of the same order.

For a wide control range it is usually necessary to divide the coil into two or more sections each with its own thermostatic expansion valve, isolating inlet solenoid valve and, sometimes, its own suction line. By this means, as each section is isolated, the rangeability of the whole is increased as far as the limit of operation of the sections remaining. For successful control, the psychrometric effect of coil section arrangement must be appreciated, as shown in Table B3.14.

It is also a common practice to connect a separate compressor or condensing unit to each section or to pairs of sections on a multi-section coil in order to increase the total control range. When this is done it is advisable to connect the compressor and coil sections such that each section performs an equal share of the duty. This avoids

**Table B3.14.** Psychrometric effect of coil section arrangement.

| Schematic diagram | Remarks | Psychometric effect |
|---|---|---|
|  | The effect is that of a complete by-passing of half the coil. |  |
|  | The effect is to reduce the contact factor of the coil as though it consisted of fewer rows of pipes. |  |
|  | Depending on coil construction and number of sections, part of the finned surface in an inoperative section is partially effective in transferring heat to an operative section. |  |

a tendency towards frosting due to unequal evaporating temperatures.

### Humidifiers

*General*

Types of humidifier are shown in Tables B3.15 and 16.

*Direct Humidifiers*

Direct humidifiers have a particular application in industrial fields and discharge water particles or vapour directly into the space to be treated. The air in the space absorbs the moisture to a degree consistent with the air movement or turbulence and the fineness of the particles created by the apparatus.

(*a*) *Compressed Air Separation*
Where compressed air is available high-pressure jets can be utilised to produce a fine water spray.

(*b*) *Hydraulic Separation*
Water separators operate direct from high-pressure mains supply, the water jet impinging on a cylindrical or volute casing, suitable ports liberating the water in spray form.

(*c*) *Mechanical Separation*
Mechanical separators operate at constant water pressure and often are of the spinning disc type in which water flows as a film over the surface of a rapidly revolving disc until thrown off by centrifugal force onto a toothed ring where it is divided into fine particles. Alternatively, water is injected into a scroll-shaped housing and separated by the action of either fan or pump.

Some mechanical separators produce molecules lighter than air, termed aerosols, which are non-wetting.

**Table B3.15.** Non-storage humidifiers.

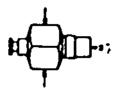
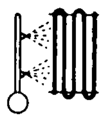
| | NON-STORAGE–DIRECT OR INDIRECT | | | NON-STORAGE–DIRECT | |
|---|---|---|---|---|---|
| | Mechanical Disc  | Mechanical Pressure  | Vapour Injection  | Compressed Air  | Hydraulic Separators  |
| Application .. | Commercial/industrial. | Commercial/industrial. | Commercial/industrial. | Industrial. | Industrial. |
| Saturation efficiency | 90%. | 90%. | Up to 80%. | Variable. | Variable. |
| Thermal efficiency .. | Low. | Low. | Restricted (humidifying only). | Low. | Low. |
| Filtration .. .. | Nil. | Nil. | Nil. | Nil. | Nil. |
| Basis of operation .. | Revolving disc. | Fan/pump. | Steam. | Air jet. | Water jet. |
| Saturating method .. | Fine spray. | Fine spray. | Vapour. | Fine spray. | Fine spray. |
| Use .. .. .. | Humidifying. | Humidifying. | Humidifying. | Humidifying. | Humidifying. |
| Advantages .. | Fineness of mist. | Fineness of mist. | Low maintenance cost. | Low initial cost. | Low initial cost. |

**Table B3.16.** Storage humidifiers.

| | STORAGE–INDIRECT | | | |
|---|---|---|---|---|
| | Spray Washers  | Capillary Washers  | Sprayed Coils  | Pan  |
| Application .. .. | Commercial/industrial. | Commercial/industrial. | Commercial/industrial. | Commercial/industrial. |
| Saturation efficiency . | 70 to 90%. | 97%. | Up to 95%. | Low. |
| Thermal efficiency .. | Up to 80%. | Up to 90%. | Up to 95%. | Low. |
| Filtration .. .. | Low under 20 $\mu$m particle size. | 90%. by weight down to 3 $\mu$m particle size. | Low. | Nil. |
| Basis of operation .. | Pump. | Pump. | Pump. | Static water. |
| Saturating method .. | Fine spray. | Surface film. | Surface film. | Surface film. |
| Use .. .. .. | Humidifying/dehumidifying. | Humidifying/dehumidifying. | Humidifying/dehumidifying. | Humidifying. |
| Advantages .. .. | Variable saturation by water control. | High efficiency/filtration/minimum space. | High efficiency. | Low initial cost. |

(*d*) *Vapour Injection*
For preheating in drying rooms and other applications direct injection of steam can provide a simple and effective method of increasing the moisture content of air provided that the rise in wet-bulb temperature from the heat in the steam does not cause control problems.

*Indirect Humidifiers*

Here, the addition of moisture to the air takes place within the apparatus itself, the air leaving in a near saturated state. Moisture is presented to the air as a mist or surface film, depending on the type of apparatus.

(*a*) *Spray Washers*
The efficiency of spray washers is governed by the fineness of atomisation achieved by the sprays, the quantity of water sprayed into the chamber in relation to the air capacity and the overall length of the unit and consequent time for which the air is in contact with the water mist. To obtain maximum efficiency the face velocity is limited to 2·5 m/s. Efficiencies to be expected are, for a single bank spray washer 70%, a double bank spray washer 85% and a treble bank spray washer 95%, with a distance between each bank of about 1 m.

Suggested rates of water flow are approximately 5 litre/s of water per 3 $m^2$ of face area of the spray chamber per bank of sprays, equivalent to approximately 7 litres of water per 10 $m^3$ of air per second. To provide the fine degree of atomisation required, gauge pressures in the region of 200 kPa are required at the spray nozzles.

This type of humidifier is particularly prone to bacteriological growth and other forms of contamination since water storage ponds may remain still for long periods during warm weather.

(*b*) *Capillary Type Washers*
In principle capillary type washers are built up from unit cells, each cell packed with filaments of glass specially orientated to give the minimum resistance to air flow with the highest efficiency.

The cells are sprayed from nozzles at a gauge pressure of 40 kPa, producing coarse droplets of water which, by capillary attraction, produce a constant film of moisture over each glass filament. The air passing through the cell is broken up into innumerable finely divided air streams providing maximum contact between water and air, resulting in high efficiency of saturation. Most dust particles down to 3 $\mu$m in size are also eliminated from the air stream, and it is therefore necessary to provide a constant flush of water through the cells to eliminate the danger of blockage.

Alternatively, an intermittent supply, controlled by time clock, may be used to flush the cells with water at pre-determined intervals. The face velocity through the washer chamber is similar to the spray type, i.e. 2·5 m/s with a maximum of 2 m/s through the cells.

Saturation efficiency of 97% is given by this type of equipment with as little as 0·8 litres of water per 10 $m^3$ of air per second, although a minimum of 4.5 litre per 10 $m^3$ of air per second is required for flushing purposes. The cells have a maximum water capacity of 11 litre/s per 10 $m^3$/s of air duty.

Capillary cells are arranged in parallel flow formation where the air and water pass through the cell in the same direction, or in a contra-flow arrangement, with water and air passing through the cell in opposite directions. Selection is governed by the humidifying or dehumidifying duty required from each cell and also the degree of cleanliness of the air handled.

Prevention of bacteriological and other contamination must be considered.

(*c*) *Sprayed Coils*
Coils fitted into casings and sprayed from low-pressure nozzles provide an efficient means of humidification. The efficiencies obtained are in direct relation to the contact factor of the coil and thus depend on the number of rows provided, fin spacing, etc.

The recommended rate of spray is about 0·8 litre/s per $m^2$ of face area with a gauge pressure at the spray nozzles of 50 kPa.

Precautions must be taken to prevent bacteriological and other contamination. Ideally, water circulation should be continuous.

(*d*) *Pan Humidifiers*
The simplest form of indirect humidifier is the pan type which consists of a shallow tank in which the water is kept at constant level by a ball float valve.

The air passing over the surface of the water picks up moisture and the water may be warmed to increase effectiveness. Efficiencies are low and depend upon the area of water surface presented to a given volume of air and disadvantages arise from the odours which can result from the static water surface.

Use of this type of humidifier is discouraged because of the high risk of bacteriological contamination.

(*e*) *Mechanical Separators*
Mechanical separators of the revolving disc type, in addition to their usefulness as direct humidifiers, can be mounted into a chamber similar to a spray washer, taking the place of the spray system and pumping set. Water treatment should be considered in hard water localities as any free aerosols not absorbed in the plant may be carried through into the conditioned space, evaporating and precipitating salts on surfaces in the form of a white dust.

(*f*) *Steam Humidifiers*
Steam provides a relatively simple and hygienic method of humidification providing the heat in the system can be absorbed. Generally the use of main boiler steam is limited in application to industry due to the characteristic odour and traces of oil which may be present.

For application to ventilating plants secondary steam can be generated at low or atmospheric pressure from mains steam or electrical supply.

*Excess Moisture Elimination*

Normally indirect water-type humidifiers induce more moisture than that required to saturate the air. To prevent excess moisture entering the ducting system an eliminator section is generally incorporated in the humidifier, comprising either a series of vertical plates profiled to cause directional changes of the air, or alternatively mats of interlaced plastic or metal fibres retained in suitable frames.

Depending upon the depth of the coil an eliminator section is not required with sprayed coil coolers if the face velocity is below 2·25 m/s.

*Materials*

Pollution in the air handled and the nature of water used for humidification purposes can create chemical conditions which may require the use of protective coatings, plastic materials or other metals in preference to steel. However, some materials provide suitable conditions for growth of bacteria and these should be avoided. A list of such materials is given elsewhere[46].

*Water Supply*

Treatment of water may be necessary where available water supplies contain a high degree of temporary hardness or calcium salts in free suspension. Any precipitation which does take place can be dealt with by the use of special acids which dissolve the scale without causing damage to metal.

Because of the potential risks resulting from the growth of micro-organisms in the water, humidifiers which require water storage should be avoided. If such humidifiers are used, consideration must be given to regular sterilisation with an appropriate biocide. Continuous water circulation should be maintained and tanks and headers should be drained when not in use.

*Evaporative Cooling*

In adiabatic saturation the increase of moisture content reduces the sensible heat in the air with a corresponding increase in the latent heat, total heat remaining the same. Through an efficient humidifier, the air is cooled almost to its entering wet-bulb temperature and can then effectively remove sensible heat gains from the building. In practice internal temperatures may be maintained at or near the external dry-bulb temperature and, in an air conditioning installation, the cost of many hours of operation of refrigerating plant can be saved.

**Fans**

*Definitions*

The following definitions should be used in relation to fan components:

*Casing*
Those stationary parts of the fan which guide air to and from the impeller.

*Guide Vanes*
A set of stationary vanes, usually radial, on the inlet or discharge side of the impeller, covering the swept annulus of the impeller blades (or wings). Their purpose is to correct the helical whirl of the airstream and thus raise the performance and efficiency of the fan.

*Impeller*
That part of a fan which, by its rotation, imparts movement to the air.

*Axial-flow Fan*
A fan having a cylindrical casing in which the air enters and leaves the impeller in a direction substantially parallel to its axis.

*Centrifugal or Radial-flow Fan*
A fan in which the air leaves the impeller in a direction substantially at right angles to its axis.

*Cross-flow or Tangential Fan*
A fan in which the air is caused to flow through the impeller in a direction substantially at right angles to its axis both entering and leaving the impeller through the blade passages.

*Mixed-flow Fan*
A fan having a cylindrical casing and a rotor followed by a stator in which the air flowing through the rotor has both axial and radial velocity components.

*Propeller Fan*
A fan having an impeller other than of the centrifugal type rotating in an orifice; the air flow into and out of the impeller not being confined by any casing.

*Terms Relating to Fan Performance*

Fan performance is expressed in terms of fan size, air delivery, pressure, speed and power input at a given air density. Efficiency will be implied or specifically expressed. The size of a fan depends on the individual manufacturer's coding but is directly expressed as, or is a function of, either the inlet diameter or the impeller diameter. Other terms are defined by BS 4856[47] and BS 848: Part 1[48] as follows:

*Reference Air*
For the purposes of rating fan performance, reference air is taken as having a density of 1.200 kg/m$^3$. This value corresponds to atmospheric air at a temperature of 20°C, a pressure of 101·325 kPa and a relative humidity of 43%.

*Fan Total Pressure*
The algebraic difference between the mean total pressure at the fan outlet and the mean total pressure at the fan inlet.

*Fan Velocity Pressure*
The velocity pressure corresponding to the average velocity of the fan outlet based on the total outlet area without any deduction for motors, fairings or other bodies.

*Fan Static Pressure*
The difference between the fan total pressure and the fan velocity pressure.

*Fan Duty (Total)*
The inlet volume dealt with by a fan at a stated fan total pressure.

*Fan Duty (Static)*
The inlet volume dealt with by a fan at a stated fan static pressure.

*Shaft Power*
The energy input, per unit time, to the fan shaft including the power absorbed by such parts of the transmission system as constitute an integral part of the fan, e.g. fan shaft bearings.

*Fan Total Efficiency*
The ratio of the air power (total) to the shaft power.

*Fan Static Efficiency*
The ratio of the air power (static) to the shaft power.

*The Fan Laws*

For a given system in which the total pressure loss is proportional to the square of the volume flow the performance of a given fan at any changed speed is obtained by applying the first three rules. The air density is considered unchanged throughout.

*Rule 1:*
The inlet volume varies directly as the fan speed.

*Rule 2:*
The fan total pressure and the fan static pressure vary as the square of the fan speed.

*Rule 3:*
The air power (total or static) and impeller power vary as the cube of the fan speed.

For changes in density:

*Rule 4:*
The fan total pressure, the fan static pressure and the fan power all vary directly as the mass per unit volume of the air which in turn varies directly as the barometric pressure and inversely as the absolute temperature.

For geometrically similar airways and fans operating at constant speed and efficiency the performance is obtained by applying the following three rules, and again the air density is considered unchanged throughout:

*Rule 5:*
The inlet varies as the cube of the fan size.

*Rule 6:*
The fan total pressure and the fan static pressure vary as the square of the fan size.

*Rule 7:*
The air power (total or static) and impeller power vary as the fifth power of the fan size.

*Axial-flow Fans*

Axial-flow fans comprise an impeller with a number of blades, usually of aerofoil cross section, operating in a cylindrical casing. The fineness of the tip clearance between impeller blades and casing has a marked effect on the pressure development of the fan and, in turn, its output and efficiency. The blades may also have "twist", i.e. the pitch angle increases from tip to root.

The pitch cannot be increased beyond the stall point of the aerofoil and the centre of the impeller has to be blanked-off by a hub to avoid recirculation. The hub is of aerodynamic form and also acts as a fairing for the motor. Large hubs and short blades characterise a high pressure to volume ratio and *vice versa* . There are various refinements, such as guide vanes to correct whirl at inlet or discharge and fairings and expanders to recover a greater proportion of the velocity head in the blade-swept annulus.

Axial-flow fans are of high efficiency and have limiting power characteristics, but as the highest pressure single-stage axial-flow fans develop only about one-fifth of the pressure produced by a forward curved (multivane) fan, they are best suited for high volume/pressure ratios. However, axial-flow fans may be staged or placed in series and when fitted with guide vanes the aggregate pressure developed is proportional to the number of stages for a given volume. A two-stage fan can be contra-rotating, and without the use of guide vanes the pressure developed may be up to 2·75 times greater than that of a single stage.

*Centrifugal Fans*

Centrifugal fans comprise an impeller which rotates usually in an involute casing. The air flows into the impeller axially, turns through a right angle within it and is discharged radially by centrifugal force. The scroll acts as a collector which permits vortex flow to the casing outlet and converts some of the high velocity pressure at the blade tips into static pressure. There are several variations of the basic form, the classified types being:

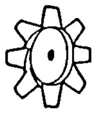
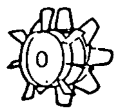
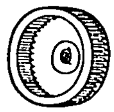
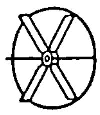
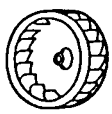
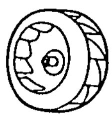
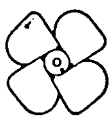
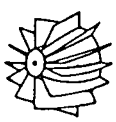
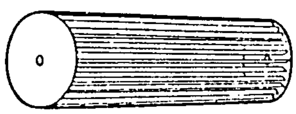
(*a*) *Forward-curved or Multi-vane*
The impeller has a relatively large number of short forward-curved blades. The air is impelled forward in the direction of rotation at a speed greater than the impeller tip speed. For a given duty this type of fan is the smallest of the centrifugal types. It operates with the lowest tip speed and is often referred to as a low-speed fan. As the velocity of the air does not decrease within the blade passages, the efficiency is not high and the motor can easily be overloaded if the system resistance is overestimated.

(*b*) *Straight-radial or Paddle Blade*
The impeller has a few (typically six) straight blades which may be fixed by the roots to a spider, or may have a back-plate and shroud-plate. This is the simplest, and least efficient, of all fan types but is well suited to applications where airborne material is present as the blades are unlikely to clog. The impeller is of high mechanical strength and is cheap to refurbish. Renewable blades or wear plates are often fitted.

(*c*) *Backwards-curved Blade*
In this type the air leaves the impeller at a speed less than the impeller tip speed and the rotational speed for a given duty is relatively high. The impeller has from ten to sixteen blades of curved or straight form inclined away from the direction of rotation. Because the blades are deep, good expansion within the blade passages takes place and this, coupled with a relatively low air speed leaving the impeller, ensures high efficiency and a non-limiting power characteristic.

(*d*) *Aerofoil Blade*
This is a refinement of the backwards-curved fan in which the impeller blades are of aerofoil contour with a venturi throat inlet and fine running clearance between inlet and impeller. The casing is compact and the volumetric output is

**Table B3.17.** Summary of fan types.

| Fan type | Fan static efficiency | Advantages | Disadvantages | Applications |
|---|---|---|---|---|
| 1. Axial-flow (without guide vanes).  | 60–65% | Very compact, straight-through flow. Suitable for installing in any position in run of ducting. | Hish tip speed. Relatively high sound level comparable with 5. Low pressure development. | All low pressure atmospheric air applications. |
| 2. Axial-flow (with guide vanes).  | 70–75% | Straight-through flow. Eminently suitable for vertical axis. | Same as 1 but to less extent. | As for 1 and large ventilation schemes such as tunnel ventilation. |
| 3. Forward-curved or multivane centrifugal.  | 50–60% | Operates with low peripheral speed. Quiet and compact. | Severely rising power characteristic requires large motor margin. | All low and medium pressure atmospheric air and ventilation plants. |
| 4. Straight or paddle-bladed centrifugal.  | 45–55% | Strong simple impeller, least clogging, easily cleaned and repaired. | Inefficiency. Rising power characteristic. | Material transport systems and any application where dust burden is high. |
| 5. Backwards-curved or backwards-inclined blade centrifugal.  | 70–75% | Good efficiency. Non-overloading power characteristic. | High tip speed. Relatively high sound level compared with 3. | Medium and high pressure applications such as high velocity ventilation schemes. |
| 6. Aerofoil-bladed centrifugal.  | 80–85% | Highest efficiency of all fans. Non - overloading power characteristic. | Same as 5. | Same as 5 but higher efficiency justifies its use for higher power applications. |
| 7. Propeller.  | Less than 40% | First cost and ease of installation. | Low efficiency and very low pressure development. | Mainly non-ducted low pressure atmospheric air applications. Pressure development can be increased by diaphragm mounting. |
| 8. Mixed-flow.  | 70–75% | Straight through flow. Suitable for installing in any position in a run of ducting. Can be used for higher pressure duties than 2. Lower blade speeds than 1 or 2 hence reduced noise. | Stator vanes are generally highly loaded due to higher pressure ratios. Maximum casing diameter is greater than either inlet or outlet diameters. | Large ventilation schemes where the somewhat higher pressures developed and lower noise levels give an advantage over 2. |
| 9. Cross-flow or tangential-flow.  | 40–50% | Straight across flow. Long, narrow discharge. | Low efficiency. Very low pressure development. | Fan-coil units. Room conditioners. Domestic heaters. |

*Note:*
Fan static efficiencies are for low and medium pressure applications and do not assume the fitting of expanders.

high. The static efficiency is the highest of all fans, but it is a relatively high-speed fan due to the low pressure development.

*Propeller Fans*

Propeller fans comprise an impeller of two or more blades of constant thickness, usually of sheet steel, fixed to a centre boss and are designed for orifice or diaphragm mounting. They have high volumetric capacity at free delivery, but very low pressure development. However, this may be increased by fitting the fan in a diaphragm which in turn may be installed in a circular or rectangular duct of area greater than the blade-swept area. The efficiency is low but the power characteristic is relatively flat so that the motor cannot be overloaded.

*Cross-flow or Tangential Fans*

These comprise a forward-curved centrifugal type impeller but with greatly increased blade length and the conventional inlets blocked off. The impeller runs in a half casing with conventional discharge but no inlet. Air is scooped inwards through the blade passages on the free side, but at the opposite side of the impeller, due to the influence of the casing, the air obeys the normal centrifugal force and flows out of the impeller and through the fan discharge.

The principle of operation relies on the setting up of a long cylindrical vortex stabilised within the impeller which, being much smaller in diameter than the impeller, rotates at high angular velocity. This in turn drives the main airstream past the blades of the fan with higher velocity than the peripheral speed of the blades themselves. In effect the air flows “across” the impeller almost at right angles to the axis.

Because this fan is so different from other types direct comparisons are not valid. A serious disadvantage of this type is that it cannot be operated at shaft speeds widely different from that for which it has been designed. Consequently it obeys the fan laws only within narrow limits of speed change. It operates with a high discharge velocity and an expander is desirable when connected to ductwork, especially as the efficiency (rather less than that for the multi-vane fan) peaks at near-free-delivery conditions. The discharge opening is characteristically narrow so the fan is not easily applicable to ducting but is well suited to fan coil units and electric space heaters.

*Mixed Flow In-line Fans*

Mixed flow fans comprise an impeller with a number of blades, often of aerofoil section, similar to the axial flow fan. The hub is of conical shape such that the passage of air through the impeller has both axial and radial components, hence the term mixed flow. The mixed flow fan is of high efficiency and can be designed for higher pressure duties than axial flow fans.

To remove the swirl generated by the passage of air through the impeller, stator guide vanes are fitted downstream. These vanes are generally highly loaded due to the high pressure ratios. If the inlet and outlet flanges are to be of the same diameter a change in casing profile is necessary in the region of the guide vanes. Separation of air flow can occur if the conditions for which the fan was designed are not maintained in practice.

**Air Control Units**

When various areas to be air conditioned have differing heat gain patterns with respect to time, these can be met from a central plant in which either the temperature or volume (or both) of the air supplied to each area is varied to meet the particular requirements of the area. Such temperature or volume control may be carried out in ductwork serving a number of rooms or zones, or may be carried out in the terminal units feeding individual rooms.

*Control of Volume*

This may be achieved by:

(*a*) *Damper*
Normally of the butterfly or multileaf type and capable of controlling the volume, providing the pressure drop across the damper does not exceed about 40 Pa. If the pressure drop is higher, there will be a tendency to generate excessive noise. Normally the damper is supplied as a separate component for direct installation in the ductwork and not as part of a terminal unit. Final adjustment is carried out manually on site.

(*b*) *Pressure Regulating Valve*
A dampering assembly consisting of one or two rows of shaped blades the size of which alters when volume adjustment is required. Because of the particular blade shape, the device gives volume adjustment up to pressure drops of about 630 Pa without generating excessive noise. The majority of dampers are set on site, but they can be controlled from a static pressure sensing element. Such units are generally supplied as a separate component for direct installation in the ductwork and not as part of a terminal unit.

(*c*) *Mechanical Volume Controller*
A device which is self-actuating and capable of automatically maintaining a constant preset volume through it, provided that the pressure drop across it is above a minimum of about 120 Pa and below a maximum of about 250 Pa. As the supply air pressure increases, most devices of this type tend to close progressively by means of a flexible curtain or solid damper; a multi-orifice plate fixed across the complete airway of the unit.

As such a unit achieves volume reduction by reducing the airway, there is a tendency to generate noise, particularly when working at high air pressures. For this reason, the volume controller is generally supplied in an acoustically treated terminal unit. It is factory preset to pass a specific volume and, when installed, will automatically give a pre-balanced air distribution system up to and including the terminal unit. It can be adjusted on-site, if desired.

*Control of Temperature*

This may be achieved by:

(*a*) *Blending*
Two separate airstreams, one warm, the other cool, may be supplied to a zone and mixed in a terminal unit to produce a supply air temperature which offsets the zone cooling or healing loads.

(*b*) *Reheat*
Controlled reheat of a pre-conditioned, low temperature air supply by means of hot water, steam or electric coils, may be used to give a resultant supply air temperature which will satisfy the zone requirement.

**Heat Recovery Devices**

*Thermal Wheel*

A thermal wheel comprises a packed cylinder which rotates slowly within an air-tight casing which bridges the ducts between which heat is to be transferred, see Fig. B3.59. The heat transfer properties are determined by the material contained in the wheel, i.e.:

(*a*) Corrugated, inorganic, fibrous material which is hygroscopic and transfers both sensible and latent heat. Air flows through the channels formed by the corrugations.

(*b*) Corrugated metal; aluminium, stainless steel or monel. Latent heat transfer is restricted to that resulting from condensation when the temperature of the heat transfer medium falls below the dew-point temperature of the warm airstream.

There must be an effective seal between the two airstreams and both must be free of particulate matter to avoid clogging. Hygroscopic media may also transfer toxic gases or vapours from a contaminated exhaust to a clean air supply.

A typical rotational speed is 20 revolutions per minute, whilst efficiency depends on face velocity and nature of the heat transfer medium. For the hygroscopic type, efficiencies from 65 to 90% (with decreasing velocity) are claimed for transfer of total heat (sensible plus latent). Maximum efficiencies for the non-hygroscopic type are about 65% for total heat and 80% for sensible heat.

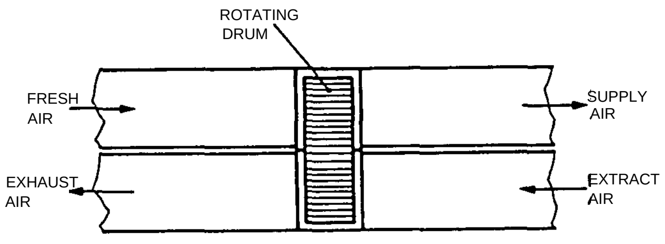



**Fig. B3.59.** Thermal wheel.

*Run-around-coil*

Finned air-to-water heat exchangers are installed in the ducts between which heat is to be transferred. A water or water/glycol circuit is used to transfer heat from the warm vitiated air to the cooler incoming air, see Fig. B3.60. Typical heat transfer efficiencies are about 50%.

Some additional energy is absorbed by the pump and fairly deep coils are required (typically 4 to 8 rows) which increases fan power requirements.

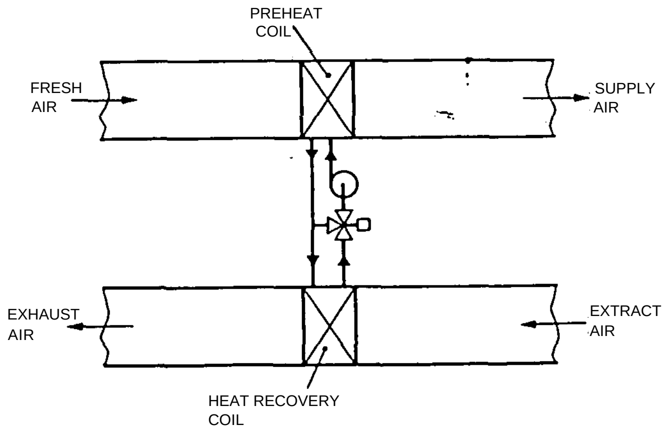



**Fig. B3.60.** Run-around-coil arrangement.

*Heat-pipe*

The heat-pipe is a passive heat exchanger of which there are two main types:

(*a*) *Horizontal:* in which a wick within the tubes transfers liquid by capillary action.

(*b*) *Vertical:* in which heat from the warm lower duct is transferred to the cold upper duct by means of a phase change in the refrigerant (see Fig. B3.61).

Finned tubes are mounted in banks in a similar manner to a cooling coil. Face velocities tend to be low (1·5 to 3·0 m/s) in order to improve efficiency. Typical efficiencies range from 50 to 65%.

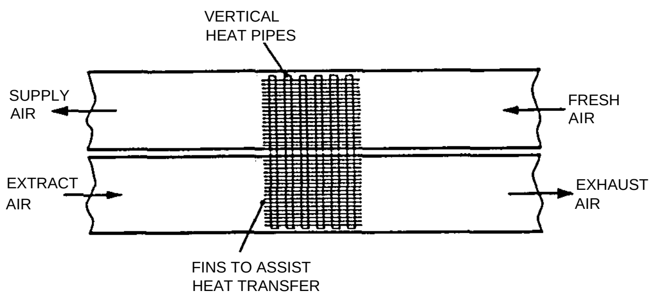



**Fig. B3.61.** Vertical heat pipe arrangement.

*Recuperator*

Recuperators usually take the form of simple and robust air-to-air plate heat exchangers, see Fig. B3.62. Their efficiencies depend on the number of air passages and hence, the area of heat transfer between the two airstreams. If the passages are large the heat exchanger may be easily cleaned and will be suitable for heat transfer from particulate-laden exhaust air. Condensation may occur in warm passages necessitating provision for draining away the condensate.

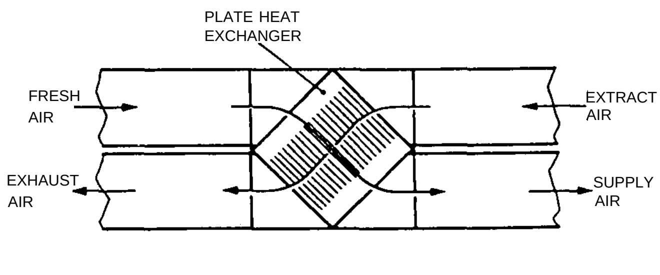



**Fig. B3.62.** Recuperator using a plate heat exchanger.

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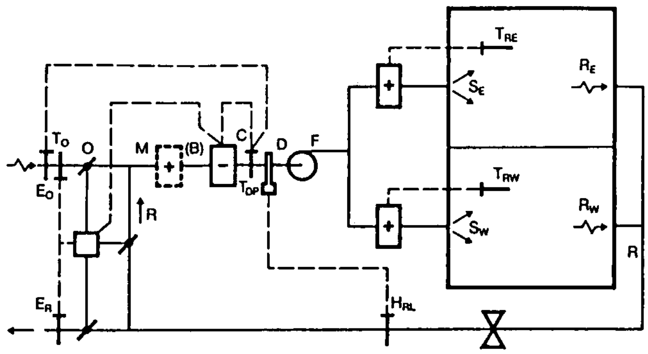



**Fig. B3.63.** System schematic: terminal reheat.

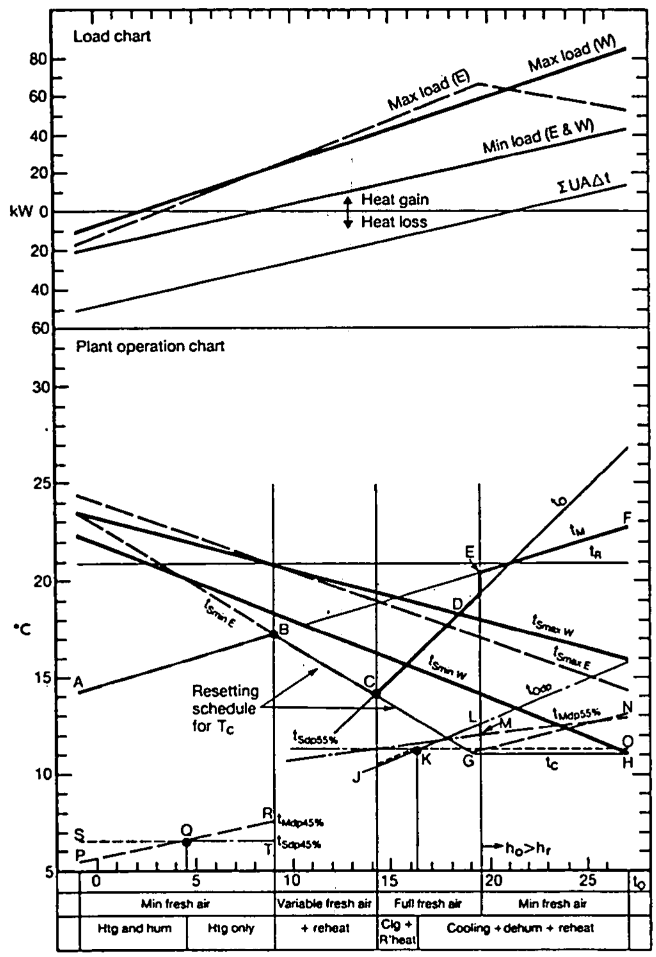



**Fig. B3.64.** Load/plant operation chart: terminal reheat.

## APPENDIX 1

### Load and Plant Operation Charts

Modelling the thermal loads on a building and its systems and the prediction of the response to changes in these loads is a complex process. Much computer time has been spent attempting to produce programmes to predict energy consumption and/or simulate the building/systems/controls interaction. Software will soon be available which will predict, with reasonable confidence, part load operation and year-round energy consumption for any combination of building load characteristic, air conditioning system and control arrangement.

However, it is useful to have a simple design tool that provides a visual representation of the variation of thermal load with outside temperature and the consequent variations in temperature and humidity throughout the system.

*Load Charts*

The load chart is a plot of the estimated sensible loads for each zone requiring separate control due to zonal differences in load profile. For each zone, it is necessary to produce plots of both maximum (MAX LOAD) and minimum loads (MIN LOAD) against a common axis of outdoor dry-bulb temperature. The maximum load line will normally include transmission loads ($\Sigma UA\Delta t$), maximum simultaneous casual gains and solar gains. The minimum load line represents the minimum load which the central plant and control system are likely to experience during normal operation.

These lines can be produced as a result of computer analysis of year-round loads, the relationship between solar gain and outside temperature for each orientation being established from a statistical analysis of weather data. That is, loads calculated on the basis of the frequency of occurence of clear sky peak solar intensity against the full range of outdoor temperatures from summer to winter design conditions.

Alternatively, a series of straight lines can be produced graphically by plotting variations in steady-state transmission losses, estimating casual loads and estimating solar gains using the data given in Sections A8 and A9, as shown in the following example.

The ($\Sigma UA\Delta t$) line in Fig. B3.64 is plotted by calculating the heat transmission through the fabric for a given value of outside temperature (-1°C in this case) and drawing a line through zero load at the point where outside and room temperatures are equal. Internal gains are estimated and added to this line.

**Table B3.18.** Typical average outdoor air temperatures for 5% of days of highest solar radiation (51.7°N).

| Month | Daily mean (°C) | Typical temperatures, hour by hour (°C) | | | | | | | | | | | | | Mean peak swing (K) |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Sun time | | | | | | | | | | | | | |
| | | 0600 | 0700 | 0800 | 0900 | 1000 | 1100 | 1200 | 1300 | 1400 | 1500 | 1600 | 1700 | 1800 | |
| March | 7·0 | 2·8 | 4·0 | 5·4 | 7·0 | 8·6 | 10·0 | 11·2 | 12·2 | 12·8 | 13·0 | 12·8 | 12·2 | 11·2 | 6·0 |
| April | 9·0 | 4·8 | 6·0 | 7·4 | 9·0 | 10·6 | 12·0 | 13·2 | 14·2 | 14·8 | 15·0 | 14·8 | 14·2 | 13·2 | 6·0 |
| May | 13·0 | 8·1 | 9·5 | 11·2 | 13·0 | 14·8 | 16·5 | 17·9 | 19·1 | 19·8 | 20·0 | 19·8 | 19·1 | 17·9 | 7·0 |
| June | 16·5 | 11·2 | 12·8 | 14·6 | 16·5 | 18·4 | 20·2 | 21·8 | 23·0 | 23·8 | 24·0 | 23·8 | 23·0 | 21·8 | 8·5 |
| July | 19·0 | 14·4 | 15·7 | 17·3 | 19·0 | 20·7 | 22·3 | 23·6 | 24·6 | 25·3 | 25·5 | 25·3 | 24·6 | 23·6 | 6·5 |
| August | 17·0 | 12·8 | 14·0 | 15·4 | 17·0 | 18·6 | 20·0 | 21·2 | 22·2 | 22·8 | 23·0 | 22·8 | 22·2 | 21·2 | 6·0 |
| September | 13·5 | 9·6 | 10·7 | 12·1 | 13·5 | 14·9 | 16·3 | 17·4 | 18·3 | 18·8 | 19·0 | 18·8 | 18·3 | 17·4 | 5·5 |

*Notes:* Peaks occur at 1500, means at 0900 and 2100, sinusoidal variations are assumed. Temperatures quoted are fairly typical of most populous areas in the United Kingdom.

Although there is not an exact correlation between solar gains and outside temperatures, consistent patterns emerge by appropriate interpretation of the available data. Table B3.18 gives typical outdoor temperatures for the 5% of days of highest solar radiation (for latitude 51.7°N) for the months March to September. Appropriate values for a west facing orientation are plotted, Fig. B3.65, against the maximum cooling load due to solar gain through vertical glazing using Section A9, Table A9.14 (assuming a lightweight building and constant air temperature).

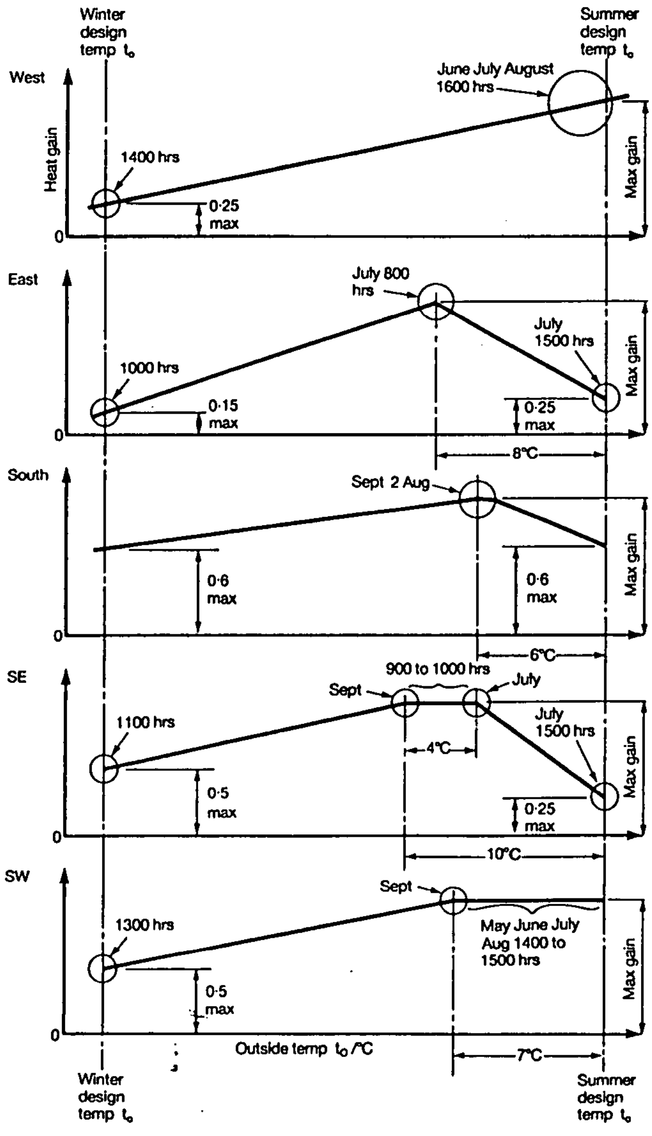



**Fig. B3.65.** Approximate solar gain for various orientations plotted against outside temperature (latitude 51·7°N).

During the winter months high isolation coincides with clear skies, often following high radiation loss at night, resulting in low daytime air temperatures. For December, maximum solar gain coincident with 0°C outside air temperature is assumed. Points for October, November, January and February are more difficult to estimate but experience shows that the accuracy of the load chart in this region is not critical.

The results of this and similar analyses for other orientations are shown in Fig. B3.66. This figure may be used in conjunction with a knowledge of the design external temperature and peak solar gain for each orientation to determine the likely solar gain which, along with the estimated internal gains, is added to the (*SUADt*) line to produce maximum load lines for each zone.

This approximation is sufficiently good to give an overall indication of plant operation but should not be used for detailed design calculations. Furthermore, significant errors will arise if this technique is applied to heavyweight buildings having long time constants.

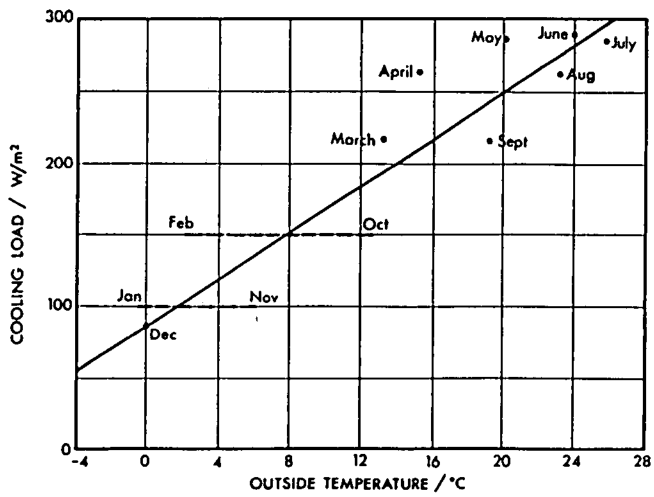



**Fig. B3.66.** Approximate solar gain for west facing orientation plotted against outside temperature.

The example load chart shown in Fig. B3.64 is for a building having two zones of equal size, occupancy level and lighting load. The transmission loss per degree temperature difference (inside to outside), $\Sigma UA$, for each zone is 2.27 kW/K. The minimum load line assumes a minimum internal gain of 30 kW. The peak solar gain for the west facing zone is 43 kW at an outside air temperature of 27°C and the east facing zone experiences the same peak solar gain but at 19°C (outside). The clear sky solar loads at -1°C (outside) are 25% and 15% of the peak for west and east zones, respectively.

Note that this example does not include an internal zone for which the maximum load profile would be a line of constant gain, based on maximum casual gains, whilst the minimum load line would be based on an estimation of minimum casual gains (probably dependent on occupancy).

*Plant Operation Chart*

The plant operation chart is a plot of likely conditions through the plant, i.e. dry-bulb temperatures and moisture contents, the latter expressed as dew-point temperatures.

In the example shown in Fig. B3.64, the room temperature ($t_R$) is assumed to be constant at 21°C and the mixing ratio ($M$) at the central point is based on the fresh air requirements for occupants in the east facing zone.

Thus, using equation B3.3:

$$\dot{m} = \frac{q_s}{C_{ph} \cdot \Delta t} \quad \text{.. .. .. .. ..} \quad \text{B3.12}$$

where:

$\dot{m}$ = rate of mass flow of air .. .. kg/s
$q_s$ = total sensible heat gain .. .. kW
$C_{ph}$ = humid specific heat of air .. .. kJ/kg K
$\Delta t$ = room air to supply air temperature differential .. .. .. .. K

The volume flow rate is given by:

$$Q = \dot{m}\ v \quad \text{.. .. .. .. ..} \quad \text{B3.13}$$

where:

$Q$ = volume flow rate .. .. .. $m^3/s$
$v$ = specific volume of air entering fan $m^3/kg$

Taking the value of $q_s$ from the load chart, for $\Delta t$ of 10 K at peak load, the rate of mass flow of air for the east facing zone is given by:

$$\dot{m}_E = \frac{65}{1{\cdot}02 \times 10} = 6{\cdot}4 \text{ kg/s}$$

From psychrometric chart:

$$v = 0{\cdot}815 \text{ m}^3/\text{kg}$$

Therefore:

$$Q_E = 5{\cdot}2 \text{ m}^3/\text{s}$$

Similarly, for the west facing zone:

$$\dot{m}_w = \frac{85}{1{\cdot}02 \times 10} = 8{\cdot}3 \text{ kg/s}$$

Hence:

$$Q_w = 6{\cdot}8 \text{ m}^3/\text{s}$$

The mixing ratio is given by:

$$M = \frac{Q_{oE}}{Q_E} \quad \text{.. .. .. .. ..} \quad \text{B3.14}$$

where:

$M$ = mixing ratio
$Q_{oE}$ = fresh air requirement for east facing zone .. .. .. .. .. $m^3/s$

Therefore:

$$M = \frac{1{\cdot}5}{5{\cdot}2} = 0{\cdot}3$$

A line ($t_M$) is drawn which represents the mixture temperature resulting from mixing the minimum quantity of outside air with return air. (A temperature rise due to lights etc., may be incorporated into this calculation if necessary.) Also, it is useful to provide a plot of outside dry-bulb temperature ($t_o$) as an indication of the temperature of the air entering the plant when the mixing dampers cycle to full fresh air.

Each zone will require a particular supply air temperature, dependant upon the prevailing room load, under the dictates of room temperature sensors. The lines $t_{sminE}$ and $t_{sminW}$ represent the minimum temperatures required to deal with the maximum load for east and west facing zones respectively. As expected, the east facing zone will require its lowest temperature of 11°C at an outside temperature of 19°C whilst the supply air for the west facing zone needs to deal with peak load only when the outside temperature has risen to 27°C.

In this example, although each zonal supply has to be able to deal with the same minimum load profile, the plots of maximum supply temperature ($t_{smaxE}$ and $t_{smaxW}$) will differ due to the differing mass flow rates. Therefore, these temperature profiles represent the possible changes in dry-bulb temperature of the air entering and leaving the air handling system.

The plant operation chart is also used to plot changes in dew-point temperature (corresponding to moisture content) through the system in order to assess the likely extent of humidification or dehumidification, and the range of outside temperatures over which the room humidities are likely to exceed a prescribed level. The implications of resetting the off-coil temperature sensor on humidity control may also be evaluated.

In order to plot the dew-point temperatures of the air entering and leaving the system an appropriate relationship between the outside dew-point and dry-bulb temperatures must be determined. Hence, a mean condition line is drawn between the summer and winter design conditions on the psychrometric chart, see Fig. B3.67. Normally this will be a straight line, however if a particular application involves evening operation only, humidities will be higher than average and the upper half of the psychrometric envelope may be used.

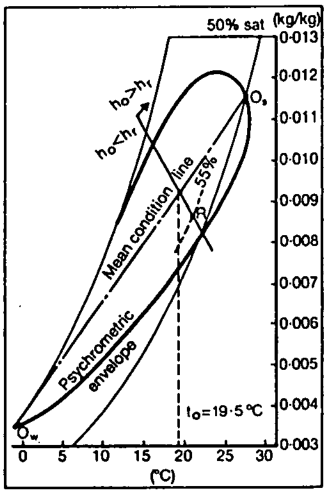
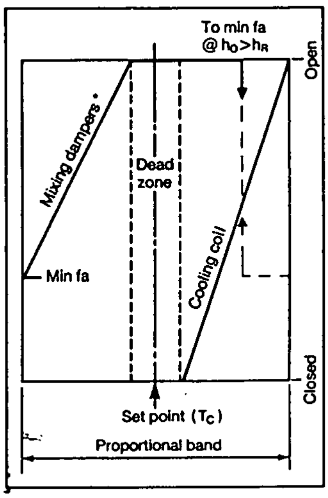



**Fig. B3.67.** External conditions for Kew.

Fig. B3.67 shows the conditions expected at Kew with a mean condition line joining 27°C dry-bulb temperature, 50% saturation and -1°C, 100% saturation. The $t_{odp}$ curve has been constructed by dividing the mean condition line into appropriate increments and determining the corresponding dew-point temperature for each outside condition.

In most cases a different level of humidity will be acceptable in summer to that acceptable in winter. In Fig. B3.68, 55% saturation and 45% saturation have been adopted as design limits for summer and winter respectively. Obviously this requires fairly close control, closer than would be required for most comfort applications.




* In full fresh air systems, mixing damper control can be replaced by air-to-air heat recovery.

**Fig. B3.68.** Proportional band.

The mixture dew-point temperatures ($t_{Mdp55\%}$ and $t_{Mdp45\%}$) corresponding to these room conditions are plotted by joining room and outside conditions for each of the increments used to plot $t_{odp}$, establishing the mixture point in each case and hence the corresponding mixture dew-point temperature. Lower room humidities would be expected below 10°C outside temperature, whilst higher humidities will be experienced at warmer external conditions.

Finally, the supply air moisture contents required to achieve room humidities of 55% and 45% with full occupancy can be represented as lines of constant supply dew-point temperature. These are determined from the psychrometric chart by calculating the moisture gain to the supply air from the occupants and subtracting it from the room moisture content, as follows:

$$Dg = \frac{q_l}{\dot{m}\, h_w} \quad \text{.. .. .. .. ..} \quad \text{B3.15}$$

where:

$Dg$ = difference between room air and supply air moisture contents .. .. kg/kg
$q_l$ = latent heat gain .. .. .. kW
$h_w$ = latent heat of evaporation of water vapour at 20°C (2450 kJ/kg) .. .. kJ/kg

The largest $Dg$ will be required for the east facing zone since this has the lowest supply mass flow rate. If the latent heat gain from the occupants is 7·5 kW per zone:

$$Dg = \frac{7·5}{6·4 \times 2450} = 0·0005 \text{ kg/kg}$$

This corresponds to a maximum supply dew-point temperature ($t_{sdp55\%}$) 11·3°C and a winter supply dew-point temperature ($t_{sdp45\%}$) of 6·5°C.

The above example is based on constant temperature but in many cases room temperature may be allowed to vary from, say, 23°C at 27°C outside to 19°C at -1°C outside. This will change the slope of the lines to allow for lower transmission losses.

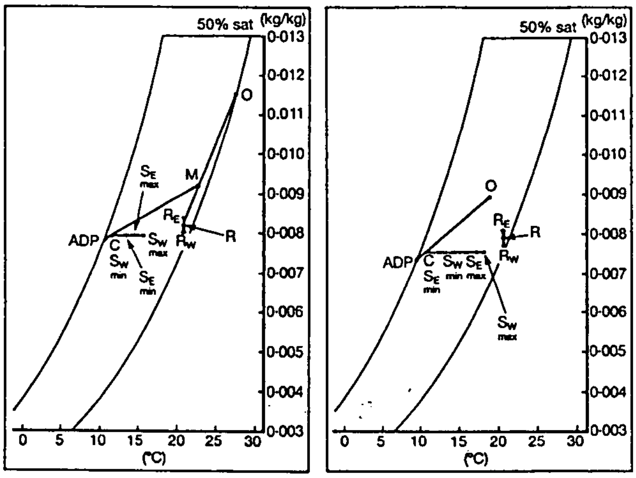
*Analysis*

In the above example it has been assumed that enthalpy comparator sensors ($E_o$ and $E_R$) will be used to cycle the mixing dampers between full and minimum fresh air when the enthalpies of the room and outdoor air are equal (the "economiser" cycle). On the psychrometric chart, Fig. B3.67, it can be seen that this corresponds to an outdoor temperature of 19°C. Dampers are likely to be on minimum fresh air above this temperature and will be held on full fresh air at lower temperatures by the influence of the off-coil sensor ($T_c$), which controls the dampers and cooling coil in sequence, see Fig. B3.68. Note that fan gain is ignored in this instance, hence $t_c$ is equal to $t_s$.

$T_c$ will hold the dampers on full fresh air until the on-coil temperature ($t_o$) drops below its set point after which, as to falls, it mixes room air with outside air in increasing proportions until the minimum fresh air setting is reached.

Hence the schedule by which the outdoor air temperature sensor, $T_O$ resets the off-coil sensor ($T_C$) set point is determined as follows.

As outside temperature falls from 27°C to 19°C (see Fig. B3.69) the supply (hence, off-coil) temperature is held constant at 11°C since the east facing zone requires its lowest supply temperature to deal with peak gains at an outside temperature of 19°C.




(a) $t_o$ = 27°C (b) $t_o$ = 19°C

**Fig. B3.69.** Psychrometric chart: terminal reheat.

As outside temperatures continue to fall the off-coil temperature may be allowed to rise, or held down to ensure that the supply moisture content is sufficiently low to maintain room humidity below the design maximum (55%). This is checked by plotting the cooling cycle at an outside temperature of, say, 17·5°C. In this case, the off-coil temperature can be as high as 12°C without causing humidity problems, although continued rise in the setting of $T_C$ could result in room humidity in excess of 55% until $t_{odp}$ falls below $t_{sdp55\%}$.

This may be prevented by holding $T_C$ down (resulting in increased reheat), or by providing an overriding humidity sensor in the zone. In this case $T_C$ is reset along the $t_{sminE}$ line until it crosses the $t_M$ line. Analysis of the plant operation chart (Fig. B3.64) shows that as $T_C$ is reset the mixing dampers will be cycled from full fresh air as the resetting schedule crosses the $t_O$ line to minimum fresh air as it crosses $t_M$. As outside temperature falls below 9°C the central plant will no longer be supplying air at the mixture temperature and the room sensors ($T_{RE}$ and $T_{RW}$) will call for reheat in these zones, as appropriate.

The resetting schedule obtained by this method is based on simplified linear load profiles and outside conditions, hence the supply temperature will not be the optimum value for minimum energy consumption at all times. However, control systems are available which monitor the position of the zonal control devices (reheater valves, dual duct dampers etc.) and reset the central plant to meet the needs of the zone requiring the lowest duct temperature, as dictated by zonal sensors.

In this example, control over low humidities is dealt with by means of a steam humidifier which will operate at outside temperatures below 4·5°C, i.e. when $t_{Mdp}$ falls below $t_{sdp45\%}$.

It may be inferred from the plant operation chart that mechanical sensible cooling is likely to occur between the line CDEF and the off-coil temperature line CGH, when outside temperatures vary between 27°C and 14·5°C.

Dehumidification is likely to occur between the lines KLMN and KH down to 16·5°C outside temperature, when $t_{odp}$ drops below $t_{sdp55\%}$. The zonal room sensors may call for reheat up to a maximum of $t_{smax}$ at any outside temperature, even when refrigeration is required.

Similar analyses can be performed for other air conditioning systems involving all-air or primary air/water distribution[29].

## APPENDIX 2

### Basic Psychrometric Processes

Table B3.19 (overleaf) illustrates the basic psychrometric processes and lists the equipment concerned. For details of the various items of equipment, see *Equipment*, page B3.39.

**Table B3.19.** Basic psychrometric processes.

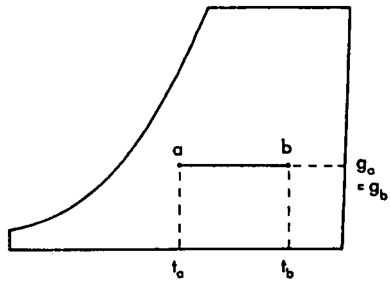
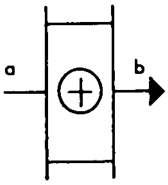
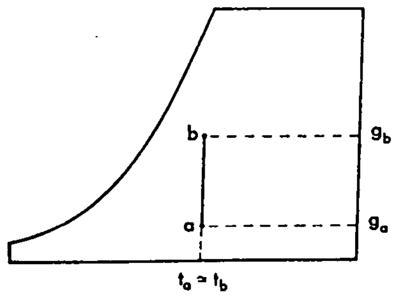
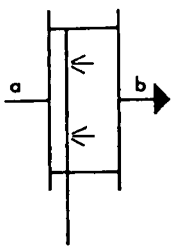
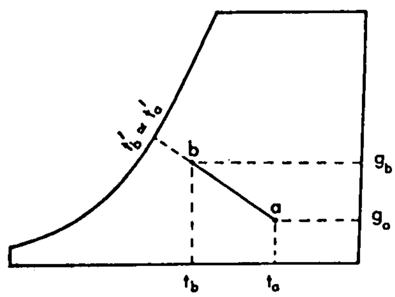
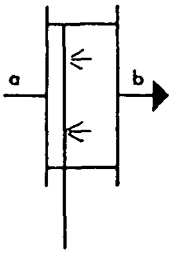
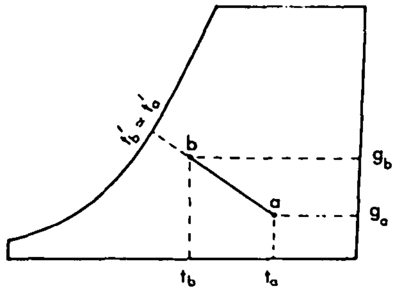
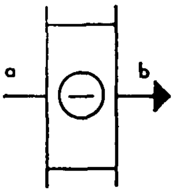
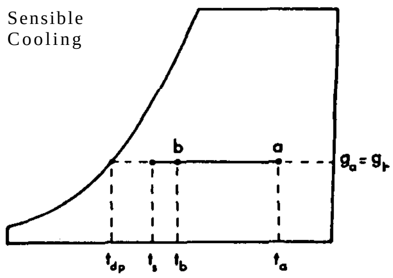
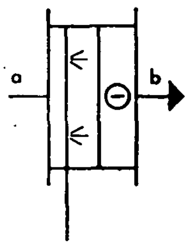
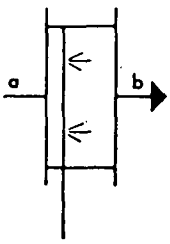
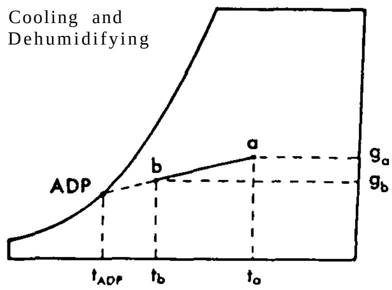
| Process and schematic diagram | Method | Remarks | Psychrometric effect |
|---|---|---|---|
| HEATING | Electric | No additional plant required. High energy cost. Wiring and switch gear costs high for large duties. Usually only step control available. | |
| | Steam | Small heat transfer surface. Plant cost high unless required for other services. Condensate return can involve difficulties. Modulating control available (2-way valve). |  |
|  | Hot water | Simple and reasonably cheap plant and distribution system. Integrates well with other heating systems. Some simplicity sacrificed to decrease heat surface with HTHW. Modulating control available (2- or 3-way valve). | |
| | Direct firing | Least expensive in specific cases. Can involve problems of combustion air and flue requirements. On/off control is common for smaller units, whilst high/low flame is usually available for larger units. | |
| HUMIDIFICATION | Steam injection | Electrically heated, self-contained unit or unit supplied by mains steam. Water treatment advisable. Small space occupied. Mains units have modulating control (2-way valve), electric units are normally on/off. Mains units may require condensate drain. |  |
|  | Water injection | Involves atomising process (spinning disc, compressed air etc.). Some types are non-recirculatory and require drainage. Air is sensibly cooled as water evaporates. Contaminants from untreated water will enter airstream. Water treatment including biocidal control is essential. Space occupied depends on type. Some units mount on duct wall, other in duct line. Control is usually on/off by stopping atomiser or water supply, larger units in multiple form may be stepped. Normally modulation is not recommended unless water flow is large. | |
| | Spray washer | Bulky equipment requiring good service access to tray and sprays. Also dehumidifies if supplied with chilled water (see Cooling–Air washer). Air sensibly cooled as water evaporates unless water is heated (not normal). Requires water treatment (including biocidal control) and bleed and recirculating pump. Removes both gaseous and particulate air contaminants but with low efficiency. Control indirect by modulation of inlet air condition (preheater or mixing dampers) or by by-pass and mixing. Saturation efficiencies range from approximately 70%, for one bank facing upstream, to 85-90% for two banks opposed. Water quantity per bank is of the order of 0·4 litre/s per $m^3/s$ of air flow. Air velocity is of the order of 2·5 m/s. |  |
| | Capillary washer | Similar to spray washer but less bulky and provides better air filtering. Has smaller cooling capacity than spray washer when used with chilled water. May require addition of cooling coil. Filtration efficiency is good. | |

**Table B3.19.** Basic psychrometric processes–*continued.*

| Progress and schematic diagram | Method | Remarks | Psychrometric effect |
|---|---|---|---|
| HUMIDIFICATION   | Sprayed cooling coil (not subject to refrigeration) | Utilizes cooling coil as wetted pack for humidifying. Action as washer but sprays less prone to blocking. Eliminators not required unless air velocity high. Requires more space than non-sprayed coil but less space than washer. Water treatment advisable, bleed essential (see cooling coil). Control as for spray. Can be used to cool coil water circuit with low air on temperature, thus making $t'_b$ greater than $t'_a$. This is sometimes used in an induction system primary plant. Saturation efficiency is of the order of 80-90%. Water quantity of the order of 0·5-1·0 litre/s per $m^3/s$ of air flow. Air velocity is of the order of 2·5 m/s. |   |
| COOLING   | Indirect cooling coil | Supplied with chilled water or brine (usually 2 or 3°C below apparatus dewpoint required). As water is in closed circuit (except for head tank) there is no water contamination from air or evaporation. Contact factor depends on number of rows of pipes deep. Chilled water enters at air off side. Drain is required. Control by modulating water temperature or flow rate (3-way valve). Normal to keep constant flow rate through chiller. | |
| | Direct cooling coil (direct expansion coil) | Coil is evaporator of refrigeration circuit. May be cheaper overall than indirect system, but usually involves refrigerant circuit site work. Control by steps, or modulated, depending on refrigeration system. May need special circuitry. Drain is required. Complex and costly for larger installations. May be excluded by local legislation for some applications. | Sensible Cooling   |
|   | Sprayed cooling coil (subject to refrigeration) | With spray off, coil operates exactly as cooling coil. Spray sometimes used to increase surface in contact with air, results in larger contact factor. Saturation efficiency of the order of 80-90%. Water quantity of the order of 0·5-1·0 litre/s per $m^3/s$ of air flow. Air velocity of order of 2·5 m/s. | |
|   | Air washer (spray washer) | See general remarks on Humidification—Spray washer. Sprays supplied with chilled water which is liable to contamination through air washing and evaporation if also humidifying. Use with normal, non-cleanable direct expansion chiller not recommended. Overflow required. Contact factor determined by spray design and number of banks. Control by change of spray water temperature (diverting chilled water back to chiller). Saturation efficiencies range from approximately 70% for one bank facing upstream to 85-90% for two banks opposed. Water quantity per bank is of the order of 0·4 litre/s per $m^3/s$ of air. Air velocity is of the order of 2·5 m/s. | Cooling and Dehumidifying   |

# SECTION B4 WATER SERVICE SYSTEMS




*This edition of Section B4 first published: 1986*

# SECTION B4 WATER SERVICE SYSTEMS

## INTRODUCTION

This section deals with the supply of water to, and distribution within buildings; its rate of consumption and storage and some precautions necessary to maintain the supply.

The principal legislation relating to water supply is contained in the following:

Water Act 1945 and 1948

Water Supplies and Sewerage Act (N. Ireland) 1945

Water (Scotland) Act (1946)

Water Resources Act (1963)

Water Act (1973)

Local Government (Scotland) Act (1973)

The distribution of water and the installation of systems are subject to the byelaws of individual water authorities, and, as a statutory requirement, the Building Regulations 1985[1]. Water authority byelaws are based on the Model Water Byelaws 1986[2].

## WATER SOURCES

Information concerning the properties of any particular water supply may be obtained from the relevant water supply authority and full details of these are listed in the Water Services Year Book[3].

### Hardness

Analysis of the total solids taken into solution as water percolates through a catchment area reveals various mineral salts. Only salts having soap-destroying properties, namely those of calcium and magnesium, are considered in the quantitative evaluation of hardness.

Hardness has traditionally been expressed in terms of temporary and permanent components, the former being that proportion of the total precipitated by boiling. Current practice tends towards the substitution of a more precise evaluation of the water characteristics as carbonate and non-carbonate hardness. Numerically, the carbonate hardness is usually identical with the alkalinity of the water.

The traditional unit of hardness, the degree Clark (grains of calcium carbonate per imperial gallon), has been replaced by mg/litre. Table B4.1 provides conversion data for various units.

**Table B4.1.** Conversion factors for scales of hardness.

| Units | Parts per million (mg/litre) | Parts per 100 000 | Grains per imperial gallon | Grains per U.S. gallon |
|---|---|---|---|---|
| One part per million as $CaCO_3$* .. .. | 1·0 | 0·10 | 0·07 | 0·058 |
| One part per 100 000 as $CaCO_3$ .. .. | 10·0 | 1·00 | 0·70 | 0·58 |
| One grain per imperial gallon as $CaCO_3$ (=1 degree Clark) .. | 14·3 | 1·43 | 1·00 | 0·83 |
| One grain per U.S. gallon as $CaCO_3$ .. | 17·1 | 1·71 | 1·20 | 1·00 |

* The convention 'as $CaCO_3$' is arbitrary and expresses the soap-destroying properties of the water, due to both calcium and magnesium salts, as though they were all due to the presence of calcium carbonate.

### Water Examination

The suitability of a water for any particular purpose can be determined only after appropriate chemical and bacteriological examination. Sampling should be arranged by the water analyst using sterilised Winchester bottles; a schedule of information regarding the characteristics of the supply must be provided with the same. Section B7 of the *Guide* provides information on water treatment techniques.

Department of Health and Social Security Report No. 71[4] contains detailed information on the bacteriological contamination of potable water supplies.

## WATER UNDERTAKING MAINS

Mains laid for the distribution of water from one district to another and for the distribution of water within a district are usually installed by the water undertaking. Supplies for domestic purposes must be provided by statute, whereas for industrial uses this is by agreement.

Although no clear definition is given in the Water Act 1945, mains are generally considered to be divided into three categories:

(*a*) Trunk mains, which are generally described as those which convey water from a source of supply (reservoir, pumping station etc.) to an outlying district without supplying consumers *en route.*

(*b*) Secondary mains, which are the distribution mains in any district, usually fed from a trunk main and supplying the consumers' connections in the district.

(*c*) Service pipes, which are the branch supplies from the secondary mains and serve individual consumers or premises.

It is of the utmost importance that a potable supply for dietetic purposes is not liable to contamination. There should be no inter-connection or cross-connection of the supply with any other water supply, e.g. water of uncertain quality or water already used for some purpose. Reflux valves or stop valves should not be considered adequate to prevent cross-contamination.

The design of water services should be arranged to prevent the possibility of backflow (or back siphonage) from any terminal outlet, cistern, or sanitary appliance.

Some water undertakings maintain separate distribution systems of non-potable water for industrial and agricultural purposes; they should not be confused with a potable water system.

**Connections**

Connections to a trunk or secondary main are normally carried out by the water undertaking. It is not usual practice to allow a service pipe to be connected to a trunk main except in extreme cases of deficiency of the water supply in the area through which the main passes. Connections to secondary mains may be made under pressure to connect pipes of 50 mm diameter and below, whereas for larger pipes a shutdown of the main is usual.

Service pipes are normally fitted by the water undertaking from the main up to the boundary of the premises to be supplied. At this point a valve is provided for their sole use. Responsibility for the supply and installation of this valve and the service pipe should be agreed with the relevant water undertaking. From the valve the service pipe continues inside the boundary to the building, where a second valve is fitted for the use of the consumer. Where a building is divided into separately occupied parts supplied from a common service pipe, a valve is required to control the supply to each part; fixed inside and under the control of the user.

A service pipe to any one consumer may be limited in size such that it is not capable of supplying more than about 2000 $m^3$ per day.

**Pressures**

A water undertaking will provide an indication of the pressures likely to be available within the mains but is not required to guarantee these above the static pressure that exists from mean low water level of the reservoir or source of supply. There is no upper limit to the pressure, although good practice and economic considerations dictate limits.

Water undertakings may have high and low pressure mains in their area of supply, zoned to avoid interconnection. Pressure zones are usually spaced at 150 to 300 kPa intervals to give a clear indication of variation. Although pressures of 1.2 to 1.5 MPa are possible, these may lead to excessive use of water.

Generally, pressure zones are controlled by service reservoirs or water towers and are at gauge pressures of 300 kPa, which may be considered a reasonable working pressure for most mains.

*Pressure Fluctuations*

Pressure fluctuations in mains will occur during periods of heavy demand, particularly in built-up areas. Night and daytime pressures can vary considerably, this being one reason advanced for providing storage.

In some areas of high pressure, the introduction of pressure reducing valves will enable flow and pressures to be adjusted to balance the system demand. Similarly, pressure sustaining valves may be fitted to maintain upstream pressures and prevent starvation of supply to higher zones.

*Pressure Boosting*

In instances where the water undertaking's main cannot provide sufficient pressure to service all of the equipment in a building, pumping equipment must be installed. Some water undertakings will allow such pumping equipment to be connected directly to the incoming service pipe, specifically in the case of a drinking water service, but the majority insist upon the installation of a break tank.

The disposition of water stored within a building depends upon many factors, including the structural advantage of storing the maximum amount at or below ground level, the need to ensure that any water stored does not stagnate and the potential vulnerability of water supply in the event of power failure. A common compromise is to store two thirds of the required capacity at or below ground level with the remaining one third held elevated.

Pressure boosting equipment falls into three categories:

(*a*) The pneumatic system, as illustrated in Fig. B4.1, which is essentially pressure controlled.

(*b*) The intermittent pump system, as illustrated in Fig. B4.2, which is essentially float switch controlled.

(*c*) The continuously running pump system, as illustrated in Fig. B4.3, which is a useful application for smaller systems and relies for its control upon careful choice of pump characteristic and a small pressure bleed from delivery to suction.

For buildings higher than about 10 storeys it is necessary to consider pressure balancing at individual draw-off fittings by means of orifice plates, or by arranging the piping system in vertical zones with reducing valves. For very high buildings, the provision of intermediate water storage and further pumping equipment may be necessary.

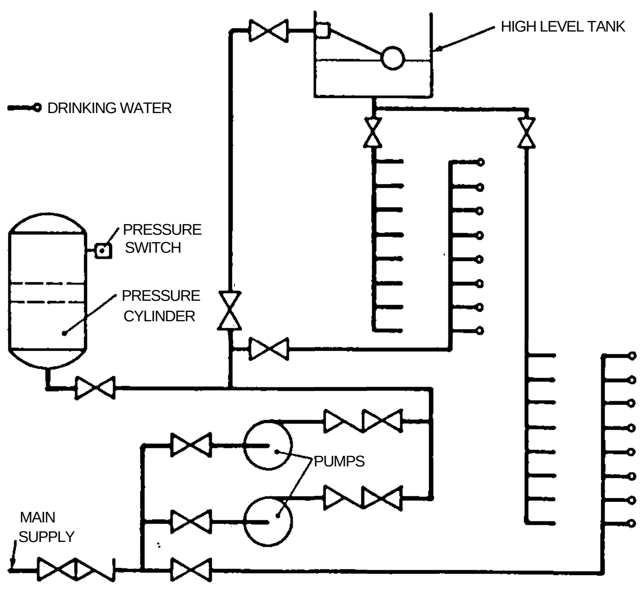



**Fig. B4.1.** Pneumatic boosting system.

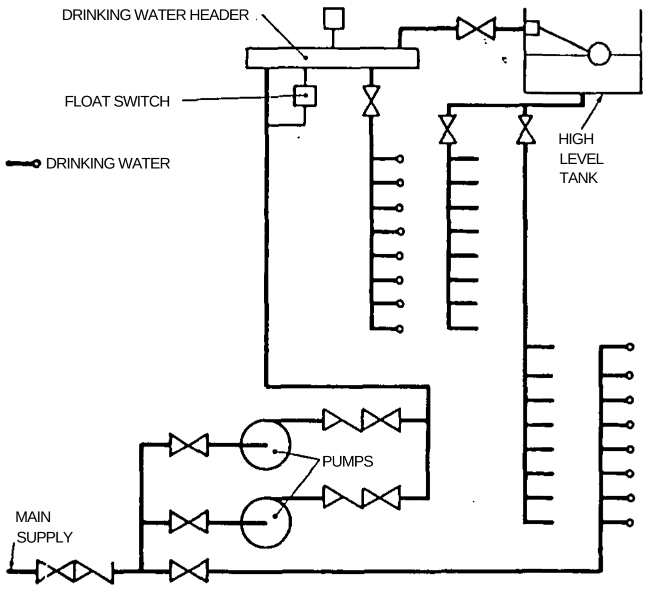



**Fig. B4.2.** Intermittent pumping system.

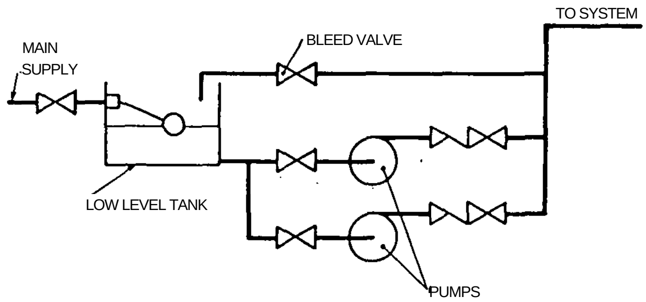



**Fig. B4.3.** Continuous pumping system.

### Charges

Installation of mains by a water undertaking to any development whether for a private user, industrial user or for a local authority, is carried out by the undertaking using its own staff or contractors; the costs being charged to the developer.

*Consumption*

A water undertaking must provide a consumer with a potable supply of water for domestic use, providing the user undertakes not to waste, misuse or contaminate that supply and to conform to the byelaws applied thereto.

Charges for water are levied on the consumer for domestic and industrial uses, but generally no charge is made for providing water for main fire-fighting purposes, except in the case of sprinkler installations, hose-reels etc.

### Metering

Water undertakings usually provide zone meters within their trunk and distribution mains to check consumption within given areas. Mains leaving reservoirs, pumping stations and water towers are metered to record the outflow. Venturi meters with recorders are often used on large mains, and orifice-type devices on medium sized mains.

Supplies to dwellings have not generally been metered in the past, but charges based on metering can be lower than those based on standing charges and rateable value, where consumption is relatively low. Supplies to industrial and commercial buildings, institutions and schools are metered.

### Fire Protection Installations

In the case of automatic fire protection installations, insurers may insist on a separate supply from the undertaking's main exclusively for fire protection. Surface boxes should be readily accessible to enable the meter to be read, and the stop valves to be operated. A meter to premises will generally be installed with a valved by-pass to enable an unmetered supply to be taken through the by-pass for a fire protection installation.

## COLD WATER STORAGE AND CONSUMPTION

### Cold Water Storage

The Model Water Byelaws[2] and a guide to these byelaws[5] give detailed information on the systems and devices necessary to prevent waste, undue consumption and contamination of water.

Care must be taken when assessing water storage as some water undertakings have special powers and, in all cases, they must be consulted. In some areas the undertaking considers that it is necessary to provide storage for one or more of the following reasons:

(*a*) to provide against interruptions of the supply caused by repairs to mains etc.,

(*b*) to reduce the maximum rate of demand on the mains,

(*c*) to limit the pressure on the distribution system, so reducing noise and waste of water due to high pressure mains, and enabling lighter and cheaper materials to be used,

(*d*) to provide a reserve of water for fire-fighting purposes during loss of a main.

Where required, water storage is usually provided to meet the 24 hour demand, although some undertakings consider reducing this to a lesser amount on application. Conversely, where the supply is irregular or inadequate or where exhaustion of the storage would have serious consequences, consideration should be given to increasing the storage capacity.

### Cold Water Consumption

Consumption of cold water in a building or group of buildings depends upon:

(*a*) the use to which the water is put,

(*b*) the number and type of fittings connected,

(*c*) the number of consumers served.

For domestic, as distinct from industrial use, storage requirements per consumer are set out in BS CP 310[6]. Further data are listed in Table B4.2. The practice of allocating storage per fitting or appliance is not now favoured and where the provision of fittings but not the extent of usage is given, an estimate of the number of consumers should be made. Table B4.3, which is a digest of the scales in BS 6465: Part 1[7] may be used for this purpose.

For industrial and commercial usage each case must be considered individually, care being taken to ensure that the demands of processes requiring large quantities of water at infrequent intervals can be met, and that this demand is referred to the water undertaking.

**Table B4.2.** Provision of domestic storage to cover 24 hours' interruption of supply.

| Type of building | Storage litres |
|---|---|
| Dwellings, houses and flats (up to 4 bedrooms) | 120/bedroom |
| Dwellings, houses and flats (more than 4 bedrooms) | 100/bedroom |
| Hostels | 90/bed |
| Hotels | 200/bed |
| Nurses' homes and medical quarters | 120/bed |
| Offices with canteens | 45/person |
| Offices without canteens | 40/person |
| Restaurants | 7/meal |
| Schools*: | |
| boarding | 90/person |
| day – primary | 15/person |
| day – middle and secondary | 20/person |

* Reference should also be made to the publications of the Department of Education and Science.

## Hospital Ward Units

Water consumption in hospitals is extremely varied, depending on the condition of the hospital and the extent of outpatient, casualty and clinic provision. Consumption has been shown to vary, for different conditions in hospitals, from 180 litre/bed day to over 1800 litre/bed day. What follows provides the basis for an assessment of cold water storage and consumption, pending further research and data acquisition on current water use. Reference should also be made to DHSS Health Technical Memorandum 27[8].

The Hospital Engineering Research Unit of the University of Glasgow carried out an investigation into the water consumption of ward units. (A ward unit is defined here as comprising all rooms which make up the working area for patient care, i.e. patient bedrooms or wards, day spaces, treatment and utility rooms, ward kitchens, staff rooms, stores and circulation spaces, etc.). Records of the investigation, together with details of the statistical analysis to which they were subjected are included in a published report[9]. Part of this report comprises a series of nomograms based upon the following equations:

Cold tank water (excluding drinking water)

$$V_{ca} = A + 243\,n \quad \ldots \ldots \ldots \ldots \quad \text{B4.1}$$

$$V_{cs} = B + 150\,n \quad \ldots \ldots \ldots \ldots \quad \text{B4.2}$$

$$M = 1 + 0{\cdot}014\,n \quad \ldots \ldots \ldots \ldots \quad \text{B4.3}$$

where:

**Table B4.3.** Allocation of sanitary appliances.

<table>
<tr><th colspan="9">Numbers of Sanitary appliances recommended for use in various types of building</th></tr>
<tr><th rowspan="2">Type of Building or Accommodation</th><th rowspan="2">Occupants</th><th colspan="2">Water Closets</th><th rowspan="2">Urinals</th><th colspan="2">Wash Basins</th><th colspan="2">Baths</th></tr>
<tr><th>Male</th><th>Female</th><th>Male</th><th>Female</th><th>Male</th><th>Female</th></tr>
<tr><td rowspan="2">Accommodation for staff in most Buildings</td><td>1-100</td><td>1 + 1 per 25</td><td>1 + 1 per 14</td><td>1 + 1 per 25</td><td>1 + 1 per 25</td><td>1 + 1 per 14</td><td rowspan="2">–</td><td rowspan="2">–</td></tr>
<tr><td>over 100</td><td>add 1 per 30 to the above</td><td>add 1 per 20 to the above</td><td>add 1 per 30 to the above</td><td>add 1 per 30 to the above</td><td>add 1 per 20 to the above</td></tr>
<tr><td rowspan="3">Accommodation for transient public</td><td>1-200</td><td rowspan="2">1 per 100</td><td>2 per 100</td><td rowspan="3">1 per 50</td><td colspan="2" rowspan="3">1 per W.C. or of W.C.s</td><td rowspan="3">–</td><td rowspan="3">–</td></tr>
<tr><td>200-400</td><td rowspan="2">add 1 per 100 to the above</td></tr>
<tr><td>over 400</td><td>add 1 per 250 to the above</td></tr>
<tr><td rowspan="3">Restaurants (Public)</td><td>1-200</td><td rowspan="2">1 per 100</td><td>2 per 100</td><td rowspan="3">1 per 25</td><td colspan="2">1 plus 1 per 25</td><td rowspan="3">–</td><td rowspan="3">–</td></tr>
<tr><td>200-400</td><td rowspan="2">add 1 per 100 to the above</td><td colspan="2" rowspan="2">add 1 per 30 to the above</td></tr>
<tr><td>over 400</td><td>add 1 per 250 to the above</td></tr>
<tr><td rowspan="4">Day Schools (Children)</td><td>1-100</td><td rowspan="2">1 per 25</td><td>1 per 10</td><td rowspan="2">1 per 10</td><td colspan="2">1 per 8</td><td rowspan="4">–</td><td rowspan="4">–</td></tr>
<tr><td>100-200</td><td rowspan="2">add 1 per 15 to the above</td><td colspan="2">add 1 per 10 to the above</td></tr>
<tr><td>200-300</td><td rowspan="2">add 1 per 30 to the above</td><td rowspan="2">add 1 per 12 to the above</td><td colspan="2" rowspan="2">add 1 per 12 to the above</td></tr>
<tr><td>over 300</td><td>add 1 per 25 to the above</td></tr>
<tr><td rowspan="2">Boarding Schools (extra over Day)</td><td>1-100</td><td colspan="2" rowspan="2">1 per 5</td><td>1 per 10</td><td colspan="2" rowspan="2">1 per 3</td><td rowspan="2">1 per 30</td><td rowspan="2">1 per 5</td></tr>
<tr><td>over 100</td><td>add 1 per 12</td></tr>
<tr><td>Nursery Schools</td><td>–</td><td colspan="2">1 per 6</td><td>–</td><td colspan="2">1 per 5</td><td colspan="2">1 per 40</td></tr>
<tr><td>Hotels (Residential Areas)</td><td>–</td><td colspan="2">1 per 9 rooms + any en suite</td><td>–</td><td colspan="2">1 per bathroom, 1 per bedroom + 1 per W.C.</td><td colspan="2">1 per 9 rooms + any en suite</td></tr>
</table>

$V_{ca}$ = average daily cold water consumption litre

$V_{cs}$ = cold water store .. .. .. litre

$M$ = rate of mains water supply .. .. litre/s

$A,B$ = constants for different types of ward, see Table B4.4

$n$ = number of beds in ward

By making certain asssumptions as to the average relationship between number of beds and quantity of equipment it has been found possible to modify the published equations to produce data more suitable for inclusion in this Section. Tables B4.5 and 6 list consumptions etc., for a series of ward units.

**Table B4.4.** Numerical constants for ward types.

| Type of ward | Values for constants | |
|---|---|---|
| | *A* | *E* |
| Surgical and gynaecological | +664 | +518 |
| Medical | −627 | −750 |
| Obstetric | +3987 | +1691 |
| Geriatric and chronic | −2482 | −786 |
| Thoracic | −941 | −1277 |
| Paediatric and infectious | +1405 | +868 |
| Orthopaedic | −1346 | −368 |

## HOT WATER STORAGE, CONSUMPTION AND PLANT SIZING

### Statutory Requirements and Codes of Practice

The Education (School Premises) Regulations require a hot water temperature not greater than 43·5°C at the point of use. This can be achieved by storage at a higher temperature and mixing at the outlet.

Both the Factories Act and the Offices, Shops and Railway Premises Act include a requirement for hot water but no particular temperature or quantity to be stored is stated. The mandatory requirements of these Acts are monitored by the Health and Safety Executive.

BS CP 324.202[10] and BS CP 342[11] include some guidance on storage quantities, but more recently acquired data are set out within this Section. BS 5918[12] is relevant for sizing solar heating or preheating systems.

*Unvented Systems*

Recent changes in the Model Water Byelaws[2] and the Building Regulations[1] permit the use of unvented hot water systems subject to safety considerations. A summary of the differences between vented and unvented systems and an explanation of the need for safety devices are given in BRE Digest 308[13], and details of the design, installation, testing and maintenance of such systems will be contained in BS 6700[14]. A method

**Table B4.5.** Ward units – average daily consumption of cold tank water.

| Number of beds | Consumption per ward unit ($m^2$/day) | | | | | | |
|---|---|---|---|---|---|---|---|
| | Surgical and gynaecological | Medical | Obstetric | Geriatric and chronic | Thoracic | Paediatric and infectious | Orthopaedic |
| 20 | 5·55 | 4·27 | 8·86 | 2·41 | 3·95 | 6·32 | 3·54 |
| 25 | 6·77 | 5·50 | 10·1 | 3·64 | 5·18 | 7·55 | 4·77 |
| 30 | 8·00 | 6·73 | 11·3 | 4·86 | 6·41 | 8·77 | 6·00 |
| 35 | 9·23 | 7·95 | 12·5 | 6·09 | 7·64 | 10·0 | 7·23 |
| 40 | 10·4 | 9·18 | 13·8 | 7·32 | 8·86 | 11·2 | 8·46 |
| 45 | 11·7 | 10·4 | 15·0 | 8·55 | 10·1 | 12·5 | 9·68 |
| 50 | 12·9 | 11·6 | 16·2 | 9·77 | 11·3 | 13·7 | 10·9 |
| 55 | 14·1 | 12·9 | 17·4 | 11·0 | 12·5 | 14·9 | 12·1 |
| 60 | 15·4 | 14·1 | 18·7 | 12·2 | 13·8 | 16·1 | 13·4 |

**Table B4.6.** Ward units – cold tank storage and mains supply.

| Number of beds | Mains supply to tank per ward unit (litre/s) | Cold tank storage per ward unit* ($m^2$) | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | | Surgical and gynaecological | Medical | Obstetric | Geriatric and chronic | Thoracic | Paediatric and infectious | Orthopaedic |
| 20 | 1·26 | 3·50 | 2·27 | 4·68 | 2·23 | 1·73 | 3·86 | 2·64 |
| 25 | 1·33 | 4·25 | 3·02 | 5·43 | 2·98 | 2·48 | 4·61 | 3·39 |
| 30 | 1·39 | 5·00 | 3·77 | 6·18 | 3·73 | 3·23 | 5·36 | 4·14 |
| 35 | 1·47 | 5·75 | 4·52 | 6·93 | 4·48 | 3·98 | 6·11 | 4·89 |
| 40 | 1·54 | 6·50 | 5·27 | 7·68 | 5·23 | 4·73 | 6·86 | 5·64 |
| 45 | 1·61 | 7·25 | 6·02 | 8·43 | 5·98 | 5·48 | 7·61 | 6·39 |
| 50 | 1·68 | 8·00 | 6·77 | 9·18 | 6·73 | 6·23 | 8·36 | 7·14 |
| 55 | 1·75 | 8·75 | 7·52 | 9·93 | 7·48 | 6·98 | 9·11 | 7·89 |
| 60 | 1·82 | 9·50 | 8·27 | 10·7 | 8·23 | 7·73 | 9·86 | 8·64 |

* The cold water storage quantities listed are those required to provide standby against failure of mains supply over the 8 hours of heaviest demands. For outlying hospitals, standby against failure over 24 hours should be considered.

of assessment and testing of the installation and performance of unvented systems is published by the British Board of Agrement[15].

**Measured Consumption Data**

*Domestic Premises*

Several studies have measured use of hot water in domestic premises and some results are presented in Table B4.7. Individual households have a wide variation in consumption, though household size has been shown to be a factor. Fig. B4.4 gives some guidance to allow for this variation, in particular giving the consumption which would satisfy the average needs of about 90% of households. This line is derived from the three most recent house monitoring studies.

*Commercial Premises*

Table B4.8 shows the average and maximum consumptions of hot water measured in recent field trials[24]. The results are given in terms of building type and are separated into total hot water and that used for service and catering purposes. The figures for service provide a guide to the amounts used for washing, bathing and cleaning. The number of people in shops was taken as the number of staff.

**Table B4.7.** Measured daily hot water consumption in dwellings.

| Study | Normalised average consumption (litre/person day) | Notes on original data |
|---|---|---|
| NBS[16] | | |
| independent boiler | 48 | 43 litre/person day normalised to 60°C |
| BRE[17] | | |
| large flats | 49 | 53 litre/person day at 52°C |
| small flats | 47 | 50 litre/person day at 52°C |
| BRE[18] | | |
| site A | 41 | 27 litre/person day at 78°C |
| site B | 69 | 71 litre/person day at 54°C |
| site C | 61 | 43 litre/person day at 74°C |
| BRE[19] | | |
| site A | (36)* | 94 litre/dwelling day |
| site B | (28)* | 72 litre/dwelling day |
| BRE[20] | | |
| 1/2 person | 44 | Delivered temperatures approximately 55°C |
| 3 person | 37 | |
| 4+ person | 30 | |
| British Gas[21] | | |
| 1 person | 63 | Temperatures not published. Results here assume water was delivered at 55°C |
| 2 person | 44 | |
| 3 person | 38 | |
| 4 person | 34 | |
| Electricity Council[22] | | |
| 10 terraced | 37 | Normalised to 55·5°C hot/cold temperature difference. |
| 106 dwellings | 43 | From energy consumption |
| BRECSU[23] | | |
| 62 1 and 2 person flats for the elderly | 44 | 43 litre/person day at mean temperature of 54°C |

* Assuming 3 persons per dwelling and hot water at 62°C.

*Note:* Normalised assuming hot delivery at 55°C and cold supply temperature of 10°C.

**Daily Demand Data**

*Domestic Premises*

In studies, large variations in demand have been found to occur between similar sized households, and on different

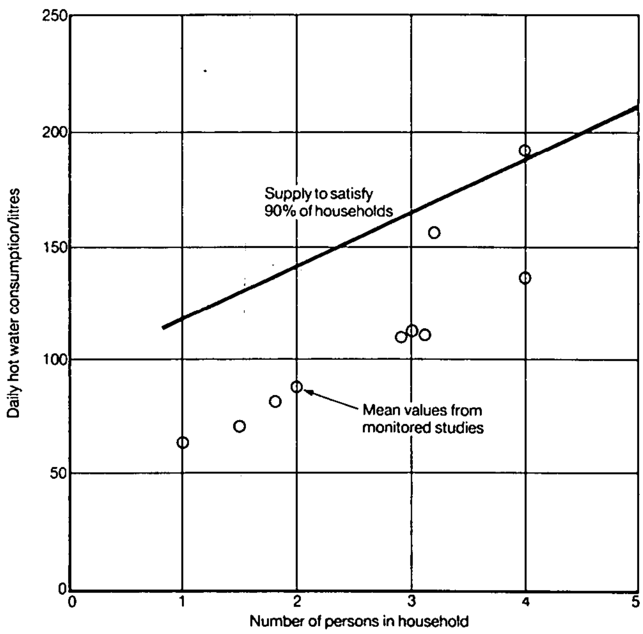



**Fig. B4.4.** Relationship between measured daily consumption and household size.

**Table B4.8.** Daily hot water consumption for various types of commercial buildings.

| Building type | | Total daily hot water consumption litre/person | Service litre/person | Catering litre/meal | Catering As % of total |
|---|---|---|---|---|---|
| Schools and colleges | maximum | 13 | 7 | 18 | 85 |
| | average | 6 | 3 | 6 | 53 |
| Hotels and hostels | maximum | 464 | 303 | 62 | 70 |
| | average | 137 | 80 | 14 | 28 |
| Restaurants | maximum | 17 | 10 | 73 | 95 |
| | average | 7 | 3 | 8(4)* | 60 |
| Offices | maximum | 26 | 10 | 33 | 87 |
| | average | 8 | 3 | 10 | 48 |
| Large shops | maximum | 25 | 6 | 45 | 91 |
| | average | 10 | 4 | 8 | 57 |

* 4 litre/meal is the average consumption in restaurants without large bar facilities.

*Note:* Normalised assuming 65°C storage and 10°C cold feed temperature.

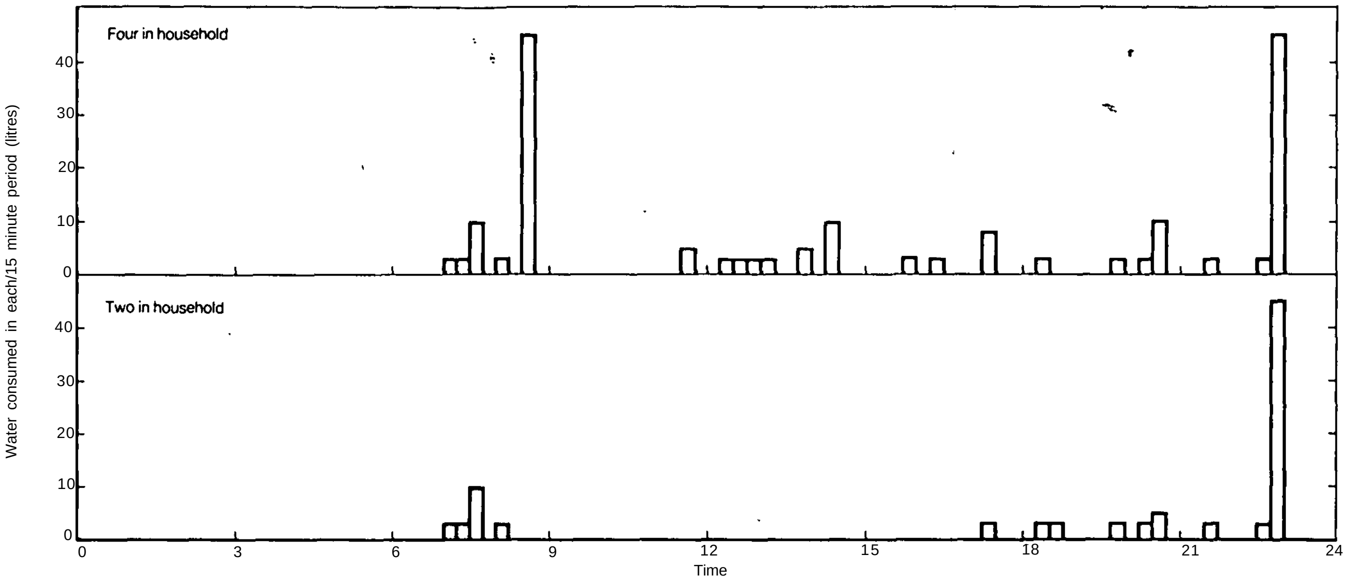



**Fig. B4.5.** Examples of draw-off patterns used in experimental studies.

days for the same household. Therefore, to satisfy the majority of needs, some general oversizing will be inevitable. When averaging the use of a large number of households a demand pattern may emerge, but any typical pattern chosen for illustrative or experimental purposes will only roughly represent a small minority of individual households. Even so, for studying the energy efficiency and adequacy of systems, it is necessary to assume some typical patterns. Two such patterns are shown in Fig. B4.5. The first represents a household of four where at least one person is present for much of the day; the second, a household of two, both of whom are assumed to be away during working hours.

Usually, a system would only be considered adequate if it could meet both these types of demand, with sufficient capacity for occasional needs such as being able to cope with two baths (equivalent to providing 90 litres at 55°C) in a relatively short time.

*Commercial Premises*

Measured quantities of hot water consumption should not stand alone as a sizing guide. The rate at which these amounts are drawn off must also be considered. This is done by relating the patterns of consumption in each day to the size of plant required. Using a mathematical mode1[24], the optimum size of boiler and storage can be matched to each demand pattern. Typical examples of daily demand are shown in Fig. B4.6, and these show the large differences in the way hot water is used in various establishments.

**Medical Aspects**

Many types of bacteria, most of them quite harmless to people, can multiply prolifically in “clean” water. This is neither a new observation nor should it cause any general concern. However, in recent years hot water supply systems have been implicated in outbreaks of legionnaires’ disease.

Although legionnaires’ disease can be associated with the cooling towers of air conditioning systems, evidence from the UK[25] and overseas[26] indicates that hot water service systems can also be a source of infection.

The following design parameters or operational characteristics of hot water systems are thought favourable to significant colonisation and should be avoided:

(*a*) dirt and sludge in cisterns and calorifiers,

(*b*) storage and/or distribution temperatures in the range 25–50°C,

(*c*) large volumes of static water or small ratios of water use to system volume.

Guidance on minimising the risk of legionnaires’ disease is given in CIBSE Technical Memorandum 13[27].

**Energy Efficiency**

In modern, well insulated dwellings hot water services can consume as much energy annually as space heating. In larger buildings the energy saving potential of separating hot water and space heating services is well recognised[28].

Factors to be considered include:

*System Design and Choice of Heating Plant*

To a considerable extent these are influenced by the types of fuel available. Also highly relevant are the expected load pattern and total usage of hot water. Whilst instantaneous heaters (usually gas or electric) can offer low standing losses and a short recovery period, and may be ideal for infrequently used and relatively isolated demand points, more conventional central plant can offer lower capital costs for high volume applications.

Consideration should be given to providing each demand point or group of demand points with its own heating and storage system rather than incur the heat loss penalties of long deadlegs or secondary circulation loops.

*Control*

Appropriate control of stored water temperature and time of heating or circulation are essential. Depending on the system type, control may achieve:

(*a*) lock out of electrical supplies during hours of zero demand by using time clocks.

(*b*) avoidance of excessive temperatures (but see *Medical Aspects*) by using cylinder thermostats,

(*c*) avoidance of boiler short cycling.

*Thermal Insulation*

An appropriate level of thermal insulation[28] should be provided to all storage vessels and primary pipework (see Section C3).

*Avoidance of Waste and Good Housekeeping*

In some building types considerable economy may be obtained by ensuring that users do not waste hot water — timed flow, self-closing, non-concussive taps or flow restrictors are possible aids. Spray taps and showers should be considered, bearing in mind medical aspects.

Boiler plant should be checked at least annually to ensure optimum combustion efficiency, and regular inspection to check for leaking taps etc. is worthwhile. Other factors that may be relevant in some buildings are descaling and water softening. Metering the use of hot water can help to identify any wasteful practices and is recommended for large buildings.

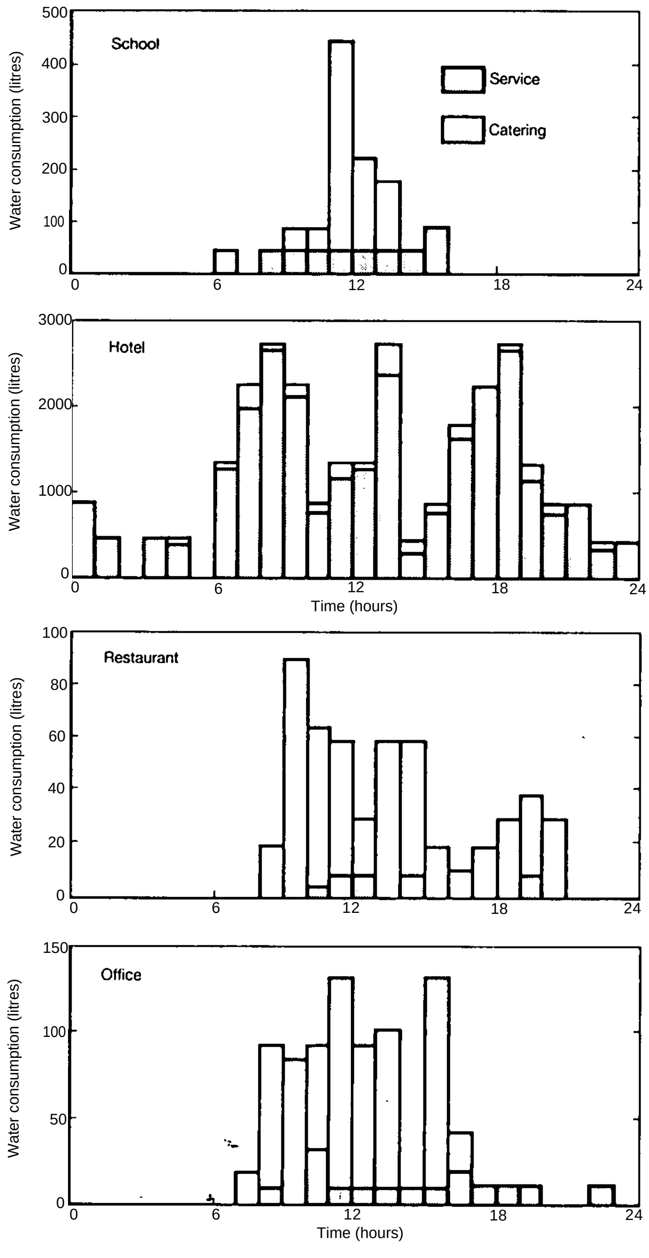



**Fig. B4.6.** Examples of daily demand patterns.

*Mixing and Stratification*

Efficient hot water storage depends on stratification within the vessel such that the lower density hot water remains at the top of the vessel ready for use, whilst the cold replenishment water enters and remains at lower levels. Without proper design even short bursts of draw-off at high flow rates may produce sufficient turbulence to cause mixing. Mixing will tend to be less in larger, especially taller vessels. Thus the traditional cylinder with a height at least twice the diameter is highly suitable.

Devices such as baffles within vessels or cold water inlets which cause less turbulence and reduce mixing are available. Where supplies to two or more appliances with high demands are required simultaneously, e.g. two baths fed from one cylinder, such devices may be essential.

*Domestic Installations*

The typical hot water system for a dwelling with a gas, solid fuel or oil-fired boiler comprises a separate hot water storage vessel which is heated indirectly by a heat exchanger within the vessel. A pump or gravity feed is used to drive the primary hot water from boiler to exchanger. Often the boiler will also provide space heating and therefore the specified output greatly exceeds that required for hot water service.

The sizing of heat exchangers not only influences the recovery rate of the storage, but also affects the system efficiency. Exchangers which can transfer only a small fraction of the boiler output can lead to excessive cycling of the boiler, particularly when the space heating is not in use. Full benefit of high transfer heat exchangers requires correct application of controls.

A number of manufacturers now produce cylinders to meet the requirements of an electricity supply industry specification[29]. This requires cylinders to be made in accordance with the appropriate British Standards (BS 699[30] and BS 1566[31]) with the following improvements:

(*a*) The cylinder is fitted with two side entry immersion heater bosses.

(*b*) A baffle is fitted to the cold feed entry to minimise mixing of hot and cold water.

(*c*) The cylinder is fitted with an aluminium protector rod to minimise corrosion.

(*d*) For a difference of 45 K between storage and ambient temperatures, the 24 hour standing losses shall not exceed those given in Table B4.9.

**Table B4.9.** Standing losses from cylinders meeting specification EC4665[29].

| Capacity of hot water storage litres | Maximum 24 hour heat loss kWh |
|---|---|
| 120 | 2·4 |
| 144 | 2·5 |
| 210 | 2·7 |

**Method of Plant Sizing**

*Commercial Premises*

Sizing of hot water plant is based essentially on the load anticipated for both hot water use and system heat losses. These are in turn affected by the storage and delivery temperatures.

In some systems, heat losses are small, with most of the input energy delivered at the point of use. Others may lose up to 90% of input energy in heat losses. In general, well designed systems should have low heat losses, but exceptions can occur where existing pipework cannot be extensively insulated and new hot water plant is to be installed.

*System Design*

Information on the likely consumption of hot water for various purposes within different building types is given under *Measured Consumption Data.*

At an early stage it must be judged whether or not the particular building is likely to have a below or above average consumption in terms of both average demand and peak demand. The consequences of occasional failure to provide water at the required temperature must also be assessed. Inherent in these judgements must be a consideration of the system recovery time.

The most appropriate delivery temperatures for the hot water, and whether a unique temperature is suitable for all delivery points should be considered. Relevant considerations are energy efficiency (including consideration of system losses), health and safety aspects.

The proposed control of the system must be carefully considered. This may be exercised by local stand-alone controllers or as part of an integrated management system. In either case, the requirements will usually be to provide hot water only during those hours that the building occupants may reasonably require it, and, if possible, to utilise fuel at the most favourable tariffs.

Following selection of several system designs from those available, the designer should calculate, for each option, both the capital and likely running costs. For each option it should be clear what is:

(*a*) the recovery time,

(*b*) the capital cost,

(*c*) the total likely running costs and how sensitive these are to actual hot water demand,

(*d*) the likelihood of failure to provide service owing to abnormal loading,

(*e*) implications for siting of plant, wash rooms etc., and any savings that could be made by alterations in building layout.

*Use of Plant Sizing Curves*

Figs. B4.7 to 17 have been prepared to provide guidance on the total hot water requirements of specific building types, within the stated limits. Non-standard building uses will require special consideration, and data for hospitals is given below under *Hospitals.* Separate curves are provided for catering and service (i.e. uses other than catering).

Different methods of using the curves apply for constant tariff fuels (e.g. gas) compared with those fuels incorporating off-peak tariffs (e.g. electricity).

On selection of a recovery period, the curves provide the basis to calculate (for a storage temperature of 65°C):

(*a*) boiler power (excluding system heat losses),

(*b*) hot water storage capacity.

*Procedure for Constant Tariff Fuel Systems*

1. Select a recovery period (a recovery period of less than half an hour can seldom be achieved with indirectly heated systems).

2. Apply selected recovery period to curve (*a*) for building type and area of demand (i.e. catering or service).

   Read off:

   (*a*) boiler output, $q$, in kW per person or per meal,

   (*b*) hot water storage capacity, $v$, in litres per person or per meal.

3. Adjust the boiler output if the required hot water temperature differs from 65°C:

$$q' = q \frac{(y - a)}{(65 - a)} \quad \text{.. .. ..} \quad \text{B4.4}$$

   where:

   $q'$ = adjusted boiler output rating kW/person or meal

   $y$ = hot water storage temperature required .. .. .. °C

   $a$ = cold feed temperature .. °C (if unknown assume 10°C)

4. (*a*) Calculate boiler output rating, $Q$, (in kW) and total storage capacity, $V$, (in litres), using numbers of people or meals applicable. Where one central plant is provided for both catering and service demands, the ratings and capacities should be added.

   (*b*) Add rate of system heat loss to $Q$ to give an adjusted rate, $Q'$.

5. It is assumed in the above stages that the design of the vessel incorporates devices to inhibit mixing of incoming cold water with hot water. If the vessel does not have such a device then an increase in the storage volume is necessary to allow for mixing. Previous practice included a 25% allowance to take account of this effect, but this may be larger than necessary.

*Example 1 (Constant Tariff Fuels)*

It is required to size the hot water plant for a school which has 500 pupils and staff and serves 400 meals per day. The required hot water temperature for service is 55°C and for catering, 65°C. The kitchens are in a separate building and therefore require separate plant.

1. A recovery period is chosen to suit the type of building. For the purposes of this example the recovery time will be taken as two hours for service and one hour for catering.

2. From the schools curve (*a*) in Fig. B4.7:

   $q$ = 0·035 kW/person
   $v$ = 1·1 litre/person

   From the schools curve (*a*) in Fig. B4.8:

   $q$ = 0·1 kW/meal
   $v$ = 1·6 litre/meal

3. $$q' = 0{\cdot}035 \times \frac{(55 - 10)}{(65 - 10)} = 0{\cdot}029 \text{ kW/person}$$

4. (*a*) For the service requirement:

   $Q$ = 0·029 × 500 = 14.5 kW
   $V$ = 1·1 × 500 = 550 litres

   For the catering requirement:

   $Q$ = 0·1 × 400 = 40 kW
   $V$ = 1·6 × 400 = 640 litres

   (*b*) System heat losses would be calculated for the pipework involved. For the purpose of this example the following are assumed:

   service system heat loss = 4·0 kW
   catering system heat loss = 2·5 kW

   Adjusted boiler output ratings:

   For service $Q'$ = 14·5 + 4 = 18.5 kW
   For catering $Q'$ = 40 + 2.5 = 42.5 kW

5. Add 25% allowance for mixing of incoming water with hot water, following consultation with vessel manufacturer. Final hot water storage capacities are:

   550 + 137 = 687 litres for service
   640 + 160 = 800 litres for catering

*Procedure for Fuel Tariff Systems incorporating Off-peak Periods*

1. Select a recovery period.

2. Apply selected recovery period to curve (*a*) for building type and area of demand (i.e. catering or service).

   Read off:

   (*a*) upper element rating, $q_u$, in kW per person or per meal.

   (*b*) hot water storage capacity above the upper element, $v_u$, in litres per person or per meal.

3. Adjust upper element rating where temperature of hot water is required to differ from that of the curves.

$$q_u' = q_u \frac{(y - a)}{(65 - a)} \quad .. \quad .. \quad .. \qquad \text{B4.5}$$

where:

$q_u'$ = adjusted upper element rating kW/person or meal
$y$ = hot water storage temperature required .. .. .. °C
$a$ = cold feed temperature .. °C (if unknown use 10°C)

4. Apply $v_u$ to curves (*b*) and project horizontally. This horizontal line intersects sloping lines to represent values of percentage of annual consumption taken at on-peak rate. Each intersect represents an option which can be selected after a comparison of associated capital costs, operating costs and spatial considerations.

   For the selected on-peak energy percentage value, project upwards and downwards from the intersect and read off:

   (*a*) lower element rating, $q_l$, in kW per person or per meal,

   (*b*) hot water storage capacity above lower element, $v_l$, in litres per person or per meal. This volume of stored water includes that stored above the upper element, $v_u$.

5. Calculate, in each case using the number of people or meals applicable:

   (*a*) storage capacity above lower element, $V_l$, in litres,

   (*b*) storage capacity above upper element, $V_u$, in litres,

   (*c*) lower element rating, $Q_l$, in kW,

   (*d*) upper element rating, $Q_u$, in kW.

   Record the percentage annual consumption taken at on-peak rate from which these values derive.

6. Add rate of system heat loss, $L$, to $Q_u$ to give an adjusted upper element rating, $Q_u'$.

7. Express system heat loss in terms of “equivalent volume”, $E$, where:

$$E = \frac{L\ n\ 3600}{4{\cdot}2\ (y - a)} \quad .. \quad .. \quad .. \qquad \text{B4.6}$$

where:

$E$ = equivalent volume .. .. litre
$L$ = rate of system heat loss .. kW
$n$ = time of operation of system h
(4·2 is the specific heat of water in kJ/kg K, 3600 is the number of seconds in an hour)

8. Calculate the increase in $V_l$ to allow for system heat losses:

$$I = \frac{(100 - P) \times E}{100} \quad .. \quad .. \quad .. \qquad \text{B4.7}$$

where:

$I$ = increase in hot water storage capacity .. .. .. litre

$P$ = annual on-peak proportion selected .. .. .. %

Add $I$ to $V_l$, to give the adjusted capacity above lower element, $V_l'$.

9. Calculate the adjustment necessary to $Q_l$:

$$Q_l' = Q_l + \left( \frac{I}{V_l} \text{ X } Q_l \right) \quad .. \quad .. \qquad \text{B4.8}$$

where:

$Q_l'$ = adjusted lower element rating kW

10. It is assumed in the above stages that the design of the vessel incorporates devices to inhibit mixing of incoming cold water with hot water. If the vessel does not have such a device then an increase in the storage volume is necessary to allow for mixing. Previous practice included a 25% allowance to take account of this effect, but this may be larger than necessary.

*Example 2 (Off-peak Tariff)*

An office with 120 staff requires a hot water heater to supply water at 55°C to wash basins. It is estimated that 20% of annual consumption will be taken at on-peak rate.

1. A two hour recovery period is selected.

2. From the office curve (*a*) in Fig. B4.13:

$q_u$ = 0·04 kW/person
$v_u$ = 1·2 litre/person

3. $$q_u' = 0{\cdot}04 \times \frac{(55 - 10)}{(65 - 10)} = 0{\cdot}033 \text{ kW/person}$$

4. From the office curve (*b*) in Fig. B4.13 and for 20% on-peak rate:

$q_l$ = 0·042 kW/person
$v_l$ = 4·2 litre/person

5. $V_l$ = 4·2 × 120 = 504 litres
$V_u$ = 1·2 × 120 = 144 litres
$Q_l$ = 0·042 × 120 = 5·04 kW
$Q_u$ = 0·033 × 120 = 3·96 kW

6. Assume system heat loss, $L$, has been calculated as 0·8 kW

$Q_u'$ = 3·96 + 0·8 = 4·76 kW

7. For a 10 hour day,

$$E = \frac{0{\cdot}8 \times 10 \times 3600}{4{\cdot}2 \times (55 - 10)} = 152 \text{ litres}$$

8. $$I = \frac{(100 - 20)}{100} \times 152 = 122 \text{ litres}$$

$V_l'$ = 504 + 122 = 626 litres

9. $$Q_l' = 5{\cdot}04 + \left( \frac{122}{504} \times 5{\cdot}04 \right) = 6{\cdot}26 \text{ kW}$$

10. Add 25% allowance for mixing of incoming cold water with hot water. Final hot water storage capacity is:

626 + 152 = 778 litres

## Plant Sizing Curves

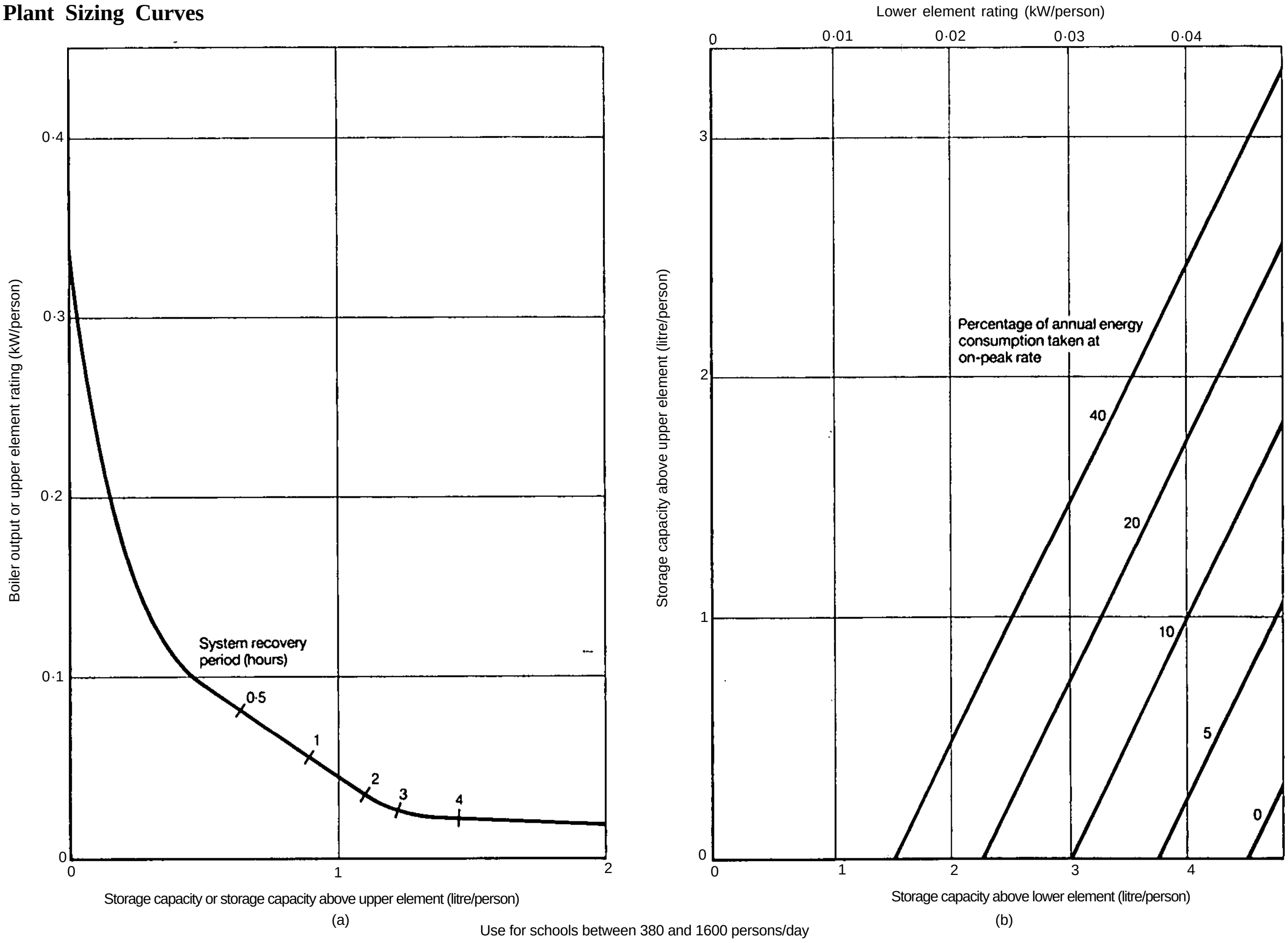



**Fig. B4.7.** Plant sizing guide for schools – service.

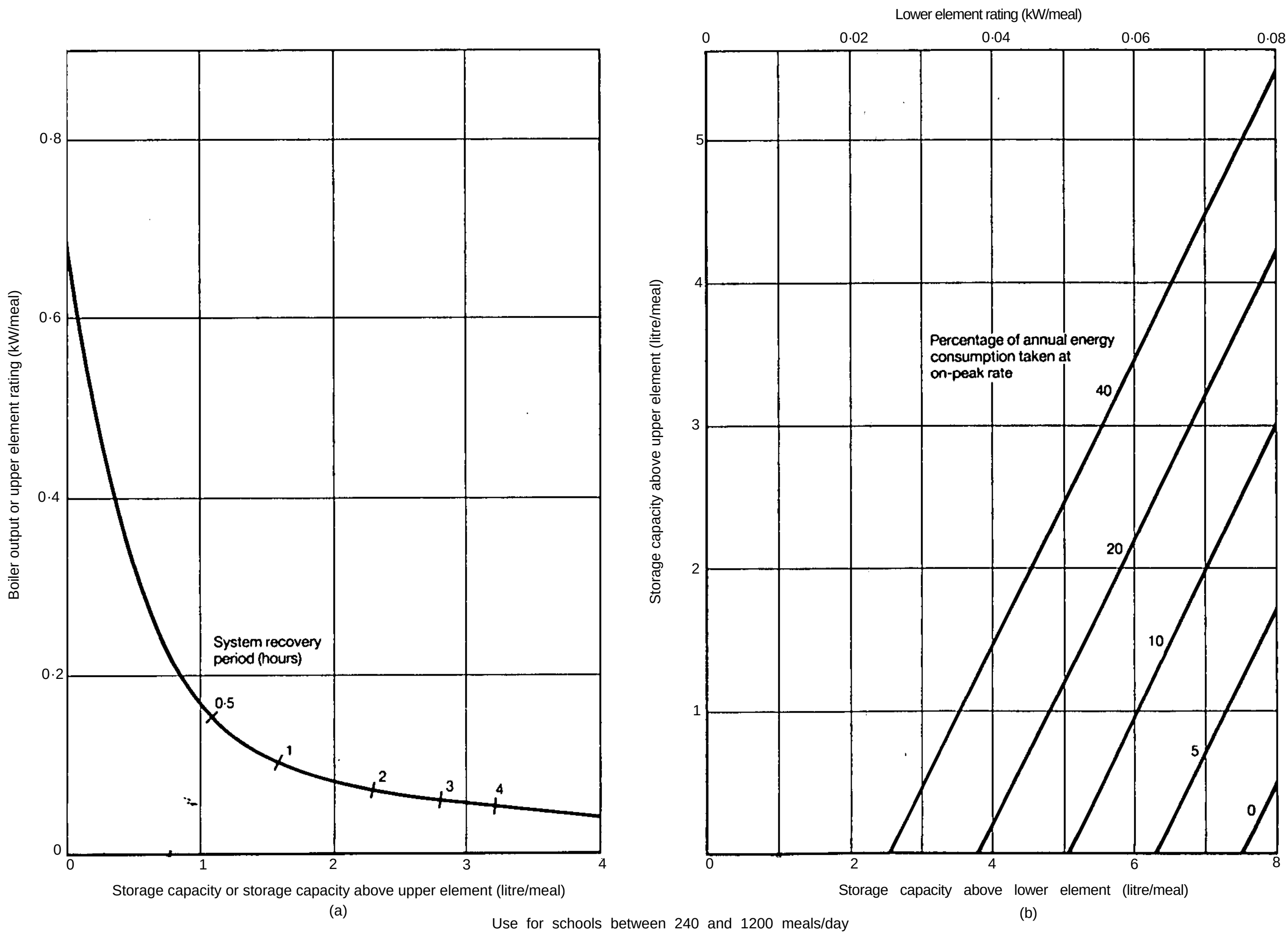
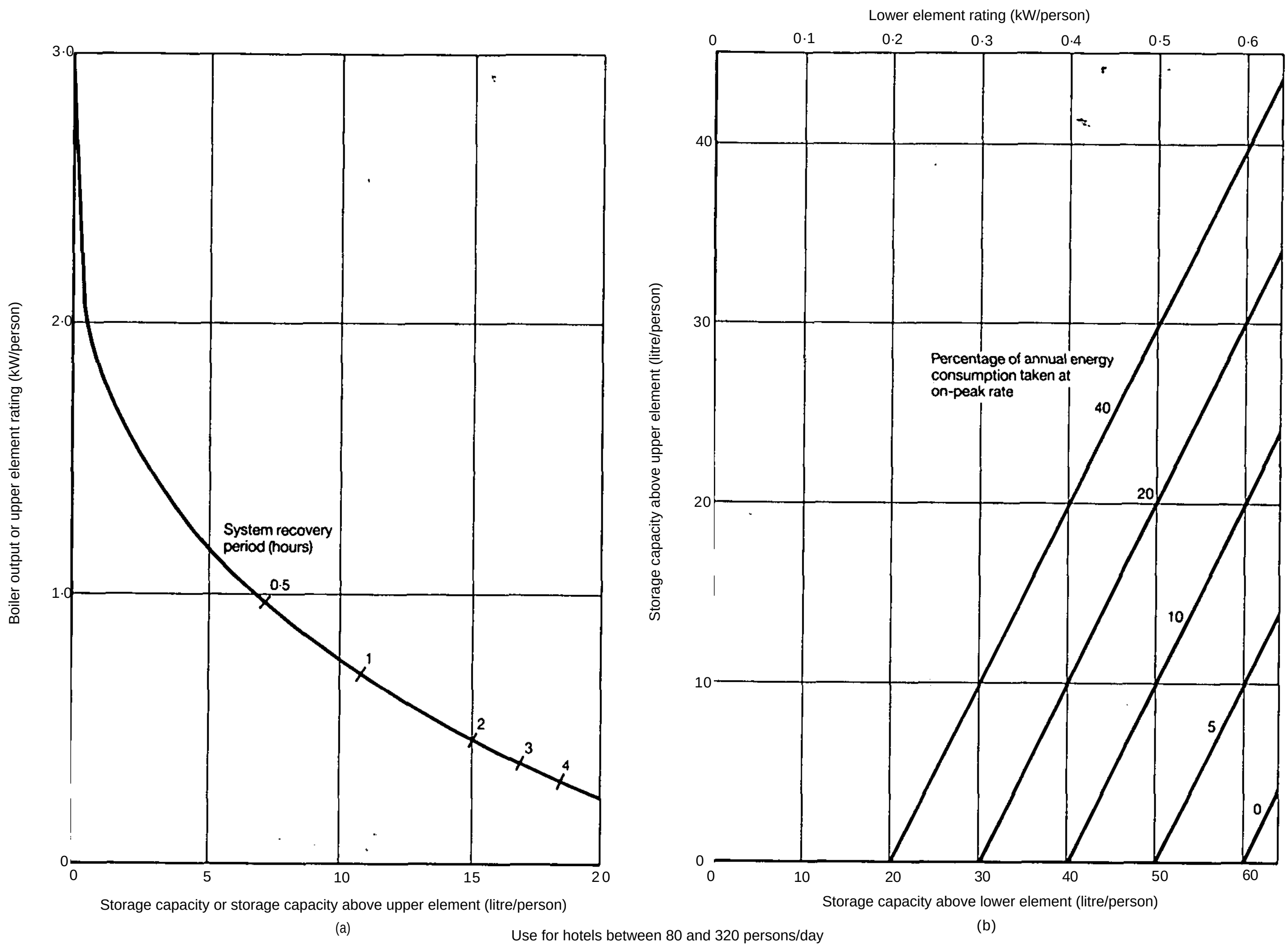
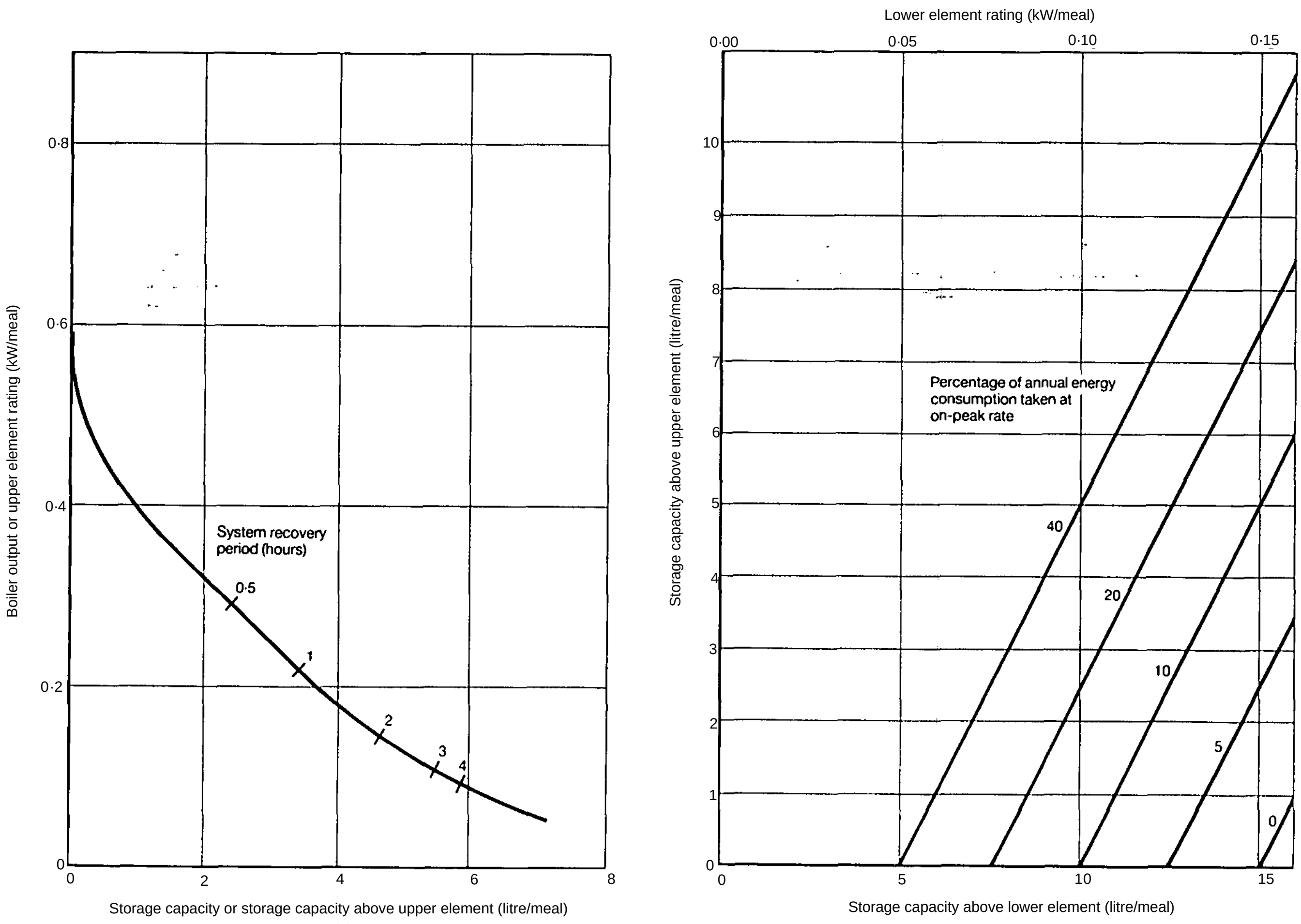



**Fig. B4.8.** Plant sizing guide for schools – catering.



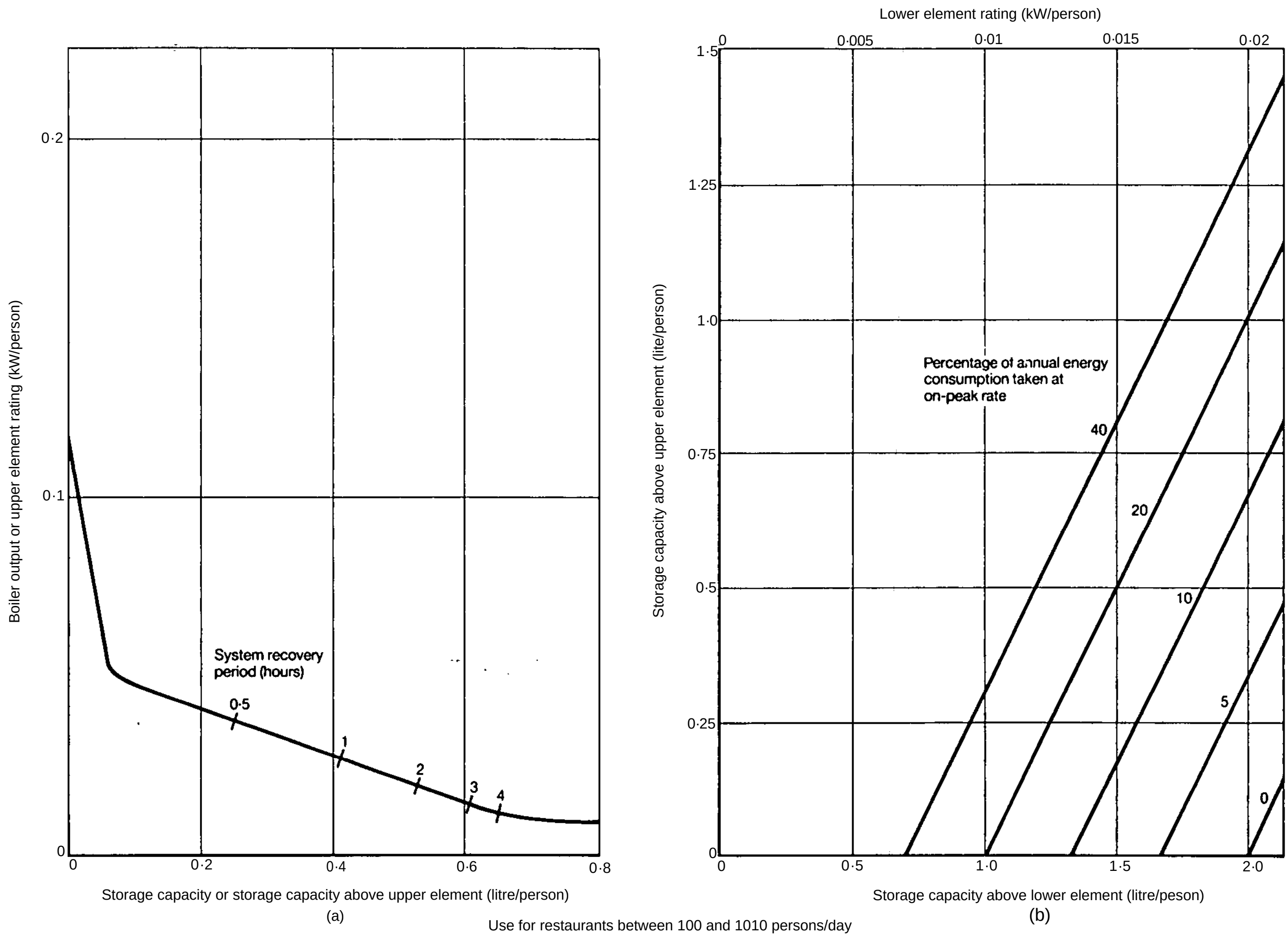

Use for hotels between 80 and 320 persons/day

**Fig. B4.9.** Plant sizing guide for hotels – service.



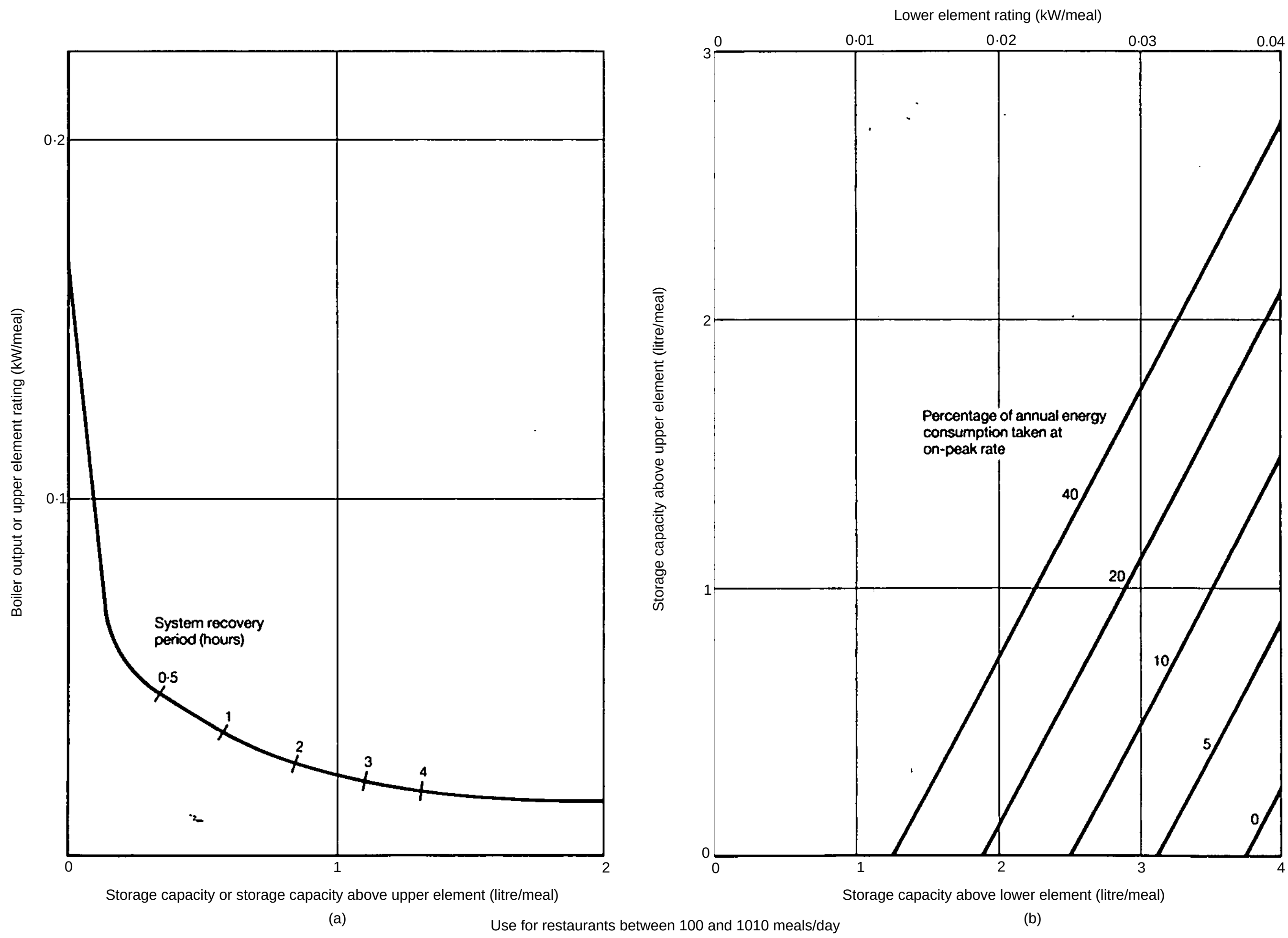

Use for hotels between 140 and 640 meals/day

**Fig. B4.10.** Plant sizing guide for hotels – catering.




Use for restaurants between 100 and 1010 persons/day

**Fig. B4.11.** Plant sizing guide for restaurants – service.




Use for restaurants between 100 and 1010 meals/day

**Fig. B4.12.** Plant sizing guide for restaurants – catering.

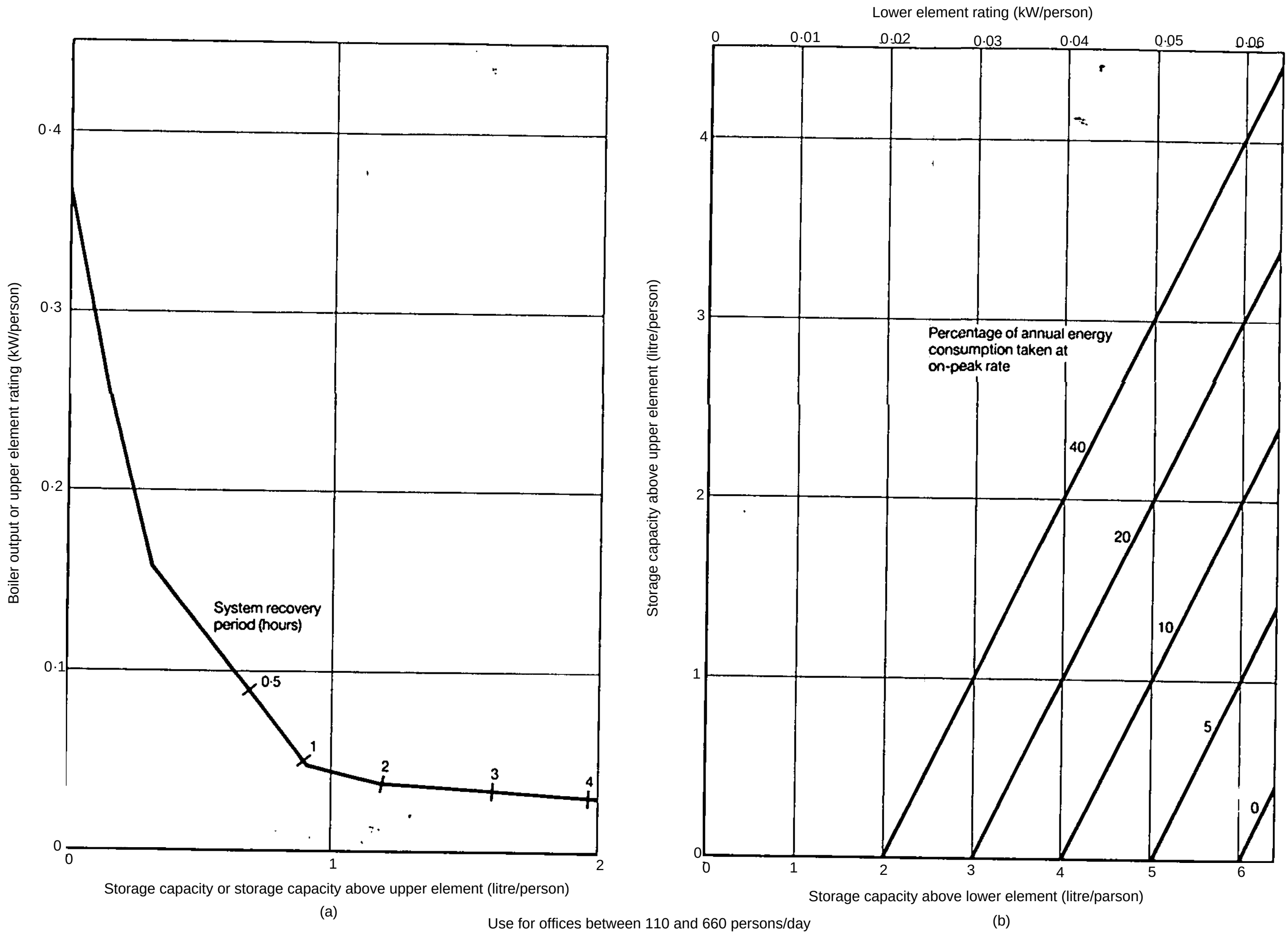



**Fig. B4.13.** Plant sizing guide for offices – service.

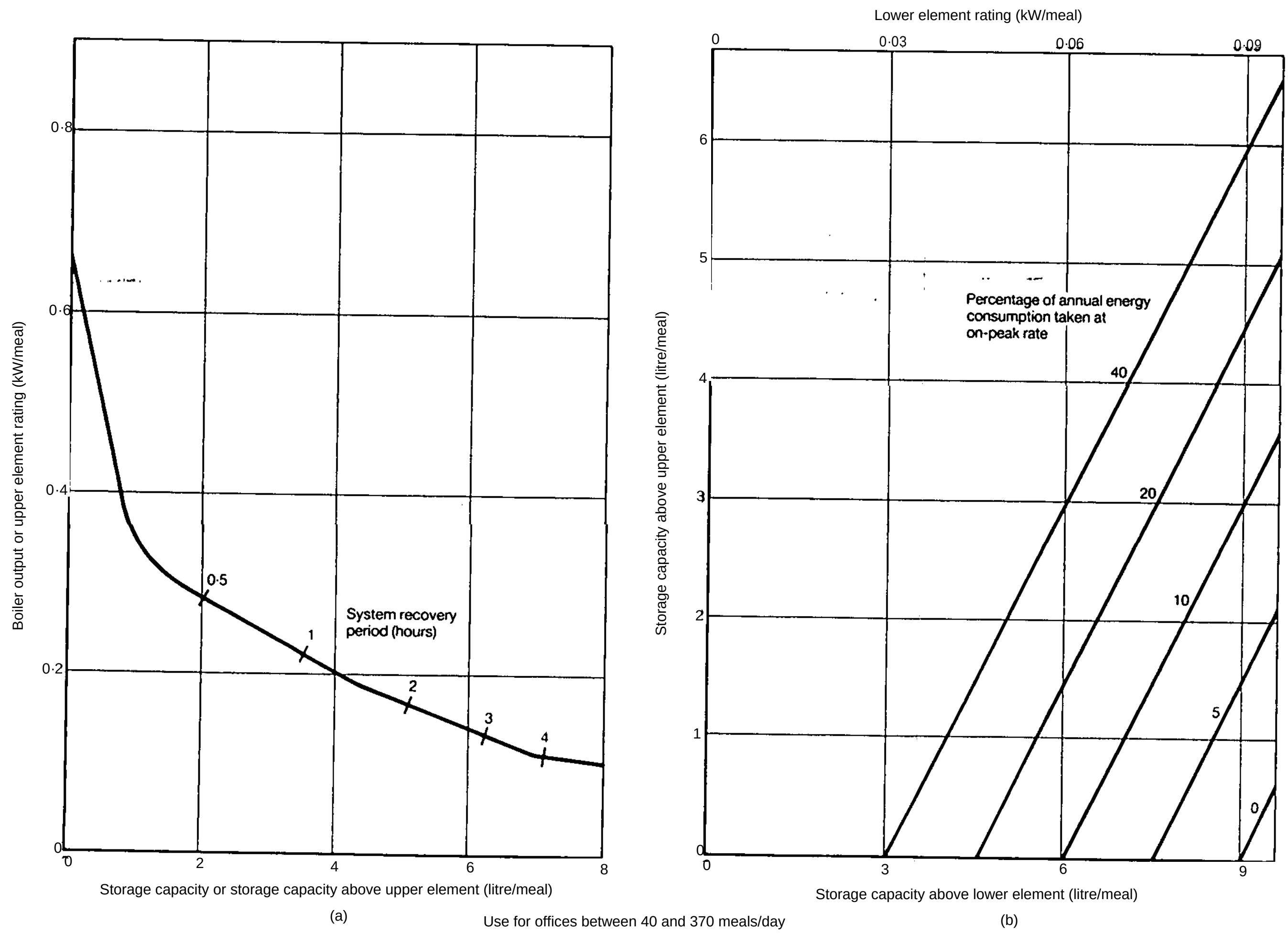



**Fig. B4.14.** Plant sizing guide for offices – catering.

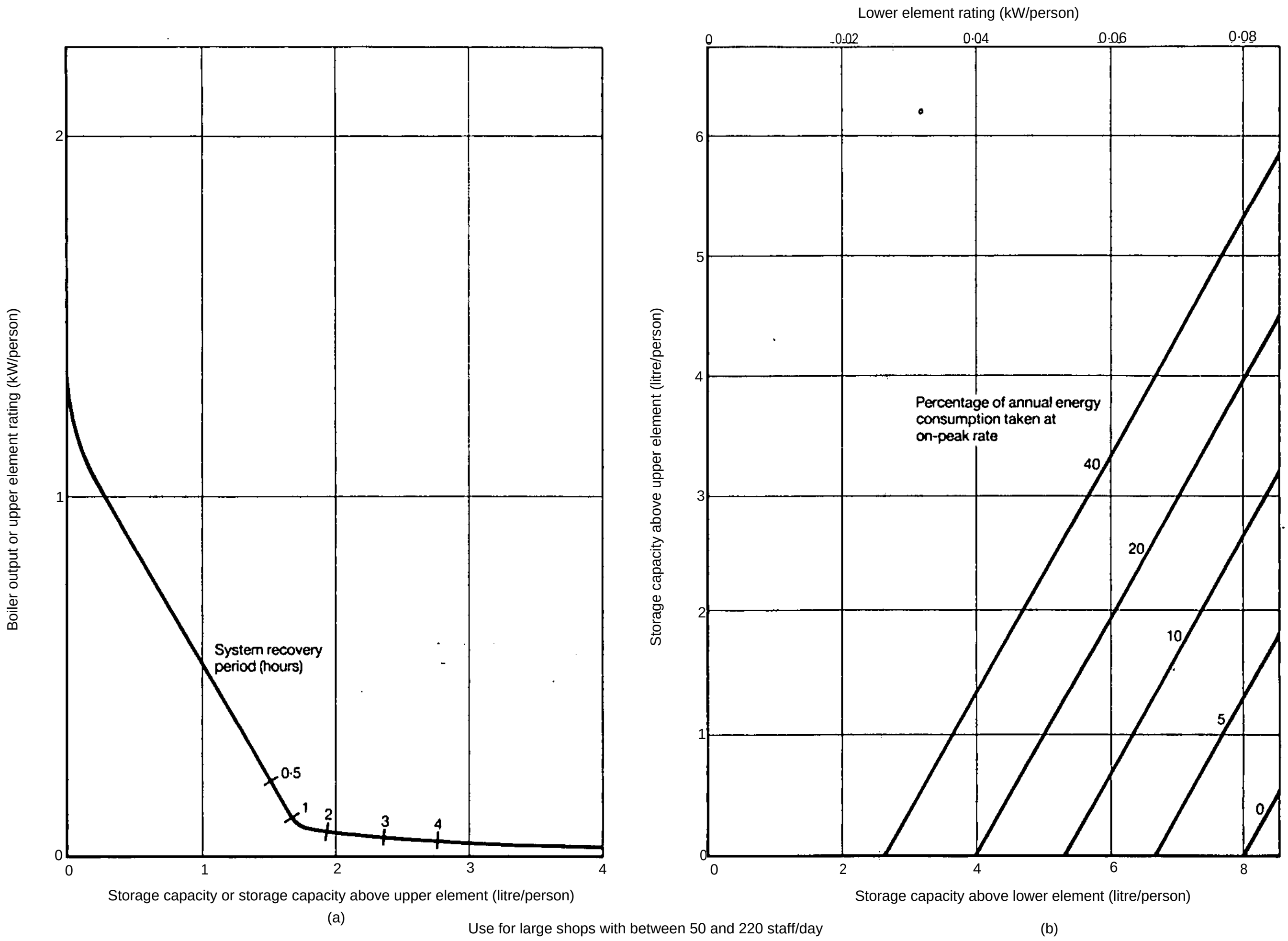



**Fig. B4.15.** Plant sizing guide for large shops – service.

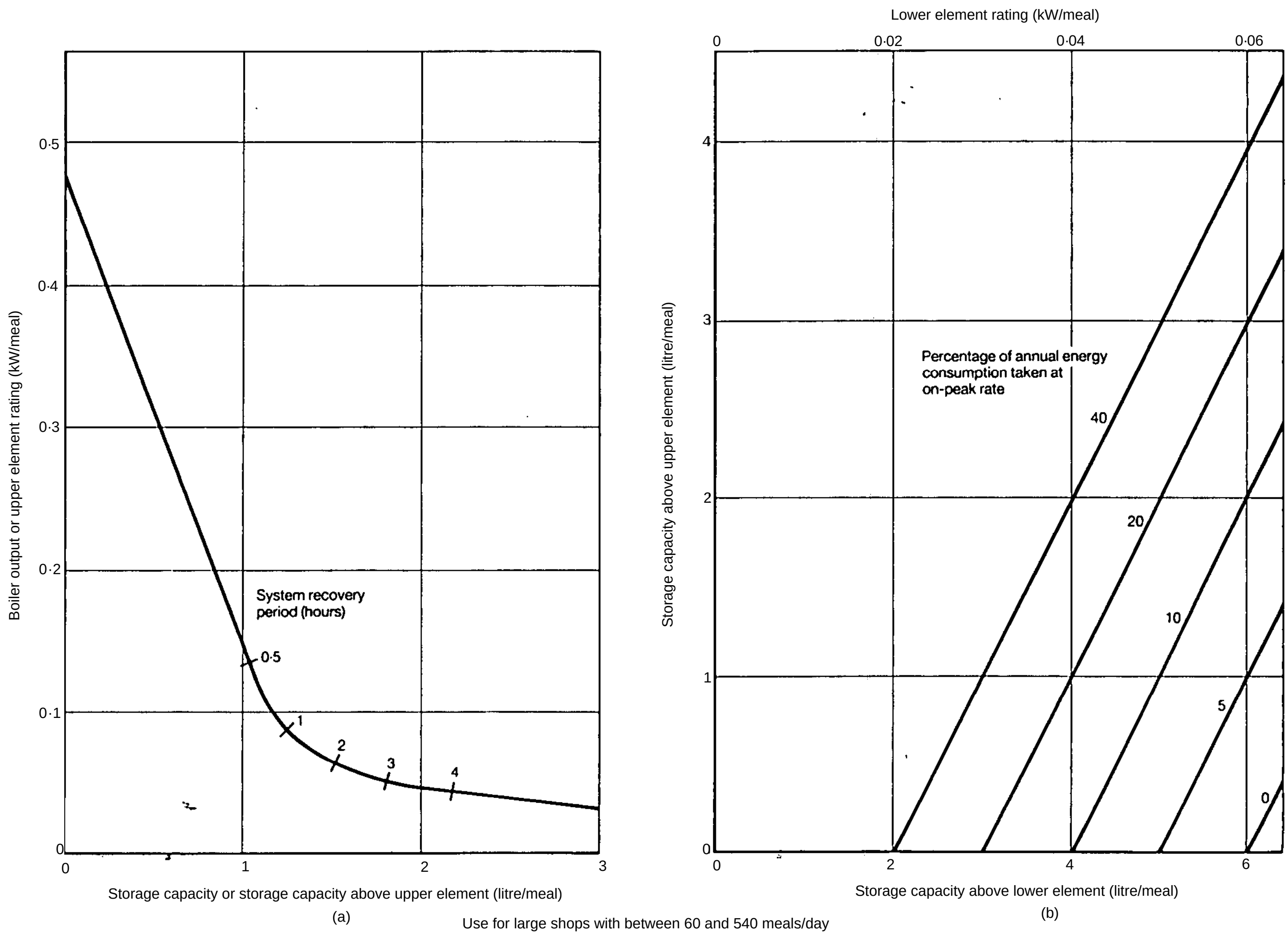
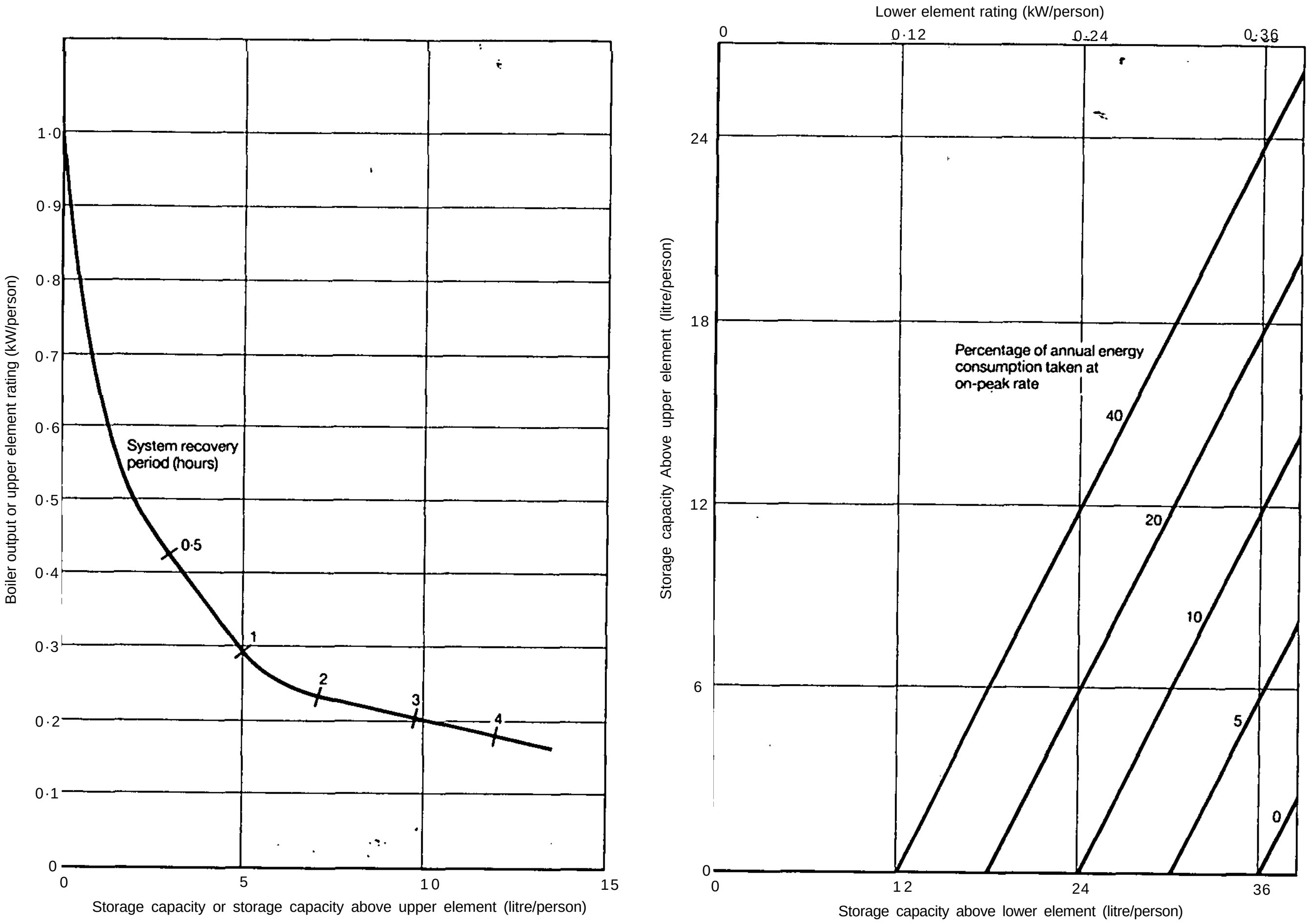



**Fig. B4.16.** Plant sizing guide for large shops – catering.




Use for student hostel with between 80 and 320 persons/day

**Fig. B4.17.** Plant sizing guide for student hostels — service

## Hospitals

### *Hot Water Storage*

The relationship between hot water storage and heating equipment in hospitals is complex and depends upon the particular circumstances. The historically accepted method is to produce a histogram of peak periods of use, taking account of water inflow into storage which reduces the temperature of stored water. The data listed in Table B4.10 should therefore be considered as representative only of capacities which have not given rise to complaints of inadequacy.

**Table B4.10.** Hot water storage and boiler power.

| Building | Storage at 65°C (litre/person) | Boiler power to 65°C (kW/person) |
|---|---|---|
| Hospitals: | | |
| General | 30 | 1·5 |
| Infectious | 45 | 1·5 |
| Infirmaries | 25 | 0·6 |
| Infirmaries (with laundry) | 30 | 0·9 |
| Maternity | 30 | 2·1 |
| Mental | 25 | 0·7 |
| Nurses homes | 45 | 0·9 |

*Notes:*

The temperature of the cold feed to the hot water storage is assumed to be 10°C. For temperatures other than 10°C the boiler power given above should be increased or reduced as appropriate.

The boiler should provide for both hot water used and for the heat loss from the towel rails, coils and circulating pipes.

### *Hot Water Consumption*

The quantities listed in Table B4.11 have been found to be representative.

**Table B4.11.** Hot water consumption.

| Building | Maximum daily demand (litre/person) |
|---|---|
| Hospitals: | |
| General | 135 |
| Infectious | 225 |
| Infirmaries | 70 |
| Infirmaries (with laundry) | 90 |
| Maternity | 230 |
| Mental | 90 |
| Nurses homes | 135 |

### *Hospital Ward Units*

The Hospital Engineering Research Unit carried out an investigation into the water consumption of ward units. Records of the investigation, together with details of the statistical analysis to which they were subjected are included in a published report[9]. Part of this report comprises a series of nomograms based upon the following equations:

$$V_{ha} = C + 100n + 73e \quad \text{B4.9}$$

$$V_{hm} = D + 203n \quad \text{B4.10}$$

$$V_{hs} = E + 41{\cdot}4n + 24{\cdot}5e \quad \text{B4.11}$$

$$Q = 3{\cdot}52 + 0{\cdot}75n \quad \text{B4.12}$$

where:

$V_{ha}$ = average daily hot water consumption .. litre

$V_{hm}$ = predicted maximum daily hot water consumption .. .. .. .. litre

$V_{hs}$ = net hot water store .. .. .. litre

Q = rate of heat supply .. .. .. kW

$C$, $D$, $E$ = constants for different types of ward, see Table B4.12

$n$ = number of beds in ward

$e$ = $\Delta$(number of items of equipment multiplied by the appropriate index), see Table B4.13

The net hot water store is that quantity of hot water actually capable of being drawn off at a temperature of about 65°C. Due to the "loss" of storage capacity within a vessel due to mixing with the cold water feed, this net hot water store value must be increased. In the absence of reliable data as to the margin required it is suggested that an addition of 25% be made.

By making certain assumptions as to the average relation between number of beds and quantity of equipment it has been found possible to modify the published equations to produce data more suitable for inclusion in this Section. Tables B4.14 and 15 list consumptions, etc., for a series of ward units; the values quoted in Table B4.15 being representative of required storage capacity rather than the net hot water store. For multiple ward blocks the multiplying factors quoted in Table B4.16 should be used; the weighted average consumption being taken where the wards are of different types.

**Table B4.12.** Numerical constants for ward types.

| Type of ward | Values for constants | | |
|---|---|---|---|
| | *C* | *D* | *E* |
| Surgical and gynaecological | -345 | +1973 | -182 |
| Medical | -482 | +1523 | -145 |
| Obstetric | -1400 | +1050 | -368 |
| Geriatric and chronic | -1573 | +386 | -536 |
| Thoracic | -600 | +2673 | -309 |
| Paediatric and infectious | +114 | +3014 | +259 |
| Orthopaedic | -609 | +1905 | -64 |

**Table B4.13.** Numerical indices for ward equipment.

| Item | Index | Item | Index |
|---|---|---|---|
| Bath | 1·2 | Milk sink | 0·3 |
| Bedpan sterilizer | 0·8 | Sluice sink | 0·3 |
| Bowl sterilizer | 0·8 | Spray | 0·3 |
| Crockery sterilizer | 0·8 | Baby bath | 0·2 |
| Mechanical sluice | 0·8 | Bedpan washer | 0·2 |
| Instrument sterilizer | 0·4 | Dish washer | 0·2 |
| Kitchen sink | 0·4 | Scrub-up | 0·2 |
| Treatment sink | 0·4 | Sluice hopper | 0·2 |
| Utility sink | 0·4 | Lavatory basin | 0·1 |
| Shower | 0·4 | Surgeons' basin | 0.1 |

## WATER SUPPLY SYSTEM DESIGN

The data contained in the following paragraphs apply to both cold and hot water pipework system design; the demand on draw-off points being listed in Table B4.17. Capacities of sanitary appliances in normal use, with supply temperatures, are set out in Table B4.18. Information on pipe sizing is contained in Section C4.

### Simultaneous Demands

Section B6 contains a full treatment of the general theoretical background. The details contained here are those relevant to hot and cold water pipe sizing, and comprise a simplified method of assessing the demand from a number of draw-off points.

Draw-off rates for appliances are given in terms of "demand units". If it is determined that unity in the scale of demand units represents a flow rate of 0·15 litre/s at intervals of 5 minutes; the cold tap of a lavatory basin being chosen for this basis, then a scale of "demand units" as shown in Table B4.19 may be calculated. The usage ratio, here taken as 0·1, is defined as the average time that the demand on a draw-off point is imposed divided by the average time between occasions of use.

These theoretical data may be rounded and multiplied by 10 to avoid fractions in summation and, at the same time, grouped under headings indicative of the frequency of usage as shown in Table B4.21.

**Table B4.14.** Ward units - daily consumption of hot water.

| Number of beds | Maximum and average consumption per ward unit ($m^3$/day) | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Surgical and gynaecological | | Medical | | Obstetric | | Geriatric and chronic | | Thoracic | | Paediatric and infectious | | Orthopaedic | |
| | Max. | Ave. | Max. | Ave. | Max. | Ave. | Max. | Ave. | Max. | Ave. | Max. | Ave. | Max. | Ave. |
| 20 | 6·05 | 2·23 | 5·59 | 2·09 | 5·14 | 1·18 | 4·45 | 1·00 | 6·73 | 1·95 | 7·05 | 2·68 | 5·95 | 1·95 |
| 25 | 7·05 | 2·79 | 6·59 | 2·66 | 6·14 | 1·75 | 5·45 | 1·57 | 7·77 | 2·52 | 8·05 | 3·25 | 7·00 | 2·52 |
| 30 | 8·09 | 3·36 | 7·64 | 3·23 | 7·18 | 2·32 | 6·50 | 2·14 | 8·77 | 3·09 | 9·09 | 3·82 | 8·00 | 3·09 |
| 35 | 9·09 | 3·93 | 8·64 | 3·80 | 8·18 | 2·89 | 7·50 | 2·70 | 9·77 | 3·66 | 10·1 | 4·39 | 9·05 | 3·66 |
| 40 | 10·1 | 4·50 | 9·68 | 4·36 | 9·18 | 3·45 | 8·55 | 3·27 | 10·8 | 4·23 | 11·1 | 4·95 | 10·0 | 4·23 |
| 45 | 11·1 | 5·07 | 10·7 | 4·93 | 10·2 | 4·02 | 9·55 | 3·66 | 11·9 | 4·80 | 12·1 | 5·52 | 11·0 | 4·80 |
| 50 | 12·1 | 5·64 | 11·7 | 5·50 | 11·2 | 4·59 | 10·5 | 4·41 | 12·9 | 5·36 | 13·1 | 6·09 | 12·1 | 5·36 |
| 55 | 13·2 | 6·20 | 12·7 | 6·07 | 12·2 | 5·16 | 11·6 | 4·98 | 13·7 | 5·93 | 14·2 | 6·66 | 13·1 | 5·93 |
| 60 | 14·2 | 6·77 | 13·7 | 6·64 | 13·3 | 5·73 | 12·6 | 5·55 | 14·9 | 6·50 | 15·2 | 7·23 | 14·1 | 6·50 |

**Table B4.15.** Ward units – hot water storage and heat supply.

| Number of beds | Rate of heat* supply per ward unit (kW) | Hot water storage capacity per ward unit† ($m^3$) | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | | Surgical and gynaecological | Medical | Obstetric | Geriatric and chronic | Thoracic | Paediatric and infectious | Orthopaedic |
| 20 | 18·5 | 1·05 | 1·09 | 0·82 | 0·61 | 1·66 | 1·59 | 1·20 |
| 25 | 22·3 | 1·34 | 1·39 | 1·11 | 0·91 | 1·95 | 1·89 | 1·50 |
| 30 | 26·0 | 1·64 | 1·68 | 1·39 | 1·18 | 2·25 | 2·18 | 1·78 |
| 35 | 29·8 | 1·93 | 1·95 | 1·68 | 1·48 | 2·54 | 2·45 | 2·05 |
| 40 | 33·5 | 2·23 | 2·25 | 1·98 | 1·78 | 2·82 | 2·75 | 2·36 |
| 45 | 37·3 | 2·50 | 2·54 | 2·27 | 2·05 | 3·11 | 3·05 | 2·64 |
| 50 | 41·0 | 2·80 | 2·84 | 2·54 | 2·34 | 3·41 | 3·32 | 2·93 |
| 55 | 44·8 | 3·09 | 3·11 | 2·84 | 2·64 | 3·68 | 3·61 | 3·23 |
| 60 | 48·5 | 3·36 | 3·41 | 3·14 | 2·93 | 3·98 | 3·91 | 3·50 |

* In addition to this heat supply requirement, provision must be made for heat loss from towel rails, coils and circulating pipes.

† This capacity has been determined by adding 25% to the approximate net hot water store as calculated from the modified equation.

**Table B4.16.** Multiplying factors for application to Tables B4.14 and B4.15.

| Number of ward units | Multiplying factors to single ward unit design values | |
|---|---|---|
| | Maximum daily hot water consumption (Table B4.14) | Store of hot and tank water (Table B4.15) |
| 1 | 1·0 | 1·0 |
| 2 | 1·8 | 1·7 |
| 3 | 2·6 | 2·3 |
| 4 | 3·4 | 2·9 |
| 5 | 4·2 | 3·5 |
| 6 | 5·0 | 4·2 |
| 7 | 5·7 | 4·7 |
| 8 | 6·4 | 5·2 |
| 9 | 7·2 | 5·8 |
| 10 | 8·0 | 6·5 |
| 15 | 12·0 | 9·2 |
| 20 | 15·5 | 12·0 |
| 25 | 19·2 | 15·0 |
| 30 | 23·0 | 17·7 |
| 35 | 26·8 | 20·5 |
| 40 | 30·5 | 23·4 |
| 45 | 34·3 | 26·2 |
| 50 | 38·2 | 29·0 |

**Table B4.17.** Approximate demand required of water points (hot and cold water).

| Sanitary appliance | Flow rate (litre/s) |
|---|---|
| Basin (spray) | 0·05 |
| Basin (tap) | 0·15 |
| Bath (private) | 0·3 |
| Bath (public) | 0·6 |
| Flushing cistern | 0·1 |
| Shower (nozzle) | 0·15 |
| Shower (100 mm rose)* | 0·4 |
| Sink (15 mm tap) | 0·2 |
| Sink (20 mm tap) | 0·3 |
| Wash fountain | 0·4 |

* The use of shower roses results in wasteful consumption of water, but emergency drench showers require high discharge rates.

**Table B4.18.** Approximate capacities of sanitary appliances in normal use.

| Sanitary appliance | Temperature range in use (°C) | Capacity (litre) |
|---|---|---|
| Basin* | 10 to 45 | 5 |
| Bath (private) | l0 to 45 | 80 |
| Bath (public) | 10 to 45 | 120 |
| Sink (small) | 10 to 65 | 12 |
| Sink (large) | l0 to 65 | 18 |
| Urinal cistern | 10 | 4·5 |
| WC (13·5 litre) | 10 | 13·5 |
| WC (9 litre) | 10 | 9 |

* The data apply to wash basins with conventional taps. Where spray taps are used, the capacity of the appliance is irrelevant, as plugs are not normally fitted.

**Table B4.19.** Theoretical demand units computed relative to a wash basin (usage ratio = 0·1).

| Fitting | | Interval of use (minutes) | | | | |
|---|---|---|---|---|---|---|
| Type | Detail | 5 | 10 | 20 | 40 | 80 |
| Basin | Hot | 0·50 | 0·26 | 0·11 | — | — |
| Basin | Cold | 1·00 | 0·50 | 0·25 | — | — |
| Bath | Hot | — | — | 2·45 | 1·23 | 0·60 |
| Bath | Cold | — | — | 4·70 | 2·45 | 1·23 |
| Sink | Small | 4·28 | 2·18 | 1·09 | — | — |
| Sink | Large | 4·28 | 2·18 | 1·09 | — | — |
| Urinal | Per stall | — | — | — | — | — |
| WC | 13·5 litre | 3·45 | 1·50 | 0·75 | — | — |
| WC | 9 litre | 2·18 | 0·99 | 0·48 | — | — |

Having established a practical scale of demand units the probability table may now be used to convert totals of demand units directly into design flow rates as given in Table B4.20.

Design procedure, therefore, involves only the totalling of system draw-off points in terms of demand units and reading off the design flow rate from the conversion table. This value is then applied to Section C4 to obtain the pipe size.

**Table B4.20.** Simultaneous demand data for design.

| Demand units | Design demand (litre/s) | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 0 | 50 | 100 | 150 | 200 | 250 | 300 | 350 | 400 | 450 | 500 | 550 | 600 | 650 | 700 | 750 | 800 | 850 | 900 | 950 |
| 0 | 0·0 | 0·3 | 0·5 | 0·6 | 0·8 | 0·9 | 1·0 | 1·2 | 1·3 | 1·4 | 1·5 | 1·6 | 1·7 | 1·9 | 2·0 | 2·1 | 2·2 | 2·3 | 2·4 | 2·5 |
| 1000 | 2·6 | 2·7 | 2·8 | 2·9 | 3·0 | 3·1 | 3·2 | 3·3 | 3·4 | 3·5 | 3·6 | 3·7 | 3·8 | 3·9 | 4·0 | 4·1 | 4·2 | 4·3 | 4·4 | 4·5 |
| 2000 | 4·6 | 4·7 | 4·8 | 4·9 | 5·0 | 5·1 | 5·1 | 5·2 | 5·3 | 5·4 | 5·5 | 5·6 | 5·7 | 5·8 | 5·9 | 6·0 | 6·1 | 6·2 | 6·3 | 6·4 |
| 3000 | 6·4 | 6·5 | 6·6 | 6·7 | 6·8 | 6·9 | 7·0 | 7·1 | 7·2 | 7·3 | 7·4 | 7·4 | 7·5 | 7·6 | 7·7 | 7·8 | 7·9 | 8·0 | 8·1 | 8·2 |
| 4000 | 8·3 | 8·3 | 8·4 | 8·5 | 8·6 | 8·7 | 8·8 | 8·9 | 9·0 | 9·1 | 9·1 | 9·2 | 9·3 | 9·4 | 9·5 | 9·6 | 9·7 | 9·8 | 9·8 | 9·9 |
| 5000 | 10·0 | 10·1 | 10·2 | 10·3 | 10·4 | 10·5 | 10·5 | 10·6 | 10·7 | 10·8 | 10·9 | 11·0 | 11·1 | 11·2 | 11·2 | 11·3 | 11·4 | 11·5 | 11·6 | 11·7 |
| 6000 | 11·8 | 11·9 | 11·9 | 12·0 | 12·1 | 12·2 | 12·3 | 12·4 | 12·5 | 12·5 | 12·6 | 12·7 | 12·8 | 12·9 | 13·0 | 13·1 | 13·1 | 13·2 | 13·3 | 13·4 |
| 7000 | 13·5 | 13·6 | 13·7 | 13·7 | 13·8 | 13·9 | 14·0 | 14·1 | 14·2 | 14·3 | 14·3 | 14·4 | 14·5 | 14·6 | 14·7 | 14·8 | 14·9 | 14·9 | 15·0 | 15·1 |
| 8000 | 15·2 | 15·3 | 15·4 | 15·5 | 15·5 | 15·6 | 15·7 | 15·8 | 15·9 | 16·0 | 16·0 | 16·1 | 16·2 | 16·3 | 16·4 | 16·5 | 16·6 | 16·6 | 16·7 | 16·8 |
| 9000 | 16·9 | 17·0 | 17·1 | 17·2 | 17·2 | 17·3 | 17·4 | 17·5 | 17·6 | 17·7 | 17·7 | 17·8 | 17·9 | 18·0 | 18·1 | 18·2 | 18·2 | 18·3 | 18·4 | 18·5 |
| 10000 | 18·6 | 18·7 | 18·8 | 18·8 | 18·9 | 19·0 | 19·1 | 19·2 | 19·3 | 19·3 | 19·4 | 19·5 | 19·6 | 19·7 | 19·8 | 19·8 | 19·9 | 20·0 | 20·1 | 20·2 |
| 11000 | 20·3 | 20·3 | 20·4 | 20·5 | 20·6 | 20·7 | 20·8 | 20·8 | 20·9 | 21·0 | 21·1 | 21·2 | 21·3 | 21·3 | 21·4 | 21·5 | 21·6 | 21·7 | 21·8 | 21·8 |
| 12000 | 21·9 | 22·0 | 22·1 | 22·2 | 22·3 | 22·3 | 22·4 | 22·5 | 22·6 | 22·7 | 22·8 | 22·8 | 22·9 | 23·0 | 23·1 | 23·2 | 23·3 | 23·3 | 23·4 | 23·5 |
| 13000 | 23·6 | 23·7 | 23·8 | 23·8 | 23·9 | 24·0 | 24·1 | 24·2 | 24·3 | 24·3 | 24·4 | 24·5 | 24·6 | 24·7 | 24·7 | 24·8 | 24·9 | 25·0 | 25·1 | 25·2 |
| 14000 | 25·2 | 25·3 | 25·4 | 25·5 | 25·6 | 25·7 | 25·7 | 25·8 | 25·9 | 26·0 | 26·1 | 26·2 | 26·2 | 26·3 | 26·4 | 26·5 | 26·6 | 26·6 | 26·7 | 26·8 |
| 15000 | 26·9 | 27·0 | 27·1 | 27·1 | 27·2 | 27·3 | 27·4 | 27·5 | 27·6 | 27·6 | 27·7 | 27·8 | 27·9 | 28·0 | 28·0 | 28·1 | 28·2 | 28·3 | 28·4 | 28·5 |
| 16000 | 28·5 | 28·6 | 28·7 | 28·8 | 28·9 | 29·0 | 29·0 | 29·1 | 29·2 | 29·3 | 29·4 | 29·4 | 29·5 | 29·6 | 29·7 | 29·8 | 29·9 | 29·9 | 30·0 | 30·1 |
| 17000 | 30·2 | 30·3 | 30·3 | 30·4 | 30·5 | 30·6 | 30·7 | 30·8 | 30·8 | 30·9 | 31·0 | 31·1 | 31·2 | 31·2 | 31·3 | 31·4 | 31·5 | 31·6 | 31·7 | 31·7 |
| 18000 | 31·8 | 31·9 | 32·0 | 32·1 | 32·1 | 32·2 | 32·3 | 32·4 | 32·5 | 32·6 | 32·6 | 32·7 | 32·8 | 32·9 | 33·0 | 33·0 | 33·1 | 33·2 | 33·3 | 33·4 |
| 19000 | 33·5 | 33·5 | 33·6 | 33·7 | 33·8 | 33·9 | 33·9 | 34·0 | 34·1 | 34·2 | 34·3 | 34·3 | 34·4 | 34·5 | 34·6 | 34·7 | 34·8 | 34·8 | 34·9 | 35·0 |

**Table B4.21.** Practical demand units for use with data listed in Table B4.20.

| Fitting | Type of application | | |
|---|---|---|---|
| | Congested | Public | Private |
| Basin* | 10 | 5 | 3 |
| Bath† | 47 | 25 | 12 |
| Sink | 43 | 22 | 11 |
| Urinal cistern‡ | – | – | – |
| WC (13·5 litre) | 35 | 15 | 8 |
| WC (9 litre) | 22 | 10 | 5 |

* These data apply to conventional taps only. If spray taps are used, demand may be continuous at 0·05 litre/s per tap.

† If a shower spray nozzle is used over the bath, demand may be continuous at 0·1 litre/s per nozzle.

‡ Demand will be continuous at 0·003 litre/s per stall.

## EQUIPMENT AND MATERIALS

### Overflows

Detailed requirements for overflow and warning pipes are contained in the Model Water Byelaws[2], and are further described in a guide to the Byelaws[5]. For definitions refer to BS 4118[32].

### Frost Precautions

Every water fitting, whether inside or outside a building, should be sited so as to reduce the risk of frost damage to the greatest extent. If a water fitting (other than an overflow pipe) is likely to suffer from damage by frost, it should be effectively protected from such damage. BS CP 99[33] gives guidance.

Wherever possible, underground water pipes not under a permanent building should be laid at a depth of 750 mm or more and underground stop valves should not be brought up to a higher level merely for ease of access.

Cisterns in unheated roof spaces should be insulated in conjunction with any ceiling insulation. Insulation should be provided over and around the cistern, but omitted from below it. Where cisterns are fitted above the roof of a flat roofed building, they should be installed, together with any associated pipework, in an insulated enclosure. This should be provided with means of heating or should be open to some heated part of the building itself.

### Fittings and Materials

Details of the scheme operated by the Water Research Centre and lists of acceptable fittings and materials are included in a cumulative Directory[34], published at regular intervals on behalf of the water industry.

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# SECTION B6 MISCELLANEOUS PIPED SERVICES



*This edition of Section B6 first published: 1986*

# SECTION B6 MISCELLANEOUS PIPED SERVICES

## INTRODUCTION

This Section presents a miscellany of information related to a variety of piped systems including gas, steam, medical and laboratory services. The various tables showing the consumption of typical apparatus and equipment give an indication of the average requirement and may be used for initial sizing, in the absence of more specific manufacturers' data.

An outline of medical piped services is presented but it is recommended that further reference be made to the relevant DHSS Hospital Technical Memoranda.

This Section also includes a more complete treatment of the mathematical background to simultaneous demand calculation than that given in Sections B4 and B8.

## GAS FUELS

The approximate rates of discharge required from domestic and miscellaneous gas-fired equipment are listed in Tables B6.1 and 2. Where only one gas rate is quoted, this applies to the majority of that type although equipment with a less or a greater consumption may be available. Where a range of gas rates is quoted, no pre-dominating level exists and the equipment available is spread more or less evenly over the range.

**Table B6.1.** Approximate connected loads for natural gas at domestic appliances.

| Appliance | Gas rate (litre/s) |
|---|---|
| Boiling ring | 0·08 |
| Cooker (four rings, grill, oven) | 0·51 |
| Drying cabinet | 0·15 |
| Poker | 0·11 |
| Wash boiler | 0·16 |
| Convector, flued | 0·2 to 0·4 |
| Convector, balanced flue | 0·2 to 0·4 |
| Convector, SE duct | 0·2 |
| Radiator, flueless | 0·02 to 0·08 |
| Radiator, flued | 0·06 to 0·25 |
| Water heater, instantaneous | |
| Single-point sink type | 0·29 |
| Single-point bath type | 0·59 |
| Multi-point type | 0·76 |
| Water heater, storage | |
| Single-point type | 0·04 to 0·14 |
| Multi-point type | 0·06 to 0·35 |

### Kitchen Equipment

For kitchen equipment approximate rates of gas discharge are quoted in Table B6.3 but should be confirmed from manufacturers' data for specific applications. Service intakes to kitchens having all gas or some gas appliances may be estimated from the data[3] listed in Tables B6.4 and 5. In the latter case it has been assumed that steam is available to serve all equipment suited to that medium. In both cases the gas rate quoted is for all equipment in full use.

**Table B6.2.** Approximate connected loads for natural gas at miscellaneous equipment.

| Item | Gas rate (litre/s) |
|---|---|
| Absorption refrigeration, per kW | 0·05 |
| Incinerator (sanitary towel) | 0·04 to 0·15 |
| Laboratory bunsen burner | 0·03 |
| Tumbler drier | 0·4 to 1·0 |

**Table B6.3.** Approximate connected loads for natural gas at kitchen equipment.

| Appliance | Maximum gas rate (litre/s) |
|---|---|
| Boiling pan, per 50 litre | 0.15 |
| Deep fat fryer | 0.5 to 1.0 |
| Bratt pan | 0.4 |
| Forced convection oven | 0.4 to 0.8 |
| Frozen food regenerator | 0.8 |
| Grill | 0.4 |
| Hot cupboard, per metre run | 0.2 |
| Range (open top, oven) | 1.0 |
| Range (fry top, oven) | 1.0 |
| Range (top section) | 0.3 to 0.6 |
| Range (roasting oven) | 0.2 to 1.2 |
| Steaming oven | 0.3 to 0.6 |
| Water boiler, per 50 litre | 0.25 |

**Table B6.4.** Approximate connected natural gas load for 'all gas' kitchen.

| Type | Connected gas rate for stated number of meals served (litre/s) | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | 50 | 100 | 200 | 300 400 | 500 600 | 700 800 | 900 1000 | 1250 | 1500 |
| Restaurants* | 2·4 | 4·4 | 7·3 | 11·2 | 15·7 | 20·8 | 25·4 | — | — |
| Institutions† | 2·7 | 5·6 | 9·3 | 12·8 | 18·3 | 24·3 | 30·7 | 33·4 | 41·1 |

* Includes hotels and restaurants serving breakfast, lunch and dinner. Also public cafeterias serving lunch plus snacks only.
† Includes holiday camps and hospitals serving breakfast, lunch and dinner. Also private canteens serving lunch plus snacks only.

**Table B6.5.** Approximate connected natural gas load for 'steam/gas' kitchens.

| Type | Connected gas rate for stated number of meals served (litre/s) | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | 50 | 100 | 200 | 300 400 | 500 600 | 700 800 | 900 1000 | 1250 | 1500 |
| Restaurants* | 1·5 | 2·7 | 4·3 | 6·7 | 9·7 | 13·1 | 15·4 | — | — |
| Institutions† | 1·7 | 3·3 | 5·1 | 7·4 | 10·4 | 14·7 | 17·4 | 19·9 | 24·0 |

* Includes hotels and restaurant serving breakfast, lunch and dinner. Also public cafeterias serving plus snacks only
† Includes holiday camps and hospitals serving breakfast lunch and dinner. also private canteens serving lunch plus snacks only.

## Simultaneous Demands

It is not possible to give any general method of determining the diversity of use appropriate to a number of gas appliances connected to a building carcass. Obviously in the case of gas boiler plant, allowance must be made for the full connected gas rate unless a proportion of the plant is specifically allocated as standby. Similarly in the case of kitchen equipment an assessment must be made of the number of appliances appropriate to the preparation of each meal, which may amount to as much as 80 per cent of the total connected.

In the case of laboratory gas points these may be considered as falling into one of the following categories:

(*a*) Those serving major apparatus such as furnaces, glass-blowing benches, etc., which are likely to be in use simultaneously and in addition to normal bench points;

(*b*) Those situated in teaching laboratories where despite an apparent excess of points, a large proportion may be in use during examination periods, etc.;

(*c*) Those situated in research laboratories where the number of points provided may be related to convenience rather than demand. Reported usage[4] of normal bunsen burners in a specific building is set out in Table B6.6.

**Table B6.6.** Simultaneous demand–laboratory bunsen burners.

| Number of burners connected | Predicted number in use | |
|---|---|---|
| | In any one laboratory | In a group of laboratories |
| 1 | 1 | — |
| 2 | 2 | — |
| 3 | 3 | — |
| 4 | 4 | — |
| 5 | 4 | 4 |
| 6 | — | 5 |
| 10 | — | 6 |
| 15 | — | 6 |
| 20 | — | 6 |
| 25 | — | 8 |
| 30 | — | 9 |
| 40 | — | 12 |
| 50 | — | 13 |
| 60 | — | 15 |
| 100 | — | 20 |

## Flow Data

### *Natural Gas*

Flow rates and pressure drops for natural gas are given in Section C4, Tables C4.45 and 46.

### *Manufactured Gas*

Since the values of density and viscosity of manufactured gas differ little from those of natural gas, Tables C4.45 and 46 may be used by applying the following correction factors.

(*a*) For constant volume flow, divide the tabulated figures for pressure loss by 1.1.

(*b*) For constant pressure loss, multiply the tabulated figures for volume flow by 1.05.

### *Liquid Petroleum Gas*

Section C4 gives equations from which flow data for butane and propane may be obtained. As an approximation, Tables C4.45 and 46 may be used by applying the following correction factors.

(*a*) For constant volume flow, multiply the tabulated figures for pressure loss by 2.2.

(*b*) For constant pressure loss, divide the tabulated figures for volume flow by 1.5.

## MEDICAL GAS SYSTEMS

Medical gas systems, conveying oxygen, nitrous oxide and nitrous oxide/oxygen mixtures are subject to stringent regulations described in detail, in DHSS Technical Memorandum 22[1]. Tables B6.7 and 8 (overleaf) give details of the flow rates required for branches and terminals respectively and Table B6.9 provides additional information with respect to hyperbaric chambers. It should be noted that flow rates are given both in SI and nominal units to avoid confusion.

**Table B6.7.** Design ('free air') flow rated for medical gases to system branches.

| Area | Accommodation | Design flow rate (lite) Nominal (per minute) | SI (per second) |
|---|---|---|---|
| Theatres* (Oxygen rate per suite) | Theatre and anaesthetic room | | |
| | First | 50 | 0.84 |
| | Second | 30 | 0·5 |
| | Remainder | 20 | 0·34 |
| | Plaster room | | |
| | Each | 20 | 0.34 |
| Theatres (Nitrous oxide rate per suite) | Theatre and anaesthetic room | | |
| | First | 30 | 0·5 |
| | Second | 20 | 0·34 |
| | Remainder | 15 | 0.25 |
| | Plaster room | | |
| | Each | 20 | 0.34 |
| Recovery wards (rate per bed) | Up to 8 | | |
| | Each | 20 | 0·34 |
| | 9 to 12 | | |
| | First 8 | 20 | 0·34 |
| | Remainder | 12 | 0·20 |
| | 13 to 16 | | |
| | First 8 | 20 | 0·34 |
| | Remainder | 10 | 0·17 |
| | Over 16 | | |
| | First 8 | 20 | 0·34 |
| | Remainder | 9 | 0·15 |
| Intensive therapy unit or area (rate per bed) | Oxygen | | |
| | Including ventilator drive | 40 | 0·67 |
| | Without ventilator drive | 20 | 0·34 |
| | Nitrous oxide | | |
| | All cases | 15 | 0·25 |
| Other terminal units (rate per terminal) | Oxygen | | |
| | Single terminal | 20 | 0·34 |
| | Remote terminal on branch | 20 | 0·34 |
| | Other terminals† | 6 | 0·1 |
| | Nitrous oxide | | |
| | Single terminal | 15 | 0·25 |
| | Remote terminal on branch | 15 | 0·25 |
| | Other terminals† | 6 | 0·1 |
| | Nitrous oxide oxygen mixtures | | |
| | Each‡ | 20 | 0·34 |

* Where piped oxygen is used to drive patient ventilators, the values quoted should be increased by 40%.
† An allowance for diversity may be applied where appropriate: 25% may be used. Diversity applies from second terminal onwards.
‡ It may be assumed that a diversity of 50% can be applied to these terminals. Allowance should be made for 6% of those in use (minimum 1) to be passing simultaneous flows at a peak rate of 4.6 litre/s (275 litre/min) each.

### Design Flow Rates

Table B6.7 gives design flow rates for various applications and it will be noted that these provide an empirical allowance for diversity in use. Fig. B6.1 provides similar data for general wards. Distribution pipework should be sized using these two sets of design flow rates but it is advisable to check the results following the routine laid down in the DHSS publication.

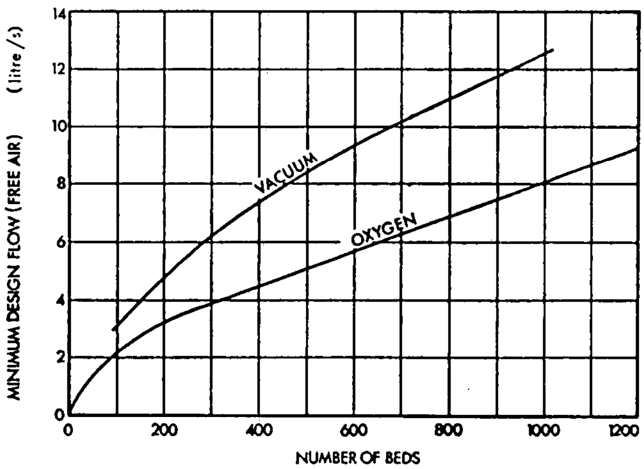


**Fig. B6.1.** Design flow rates for general wards.

**Table B6.9.** Oxygen requirements for single person hyperbaric chambers.

| Use | Approx. time per single treatment (h) | Consumption (litre) Per treatment | Additional Nominal (per minute) | Additional SI (per minute) |
|---|---|---|---|---|
| Hyperbaric beds* | 2 | 4200 | 250 | 4·2 |
| Hyperbaric chambers | | | | |
| Clinical, with recirculation | 2 | 5000 | 250 | 4·2 |
| Clinical, without recirculation | 2 | 10000 | 400 | 6·7 |
| Neo-natal | — | — | 30 | 0·5 |
| Radiotherapy | 1 | 14000 | 250 | 4·2 |

* Data relate to a single treatment. For continuous treatment flow may be taken at 4·2 litre/s (250 litre/min).

### Flow Data

Table B6.10 lists pressure drop/flow data for medical gases and has been compiled by extraction and transposition of DHSS information. The pipework distribution system should be designed so that the pressure drop from the central gas store to the point of delivery at the inlet to the terminal unit does not exceed 5 per cent of the initial pressure.

**Table B6.8.** Flow rates and pressures required at terminal units.

| Service | Flow rate at stp (litre) | | Gauge pressure at terminal unit/bar | |
|---|---|---|---|---|
| | Nominal (per minute) | SI (per second) | Nominal at entry | Max. pressure difference across unit |
| Oxygen | 40 | 0·67 | 3·9 | 0·035 |
| Nitrous oxide | 15 | 0·25 | 3·9 | 0·035 |
| Medical gas mixtures (oxygen/nitrous oxide; 1:1) | 20 (*a*) | 0·33 | 3·9 | 0·55 |
| Medical compressed air (theatres using air-operated surgical tools) | 250 (*b*) | 4·17 | 6·9 | 0·35 |
| Medical compressed air (other theatres) | 50 | 0·83 | 6·9 | 0·35 |
| Medical compressed air (dental surgeries) | 65 | 1·08 | 3·9 | 0·35 |
| Medical vacuum | 40 | 0·67 | (*c*) | (*d*) |

*Notes:*

(*a*) Analgesic gas terminal units should be capable of supplying inhalationary gasps at a peak flow rate of 4·59 litre/s without causing the available pressure to fall below 3·1 bar.

(*b*) Only a small range of surgical tools operate at flow rates exceeding 4·17 litre/s. Therefore, if it is known that the range of tools to be used will not require a flow rate exceeding this figure, the central pipe supply system may be designed for a maximum flow rate of 4·17 litre/s for each theatre terminal unit.

(*c*) DHSS Technical Memorandum 22 quotes 400 mm Hg below standard atmospheric pressure (760 mm Hg). (Equivalent to an absolute pressure of 0·48 bar.)

(*d*) DHSS Technical Memorandum 22 quotes 100 mm Hg. (In SI units this represents 13·33 kN/m$^2$.)

**Table B6.10.** Volume flow rates ('free air') for medical gases, based on an initial gauge pressure of 3·9 bar. Piping to BS 2871: Part I, Table X.

| $\Delta p$ (N/m$^3$) | Volume flow in copper pipes having following outside diam. (mm) (litre/s) | | | | | | $\Delta p$ (N/m$^3$) |
|---|---|---|---|---|---|---|---|
| | 12 | 15 | 22 | 28 | 35 | 42 | |
| 12·5 | | 0·833 | 2·46 | 5·00 | 9·06 | 15.4 | 12·5 |
| 16·0 | 0·510 | 0·958 | 2·83 | 5·74 | 10·4 | 17.6 | 16·0 |
| 20·0 | 0·580 | 1·09 | 3·21 | 6·51 | 11·8 | 19.9 | 20·0 |
| 25·0 | 0·658 | 1·23 | 3·64 | 7·37 | 13·3 | 22.6 | 25·0 |
| 31·5 | 0·751 | 1·41 | 4·14 | 8·39 | 15·2 | 25.6 | 31·5 |
| 40 | 0·860 | 1·61 | 4.74 | 9.59 | 17.3 | 29·3 | 40 |
| 50 | 0·977 | 1·83 | 5·37 | 10·9 | 19·6 | 33·1 | 50 |
| 63 | 1·11 | 2·09 | 6·12 | 12·4 | 22·3 | 37·6 | 63 |
| 80 | 1·28 | 2·39 | 7·00 | 14·1 | 25·5 | 43·0 | 80 |
| 100 | 1·45 | 2·72 | 7·94 | 16·0 | 28·9 | 48·6 | 100 |
| 125 | 1·65 | 3·08 | 9·00 | 18·2 | 32.7 | 55.0 | 125 |
| 160 | 1·89 | 3·55 | 10·3 | 20·8 | 37·5 | 63·1 | 160 |
| 200 | 2·15 | 4·03 | 11·7 | 23·6 | 42·5 | 71·4 | 200 |
| 250 | 2·44 | 4·57 | 13·3 | 26·8 | 48·1 | 80·9 | 250 |
| 315 | 2·78 | 5·21 | 15·1 | 30·5 | 54·7 | 91·9 | 315 |
| 400 | 3·19 | 5·97 | 17·3 | 34·8 | 62·5 | 105 | 400 |
| 500 | 3·62 | 6·78 | 19·6 | 39·5 | 70·8 | 119 | 500 |
| 630 | 4·13 | 7·73 | 22·4 | 44·9 | 80·5 | | 630 |
| 800 | 4·73 | 8·86 | 25·6 | 51·3 | | | 800 |
| 1000 | 5·38 | 10·1 | 29·0 | | | | 1000 |
| 1250 | 6·10 | 11·4 | 32·9 | | | | 1250 |
| 1600 | 7·02 | 13·1 | | | | | 1600 |
| 2000 | 7·98 | 14·9 | | | | | 2000 |
| 2500 | 9·06 | | | | | | 2500 |

### Central Storage

The central storage for medical gases may be in cylinders or, in the case of oxygen, in bulk liquid form. Total storage should normally be based upon one week's demand but greater storage may be required in areas remote from suppliers' depots. Annual consumption statistics for a variety of hospital sizes and types are given in Tables B6.11 and 12.

**Table B6.11.** Annual consumption of medical oxygen in hospitals.*

| Nominal bed complement | Annual consumption ($dam^3$) | | | |
|---|---|---|---|---|
| | Teaching | | Non-teaching | |
| | Av. | Max. | Av. | Max. |
| 100 | 6·5 | 6·8 | 1·4 | 9·9 |
| 200 | 11·3 | 16·1 | 4·2 | 20·4 |
| 300 | 14·4 | 28·9 | 6·8 | 31·1 |
| 400 | 17·8 | 45·3 | 9·6 | 43·0 |
| 500 | 21·2 | 65·1 | 12·5 | 55·2 |
| 600 | 24·6 | 87·8 | 15·0 | 67·1 |
| 700 | 28·0 | 114·7 | 17·6 | 79·3 |
| 800 | 31·7 | — | 20·1 | 91·7 |
| 900 | 35·7 | — | 22·9 | 104·8 |
| 1000 | 39·6 | — | 26·1 | 118·9 |
| 1100 | 43·6 | — | 29·2 | — |
| 1200 | 47·6 | — | 32·3 | — |

**Table B6.12.** Annual consumption of nitrous oxide in hospitals.*

| Nominal bed complement | Annual consumption ($dam^3$) | | | | | |
|---|---|---|---|---|---|---|
| | Type 'A' | | Type 'B' | | Type 'C' | |
| | Av. | Max. | Av. | Max. | Av. | Max. |
| 100 | — | 2·99 | 0·53 | 2·00 | 0·59 | 1·95 |
| 200 | 0·85 | 3·92 | 0·84 | 2·31 | 0·72 | 2·09 |
| 300 | 1·78 | 4·84 | 1·15 | 2·61 | 0·85 | 2·21 |
| 400 | 2·70 | 5·77 | 1·45 | 2·92 | 0·98 | 2·35 |
| 500 | 3·63 | 6·70 | 1·76 | 3·23 | 1·10 | 2·47 |
| 600 | 4·56 | 7·62 | 2·06 | 3·53 | 1·23 | 2·60 |
| 700 | 5·49 | 8·55 | 2·37 | 3·84 | 1·36 | 2·73 |
| 800 | 6·41 | 9·47 | 2·68 | 4·15 | 1·49 | 2·86 |
| 900 | 7·33 | 10·40 | 2·99 | 4·46 | 1·62 | 2·99 |
| 1000 | 8·26 | 11·32 | 3·29 | 4·76 | 1·75 | 3·12 |
| 1100 | 9·19 | 12·26 | 3·60 | 5·06 | 1·88 | 3·25 |
| 1200 | 10·12 | 13·18 | 3·91 | 5·37 | 2·01 | 3·38 |

Type A Acute teaching and 'other' teaching.
Type B Acute, maternity, maternity teaching, tuberculosis and chest, children's acute and 'other'.
Type C Mainly acute, mainly acute teaching, partly acute and orthopaedic.

*Note: 1 $dam^3$ = 1000 $m^3$.

## MEDICAL COMPRESSED AIR

The requirements for medical compressed air terminal units are given in Table B6.8. Flow rates are given both in SI and nominal units to avoid confusion.

### Design Flow Rates

Table B6.13 gives design flow rates for various applications and it will be noted that these provide an empirical allowance for diversity in use. Distribution pipework and central plant should be sized using the data listed.

**Table B6.13.** Design ('free air') flow rates for medical compressed air.

| Area | Accommodation | Design flow rate (litre) | |
|---|---|---|---|
| | | Nominal (per minute) | SI (per second) |
| Theatres* (rate per suite) | Up to 8 | | |
| | First 1 | 300 | 5 |
| | Remainder | 50 | 0·84 |
| | 9 to 16 | | |
| | First 2 | 600 | 10 |
| | Remainder | 30 | 0.5 |
| | Over 16 | | |
| | First 3 | 900 | 10 |
| | Remainder | 20 | 0.34 |
| Recovery wards (rate per bed) | Up to 8 | | |
| | Each | 50 | 0·84 |
| | 9 to 12 | | |
| | First 8 | 50 | 0·84 |
| | Remainder | 30 | 0·5 |
| | Over 12 | | |
| | First 8 | 50 | 0·84 |
| | Remainder | 25 | 0·5 |
| Intensive therapy unit or area (rate per bed) | All cases | | |
| | First 4 | 50 | 0·84 |
| | Remainder | 25 | 0·5 |
| Dental surgeries (rate per chair) | Up to 5 | | |
| | Each | 65 | 1·1 |
| | 6 to 10 | | |
| | First 5 | 65 | 1·1 |
| | Remainder | 26 | 0·44 |
| | 11 to 30 | | |
| | First 5 | 65 | 1·1 |
| | Remainder | 20 | 0·33 |
| | 31 to 80 | | |
| | First 5 | 65 | 1·1 |
| | Remainder | 16 | 0·27 |
| | Over 80 | | |
| | First 5 | 65 | 1·1 |
| | Remainder | 13 | 0·22 |
| Other terminal units (rate per unit) | All cases | | |
| | First 8 | 50 | 0·84 |
| | Remainder | 10 | 0·17 |

* Each individual terminal unit should be capable of passing 4·2 litre/s (250 litre/min).

**Table B6.14.** Volume flow rates ('free air') for medical compressed air, based on an initial guage pressure of 7.1 bar. Piping to BS 2871 : Part I, Table X.

| $\Delta p$ (N/m$^3$) | Volume flow in copper pipes having following outside diam. (mm) (litre/s) | | | | | | $\Delta p$ (N/m$^3$) |
|---|---|---|---|---|---|---|---|
| | 12 | 15 | 22 | 28 | 35 | 42 | |
| 20·0 | 0·773 | 1·45 | 4·25 | 8·62 | 15·6 | 23·3 | 20·0 |
| 25·0 | 0·877 | 1·64 | 4·81 | 9·75 | 17·6 | 29·8 | 25·0 |
| 31·5 | 0·999 | 1·87 | 5·47 | 11.1 | 20·0 | 33·8 | 31·5 |
| 40·0 | 1·14 | 2·14 | 6·25 | 12·7 | 22·8 | 38·5 | 40·0 |
| 50·0 | 1·30 | 2·43 | 7·08 | 14·3 | 25·8 | 43·5 | 50·0 |
| 63 | 1·48 | 2·76 | 8·06 | 16·3 | 29·3 | 49·4 | 63 |
| 80 | 1·69 | 3·16 | 9·21 | 18·6 | 33·5 | 56·3 | 80 |
| 100 | 1·92 | 3.58 | 10.4 | 21·0 | 37·8 | 63·7 | 100 |
| 125 | 2·17 | 4·06 | 11·8 | 23·8 | 42·8 | 72·0 | 125 |
| 160 | 2·50 | 4·67 | 13·6 | 27·3 | 49·0 | 82·4 | 160 |
| 200 | 2·84 | 5·29 | 15·4 | 30·9 | 55·5 | 93·2 | 200 |
| 250 | 3·22 | 6·00 | 17·4 | 35·0 | 62·7 | 105 | 250 |
| 315 | 3·66 | 6·83 | 19·8 | 39·8 | 71·3 | 120 | 315 |
| 400 | 4·19 | 7·81 | 22·6 | 45·4 | 8l·3 | 136 | 400 |
| 500 | 4·76 | 8·85 | 25·6 | 51·4 | 92·0 | 154 | 500 |
| 630 | 5·42 | 10·1 | 29·2 | 58·4 | 104 | 175 | 630 |
| 800 | 6·20 | 11·5 | 33·3 | 66·7 | 119 | 199 | 800 |
| 1000 | 7·03 | 13·1 | 37·8 | 75·4 | 135 | 225 | 1000 |
| 1250 | 7·98 | 14·8 | 42·8 | 85·4 | 152 | | 1250 |
| 1600 | 9·17 | 17·0 | 49·1 | 97·9 | | | 1600 |
| 2000 | 10·4 | 19·3 | 55·6 | | | | 2000 |
| 2500 | 11·8 | 21·9 | | | | | 2500 |
| 3150 | 13·4 | 24·9 | | | | | 3150 |
| 4000 | 15·4 | | | | | | 4000 |

## Flow Data

Table C4.41 gives flow rates and pressure drop values for compressed air systems. In general, design should be such that the pressure drop to any terminal, at maximum demand, does not exceed 5 to 10 per cent of the initial pressure: velocities of up to about 6 m/s represent normal practice.

For the particular case of compressed air supplies in hospitals, it is recommended that the pressure drop/flow data quoted in Table B6.14 be used. This has been extracted and transposed from the DHSS publication previously mentioned.

## MEDICAL VACUUM

All vacuum data are given in terms of absolute (laboratory) units reading in terms of positive pressure from absolute zero.

**Table B6.15.** Design ('free air') flow rates for medical vacuum.

| Area | Accommodation | Design flow rate (litre) Nominal (per minute) | SI (per second) |
|---|---|---|---|
| Theatres (rate per theatre) | Up to 8 | | |
| | Each | 80 | 1·34 |
| | 9 to 12 | | |
| | First 8 | 80 | 1·34 |
| | Remainder | 48 | 0·8 |
| | 13 to 16 | | |
| | First 8 | 80 | 1·34 |
| | Remainder | 40 | 0·67 |
| | Over 16 | | |
| | First 8 | 80 | 1·34 |
| | Remainder | 36 | 0·6 |
| Anaesthetic room (rate per room) | Up to 8 | | |
| | Each | 40 | 0·67 |
| | 9 to 12 | | |
| | First 8 | 40 | 0·67 |
| | Remainder | 24 | 0·4 |
| | 13 to 16 | | |
| | First 8 | 40 | 0·67 |
| | Remainder | 20 | 0·34 |
| | Over 16 | | |
| | First 8 | 40 | 0·67 |
| | Remainder | 18 | 0·3 |
| Recovery wards (rate per bed) | Up to 8 | | |
| | Each | 40 | 0·67 |
| | 9 to 12 | | |
| | First 8 | 40 | 0·67 |
| | Remainder | 24 | 0·4 |
| | 13 to 16 | | |
| | First 8 | 40 | 0·67 |
| | Remainder | 20 | 0·34 |
| | Over 16 | | |
| | First 8 | 40 | 0·67 |
| | Remainder | 18 | 0·3 |
| Intensive therapy unit or area (rate per bed) | All cases | | |
| | Each | 20 | |
| Wards (rate as stated) | High dependency areas | | |
| | Each bed* | 12 | 0·2 |
| | Intermediate care | | |
| | First 6 beds, total | 40 | 0·67 |
| | Next 9 beds, each | 3·5 | 0·06 |
| | Remainder, each | 2·5 | 0·05 |
| | Self care | | |
| | First 6 beds, total | 40 | 0·67 |
| | Next 9 beds, each | 3·5 | 0·06 |
| | Remainder, each | 2·5 | 0·05 |
| Other terminal units† (rate per unit) | All cases | | |
| | Each | 20 | 0·34 |

* A minimum of 0·67 litre/s (40 litre/min) should be allowed for in designing any individual pipeline.

† It may be assumed that a diversity of 50% can be applied to terminal units in other departments.

The requirements for medical vacuum terminal points are given in Table B6.8. Flow rates are given both in SI and nominal units to avoid confusion.

### Design Flow Rates

Table B6.15 gives design flow rates for various applications and it will be noted that these provide an empirical allowance for diversity in use. Fig. B6.1 provides similar data for general wards. Distribution pipework should be sized using these two sets of design flow rates but it is advisable to check the results following the routine laid down in the DHSS publication. The central plant should be sized using these data, with due allowance for standby equipment and it should be noted particularly that equipment must be specified with clarity as to the volumetric throughput i.e. in terms of free air or air at the required reduced pressure.

### Dental Departments

Vacuum service to dental chairs is often provided by local plant and a special characteristic of high flow (approximately 5 litre/s of free air) is required, with outlet to the waste system. The operation of a pipeline pump to produce a vacuum of 0·65 to 0·75 bar absolute pressure has been found adequate.

### FIow Data

Table B6.16 (overleaf) lists pressure change/flow data for medical vacuum and has been compiled by extraction and transposition of DHSS information. The pipework distribution system should be designed so that the pressure change from the central vacuum plant to the suction point, at the outlet from the terminal unit, does not exceed 10 per cent of the suction available (or about 7 kN/m$^2$).

### Fittings Losses

Typical losses for "high duty" brazed fittings and valves may be allowed for by substituting an equivalent length of straight pipe. Table B6·17 gives suitable values based on various authorities, including manufacturers' technical literature. Figures in respect of valves refer to the fully open position.

**Table B6.17.** Equivalent length of pipe for fittings.

| Item | Length of equivalent pipe (m) for stated outside diameter* (mm) 12 | 15 | 22 | 28 | 35 | 42 | 54 | 76 |
|---|---|---|---|---|---|---|---|---|
| Gate or ball valve | 0·3 | 0·3 | 0·6 | 0·9 | 0·9 | 1·1 | 1·2 | 1·2 |
| Globe or diaphragm valve | 4·0 | 6·0 | 9·0 | 9·0 | 13·0 | 14·0 | 19·0 | 21·0 |
| Angle valve | 2·4 | 3·1 | 4·0 | 5·0 | 6·0 | 7·0 | 9·0 | 14·0 |
| Tee (through) | 0·2 | 0·3 | 0·6 | 0·6 | 0·6 | 0·9 | 1·2 | 1·6 |
| Tee (branch) | 0·9 | 1·0 | 1·9 | 1·9 | 2·2 | 2·8 | 3·5 | 4·0 |
| Elbow (90) | 0·4 | 0·5 | 0·7 | 0·9 | 1·1 | 1·2 | 1·4 | 1·6 |

* BS 2871 : Part I, Table X.

**Table B6.16.** Volume flow rates ('free air') for medical vacuum, based on an absolute pressure at pump suction of 0·48 bar (360 mm Hg). Piping to BS 2871: Part I, Table X.

| $Dp$ (N/m$^3$) | Volume flow in copper pipes having following outside diam. (mm) (litre/s) | | | | | | | | $Dp$ (N/m$^3$) |
|---|---|---|---|---|---|---|---|---|---|
| | 12 | 15 | 22 | 28 | 35 | 42 | 54 | 76 | |
| 3·15 | | | | 0·585 | 1·06 | 1·82 | 3·75 | 9·76 | 3·15 |
| 4·0 | | | | 0·672 | 1·22 | 2·08 | 4·30 | 11·2 | 4·0 |
| 5·0 | | | | 0·764 | 1·39 | 2·37 | 4·88 | 12·7 | 5·0 |
| 6·3 | | | | 0·847 | 1·59 | 2·70 | 5·57 | 14·4 | 6·3 |
| 8·0 | | | 0·505 | 1·00 | 1·82 | 3·10 | 6·39 | 16·5 | 8·0 |
| 10·0 | | | 0·574 | 1·14 | 2·07 | 3·53 | 7·26 | 18·7 | 10·0 |
| 12·5 | | | 0·652 | 1·30 | 2·36 | 4·01 | 8·24 | 21·2 | 12·5 |
| 16·0 | | | 0·751 | 1·50 | 2·72 | 4·62 | 9·49 | 24·4 | 16·0 |
| 20·0 | | | 0·853 | 1·71 | 3·09 | 5·26 | 10·8 | 27·6 | 20·0 |
| 25·0 | | | 0·969 | 1·94 | 3·52 | 5·98 | 12·3 | 31·4 | 25·0 |
| 31·5 | | | 1·11 | 2·22 | 4·02 | 6·83 | 14·0 | 35·7 | 31·5 |
| 40·0 | | | 1·27 | 2·55 | 4·62 | 7·83 | 16·0 | 40·9 | 40·0 |
| 50·0 | | 0·482 | 1·44 | 2·90 | 5·26 | 8·90 | 18·2 | 46·3 | 50·0 |
| 63·0 | | 0·552 | 1·64 | 3·32 | 6·01 | 10·2 | 20·8 | 52·8 | 63·0 |
| 80·0 | | 0·634 | 1·88 | 3·81 | 6·90 | 11·7 | 23·8 | 60·4 | 80·0 |
| 100 | | 0·722 | 2·14 | 4·33 | 7·84 | 13·3 | 27·1 | 68·5 | 100 |
| 125 | | 0·822 | 2·43 | 4·93 | 8·92 | 15·1 | 30·7 | 77·7 | 125 |
| 160 | 0·496 | 0·949 | 2·80 | 5·69 | 10·3 | 17·4 | 35·4 | 89·3 | 160 |
| 200 | 0·567 | 1·08 | 3·18 | 6·47 | 11·7 | 19·8 | 40·2 | | 200 |
| 250 | 0·648 | 1·23 | 3·61 | 7·36 | 13·3 | 22·5 | 45·7 | | 250 |
| 315 | 0·743 | 1·41 | 4·13 | 8·42 | 15·2 | 25·7 | | | 315 |
| 400 | 0·858 | 1·62 | 4·73 | 9·67 | 17·5 | 29·4 | | | 400 |
| 500 | 0·980 | 1·84 | 5·37 | 11·0 | 19·9 | | | | 500 |
| 630 | 1·13 | 2·11 | 6·13 | 12·6 | | | | | 630 |
| 800 | 1·30 | 2·42 | 7·03 | | | | | | 800 |

## NON-MEDICAL COMPRESSED AIR

Under no circumstances should non-medical and medical compressed air systems be mixed, either by pipework interconnections or the common use of compressors. It is preferable to keep medical and non-medical compressed air systems in separate compartments or rooms.

### Pathology Departments

For pathology departments, compressed air at a gauge pressure of approximately 3·5 bar is required to serve bench outlets and equipment such as that listed in Table B6.18. For the purpose of calculating compressor size, the first general bench outlet should be assumed to require 1·34 litre/s and all others 0·34 litre/s.

**Table B6.18.** Typical pathology laboratory equipment. Compressed air requirements.

| Equipment | Gauge pressure (bar) | Design flow rate (litre) | |
|---|---|---|---|
| | | Nominal (per minute) | SI (per second) |
| De-ionizing plant | 0·4 | 60 | 1·0 |
| Flame photometer | 1·4 | 20 | 0·34 |
| Glassware drying machine | 2·8 | 85 | 1·42 |

### Industrial and General

The rate of discharge for compressed air points in industrial use may best be obtained after consultation with the machine or tool manufacturer. Emphasis should be placed, in such an enquiry, upon exactitude of definition, i.e. quantity in terms of air at the operating pressure or under free air conditions. Table B6.19 gives examples from the vast range of equipment in use.

**Table B6.19.** Compressed air requirements for typical industrial applications.

| Equipment | Gauge pressure (bar) | Consumption of free air (litre/s) |
|---|---|---|
| Air motors | | |
| Per kW | — | 16 to 22 |
| Contractors' tools | | |
| Breakers, diggers etc. | 5·5 | 2 to 35 |
| Controls | | |
| Typical | 1·0 | 0·005 to 0·01 |
| Fluidics devices | | |
| Typical | 0·05 to 1·5 | 0·05 |
| Laboratories | | |
| Typical bench point | 1·5 | 5 |
| Typical bench point | 5·5 | 15 |
| Workshop tools | | |
| Blast cleaners, small | 6·5 | 10 to 13 |
| Blast cleaners, large | 6·5 | 100 to 110 |
| Percussive, light | 5·5 | 2 to 8 |
| Percussive, heavy | 5·5 | 10 to 15 |
| Rotary (e.g. drills etc.) | 3·5 to 5·0 | 2 to 15 |
| Spray guns | 2·0 to 5·0 | 0·5 to 10 |

### Simultaneous Demands

It is not possible to generalize on the subject of the probable simultaneous demand upon the central plant. Tools have not only varying use but also a varying consumption depending on the power output. For laboratories, reports indicate that simultaneous demand may be very small, a figure of 5 per cent being quoted in one instance.[2]

### Pipework Distribution

In an ideal compressed air installation, all cooling and condensing should be carried out before the air leaves the receiver. However, this is seldom achieved in practice. To ensure effective operation of any compressed air equipment it is essential to ensure that the mains are installed with an adequate fall to purpose-made drainage points. The general layout of the building will usually dictate the best positions for drainage points but, in general, the main should be given a fall of not less than 1 in 100 in the direction of flow, and the maximum distance between drainage points should not exceed 30 metres.

Provided that care is exercised in design to prevent excessive pressure drops, it is common to size compressed air mains on a velocity basis. For practical purposes, a reasonable velocity is 6 m/s which is sufficiently low to prevent excessive pressure drops on most systems and to minimise the entrainment of moisture which may collect at the bottom of the pipe.

Consideration should be given to a ring main system within the distribution network to enhance system flexibility and capacity and to reduce the overall velocity and pressure losses due to the pipework. All branches should be taken off the top of the main and drop to low level with a strainer at the base of the dropper.

Manual operation of drain points is seldom employed on the distribution mains, the majority of installations requiring the provision of automatic air traps. Air binding of traps can occur on initial start-up or where there is the possibility of large quantities of water collecting at the trap. This problem can be eliminated by the use of pressure balance pipes between the main and the trap. A strainer should be installed before each trap.

Airborne water particles will be carried along with the air and these will not be removed by the normal drain points. Therefore separators should be incorporated in the system to remove airborne water droplets. An in-line separator should be fitted in the main as it leaves the receiver and also at any terminals serving equipment prone to damage by water.

The use of a suitable air dryer adjacent to the receiver in the plant room will often eliminate the need for traps on the distribution system. However, each system must be evaluated separately.

For compressed air equipment serving automatic control systems, it is necessary to consider the use of filters and air dryers in conjunction with the compressor (see CIBSE Applications Manual, Automatic Controls and their Implications for Systems Design).

Advice on air quality requirements for equipment operated by compressed air should be sought from the equipment manufacturer. Many sensitive pieces of equipment require terminal filtration in addition to filtration at the compressor.

Compressed air plant is dealt with in Section B16.

## NON-MEDICAL VACUUM

### Suction Requirements

To place quantities and suction levels in perspective, it is of interest to note that the common laboratory filter pump will evacuate to an absolute pressure of about 1·5 kN/m$^2$ when handling about 0·05 litre/s of free air.

The requirements imposed upon a piped vacuum system vary widely in practice, from 0·1 litre/s free air in laboratories to very small quantities in specialist use, with absolute pressures from 6 kN/m$^2$ down to 100 N/m$^2$. The question of diversity in use is seldom considered in installation design, pipes being sized on an empirical basis as illustrated in Table B6.20.

**Table B6.20.** Approximate pipe sizes for vacuum plants. (For laboratory applications only.)

| Number of points connected | Pipe size (mm) |
|---|---|
| Up to 24 .. .. .. .. .. | 28 |
| 25–50 .. .. .. .. .. | 42 |
| 50–150 .. .. .. .. .. | 54 |
| 150–200 .. .. .. .. .. | 67 |
| Over 200 .. .. .. .. .. | 76 |

*Notes:*
(1) No outlet branch should be less than 15 mm.
(2) Each outlet should be fitted with an integral restriction to prevent implosion of thin-walled glass vessels due to 'snap vacuum'.

### Pathology Departments

For pathology departments, a vacuum at bench fittings of 0·48 bar absolute pressure is normally adequate, the flow rate being 0·5 litre/s of free air. Bench fittings are often throttled to either 0·1 or 0·34 litre/s to limit the size of the plant required, the basis of determination being to allow 0·5 litre/s for the first fitting connected and 0.1 litre/s for each of the remainder.

## STEAM

Where steam supplies are required to serve engines, presses and industrial equipment generally, confirmation of the relevant consumptions and requisite pressures should be obtained from the manufacturers. The following details give approximate data with respect to equipment commonly encountered in hospitals, laundries and kitchens.

### Sterilizers

Specification data for sterilizers is contained in DHSS Technical Memorandum 105 and Table B6.21 lists the items relevant to steam supply. It must be emphasised that a constant and unfluctuating supply of dry saturated steam, with a dryness fraction of approximately 0·9, at a gauge pressure between 3·5 and 4 bar is essential if sterilizers are to operate at maximum efficiency.

### Laundries

The approximate steam requirement of a commercial or institutional laundry,[6] including washing, drying and finishing, is 3·5 to 4·5 kg per kg dry mass for heat recovery machines and 4 to 5 kg per kg dry mass for conventional machines. For domestic work the average dry mass per piece is 0·3 kg and for institutional work is 0·33 kg per piece.

For laundry machinery average steam consumptions and supply pressures are listed in Table B6.22

**Table B6.22.** Approximate steam requirements for laundry machinery.

| Machine | Steam gauge pressure (bar) | Approx. consumption: Dry weight (g/g) | Approx. consumption: (g/s) |
|---|---|---|---|
| Continuous driers | | | |
| (typical) | 7 | 1·5 | — |
| 4·4 m × 2·1 m | — | — | 52 |
| 4·7 m × 1·1 m | — | — | 35 |
| 5·8 m × 2·1 m | — | — | 65 |
| 7·6 m × 2·1 m | — | — | 78 |
| Flatwork ironers | | | |
| (typical) | 7 | 1·0 | — |
| 0·15 m/s | — | — | 30 to 39 |
| 0·22 m/s | — | — | 45 to 72 |
| 0·30 m/s | — | — | 71 to 98 |
| 0·37 m/s | — | — | 97 to 117 |
| 0·45 m/s | — | — | 117 to 143 |
| Presses (typical) | 7 | — | — |
| Cabinet shirt | — | — | 10 to 13 |
| Free steam | — | — | 5 |
| Pony or yoke | — | — | 2 |
| Scissors | — | — | 4 |
| Sleeve form | — | — | 1 to 2 |
| Twin rotary | — | — | 8 |
| Tumbler driers | | | |
| (typical) | 7 | 1·5 | — |
| 7 kg capacity | — | — | 6 to 9 |
| 14 kg capacity | — | — | 15 to 20 |
| 28 kg capacity | — | — | 30 to 46 |
| 45 kg capacity | — | — | 45 to 72 |
| 59 kg capacity | — | — | 75 |
| Washing machines | | | |
| (typical) | — | 1·0 | — |
| Counter flow | 4 | 0·5 | — |
| End loading | 3·5 | 1·0 to 1·5 | — |
| Side loading | 3·0 | 1·0 to 2·0 | — |
| Washer extractors | 4 or 7 | 0·5 | 40 |

**Table B6.21.** Approximate steam and condensate requirements for sterilizers.

| Type | Chamber size: Dimensions (h × w × l) /mm | Chamber size: Nominal capacity /litre | Steam: Inlet size /mm | Steam: Peak flow /(kg/s) | Steam: Max. output /(kg/cycle) | Steam: Safety valve vent * /kg/mm) | Condensate: Jacket size /mm | Condensate: Drain Size * / mm | Condensate: Drain Max. flow /(litre/min) |
|---|---|---|---|---|---|---|---|---|---|
| Porous loads | 660 × 660 × 660 | 300 | 25 | 0·08 | 20 | 28 | 15 | 42 | 25 |
| | 910 × 660 × 660 | 400 | 25 | 0·08 | 25 | 35 | 15 | 42 | 25 |
| | 910 × 660 × 990 | 600 | 25 | 0·08 | 30 | 35 | 15 | 42 | 25 |
| | 910 × 660 × 1320 | 800 | 25 | max. 0·08 | 50 | 35 | 15 | 54 | 25 |
| Bottled fluids | 910 × 660 × 660 | 400 | 22 | 0·08 | 55 | 35 | | 42 | 40 |
| | 910 × 660 × 990 | 600 | 22 | 0·08 | 65 | 35 | | 42 | 40 |
| | 910 × 660 × 1320 | 800 | 22 | 0·08 | 80 | 35 | | 42 | 40 |
| | 910 × 660 × 1600 | 970 | 22 | 0·08 | 130 | 35 | | 42 | 40 |
| Unwrapped instruments and unwrapped utensils | 300 dia. × 500 | 35 | 22 | 0·03 | 3 | 22 | 15 | 25 | 20 |
| | 510 dia. × 510 | 100 | 22 | 0·03 | 10 | 22 | 15 | 42 | 30 |
| | 450 dia. × 760 | 130 | 22 | 0·03 | 10 | 22 | 15 | 42 | 30 |
| | 510 dia. × 910 | 180 | 22 | 0·03 | 13 | 28 | 15 | 42 | 45 |
| | 510 × 510 × 910 | 240 | 22 | 0·03 | 15 | 35 | 15 | 42 | 45 |

*Typical values

### Kitchens

Approximate rates of steam supply to individual items of kitchen equipment are quoted in Table B6.23

Service intakes to kitchens having some appliances steam fed and the remainder served by gas have been referred to in a previous paragraph. On the assumption that all items suited to steam service are so supplied the requirement at a gauge pressure of about 3 bar may be estimated from the data[3] listed in Table B6.24. The quantities listed are for all equipment in full use.

**Table B6.23.** Approximate steam requirements for kitchen appliances.

| Appliance | Steam gauge pressure (bar) | Approximate consumption (g/s) |
|---|---|---|
| Boiling pan | | |
| 90 litre capacity | 1·0 | 15·5 |
| 135 litre capacity | 1·0 | 17·5 |
| 180 litre capacity | 1·0 | 18·5 |
| Dishwasher | | |
| 2000 piece capacity | 0·7 to 2 | 15 to 30 |
| 6000 piece capacity | 0·7 to 2 | 30 to 45 |
| Hot cupboard with shelves | | |
| 1800 mm long unit | 0·3 to 3 | 4·0 |
| Sterilizing sink coil | 0·7 to 1 | 6·0 |
| Wet steaming oven | | |
| per 0·2 m compartment | 0·2 to 0·3 | 10 |
| Bains Maries | 1·0 | 1·7 to 4·2 |
| Tilting kettles | | |
| per 40 litre capacity | 1·0 | 13 |

**Table B6.24.** Approximate connected steam load for 'steam/gas' or 'steam/electric' kitchens.

| Type | Connected steam load for stated number of meals served (g/s) | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | 50 | 100 | 200 | 300 400 | 500 600 | 700 800 | 900 1000 | 1250 | 1500 |
| Restaurants* | 17 | 22 | 38 | 74 | 88 | 105 | 110 | — | — |
| Institutions† | 32 | 47 | 98 | 118 | 168 | 216 | 240 | 270 | 300 |

* Includes hotels and restaurants serving breakfast, lunch and dinner. Also public cafeterias serving lunch plus snacks only.

† Includes holiday camps and hospitals serving breakfast, lunch and dinner. Also private canteens serving lunch plus snacks only.

## SIMULTANEOUS DEMANDS

Reference has been made under previous paragraph headings in this section to empirical methods used to determine the number of fittings connected to a piped system which may be in use simultaneously. Problems involving demand fall under three broad headings:

(*a*) Those where the co-ordination between the times of use of the fittings is known as, for instance, a centrally timed supply to a number of items of process equipment.

(*b*) Those where no special co-ordination is known other than the probable ratio between the period of flow and the period of usage as, for instance, a wash-basin which is filled with water in, say, 1 minute but not used again for 5 minutes.

(*c*) Those where the co-ordination is known to involve the simultaneous use of all the fittings connected.

Of these three types of problem the first and third are capable of easy solution, but the second can be approached only by:

(*a*) Empirical means.

(*b*) The application of the theory of probability.[7.8.9]

### Probability–Basic Concept

If there are two types of event: favourable and unfavourable, then:

$$p + q = 1 \quad \text{.. .. .. .. ..} \quad \text{B6.1}$$

where:

$p$ = ratio of favourable events to total events or duration of events to frequency

$q$ = ratio of unfavourable events to total events or duration of events to frequency

If there is only one event being considered, $p$ may be taken as the probability of occurrence and $q$ as the probability of non-occurrence.

The number of different combinations that can be arranged from a given number of elements is given by:

$$C_{nr} = \frac{n!}{r!\,(n-r)!} \quad \text{.. .. .. ..} \quad \text{B6.2}$$

where:

$C_{nr}$ = number of combinations

$n$ = number of elements

$r$ = number of elements in each combination

### Estimation of Coincidence

For a number of like independent events, whose individual probability is the same, the probability of coincidence is:

$$P_c = p^x \quad \text{.. .. .. .. ..} \quad \text{B6.3}$$

where:

$P_c$ = probability of coincidence of a number of events

$p$ = probability of individual events

$x$ = number of events coinciding

This does not exclude the possibility that other events will also coincide: the probability that an event will not occur however being given also by equation B6.3. Out of $n$ possible events, the probability that $r$ events will coincide is, therefore:

$$P_r = p^r . q^{n-r} \qquad \text{B6.4}$$

or:

$$P_r = p^r . (1 - p)^{n-r} \qquad \text{B6.5}$$

where:

- $P_r$ = probability of coincidence of $r$ events
- $p$ = individual probability of occurrence
- $q$ = individual probability of non-occurrence
- $r$ = number of events coinciding
- $n$ = total number of events

**Binomial and Poisson Distributions**

In most building services' applications, the critical consideration is not $r$ particular events but any combination of $r$ events. To find the probability of this it is necessary to multiply the probability of $r$ particular events occurring simultaneously by the number of ways that $r$ events can be chosen from a total of $n$. This number, usually represented by the symbol $C_{nr}$, is given by equation B6.6. Thus the probability that any $r$ events will coincide is given by the expression:

$$P_{nr} = C_{nr} . p^r \ (1 - p)^{n-r}$$
$$= \frac{n!}{r! \ (n-r)!} \ p^r (1 - p)^{n-r} \qquad \text{B6.6}$$

where:

$P_{nr}$ = probability that any $r$ events will coincide

This is known as the binomial distribution function and tables are available from which the various elements may be evaluated.

For small values of $p$ ($p < 0{\cdot}1$) and large values of $n$ ($n > 50$) a close approximation is given by Poisson's formula:

$$P_{nr} = \frac{e^{np} \ (np)^r}{r!} \qquad \text{B6.7}$$

In many calculations where $p$ values are small, the results even for small values of $n$ are likely to be as accurate as the basic data. There are many circumstances when values of this sort will be appropriate but often it will be desirable to know the probability of selected design loads being exceeded. This gives a measure of the degree of risk (or amenity) involved in the design assumption which may be evaluated on the basis described above.

**Practical Application**

It has been shown[7, 10] that an integrated expression, accurate to within 1 per cent, may be used to approximate results obtained from equation B6.6, in terms of the Error Function:

$$\sum_{np-s}^{np+s} P_{nr} = erf \left[ \frac{s}{[2np \ (1 - p)]^{0{\cdot}5}} \right] \qquad \text{B6.8}$$

where:

$s = np + m$

$$erf x = \frac{2}{\pi^{0{\cdot}5}} \int_o^x \frac{- t^2}{exp \ (- t^2) \ dt}$$

Thence, a simple practical expression may be evolved by substitution:

$$m \simeq np + x \ [2np \ (1 - p)]^{0{\cdot}5} \qquad \text{B6.9}$$

where:

- $n$ = Total number of fittings connected
- $m$ = Number of fittings subjected to simultaneous use
- $t$ = Average time a demand is imposed on a fitting for each occasion of use
- $T$ = Average time between occasions of use
- $p$ = $\frac{t}{T}$, the usage ratio

The coefficient $x$ is equivalent to an Error Function which represents the appropriate level of acceptability. The *Guide* has, in previous editions and in Sections B4 and 8 of this edition, used a value of 1·8 as representing a 98·909 per cent acceptable service: Table B6.25 gives a range of other values.

**Table B6.25.** Values of $x$ for equation B6.9.

| x | 0 | 0·02 | 0·04 | 0·06 | 0·08 |
|---|---|---|---|---|---|
| 1·5 | 96·611 | 96·841 | 97·059 | 97·263 | 97·455 |
| 1·6 | 97·635 | 97·804 | 97·962 | 98·110 | 98·249 |
| 1·7 | 98·379 | 98·500 | 98·613 | 98·719 | 98·817 |
| 1·8 | 98·909 | 98·994 | 99·074 | 99·147 | 99·216 |
| 1·9 | 99·279 | 99·338 | 99·392 | 99·443 | 99·489 |
| 2·0 | 99·532 | 99·572 | 99·609 | 99·642 | 99·673 |
| 2·1 | 99·702 | 99·728 | 99·753 | 99·775 | 99·795 |
| 2·2 | 99·814 | 99·831 | 99·846 | 99·861 | 99·874 |
| 2·3 | 99·886 | 99·897 | 99·906 | 99·915 | 99·924 |
| 2·4 | 99·931 | 99·938 | 99·944 | 99·950 | 99·955 |
| 2·5 | 99·959 | 99·963 | 99·967 | 99·971 | 99·974 |
| 2·6 | 99·976 | 99·979 | 99·981 | 99·983 | 99·985 |
| 2·7 | 99·987 | 99·988 | 99·989 | 99·991 | 99·992 |
| 2·8 | 99·992 | 99·993 | 99·994 | 99·995 | 99·995 |
| 2·9 | 99·996 | 99·996 | 99·997 | 99·997 | 99·998 |

Table B6.26 lists results calculated from equation B6.9 for a range of values for $n$ and $p$ and a number of levels of satisfaction. It will be noted that variation in the usage ratio produces divergences at least as significant as variation in the level of satisfaction.

**Tabulated Results**

Tables B6.27 to 37 present data calculated from equation B6.9 for a range of usage ratios and a range of connected fittings. It is perhaps worth emphasis that the body of each table is phrased in *numbers of fittings* likely to be subjected to simultaneous use and not, as in Sections B4 and 8, in units of flow quantity.

**Table B6.26.** Variation in coincident use for various satisfaction rates.

| Per cent satisfaction | Number of fittings in use coincidentally for listed values of *n* and *p* | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | *n* = 2000 | | | | | *n* = 200 | | | | | *n* = 20 | | | | |
| | 0·02 | 0·05 | 0·1 | 0·2 | 0·5 | 0·02 | 0·05 | 0·1 | 0·2 | 0·5 | 0·02 | 0·05 | 0·1 | 0·2 | 0·5 |
| 99·0 | 56 | 125 | 235 | 446 | 1058 | 9 | 18 | 31 | 55 | 118 | 2 | 4 | 6 | 9 | 16 |
| 99·5 | 57 | 127 | 238 | 450 | 1063 | 9 | 19 | 32 | 56 | 120 | 2 | 4 | 6 | 9 | 16 |
| 99·8 | 59 | 130 | 242 | 455 | 1069 | 10 | 20 | 33 | 58 | 122 | 2 | 4 | 6 | 10 | 17 |
| 99·90 | 61 | 132 | 244 | 459 | 1074 | 10 | 20 | 34 | 59 | 123 | 3 | 4 | 6 | 10 | 17 |
| 99·95 | 62 | 134 | 247 | 462 | 1078 | 11 | 21 | 35 | 60 | 125 | 3 | 4 | 7 | 10 | 18 |
| 99·98 | 63 | 136 | 250 | 466 | 1083 | 11 | 22 | 36 | 61 | 127 | 3 | 5 | 7 | 11 | 18 |
| 99·990 | 64 | 138 | 252 | 469 | 1087 | 12 | 22 | 37 | 62 | 128 | 3 | 5 | 7 | 11 | 19 |
| 99·995 | 65 | 139 | 254 | 472 | 1090 | 12 | 23 | 37 | 63 | 129 | 3 | 5 | 7 | 11 | 19 |
| 99·998 | 66 | 141 | 257 | 475 | 1094 | 12 | 23 | 38 | 64 | 130 | 3 | 5 | 8 | 12 | 19 |

**Table B6.27.** Simultaneous demand.

| Number of fittings connected | Number of fittings in simultaneous use for following values of p | | | | |
|---|---|---|---|---|---|
| | 0·0010 | 0·0016 | 0·0025 | 0·0040 | 0·0063 |
| 50 | 1 | 1 | 1 | 1 | 2 |
| 100 | 1 | 1 | 2 | 2 | 3 |
| 150 | 1 | 2 | 2 | 3 | 3 |
| 200 | 1 | 2 | 2 | 3 | 4 |
| 250 | 2 | 2 | 3 | 4 | 5 |
| 300 | 2 | 2 | 3 | 4 | 6 |
| 350 | 2 | 3 | 3 | 5 | 6 |
| 400 | 2 | 3 | 4 | 5 | 7 |
| 450 | 2 | 3 | 4 | 5 | 7 |
| 500 | 2 | 3 | 4 | 6 | 8 |
| 550 | 3 | 3 | 4 | 6 | 8 |
| 600 | 3 | 4 | 5 | 7 | 9 |
| 650 | 3 | 4 | 5 | 7 | 10 |
| 700 | 3 | 4 | 5 | 7 | 10 |
| 750 | 3 | 4 | 6 | 8 | 11 |
| 800 | 3 | 4 | 6 | 8 | 11 |
| 850 | 3 | 4 | 6 | 8 | 12 |
| 900 | 3 | 5 | 6 | 9 | 12 |
| 950 | 4 | 5 | 6 | 9 | 13 |
| 1000 | 4 | 5 | 7 | 9 | 13 |
| 1050 | 4 | 5 | 7 | 10 | 14 |
| 1100 | 4 | 5 | 7 | 10 | 14 |
| 1150 | 4 | 5 | 7 | 10 | 15 |
| 1200 | 4 | 6 | 8 | 11 | 15 |
| 1250 | 4 | 6 | 8 | 11 | 16 |
| 1300 | 4 | 6 | 8 | 11 | 16 |
| 1350 | 4 | 6 | 8 | 12 | 16 |
| 1400 | 5 | 6 | 9 | 12 | 17 |
| 1450 | 5 | 6 | 9 | 12 | 17 |
| 1500 | 5 | 7 | 9 | 13 | 18 |
| 1550 | 5 | 7 | 9 | 13 | 18 |
| 1600 | 5 | 7 | 9 | 13 | 19 |
| 1650 | 5 | 7 | 10 | 14 | 19 |
| 1700 | 5 | 7 | 10 | 14 | 20 |
| 1750 | 5 | 7 | 10 | 14 | 20 |
| 1800 | 5 | 7 | 10 | 15 | 21 |
| 1850 | 5 | 8 | 10 | 15 | 21 |
| 1900 | 6 | 8 | 11 | 15 | 22 |
| 1950 | 6 | 8 | 11 | 15 | 22 |
| 2000 | 6 | 9 | 12 | 16 | 23 |

**Table B6.28.** Simultaneous demand ($p$=0·01).

| Number of fittings connected | Number of fittings in simultaneous use | | | | |
|---|---|---|---|---|---|
| | 0 | 20 | 40 | 60 | 80 |
| 0 | 0 | 1 | 2 | 3 | 3 |
| 100 | 4 | 4 | 4 | 5 | 5 |
| 200 | 6 | 6 | 7 | 7 | 7 |
| 300 | 8 | 8 | 8 | 9 | 9 |
| 400 | 9 | 10 | 10 | 10 | 11 |
| 500 | 11 | 11 | 12 | 12 | 12 |
| 600 | 13 | 13 | 13 | 14 | 14 |
| 700 | 14 | 15 | 15 | 15 | 15 |
| 800 | 16 | 16 | 16 | 17 | 17 |
| 900 | 17 | 18 | 18 | 18 | 18 |
| 1000 | 19 | 19 | 19 | 20 | 20 |
| 1100 | 20 | 20 | 21 | 21 | 21 |
| 1200 | 22 | 22 | 22 | 22 | 23 |
| 1300 | 23 | 23 | 24 | 24 | 24 |
| 1400 | 24 | 25 | 25 | 25 | 25 |
| 1500 | 26 | 26 | 26 | 27 | 27 |
| 1600 | 27 | 27 | 28 | 28 | 28 |
| 1700 | 29 | 29 | 29 | 29 | 30 |
| 1800 | 30 | 30 | 30 | 31 | 31 |
| 1900 | 31 | 32 | 32 | 32 | 32 |

**Table B6.29.** Simultaneous demand ($p$=0·016).

| Number of fittings connected | Number of fittings in simultaneous use | | | | |
|---|---|---|---|---|---|
| | 0 | 20 | 40 | 60 | 80 |
| 0 | 0 | 2 | 3 | 3 | 4 |
| 100 | 5 | 6 | 6 | 7 | 7 |
| 200 | 8 | 9 | 9 | 10 | 10 |
| 300 | 11 | 11 | 12 | 12 | 13 |
| 400 | 13 | 14 | 14 | 15 | 15 |
| 500 | 16 | 16 | 17 | 17 | 18 |
| 600 | 18 | 19 | 19 | 19 | 20 |
| 700 | 20 | 21 | 21 | 22 | 22 |
| 800 | 23 | 23 | 24 | 24 | 24 |
| 900 | 25 | 25 | 26 | 26 | 27 |
| 1000 | 27 | 28 | 28 | 28 | 29 |
| 1100 | 29 | 30 | 30 | 31 | 31 |
| 1200 | 31 | 32 | 32 | 33 | 33 |
| 1300 | 34 | 34 | 34 | 35 | 35 |
| 1400 | 36 | 36 | 37 | 37 | 37 |
| 1500 | 38 | 38 | 39 | 39 | 39 |
| 1600 | 40 | 40 | 41 | 41 | 42 |
| 1700 | 42 | 42 | 43 | 43 | 44 |
| 1800 | 44 | 45 | 45 | 45 | 46 |
| 1900 | 46 | 47 | 47 | 47 | 48 |

**Table B6.30.** Simultaneous demand ($p$=0·025).

| Number of fittings connected | Number of fittings in simultaneous use | | | | |
|---|---|---|---|---|---|
| | 0 | 20 | 40 | 60 | 80 |
| 0 | 0 | 2 | 4 | 5 | 6 |
| 100 | 7 | 8 | 8 | 9 | 10 |
| 200 | 11 | 12 | 13 | 13 | 14 |
| 300 | 15 | 16 | 16 | 17 | 18 |
| 400 | 19 | 19 | 20 | 21 | 21 |
| 500 | 22 | 23 | 24 | 24 | 25 |
| 600 | 26 | 26 | 27 | 28 | 28 |
| 700 | 29 | 30 | 30 | 31 | 32 |
| 800 | 33 | 33 | 34 | 34 | 35 |
| 900 | 36 | 36 | 37 | 38 | 38 |
| 1000 | 39 | 40 | 40 | 41 | 42 |
| 1100 | 42 | 43 | 44 | 44 | 45 |
| 1200 | 46 | 46 | 47 | 47 | 48 |
| 1300 | 49 | 49 | 50 | 51 | 51 |
| 1400 | 52 | 53 | 53 | 54 | 54 |
| 1500 | 55 | 56 | 56 | 57 | 58 |
| 1600 | 58 | 59 | 59 | 60 | 61 |
| 1700 | 61 | 62 | 63 | 63 | 64 |
| 1800 | 65 | 65 | 66 | 66 | 67 |
| 1900 | 68 | 68 | 69 | 69 | 70 |

**Table B6.31.** Simultaneous demand ($p$=0·040).

| Number of fittings connected | Number of fittings in simultaneous use | | | | |
|---|---|---|---|---|---|
| | 0 | 20 | 40 | 60 | 80 |
| 0 | 0 | 3 | 5 | 6 | 8 |
| 100 | 9 | 11 | 12 | 13 | 14 |
| 200 | 16 | 17 | 18 | 19 | 20 |
| 300 | 21 | 23 | 24 | 25 | 26 |
| 400 | 27 | 28 | 29 | 30 | 31 |
| 500 | 32 | 33 | 34 | 35 | 36 |
| 600 | 38 | 39 | 40 | 41 | 42 |
| 700 | 43 | 44 | 45 | 46 | 47 |
| 800 | 48 | 49 | 50 | 51 | 52 |
| 900 | 53 | 54 | 55 | 56 | 57 |
| 1000 | 58 | 59 | 60 | 61 | 62 |
| 1100 | 63 | 64 | 65 | 66 | 67 |
| 1200 | 68 | 69 | 70 | 71 | 72 |
| 1300 | 73 | 74 | 75 | 76 | 77 |
| 1400 | 78 | 79 | 80 | 81 | 82 |
| 1500 | 83 | 84 | 85 | 86 | 86 |
| 1600 | 88 | 89 | 89 | 90 | 91 |
| 1700 | 92 | 93 | 94 | 95 | 96 |
| 1800 | 97 | 98 | 99 | 100 | 101 |
| 1900 | 102 | 103 | 104 | 105 | 106 |

**Table B6.32.** Simultaneous demand ($p$=0·063).

| Number of fittings connected | Number of fittings in simultaneous use | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 0 | 5 | 10 | 15 | 20 | 25 | 30 | 35 | 40 | 45 | 50 | 55 | 60 | 65 | 70 | 75 | 80 | 85 | 90 | 95 |
| 0 | 0 | 2 | 3 | 3 | 4 | 5 | 5 | 6 | 6 | 7 | 8 | 8 | 9 | 9 | 10 | 10 | 11 | 11 | 12 | 12 |
| 100 | 13 | 13 | 14 | 14 | 15 | 15 | 16 | 16 | 17 | 17 | 17 | 18 | 18 | 19 | 19 | 20 | 20 | 20 | 21 | 21 |
| 200 | 22 | 23 | 23 | 23 | 24 | 24 | 25 | 25 | 26 | 26 | 26 | 27 | 27 | 28 | 28 | 28 | 29 | 29 | 30 | 30 |
| 300 | 31 | 31 | 32 | 32 | 32 | 33 | 33 | 34 | 34 | 34 | 35 | 35 | 36 | 36 | 36 | 37 | 37 | 38 | 38 | 38 |
| 400 | 39 | 40 | 40 | 40 | 41 | 41 | 41 | 42 | 42 | 43 | 43 | 43 | 44 | 44 | 45 | 45 | 45 | 46 | 46 | 46 |
| 500 | 47 | 48 | 48 | 48 | 49 | 49 | 50 | 50 | 50 | 51 | 51 | 51 | 52 | 52 | 53 | 53 | 53 | 54 | 54 | 54 |
| 600 | 55 | 56 | 56 | 56 | 57 | 57 | 57 | 58 | 58 | 59 | 59 | 59 | 60 | 60 | 60 | 61 | 61 | 62 | 62 | 62 |
| 700 | 63 | 63 | 64 | 64 | 65 | 65 | 65 | 66 | 66 | 66 | 67 | 67 | 68 | 68 | 68 | 69 | 69 | 69 | 70 | 70 |
| 800 | 71 | 71 | 72 | 72 | 72 | 73 | 73 | 73 | 74 | 74 | 75 | 75 | 75 | 76 | 76 | 76 | 77 | 77 | 77 | 78 |
| 900 | 79 | 79 | 79 | 80 | 80 | 80 | 81 | 81 | 81 | 82 | 82 | 83 | 83 | 83 | 84 | 84 | 84 | 85 | 85 | 85 |
| 1000 | 86 | 87 | 87 | 87 | 88 | 88 | 88 | 89 | 89 | 89 | 90 | 90 | 91 | 91 | 91 | 92 | 92 | 92 | 93 | 93 |
| 1100 | 94 | 94 | 95 | 95 | 95 | 96 | 96 | 96 | 97 | 97 | 97 | 98 | 98 | 98 | 99 | 99 | 100 | 100 | 100 | 101 |
| 1200 | 101 | 102 | 102 | 102 | 103 | 103 | 103 | 104 | 104 | 105 | 105 | 105 | 106 | 106 | 106 | 107 | 107 | 107 | 108 | 108 |
| 1300 | 109 | 109 | 110 | 110 | 110 | 111 | 111 | 111 | 112 | 112 | 112 | 113 | 113 | 113 | 114 | 114 | 115 | 115 | 115 | 116 |
| 1400 | 116 | 117 | 117 | 117 | 118 | 118 | 118 | 119 | 119 | 120 | 120 | 120 | 121 | 121 | 121 | 122 | 122 | 122 | 123 | 123 |
| 1500 | 124 | 124 | 124 | 125 | 125 | 126 | 126 | 126 | 127 | 127 | 127 | 128 | 128 | 128 | 129 | 129 | 129 | 130 | 130 | 130 |
| 1600 | 131 | 132 | 132 | 132 | 133 | 133 | 133 | 134 | 134 | 134 | 135 | 135 | 135 | 136 | 136 | 136 | 137 | 137 | 138 | 138 |
| 1700 | 139 | 139 | 139 | 140 | 140 | 140 | 141 | 141 | 141 | 142 | 142 | 142 | 143 | 143 | 144 | 144 | 144 | 145 | 145 | 145 |
| 1800 | 146 | 146 | 147 | 147 | 147 | 148 | 148 | 148 | 149 | 149 | 149 | 150 | 150 | 151 | 151 | 151 | 152 | 152 | 152 | 153 |
| 1900 | 153 | 154 | 154 | 154 | 155 | 155 | 155 | 156 | 156 | 156 | 157 | 157 | 158 | 158 | 158 | 159 | 159 | 159 | 160 | 160 |

**Table B6.33.** Simultaneous demand ($p$=0·010).

| Number of fittings connected | Number of fittings in simultaneous use | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 0 | 5 | 10 | 15 | 20 | 25 | 30 | 35 | 40 | 45 | 50 | 55 | 60 | 65 | 70 | 75 | 80 | 85 | 90 | 95 |
| 0 | 0 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 10 | 11 | 12 | 13 | 13 | 14 | 15 | 16 | 16 | 17 |
| 100 | 18 | 19 | 20 | 20 | 21 | 22 | 22 | 23 | 24 | 24 | 25 | 26 | 26 | 27 | 28 | 28 | 29 | 30 | 30 | 31 |
| 200 | 32 | 33 | 33 | 34 | 35 | 35 | 36 | 36 | 37 | 38 | 38 | 39 | 40 | 40 | 41 | 41 | 42 | 43 | 43 | 44 |
| 300 | 45 | 46 | 46 | 47 | 47 | 48 | 49 | 49 | 50 | 50 | 51 | 52 | 52 | 53 | 53 | 54 | 55 | 55 | 56 | 56 |
| 400 | 58 | 58 | 59 | 59 | 60 | 61 | 61 | 62 | 62 | 63 | 54 | 64 | 65 | 65 | 66 | 66 | 67 | 68 | 68 | 69 |
| 500 | 70 | 71 | 71 | 72 | 72 | 73 | 73 | 74 | 75 | 75 | 76 | 76 | 77 | 78 | 78 | 79 | 79 | 80 | 80 | 81 |
| 600 | 82 | 83 | 83 | 84 | 84 | 85 | 86 | 86 | 87 | 87 | 88 | 88 | 89 | 90 | 90 | 91 | 91 | 92 | 92 | 93 |
| 700 | 94 | 95 | 95 | 96 | 96 | 97 | 98 | 98 | 99 | 99 | 100 | 100 | 101 | 102 | 102 | 103 | 103 | 104 | 104 | 105 |
| 800 | 106 | 107 | 107 | 108 | 108 | 109 | 110 | 110 | 111 | 111 | 112 | 112 | 113 | 113 | 114 | 115 | 115 | 116 | 116 | 117 |
| 900 | 118 | 119 | 119 | 120 | 120 | 121 | 121 | 122 | 122 | 123 | 124 | 124 | 125 | 125 | 126 | 126 | 127 | 128 | 128 | 129 |
| 1000 | 130 | 130 | 131 | 131 | 132 | 133 | 133 | 134 | 134 | 135 | 135 | 136 | 136 | 137 | 138 | 138 | 139 | 139 | 140 | 140 |
| 1100 | 141 | 142 | 143 | 143 | 144 | 144 | 145 | 145 | 146 | 146 | 147 | 148 | 148 | 149 | 149 | 150 | 150 | 151 | 151 | 152 |
| 1200 | 153 | 154 | 154 | 115 | 155 | 156 | 156 | 157 | 158 | 158 | 159 | 159 | 160 | 160 | 161 | 161 | 162 | 163 | 163 | 164 |
| 1300 | 165 | 165 | 166 | 166 | 167 | 167 | 168 | 169 | 169 | 170 | 170 | 171 | 171 | 172 | 172 | 173 | 174 | 174 | 175 | 175 |
| 1400 | 176 | 177 | 177 | 178 | 178 | 179 | 180 | 180 | 181 | 181 | 182 | 182 | 183 | 183 | 184 | 185 | 185 | 186 | 186 | 187 |
| 1500 | 188 | 188 | 189 | 189 | 190 | 191 | 191 | 192 | 192 | 193 | 193 | 194 | 194 | 195 | 195 | 196 | 197 | 197 | 198 | 198 |
| 1600 | 199 | 200 | 200 | 201 | 201 | 202 | 203 | 203 | 204 | 204 | 205 | 205 | 206 | 206 | 207 | 207 | 208 | 209 | 209 | 210 |
| 1700 | 211 | 211 | 212 | 212 | 213 | 213 | 214 | 215 | 215 | 216 | 216 | 217 | 217 | 218 | 218 | 219 | 219 | 220 | 221 | 221 |
| 1800 | 222 | 223 | 223 | 224 | 224 | 225 | 225 | 226 | 227 | 227 | 228 | 228 | 229 | 229 | 230 | 230 | 231 | 231 | 232 | 233 |
| 1900 | 234 | 234 | 235 | 235 | 236 | 236 | 237 | 237 | 238 | 238 | 239 | 240 | 240 | 241 | 241 | 242 | 242 | 243 | 243 | 244 |

**Table B6.34.** Simultaneous demand ($p$=0·16).

| Number of fittings connected | Number of fittings in simultaneous use | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 0 | 5 | 10 | 15 | 20 | 25 | 30 | 35 | 40 | 45 | 50 | 55 | 60 | 65 | 70 | 75 | 80 | 85 | 90 | 95 |
| 0 | 0 | 3 | 5 | 6 | 7 | 9 | 10 | 11 | 12 | 13 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 |
| 100 | 26 | 27 | 28 | 29 | 30 | 31 | 32 | 33 | 34 | 35 | 36 | 37 | 38 | 39 | 40 | 41 | 42 | 43 | 44 | 45 |
| 200 | 47 | 48 | 49 | 50 | 51 | 52 | 53 | 54 | 55 | 56 | 57 | 58 | 59 | 59 | 60 | 61 | 52 | 63 | 64 | 65 |
| 300 | 67 | 68 | 69 | 70 | 71 | 72 | 73 | 73 | 74 | 75 | 76 | 77 | 78 | 79 | 80 | 81 | 82 | 83 | 84 | 84 |
| 400 | 86 | 87 | 88 | 89 | 90 | 91 | 92 | 93 | 94 | 95 | 95 | 96 | 97 | 98 | 99 | 100 | 101 | 102 | 103 | 104 |
| 500 | 105 | 106 | 107 | 108 | 109 | 110 | 111 | 112 | 113 | 113 | 114 | 115 | 116 | 117 | 118 | 119 | 120 | 121 | 122 | 122 |
| 600 | 124 | 125 | 126 | 127 | 128 | 129 | 130 | 130 | 131 | 132 | 133 | 134 | 135 | 136 | 137 | 138 | 138 | 139 | 140 | 141 |
| 700 | 143 | 144 | 145 | 146 | 146 | 147 | 148 | 149 | 150 | 151 | 152 | 153 | 154 | 154 | 155 | 156 | 157 | 158 | 159 | 160 |
| 800 | 161 | 162 | 163 | 164 | 165 | 166 | 167 | 168 | 168 | 169 | 170 | 171 | 172 | 173 | 174 | 175 | 176 | 176 | 177 | 178 |
| 900 | 180 | 181 | 182 | 183 | 183 | 184 | 185 | 186 | 187 | 188 | 189 | 190 | 191 | 191 | 192 | 193 | 194 | 195 | 196 | 196 |
| 1000 | 198 | 199 | 200 | 201 | 202 | 203 | 203 | 204 | 205 | 206 | 207 | 208 | 209 | 210 | 210 | 211 | 212 | 213 | 214 | 215 |
| 1100 | 217 | 217 | 218 | 219 | 220 | 221 | 222 | 223 | 223 | 224 | 225 | 226 | 227 | 228 | 229 | 230 | 230 | 231 | 232 | 233 |
| 1200 | 235 | 236 | 236 | 237 | 238 | 239 | 240 | 241 | 242 | 243 | 243 | 244 | 245 | 246 | 247 | 248 | 249 | 249 | 250 | 251 |
| 1300 | 253 | 254 | 255 | 255 | 256 | 257 | 258 | 259 | 260 | 261 | 262 | 262 | 263 | 264 | 265 | 266 | 267 | 268 | 268 | 269 |
| 1400 | 271 | 272 | 273 | 274 | 274 | 275 | 276 | 277 | 278 | 279 | 280 | 280 | 281 | 282 | 283 | 284 | 285 | 286 | 286 | 287 |
| 1500 | 289 | 290 | 291 | 292 | 292 | 293 | 294 | 295 | 296 | 297 | 298 | 298 | 299 | 300 | 301 | 302 | 303 | 304 | 304 | 305 |
| 1600 | 307 | 308 | 309 | 310 | 310 | 311 | 312 | 313 | 314 | 315 | 316 | 316 | 317 | 318 | 319 | 320 | 321 | 322 | 322 | 323 |
| 1700 | 325 | 326 | 327 | 328 | 328 | 329 | 330 | 331 | 332 | 333 | 334 | 334 | 335 | 336 | 337 | 338 | 339 | 340 | 340 | 341 |
| 1800 | 343 | 344 | 345 | 346 | 346 | 347 | 348 | 349 | 350 | 351 | 352 | 352 | 353 | 354 | 355 | 356 | 357 | 357 | 358 | 359 |
| 1900 | 361 | 362 | 363 | 363 | 364 | 365 | 366 | 367 | 368 | 369 | 369 | 370 | 371 | 372 | 373 | 374 | 375 | 375 | 376 | 377 |

**Table B6.35.** Simultaneous demand ($p$=0·25).

| Number of fittings connected | Number of fittings in simultaneous use | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 0 | 5 | 10 | 15 | 20 | 25 | 30 | 35 | 40 | 45 | 50 | 55 | 60 | 65 | 70 | 75 | 80 | 85 | 90 | 95 |
| 0 | 0 | 4 | 6 | 8 | 10 | 12 | 14 | 15 | 17 | 19 | 20 | 22 | 24 | 25 | 27 | 28 | 30 | 31 | 33 | 34 |
| 100 | 38 | 39 | 41 | 42 | 44 | 45 | 47 | 48 | 50 | 51 | 52 | 54 | 55 | 57 | 58 | 60 | 61 | 63 | 64 | 66 |
| 200 | 68 | 70 | 71 | 73 | 74 | 76 | 77 | 79 | 80 | 81 | 83 | 84 | 86 | 87 | 88 | 90 | 91 | 93 | 94 | 96 |
| 300 | 98 | 100 | 101 | 103 | 104 | 105 | 107 | 108 | 110 | 111 | 112 | 114 | 115 | 116 | 118 | 119 | 121 | 122 | 123 | 125 |
| 400 | 128 | 129 | 130 | 132 | 133 | 135 | 136 | 137 | 139 | 140 | 141 | 143 | 144 | 146 | 147 | 148 | 150 | 151 | 152 | 154 |
| 500 | 157 | 158 | 159 | 161 | 162 | 163 | 165 | 166 | 167 | 169 | 170 | 172 | 173 | 174 | 176 | 177 | 178 | 180 | 181 | 182 |
| 600 | 185 | 187 | 188 | 189 | 191 | 192 | 193 | 195 | 196 | 197 | 199 | 200 | 201 | 203 | 204 | 206 | 207 | 208 | 210 | 211 |
| 700 | 214 | 215 | 216 | 218 | 219 | 220 | 222 | 223 | 224 | 226 | 227 | 228 | 230 | 231 | 233 | 234 | 235 | 237 | 238 | 239 |
| 800 | 242 | 243 | 245 | 246 | 247 | 249 | 250 | 251 | 253 | 254 | 255 | 257 | 258 | 259 | 261 | 262 | 263 | 265 | 266 | 267 |
| 900 | 270 | 271 | 273 | 274 | 275 | 277 | 278 | 280 | 281 | 282 | 284 | 285 | 286 | 288 | 289 | 290 | 292 | 293 | 294 | 296 |
| 1000 | 298 | 300 | 301 | 302 | 304 | 305 | 306 | 308 | 309 | 310 | 312 | 313 | 314 | 316 | 317 | 318 | 320 | 321 | 322 | 324 |
| 1100 | 326 | 328 | 329 | 330 | 332 | 333 | 334 | 336 | 337 | 338 | 340 | 341 | 342 | 344 | 345 | 346 | 347 | 349 | 350 | 351 |
| 1200 | 354 | 355 | 357 | 358 | 359 | 361 | 362 | 363 | 365 | 366 | 367 | 369 | 370 | 371 | 373 | 374 | 375 | 377 | 378 | 379 |
| 1300 | 382 | 383 | 385 | 386 | 387 | 389 | 390 | 391 | 393 | 394 | 395 | 397 | 398 | 399 | 401 | 402 | 403 | 404 | 406 | 407 |
| 1400 | 410 | 411 | 412 | 414 | 415 | 416 | 418 | 419 | 420 | 422 | 423 | 424 | 426 | 427 | 428 | 430 | 431 | 432 | 434 | 435 |
| 1500 | 437 | 439 | 440 | 441 | 443 | 444 | 445 | 447 | 448 | 449 | 451 | 452 | 453 | 455 | 456 | 457 | 459 | 460 | 461 | 463 |
| 1600 | 465 | 466 | 468 | 469 | 470 | 472 | 473 | 474 | 476 | 477 | 478 | 480 | 481 | 482 | 484 | 485 | 486 | 488 | 489 | 490 |
| 1700 | 493 | 494 | 495 | 497 | 498 | 499 | 501 | 502 | 503 | 505 | 506 | 507 | 509 | 510 | 511 | 513 | 514 | 515 | 516 | 518 |
| 1800 | 520 | 522 | 523 | 524 | 526 | 527 | 528 | 530 | 531 | 532 | 534 | 535 | 536 | 537 | 539 | 540 | 541 | 543 | 544 | 545 |
| 1900 | 548 | 549 | 551 | 552 | 553 | 555 | 556 | 557 | 558 | 560 | 561 | 562 | 564 | 565 | 566 | 568 | 569 | 570 | 572 | 573 |

**Table B6.36.** Simultaneous demand ($p$=0·40).

| Number of fittings connected | Number of fittings in simultaneous use | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 0 | 5 | 10 | 15 | 20 | 25 | 30 | 35 | 40 | 45 | 50 | 55 | 60 | 65 | 70 | 75 | 80 | 85 | 90 | 95 |
| 0 | 0 | 5 | 8 | 11 | 14 | 16 | 19 | 21 | 24 | 26 | 29 | 31 | 34 | 36 | 38 | 41 | 43 | 45 | 48 | 50 |
| 100 | 55 | 57 | 59 | 62 | 64 | 66 | 68 | 71 | 73 | 75 | 78 | 80 | 82 | 84 | 86 | 89 | 91 | 93 | 95 | 98 |
| 200 | 102 | 104 | 106 | 109 | 111 | 113 | 115 | 118 | 120 | 122 | 124 | 126 | 128 | 131 | 133 | 135 | 137 | 139 | 142 | 144 |
| 300 | 148 | 150 | 152 | 155 | 157 | 159 | 161 | 163 | 165 | 168 | 170 | 172 | 174 | 176 | 178 | 181 | 183 | 185 | 187 | 189 |
| 400 | 194 | 196 | 198 | 200 | 202 | 204 | 206 | 209 | 211 | 213 | 215 | 217 | 219 | 221 | 224 | 226 | 228 | 230 | 232 | 234 |
| 500 | 239 | 241 | 243 | 245 | 247 | 249 | 251 | 254 | 256 | 258 | 260 | 262 | 264 | 266 | 268 | 271 | 273 | 275 | 277 | 279 |
| 600 | 283 | 285 | 288 | 290 | 292 | 294 | 296 | 298 | 300 | 302 | 305 | 307 | 309 | 311 | 313 | 315 | 317 | 319 | 321 | 324 |
| 700 | 328 | 330 | 332 | 334 | 336 | 338 | 340 | 343 | 345 | 347 | 349 | 351 | 353 | 355 | 357 | 359 | 362 | 364 | 366 | 368 |
| 800 | 372 | 374 | 376 | 378 | 381 | 383 | 385 | 387 | 389 | 391 | 393 | 395 | 397 | 400 | 402 | 404 | 406 | 408 | 410 | 412 |
| 900 | 416 | 418 | 421 | 423 | 425 | 427 | 429 | 431 | 433 | 435 | 437 | 439 | 442 | 444 | 446 | 448 | 450 | 452 | 454 | 456 |
| 1000 | 460 | 463 | 465 | 467 | 469 | 471 | 473 | 475 | 477 | 479 | 481 | 483 | 486 | 488 | 490 | 492 | 494 | 496 | 498 | 500 |
| 1100 | 504 | 506 | 509 | 511 | 513 | 515 | 517 | 519 | 521 | 523 | 525 | 527 | 529 | 532 | 534 | 536 | 538 | 540 | 542 | 544 |
| 1200 | 548 | 550 | 552 | 555 | 557 | 559 | 561 | 563 | 565 | 567 | 569 | 571 | 573 | 575 | 577 | 580 | 582 | 584 | 586 | 588 |
| 1300 | 592 | 594 | 596 | 598 | 600 | 602 | 605 | 607 | 609 | 611 | 613 | 615 | 617 | 619 | 621 | 623 | 625 | 627 | 630 | 632 |
| 1400 | 636 | 638 | 640 | 642 | 644 | 646 | 648 | 650 | 652 | 655 | 657 | 659 | 661 | 663 | 665 | 667 | 669 | 671 | 673 | 675 |
| 1500 | 679 | 682 | 684 | 686 | 688 | 690 | 692 | 694 | 696 | 698 | 700 | 702 | 704 | 707 | 709 | 711 | 713 | 715 | 717 | 719 |
| 1600 | 723 | 725 | 727 | 729 | 731 | 733 | 736 | 738 | 740 | 742 | 744 | 746 | 748 | 750 | 752 | 754 | 756 | 758 | 760 | 763 |
| 1700 | 767 | 769 | 771 | 773 | 775 | 777 | 779 | 781 | 783 | 785 | 787 | 789 | 792 | 794 | 796 | 798 | 800 | 802 | 804 | 806 |
| 1800 | 810 | 812 | 814 | 816 | 819 | 821 | 823 | 825 | 827 | 829 | 831 | 833 | 835 | 837 | 839 | 841 | 843 | 845 | 847 | 850 |
| 1900 | 854 | 856 | 858 | 860 | 862 | 864 | 866 | 868 | 870 | 872 | 874 | 876 | 879 | 881 | 883 | 885 | 887 | 889 | 891 | 893 |

**Table B6.37.** Simultaneous demand ($p$=0·63).

| Number of fittings connected | Number of fittings in simultaneous use | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | **0** | **5** | **10** | **15** | **20** | **25** | **30** | **35** | **40** | **45** | **50** | **55** | **60** | **65** | **70** | **75** | **80** | **85** | **90** | **95** |
| 0 | 0 | 6 | 10 | 14 | 18 | 22 | 26 | 29 | 33 | 37 | 40 | 44 | 47 | 51 | 54 | 58 | 61 | 65 | 68 | 72 |
| 100 | 79 | 82 | 86 | 89 | 92 | 96 | 99 | 103 | 106 | 110 | 113 | 116 | 120 | 123 | 127 | 130 | 133 | 137 | 140 | 143 |
| 200 | 150 | 153 | 157 | 160 | 164 | 167 | 170 | 174 | 177 | 180 | 184 | 187 | 190 | 194 | 197 | 200 | 204 | 207 | 210 | 214 |
| 300 | 220 | 224 | 227 | 230 | 234 | 237 | 240 | 243 | 247 | 250 | 253 | 257 | 260 | 263 | 267 | 270 | 273 | 277 | 280 | 283 |
| 400 | 290 | 293 | 296 | 300 | 303 | 306 | 310 | 313 | 316 | 319 | 323 | 326 | 329 | 333 | 336 | 339 | 342 | 346 | 349 | 352 |
| 500 | 359 | 362 | 365 | 369 | 372 | 375 | 379 | 382 | 385 | 388 | 392 | 395 | 398 | 402 | 405 | 408 | 411 | 415 | 418 | 421 |
| 600 | 428 | 431 | 434 | 438 | 441 | 444 | 447 | 451 | 454 | 457 | 460 | 464 | 467 | 470 | 474 | 477 | 480 | 483 | 487 | 490 |
| 700 | 496 | 500 | 503 | 506 | 509 | 513 | 516 | 519 | 522 | 526 | 529 | 532 | 536 | 539 | 542 | 545 | 549 | 552 | 555 | 558 |
| 800 | 565 | 568 | 571 | 575 | 578 | 581 | 584 | 588 | 591 | 594 | 597 | 601 | 604 | 607 | 610 | 614 | 617 | 620 | 623 | 627 |
| 900 | 633 | 636 | 640 | 643 | 646 | 649 | 653 | 656 | 659 | 662 | 666 | 669 | 672 | 675 | 679 | 682 | 685 | 688 | 692 | 695 |
| 1000 | 701 | 705 | 708 | 711 | 714 | 718 | 721 | 724 | 727 | 731 | 734 | 737 | 740 | 743 | 747 | 750 | 753 | 756 | 760 | 763 |
| 1100 | 769 | 773 | 776 | 779 | 782 | 786 | 789 | 792 | 795 | 799 | 802 | 805 | 808 | 812 | 815 | 818 | 821 | 824 | 828 | 831 |
| 1200 | 837 | 841 | 844 | 847 | 850 | 854 | 857 | 860 | 863 | 867 | 870 | 873 | 876 | 879 | 883 | 886 | 889 | 892 | 896 | 899 |
| 1300 | 905 | 909 | 912 | 915 | 918 | 922 | 925 | 928 | 931 | 934 | 938 | 941 | 944 | 947 | 951 | 954 | 957 | 960 | 964 | 967 |
| 1400 | 973 | 976 | 980 | 983 | 986 | 989 | 993 | 996 | 999 | 1002 | 1006 | 1009 | 1012 | 1015 | 1018 | 1022 | 1025 | 1028 | 1031 | 1035 |
| 1500 | 1041 | 1044 | 1047 | 1051 | 1054 | 1057 | 1060 | 1064 | 1067 | 1070 | 1073 | 1077 | 1080 | 1083 | 1086 | 1089 | 1093 | 1096 | 1099 | 1102 |
| 1600 | 1109 | 1112 | 1115 | 1118 | 1122 | 1125 | 1128 | 1131 | 1135 | 1138 | 1141 | 1144 | 1147 | 1151 | 1154 | 1157 | 1160 | 1164 | 1167 | 1170 |
| 1700 | 1176 | 1180 | 1183 | 1186 | 1189 | 1193 | 1196 | 1199 | 1202 | 1205 | 1209 | 1212 | 1215 | 1218 | 1222 | 1225 | 1228 | 1231 | 1234 | 1238 |
| 1800 | 1244 | 1247 | 1251 | 1254 | 1257 | 1260 | 1263 | 1267 | 1270 | 1273 | 1276 | 1280 | 1283 | 1286 | 1289 | 1292 | 1296 | 1299 | 1302 | 1305 |
| 1900 | 1312 | 1315 | 1318 | 1321 | 1325 | 1328 | 1331 | 1334 | 1337 | 1341 | 1344 | 1347 | 1350 | 1354 | 1357 | 1360 | 1363 | 1366 | 1370 | 1373 |

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# SECTION B7 CORROSION PROTECTION AND WATER TREATMENT




This edition of Section B7 first published: 1986

# SECTION B7 CORROSION PROTECTION AND WATER TREATMENT

## INTRODUCTION

Many engineering materials, particularly metals, are processed from natural raw materials to provide specific properties. In time, under service conditions, they tend to revert to their original state. This reversion may be a very long term process. Materials will not fail within the designated lifetime if the selection and application has had proper consideration at the design stage and the plant has been operated as envisaged by the designer. Material selection for particular conditions invariably results in a compromise between the ideal and the most economic.

To make the choice it is necessary to know the likely conditions, both environmental and functional, which are to be accommodated. In addition to the operational conditions, account must also be taken of manufacturing, installation, idle and vandal situations. Satisfactory historical service of a material has often resulted in its adoption for specific duties without conscious consideration of all the conditions to be met. It is only when complaints concerning service failures arise that the suitability of the material is questioned.

Selection of materials is undertaken with the knowledge that certain additional protective measures can be adopted to permit otherwise unsuitable materials to be used. The most obvious are the provision of surface coatings, e.g. paint or metallic, and modification of the medium handled to reduce its aggressive nature, e.g. by water treatment.

System performance must also be protected. Deterioration of the system will result from the internal build-up of corrosion products blocking flowpaths. In an inadequately protected system the formation of scale and microbiological growth will also lead to blockages and changes of heat transfer characteristics. To obviate these potential hazards surface coatings and water treatment are again the principal tools.

This Section covers the broad aspects of corrosion processes, the actions and reactions of materials and the methods of treatment in order to provide an overall picture. It then concentrates on those areas and applications which are specific to building services.

## DEFINITIONS AND GLOSSARY

*Aerobic:*
Pertaining to corrosive action, chemical or biological, in the presence of oxygen.

*Anaerobic:*
Pertaining to corrosive action, chemical or biological, in the absence of oxygen.

*Base Metal:*
The metal which has the lower electrode potential (i.e. cathode) when two or more different metals are in contact in an acidic aqueous solution. The base metal corrodes preferentially.

*Crevice Corrosion:*
Corrosion in a narrow aperture due to localised variations in the electrolytic conditions.

*Dezincification:*
The removal by corrosion of the zinc from the beta phase of duplex brass, leaving porous copper.

*Differential Aeration:*
Non uniform oxygen distribution often around crevices, which can increase corrosion locally.

*Noble Metal:*
The metal which has the higher electrode potential (i.e. cathodic) when two or more different metals are in contact in an acidic aqueous solution.

*Passivity:*
A condition, in which the anodic dissolution of a metal is strongly reduced by, e.g. a thin oxide film.

*Polarisation:*
Resistance to current flow at an electrode, produced by the product formed by electrolysis.

*Sacrificial Coatings:*
Metallic coatings which form the anode in an electrolytic cell in which the protected metal becomes the cathode.

*Total Dissolved Solids (TDS):*
The solid residue, expressed in mg/litre, produced by boiling away a sample of water.

## DETERIORATION OF MATERIALS AND SYSTEMS

### Metals

Corrosion is the principal cause of metal component failures. The processes of corrosion are discussed in detail later to indicate the precautions to be taken to obtain optimum performance from materials and systems.

### Plastics and Rubbers

The engineering and environmental capabilities of these materials have improved considerably. However, failures may result from their use in physical conditions for which they have not been formulated. When determining the suitability of plastics and rubbers the mechanical, thermal, chemical, biological and weathering abilities must all be considered in relation to the demands of the application. The wide range of these materials makes it impossible to provide detailed guidance on their selection for specific duties. The designer must consult manufacturers to obtain details of the material characteristics which should then be checked against the guidelines which follow later.

**Fluids**

The components of a system enclosing a fluid will often be subject to both internal and external environmental conditions. The medium enclosed is usually the working fluid for the system. In building services this covers various gases and liquids which have differing effects on materials. The formulation and condition of the fluid may increase the aggressiveness of otherwise inert materials. Water is an example of a liquid, the properties of which vary widely, resulting in very different effects on materials. Such factors must be taken into account when considering materials to be used with a fluid medium, see *Materials Selection.*

**Corrosion**

This action occurs due to a number of different processes which are discussed later. Failure will usually result if there is no control over the rate at which the action takes place. It is therefore essential for this to be recognised and accommodated in the design.

**Erosion**

The loss of constructional materials may result from the impingement of particulate matter in the fluid, or high fluid velocities. This loss may be either the base material itself or the surface oxides which normally provide a protective film against corrosion. The ability of both the fluid handled and the surrounding environment to generate erosion must be considered when evaluating materials for a particular application.

**Deposition**

The presence within a system of suspended or dissolved matter can lead to the formation of a surface coating or scale, which affects flow rates and heat transfer efficiencies and may also induce crevice corrosion. In addition, the sedimentation of particles in low velocity areas may give rise to sludge and corrosive action may result.

**Fatigue and Stress**

Mechanical loading of certain metallic materials may lead to corrosion processes which can result in failure. This is not dealt with in detail here since the conditions under which it occurs rarely arise in building services.

**External Conditions**

Failures often result from corrosion due to the surrounding external environmental conditions which have not been assessed for their effect on the materials of construction.

If a material which is suited to the service is not appropriate for the external conditions, some form of protection must be provided. This may take the form of a surface coating or, possibly, an electric current may be induced to adjust the corrosion action.

## CORROSION OF METALS

Corrosion is the term used to describe the process of conversion of metals into oxides or other compounds. There are two basic types of process involved:

(*a*) Reactions between metals and the atmosphere whereby a surface scale, usually oxide or sulphide, is formed; e.g. tarnishing, high temperature scaling.

(*b*) Electro-chemical reactions between metals and their environments which depend upon the presence of electrolytes to conduct electricity.

There are also areas of overlap, where the presence of one of these processes will encourage the other.

**Non Electro-chemical Corrosion**

The basic reactions between metals and the atmosphere can be extended to cover corrosion caused by the bacteria which frequently arise from atmospheric/system/fluid combination. Surface deposition and scaling are the most common effects. The various situations are described below.

*Scale Production*

Mill-scale is an oxide surface film formed during high temperature working of steels. There is a difference in electrical potential between mill-scale and steel causing electrolytic cells to be set up. Sometimes the mill-scale covers up to 95% of the steel surface so any electrolytic attack is concentrated on the remaining 5% bare steel with a correspondingly concentrated corrosion rate. Therefore, steel should be cleaned of mill-scale by chemical cleaning or blast cleaning.

When water is heated and circulated in a boiler and pipework system, scales formed from mineral salts in the water will tend to deposit on heat transfer and pipe surfaces. The type of scale will depend upon the particular salts in the water and the temperature of operation. The scales can vary from soft sludges to hard, rock-like scales. The thermal conductivity of typical scales are given in Table B7.1. From this it can be seen that a thin scale formed on, say, the flame tube of a boiler can reduce heat transfer to such an extent that the boiler metal will be in danger of failure due to excessive metal temperatures. Therefore, it is advisable to treat the water to limit the scale formation and methods are described under *Water Treatment.*

**Table B7.1.** Thermal conductivities of typical scale materials.

| Material | Thermal conductivity (W/mK) |
|---|---|
| (Boiler Steel) | (50) |
| Calcium phosphate | 3.6 |
| Iron oxide | 2.6 |
| Magnesium phosphate | 2.2 |
| Calcium sulphate | 1.4 |
| Calcium carbonate | 0.36 |
| Silica | 0.17 |
| Typical boiler scale | 2.9 |

*Microbiological Growths*

Organisms generally found in low temperature water systems (e.g. cooling water circuits) fall into three main groups:

(*a*) *Fungi*
Both yeasts and moulds; these result in slime and wood decay by consumption of the cellulose. The deposits that the organisms produce establish differential corrosion cells and shield metal surfaces from any corrosion inhibitors which may be present.

(*b*) *Algae*
Generally coloured blue, blue-green or green by the presence of chlorophyll. Sunlight is necessary for growth. They cause slime deposits and dead algae provide food for bacteria and fungi.

(*c*) *Bacteria*
Only seen when large colonies are present, or under a magnification of about 800x. Some classes of bacteria cause slime, corrosion, gas production and decay of wood. Table B7.2 identifies some types of bacteria that can cause corrosion of metals.

**Table B7.2.** Corrosion of metals by bacteria.

| Bacterium | Action | Problems caused |
|---|---|---|
| *Gallionella*<br>*Crenothrix*<br>*Spaerotilus* | Convert soluble ferrous ions to insoluble ferric ions | Produces iron oxide deposits, increasing corrosion. |
| *Desulfovibrio*<br>*Clostridium*<br>*Thiobacillier* | Hydrogen sulphide producers (sulphate reducers) | Corrodes metals.<br>Reduces chromates.<br>Destroys chlorine.<br>Precipitates zinc. |
| *Thiobacillus* | Sulphuric acid producer | Corrodes metals. |
| *Nitrobacter*<br>*Nutrosomonas* | Nitric acid producers | Corrodes metals. |

*Flue Gases*

The corrosion effects of flue gases are fuel and temperature dependent and relate to the dew-points of the combustion products.

(*a*) *Water dew-point*
When a sulphur-free fuel is burned with 30% excess air the temperature at which moisture begins to condense out in the flue gases is approximately 45°C, which is the water dew-point.

(*b*) *Acid dew-point*
If a fuel containing 1% sulphur is burned with 30% excess air, sulphur oxides are produced and, in addition to the water dew-point, an acid dew-point occurs at approximately 115°C. The acid dew-point is the highest surface temperature at which deposition can occur. Deposition will be in the form of corrosive sulphuric acid. Acid dew-point temperatures vary but are in the range 115 to 130°C for most commercial boilers. They depend upon flame temperature, excess air used and sulphur content of the fuel. There is no appreciable difference in acid dew-point from fuels containing 1 to 4% sulphur but a significant depression occurs in oil fuels containing less than 0·5% sulphur by weight. A significant depression in dew-point temperature can be achieved by limiting the excess air to 1% max. But this can only be achieved with very sophisticated burners, generally in power station applications.

(*c*) *Corrosion rate*
A peak corrosion rate occurs some 25 to 30°C below the acid dew-point but extremely rapid corrosion occurs when surface temperature falls below the water dew-point temperature.

*Fireside Deposits in Heat Generators*

The deposits formed by the products of combustion vary with the type of fuel, the excess air used and the temperature involved. The corrosion mechanism caused by sulphuric acid in the flue gases has already been described. Solid fuel fired plant can be subject to fused fly ash deposits and 'alkali matrix' types where the fly ash is enclosed in a matrix of sulphates of sodium, potassium, iron and calcium. Oil-fired plant can be subject to carbon deposits and sulphur bearing deposits.

All deposits should be removed by regular cleaning. They tend to be hygroscopic, taking up moisture from the atmosphere and forming acid corrosion cells if they are allowed to remain on the metal surface when the boiler is off-line. This corrosion is very concentrated as it occurs in the area below the water dew-point.

**Electro-chemical and Electrolytic Corrosion**

Whether metals are immersed in solutions, exposed to the damp atmosphere or buried in the soil, the corrosion mechanism is electro-chemical in nature and is similar to the process which occurs in a battery.

The corrosion cells have three constituents; an anode, a cathode, and an electrically conducting solution or electrolyte. The anode is the site at which the metal is corroded, the electrolyte is the corrosive medium. The cathode, which forms the other electrode in the cell, is not consumed in the corrosion process. The anode and cathode can both be part of the same metal surface or different metal surfaces in contact. At the anode, the corroding metal passes into the electrolyte as positively charged ions releasing electrons which participate in the cathodic reaction. Hence, the corrosion current between the anode and the cathode consists of electrons flowing within the metal, and ions flowing within the electrolyte.

Often, small corrosion cells form at many points on the same piece of metal. These cells arise from differences in the constituent phases of the metal; from discontinuities in the natural or applied protective surface coating; or from localised variations in the electrolyte.

Crevice corrosion may occur at joints or under surface deposits of extraneous material. The fluid trapped in the crevices may vary in composition and/or concentration from that of the bulk fluid which may affect the formation of any passive protective surface film. The variation of surface potential and electrolyte conductivity may then give rise to localised attack.

*Factors Affecting Corrosion and Oxidation of Metals in Air*

(*a*) *Temperature*
Oxidation or scaling is the predominating mechanism in dry atmospheres and at temperatures greater than 200°C. At temperatures less than 100°C the electro-chemical mechanism is generally dominant. Within the usual ranges of temperature found in the UK, temperature has less influence on wet corrosion than has pollution and duration of wetness.

(*b*) *Duration of wetness*
This is a factor related to pollution, condensation, and relative humidity effects. It does not directly influence the corrosion rate but does affect the overall loss of metal.

(*c*) *Air composition*
Corrosion depends on the contaminants present in the air, i.e. the dust content, the gaseous impurities and moisture content of the air. Near the coast, sea salts will be present whilst in industrial regions, the air will include appreciable quantities of carbon, carbon compounds and sulphur dioxide, along with smaller amounts of hydrogen sulphide, nitrogen dioxide and ammonia. Sulphur dioxide, which is converted to sulphuric acid, is the main accelerator of atmospheric corrosion. Dust particles also play an important role because they may be composed of a corrosive chemical and, with moisture, form a corrosion cell when they settle on a metal surface. They can also absorb sulphur dioxide from the atmosphere and so cause corrosion on contact with metals. In addition, the corrosion products can be hygroscopic and therefore increase the duration of wetness.

Moisture or relative humidity is important in considering atmospheric corrosion. In general, for any corrodible metal, there is a critical humidity below which it does not corrode. The critical humidity is determined largely by the hygroscopic nature of any solid contamination which may be present, and also by that of the corrosion product. Hence, it is greatly influenced by the pollutants in the atmosphere. Broadly, however, the range of critical humidities for steel, copper, nickel and tin can be regarded as falling within the range 50 to 70% r.h.

*Factors Affecting Corrosion of Metals in Water*

(*a*) *Acidity/alkalinity*
The acidity or alkalinity of a solution is defined by its pH value. A pH value of 7 is neutral; greater than 7 denotes alkalinity and less than 7, acidity. Alkaline solutions are generally less corrosive to ferrous systems, i.e. when pH is maintained at 8·5 or above by the addition of an alkali or alkaline salt. This also promotes polarisation of the cathodic areas, thus reducing corrosion rates.

(*b*) *Dissolved gases*
(*i*) *Oxygen:* The rate of corrosion increases with concentration, typical levels being 2 to 10 ppm. Differential aeration cells around crevices and deposits can be formed, producing a non-uniform oxygen distribution.

(*ii*) *Carbon dioxide:* Carbon dioxide can influence the corrosion rate both directly, due to its acidic nature by providing a source of hydrogen ions for the cathodic reaction, and indirectly by preventing the formation of a protective scale. Concentrations up to 40 ppm are not unusual in steam and condensate lines.

(*c*) *Dissolved salts*
(*i*) *Aggresive ions:* Sulphate and chloride ions are generally aggressive in that they inhibit the formation of a protective scale. In particular, chloride ions can break down passivity with a tendency to more localised attack. Sulphate ions under anaerobic conditions can provide an environment for micro-biological corrosion of ferrous materials.

(*ii*) *Non-aggressive ions:* Ions such as calcium and magnesium bicarbonate have inhibitive properties. The Langelier Saturation Index expresses the corrosion or inhibition potential of water in terms of free, or ‘aggressive’, carbon dioxide content by comparing the known pH to an equilibrium saturation value determined by experiment. The Index is the numerical difference between these values. Positive values indicate a tendency to protect by scale deposition whereas negative values indicate that the scale is likely to dissolve, leaving unprotected metal liable to corrode. A nomograph for the determination of saturation pH is given in the Appendix and a full explanation of the Langelier Saturation Index is given elsewhere[1,2].

(*d*) *Total dissolved solids (TDS)*
TDS values serve as a useful guide to overall purity and as a measure of conductivity, particularly when a full analysis is not available.

(*e*) *Metal ions*
Ions of more noble metals in solution can lead to the formation of local bimetallic cells by their reduction and simultaneous deposition on the metal; localised attack may follow (see *Systems–Corrosion Risks and Design*).

(*f*) *Organic matter*
Organic matter can affect the pH, deposition of any protective scale and provide a source of bacterial activity either by sulphate reducing action or by formation of deposits leading to localised attack.

(*g*) *Temperature*
Although difficult to predict or quantify, corrosion rates generally increase with temperature. Heat transfer surfaces are particularly liable to increased corrosion attack.

(*h*) *Flow velocity*
In most instances the corrosion rate increases with the velocity of flow. Erosion may occur at high flow velocities particularly in the presence of entrained gases or solids.

(*i*) *Surface condition*
Dirty surfaces, e.g. those carrying grease, production scale etc., tend to be more susceptible to localised corrosion than clean surfaces.

(*j*) *Dissimilar metals*
Corrosion is accelerated by differences in the electrical potentials of dissimilar metals immersed in an electrolyte. Fig. B7.1 shows the galvanic series of metals and alloys in flowing sea water at 5 to 25°C. The greater the difference in potential the greater the driving force for the electrochemical reaction.

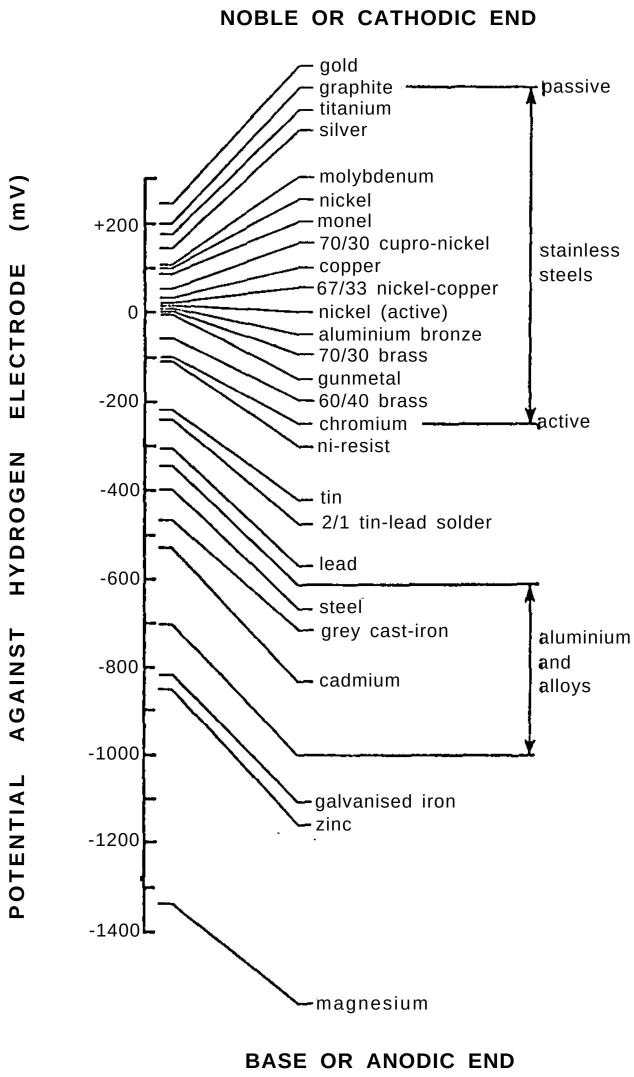


**Fig. B7.1.** Galvanic series of metals and alloys.

(*k*) *Stress*

Stresses in specific metals and alloys can cause stress/corrosion cracking when they are exposed to certain corrosive environments. Typical examples are mild steels in contact with hot chloride solutions. The criteria required to produce this effect are a susceptible alloy/environment combination, tensile stress (either applied or residual from the metal forming processes) and temperature.

*Factors Affecting Corrosion of Metals in Soils*

Corrosion in soils has many of the features of corrosion in water. The main factors determining whether the soil conditions are conducive to corrosion are moisture, aeration and electrical conductivity, the latter being influenced by the presence of dissolved salts and the pH of the soil. The electrolyte to which buried steel structures are exposed will not normally be sufficiently acidic for corrosion to occur other than by a process requiring the presence of oxygen. Corrosion normally requires a soil pervious to air.

An important exception is the attack to which iron and steel are subject in neutral water-clogged clay, to which access of oxygen from the air is precluded by the nature of the soil. In this case, corrosion may proceed by a cathodic reaction, governed by the presence of sulphate reducing bacteria in the soil.

Corrosion of buried metals may be caused by stray electric currents, e.g. from electric railways. The buried metal provides a preferential low resistance path for the stray currents, acting as a cathode where the positive current enters the metal and an anode where it leaves. The effect is worse with direct currents and corrosion can be very severe. Alternating currents may be rectified by oxide films on the metal surface.

## SYSTEMS–CORROSION RISKS AND DESIGN

This section provides a guide to the design of various building services systems, with a view to limiting potential corrosion damage. Influential features are highlighted and typical physical conditions listed.

When designing a system each of these characteristics must be taken into account to provide satisfactory reliability of the components. Particular conditions in some applications will necessitate deviation from the use of conventional solutions. Normal conditions to be expected are identified, so that variations can be recognised and accommodated at the design stage. Such situations may be referred to specialists who can assess the implications and recommend an appropriate alternative.

### Cold Water Services

Cold water services usually include distribution pipework, storage capacity and terminal units having particular functional duties. Various materials are used for these items including metals, ceramics, plastics, jointing materials and sealants. The choice is limited by commercial availability and byelaws which only permit the use of Water Research Centre approved fittings[3] to be in contact with potable water.

Materials to be used in contact with cold water supplies must be selected to combat certain corrosive conditions. In general terms, the water will pass through the system once only. Therefore, it is unlikely that water treatment can be used to eliminate corrosive characteristics at this stage due to the chemical effects on the potability of the water and the economics of treating substantial quantities of water, much of which is going to waste.

The water supplied will have a particular composition and will include impurities which will both aggravate and inhibit corrosion. The local water supply analysis should be considered to identify the corrosive potential.

Because cold water services are generally contained within the building, protection from external corrosion damage is unlikely to be necessary. If thermal insulation is used this should not increase the corrosion damage risk. Joint leakage or condensation can result in wet external surfaces and materials should be chosen to withstand this, or surface protection must be considered, see *Materials Selection.*

*Pipework*

For the distribution of cold water there is a choice of materials and the prime consideration is usually cost. Lead has been used in the past for potable water, particularly when buried in the soil. It is now recognised as a health hazard and is no longer permitted under the Model Water Byelaws (1986)[4]. Galvanised steel, copper and plastics are all in common use. Bare steel or iron are corroded substantially by natural waters and are given surface protection. Galvanised coatings provide electrolytic protection, zinc being the sacrificial anode and steel, the cathode. This protection applies to most hard waters and those which produce inert calcareous scale, but not to acidic or soft waters with a free carbon dioxide content greater than 30 $\mu g/cm^3$.

The use of copper in some soft water areas with free carbon dioxide may result in contamination by dissolved copper. Copper tube may also suffer from pitting corrosion. This usually occurs when the tube contains a surface film cathodic to the underlying metal. Any break in the surface film will result in accelerated attack of the copper beneath. This is known to occur in hard, or moderately hard, waters. Such failures have generally been associated with the presence of a carbon film in the copper tube originating from the manufacturing process. Manufacturers are now aware of the need to remove the carbon film and use mechanical cleaning to ensure a satisfactory surface.

The use of dissimilar metals in a system may create an electrolytic cell, the cathode being displaced. If a combination of metals is to be used, the electrolytic relationship between them should be examined and suitable precautions taken to limit corrosive action.

Light gauge stainless steel tube is used for cold water services. The stainless steel is of the 18% chromium 8% nickel type and is compatible with other plumbing materials. Only limited jointing techniques are suitable for this material at present but work on alternatives is progressing.

*Valves and Fittings*

Materials for these items are generally selected for their engineering properties and, in the case of fittings, for compatibility with the pipe. It is unlikely that corrodible materials will be used but the question of dissimilar metals must be considered. Conventional plumbing solders impart lead contamination to waters and tin/silver solders should now be used. Fluxes used in soldered joints may be corrosive and must be removed or neutralised as soon as possible after use.

Sealants and jointing materials may introduce corrosion hazards, their effect on both adjacent materials and the water contents should be examined.

*Cisterns*

Static pressure and storage cisterns contain water exposed to the atmosphere providing corrosive conditions. Traditionally these have been metallic; galvanised steel, cast iron and copper, being the usual materials. Asbestos cement has also been in service for some time. Certain areas with acidic and soft waters have revealed problems with dissimilar metals, i.e. copper tube with galvanised cisterns, and a non-tainting bituminous paint has been used to protect the cistern surface. Plastic materials are now available for these duties, including polyethylene, polypropylene, unplasticised PVC and GRP.

**Domestic Hot Water Services**

The system supplying domestic hot water will include a heat generator or exchanger, storage capacity, distribution pipework, controls, pumps, terminal outlets and recirculation circuits for larger systems. If the hot water is accessible for human consumption all materials and fittings must comply with the Water Byelaws relating to the supply of potable water.

The cold water supply to the heat generator or exchanger will come either direct from mains or from storage. The composition and quality of water together with the requirements of the cold supply system are as set out under *Cold Water Services.* Conditions and materials for the primary heating circuit are given under *Heating or Steam Services and Condensate Return.*

Like cold water services, domestic hot water is consumable and not continuously recirculated. Therefore, water treatment is not usually a practical or economic solution to corrosion and scaling effects. Potential problems are best dealt with by careful selection of materials and controlled operating conditions. Components may need occasional servicing and should be accessible for inspection and maintenance.

The composition of the water will influence the corrosion potential within the system. Hard waters may be associated with a reversal of the electrical potential between zinc and steel. The protective zinc becomes cathodic in some hard waters at temperatures above 60°C and corrosion of the steel may result. Formation of a protective scale early in the life of a system may overcome this problem but high temperatures and wide temperature variations can displace the scale and expose fresh steel surfaces to corrosion. Scale formation is temperature dependent and operation at lower temperatures in waters with high temporary hardness reduces the rate of build-up of scale.

Mixtures of metals in domestic hot water systems with aggressive waters have led to failures. Small amounts of copper picked up by the flow of water, if deposited on new galvanised steel, will set up an electrolytic cell. The zinc will rapidly corrode and expose the steel base to attack. If copper and galvanised steel are to be used in the same system the copper should only be used downstream of the steel.

Because the system is usually contained entirely within the building it is not subject to weathering damage. Pipework and plant may be thermally insulated and the insulation materials should be compatible with the equipment, even if wetted. Suitable protection must be provided to protect the insulation from the surrounding environment. Leakage should be prevented from saturating the insulation as hidden corrosion may occur.

*Pipework*

The choice of materials for the distribution of hot water is limited. Copper is often the first choice but cost may be prohibitive in certain situations. Soft supply waters from moorland sources may have an adverse effect on copper and supply authorities should be consulted in suspect areas. The second material usually considered is galvanised steel. Generally cheaper than copper, the cost and time taken to install may offset the savings. High temperatures may result in the protective mechanism being upset. The zinc anodic coating will provide protection to the steel base but, in certain hard waters above 65°C, the electrical potential is reversed and corrosion of the steel may result. The mixture of copper and galvanised steel in a system is not desirable, see above.

Increasing use is being made of light gauge stainless steel for hot water pipework. This material is of a similar order of cost to copper but is not so workable. Jointing is the main drawback, although capillary soldered and compression fittings are available. Stainless steel is not aggressive to other metals and may be used with copper and galvanised steel without risk of accelerated failures.

Plastic materials have not yet progressed to a stage of general acceptance for hot water pipework. Activity by a number of manufacturers to develop appropriate material to withstand the temperature range for a reasonable life is likely to result in success in the near future.

*Heat Generators and Exchangers*

Water may be heated either directly through heat generators or indirectly, using either hot water or steam passing through a heat exchanger to raise the domestic hot water temperature. Since hardness will form scale in the hottest parts of a system, care must be taken to avoid blockages. Materials used must resist deterioration from both internal and external conditions.

Heat generators may be of the cast iron sectional type suitable for solid, liquid and gas fuels. Welded mild steel generators are more likely to corrode and are generally avoided for this application.

*Hot Water Storage*

A supply of water is retained ready for use in a cylinder at a suitable temperature and in sufficient quantity to accommodate a reasonable instantaneous demand. The cylinder must withstand the temperature and pressure and must be compatible with other system materials. Copper and galvanised steel cylinders are commonly used. Glass fibre reinforced plastics have been used but operating conditions and cylinder capability must be matched.

The use of electric immersion heaters with thermostatic control can, in the event of thermostat failure, result in excess temperatures which should be taken into account when selecting materials. Water velocities will generally be low and sedimentation may occur.

*Controls, Pumps and Terminal Outlets*

Selection of all plant items must include consideration of the possible contamination of the water and deterioration of the items due to corrosion or scale formation. Water Byelaws must be consulted to ensure that the system is acceptable to the Local Water Authority.

**Heating**

Heating systems are often designed on the assumption that some major elements will last as long as the process or the building they serve. Where components are likely to be subjected to more concentrated wear and/or corrosion hazards, the design should ensure that maintenance or replacement can be carried out.

The heating medium considered here is water. Its composition varies depending on the source and appropriate water treatment must be provided to limit its corrosion potential. The requirements for fresh make-up water should be minimised.

Plant and pipework are normally housed within buildings, so that external corrosion hazards are limited to that occurring from the leaks. Buried or underground mains may be subject to corrosion depending upon the ground conditions and the condition of the protective casing, see *Corrosion of Metals.*

*Boilers*

LTHW systems can use boilers constructed of mild steel or cast iron. MTHW and HTHW systems use mild steel boilers due to the pressures involved. Appropriate water treatment must be employed to eliminate scale formation and corrosion.

Low return water temperatures can cause severe corrosion where sulphur bearing fuels are burned. In addition, the use of wide burner turndown ratios can cause corrosion problems, particularly in flues and chimneys with sulphur bearing fuels when the temperature falls below the dew-point condition.

*Pipework System and Cisterns*

In domestic systems, the pipework usually consists of small bore copper tubing connected to mild steel radiators, with either a mild steel or cast iron boiler. Usually in these closed systems there is no problem with the copper pipes.

There can be a hazard where water is continuously changed as in domestic HWS systems. In rare cases, pitting corrosion may occur with soft waters containing manganese, which deposits manganese dioxide rich scales in the hottest part of the system. This is cathodic to copper and can cause pitting of the tube. With proper water treatment this should not be a problem.

Larger heating systems generally have mild steel distribution pipework and the correct treatment must be provided if internal corrosion is to be avoided. The corrosion hazards are greater where water temperatures are above 100°C and these systems consequently require more comprehensive treatment. The trend is towards complete demineralisation of the make up water with subsequent chemical dosing to correct the pH and eliminate dissolved oxygen. This procedure also reduces the dangers of external corrosion of pipes at leakage points, since there are no dissolved solids in the water to be deposited and concentrated on the metal surface as the leaking water flashes to steam. External treatment may be necessary for buried pipes, e.g. cathodic protection or special external coatings on the pipes.

The use of galvanised steel with copper is dealt with under *Domestic Hot Water Services*. Galvanised steel cisterns are commonly used with copper piped systems. The temperature at this point is low enough to limit electrolytic action. Some cistern failures have been caused by drips of hot, copper bearing, water from vent pipes or flow of copper bearing water through the feed pipe into the cistern. Plastic cisterns eliminate this problem but must be able to withstand any high temperature discharges.

Dosing chemicals must be added so that they reach every part of the system. (A common site for corrosion is the connecting pipe between a pressurisation unit and the system pipework, particularly if there is little temperature change and the water remains fairly static in the pipe.) For large systems it is advisable to install a corrosion monitoring system. All systems should be subject to expert check and chemical adjustments, on a regular basis.

*Valves*

Valves for water systems may include the following materials; brass, non-dezincable brass, gunmetal, cast iron, bronze, cast steel and stainless steel. With correct water treatment these materials should not be affected in the water systems considered. Attention should be paid to gland maintenance since leaks from valve glands can cause deposits to form on spindles resulting in corrosion, thus increasing the leakage rate and rendering the valves useless.

*Heat Exchangers*

These usually consist of mild steel or cast iron shells with copper tubes forming the heat exchange surface. If dissolved oxygen is removed from the circulating water in a closed system there is little problem with corrosion.

Erosion and impingement attack can occur, particularly near to the tube inlet ends where turbulence is greatest. This problem should not occur, provided that the pH has been suitably corrected and the oxygen eliminated.

**Steam Services and Condensate Return**

Materials and components must be selected with due regard to the proposed design life of the system(s).

*Boilers*

Steam boilers are made of mild steel. Before being put into use they should be thoroughly cleaned internally in order to remove all debris and grease.

Boiler feed water can be made up of raw water, suitably softened or treated, or returned condensate plus softened/treated make-up raw water, depending upon the particular process served. Water treatment and conditioning are essential to provide the following water conditions:

(*a*) Fully softened to prevent scale formation on heat exchange metal surfaces.

(*b*) De-aerated, or treated with oxygen scavenging chemicals in order to limit oxygen corrosion of heating surfaces.

(*c*) Maintained in an alkaline condition, in order to sustain a protective oxide film on the metal surfaces and limit the effect of oxygen, chlorides and carbon dioxide introduced with the feed water. The alkalinity should be maintained at not less than 10 to 20% of the total dissolved solid content of the boiler water.

(*d*) The dissolved solids concentration in the boiler must be kept within limits recommended by the manufacturer. This entails a blowdown procedure the amount of which can be assessed from:

$$B = \left(\frac{S_1}{S_2 - S_1}\right) E \quad \ldots \quad \text{B7.1}$$

where:

$B$ = blowdown volume .. .. litre/s
$E$ = boiler evaporation .. .. kg/s
$S_1$ = solids in boiler feed .. .. ppm
$S_2$ = permissible solids concentration in boiler .. .. .. .. ppm

The dissolved solids concentration in the feed water must take into account the dilution effect of recovered clean condensate and any increase in solids due to the addition of conditioning chemicals.

If the calculated blowdown is high, say 8%, then continuous blowdown systems with heat recovery may be required to avoid reducing boiler output, and efficiency, or alternatively a more elaborate feed treatment system may reduce blowdown requirements and consequent heat loss.

(*e*) The recommended boiler water conditions which should be maintained are given under *Chemical Cleaning and Passivation* and *Water Treatment.*

When the plant is burning sulphur-bearing fuels, the temperature of metal surfaces must be maintained above the acid dew-point in order to avoid excessive corrosion.

*Pumps*

Modern boiler feed pumps are usually of the centrifugal type. Adequate static head must be provided above the pump suction in order to avoid cavitation problems when pumping hot feedwater. The pump materials, and the impeller, must be suitable for chemically treated water.

*Feed Tanks*

These are usually fabricated from welded mild steel, treated internally with corrosion protection paints or coatings. Sectional cast iron tanks are also used. Arrangements should be made to maintain the feed tank at a temperature greater than 80°C in order to reduce the dissolved oxygen concentration in the feedwater. It is usually better to have the treated make-up water in a separate cold tank, connected by a balance pipe and a non-return valve to the hot feed tank. Flash steam should be vented from the hot tank away from buildings.

*Feedwater Pipework*

From the feedtank to the boilers, the water should be treated with oxygen scavenging chemicals with the pH corrected to an alkaline condition.

*Blowdown Systems*

The pipework normally carries the blowdown from the boiler into a blowdown tank, thence to drain when cooled. The pipework/blowdown tank system carries solids in the water which will settle out and provision should be made for draindown and cleaning. Internal corrosion is not normally a problem.

*Steam Pipework*

This should be designed to be self-draining of condensate and with provision to vent all the air from the pipework system. Over stressing on expansion/contraction must be avoided by correct pipework design. If the boiler water has not been correctly treated there may be corrosion in the steam pipework due to carry over of solids, $CO_2$ etc. The problem is aggravated where plant is shut down and restarted daily and in such cases extra attention should be paid to the design of air vents, draining, expansion and contraction provisions etc.

*Water Treatment Provisions*

The water treatment plant used will vary according to the raw water quantity and quality. Certain treatment plant requires plastic piping and other corrosion resistant materials to reduce corrosion in the plant. Effluent discharge from water treatment plant may require neutralisation before being discharged into drains in order to avoid corrosion and/or deposition problems.

*Condensate Pipework*

This pipework may be fabricated in mild steel or copper. In the former, corrosion can be very severe where the raw make-up water has a high alkaline hardness and is treated with a base exchange softener. This combination causes bicarbonates in the boiler water to break down and allow carry over of $CO_2$ in the condensate system, resulting in acidic conditions. Amines and other chemical dosing can be used where toxicity is not a problem, or an alternative method of water treatment for the feed-water could be used, e.g. 'de-alkalisation. Although more expensive, copper pipework systems are more resistant to these conditions.

The overall situation must be studied before the pipe material is selected. This study must include analysis of the costs of water treatment. Arrangements should be made to ensure that condensate return pipework is run in a fully flooded condition in order to avoid corrosion at the water/air interface.

*Condensate Receiver Pumping Sets*

Much of the above applies to these units. Since receivers are vented to the atmosphere, oxygen is admitted to the system which can cause corrosion at the condensate/air interface. If receivers are incorrectly sized and pumped mains are allowed to run in conditions other than fully flooded, it is difficult to avoid some corrosion at the water/air interface. When pumping condensate, adequate static head must be provided in order that the pumps function without cavitation.

*Traps*

The correct steam trap must be selected for the particular application, see Section Bl6. In addition to removing all the condensate at all load conditions, the trap should also remove air under start up conditions or, alternatively, suitable air venting arrangements must be made in the equipment served or the steam mains.

**Air Conditioning and Mechanical Ventilation**

Systems can range from simple mechanical ventilation to full air conditioning, incorporating humidity and temperature control. Air conditioning systems are usually designed on the assumption that they should have a working life of not less than twenty years. Such systems obviously demand the maximum attention to prevent corrosion during this or any extended period.

*System Fans and Motors*

The materials of construction of components, e.g. motors, must be suitable for the prevailing conditions of temperature and humidity.

*Ductwork*

The materials of construction of the ductwork and supports must be resistant to corrosion from both internal and external conditions. HVCA Ductwork Specification DW 142[5] covers materials requirements for most conventional air conditioning and mechanical ventilation systems.

*Compressors and Chilled Water Systems*

The basic requirements for ensuring that corrosion and deposition are inhibited are covered under *Water Treatment.* In addition, corrosion can arise from condensation on the external surfaces of these elements. This may be overcome by insulation of the pipework, when in a dry state, using a suitable vapour seal.

The insulation should be installed under supervision since any failure in the seal will quickly render the complete section of insulation useless and create corrosive conditions. Valves, changes in pipework sections and connections to heat exchangers or storage vessels should be defined as boundary points for each vapour sealed section of insulation.

*Humidifiers*

These consist of spray washers, steam humidifiers, or atomising humidifiers:

(*a*) *Spray washers:* These are similar to cooling tower circuits and suffer from the same problems of scale formation, corrosion, and biological fouling. Chemical treatment of the make-up water is necessary to control scale and corrosion and to maintain pH values between 6·5 and 8·0. Arrestor valves are prone to bacterial slime formation and biocide treatment must be carfully selected to avoid any toxic vapours in the conditioned air.

(*b*) *Steam humidifiers:* Individual, unit type, humidifiers contain only small amounts of water, heated by thermostatically controlled elements. Scale preventative dosing should be used with "timed" bleed-off to maintain the concentration of solids within the limits specified by the manufacturer. The bleed-off volume may be calculated using equation B7.1, by substituting "bleed-off" for "blow down" and "humidifier" for "boiler".

(*c*) *Atomising: humidifiers:* These normally use a spinning disc. Any solids in the water result in dust deposits in the conditioned rooms, thus the water to these systems should be fully de-mineralised and treated to maintain pH values between 8 and 9.

**Cooling Water**

The cooling water system includes the cooling tower(s), circulating pumps, distribution pipework and heat exchangers. The type of cooling tower and its location affect the treatment required to prevent corrosion and/or blockages. Roof mounted units tend to be highly rated to reduce weight. Consequently, water treatment methods and control should be more precise since the daily 'turnover' is several times the system capacity.

Materials of construction should be selected to resist corrosive conditions. Mixing of metals should be avoided. Most cooling water systems will be expected to last the life of the air conditioning system. Therefore, proper servicing is essential with adequate provision for replacing components, e.g. tower packing.

The water treatment required will depend upon the make-up water conditions, the concentration factor and the atmospheric conditions.

The following relates to open evaporative systems since these give the greatest problems. Closed cooling systems are easier to protect and are similar to LTHW heating systems with regard to water treatment.

*Cooling Towers*

Many cooling towers contain no more than 300mm depth of water in the basin and, with induced draft, the resultant temperature reduction relates to a make-up rate l-l½ times that of the circulation rate. With high circulation rates there can be a complete change of water every 1½ to 3 hours resulting in the concentration of solids of 4-8 times per 13 hour day. If the water is hard, several kilograms of sludge will be deposited daily. If the water is soft and corrosive, acidic conditions develop. In addition, the cooling tower is washing the atmosphere of airborne deposits. Thus correct water treatment is essential.

If the cooling tower is sited down-wind of boiler chimneys, contamination by sulphur dioxide is likely. This will create acidic conditions in the circulating water causing corrosion. The tower should have facilities for regular washing out of the basin and be fitted with adequate inspection panels. Permanent bleed-off controls should be provided on the return water line. The number of different metals should be minimised. The use of galvanising and copper in the same system must be avoided since this can cause corrosion, particularly if the pH falls due to sulphur dioxide dissolution. To prevent scaling of surfaces there should be a minimum of temporary hardness in the circulating water, the pH should be maintained at between 8·5 and 9·0 and a suitable biocide should be employed to prevent algae, bacteria or slime developing in the tower.

The water quantity for bleed-off may be calculated from:

$$Q = \left(\frac{h_1}{h_2 - h_2}\right) E \qquad \text{B7.2}$$

where:

$Q$ = bleed-off quantity .. .. .. litre/s
$E$ = evaporation .. .. .. .. litre/s
$h_1$ = supply water hardness .. .. .. ppm
$h_2$ = sump water hardness .. .. .. ppm

In relation to biocides, particular attention must be given to the treatment necessary to restrict the growth of the bacterium *Legionella pneumophila* in the cooling tower water systems. Regular chlorination of the system is essential, coupled with the use of other suitable chemical treatments.

*Pumps*

A common fault in these installations is to allow insufficient net positive suction head for the system circulating pump from the water level in the water tower basin. This causes cavitation in the pump, together with air admission, which increases the oxygen content in the circulating water and hence the risk of corrosion. In some cases, air admission can be eliminated by fitting lantern rings to the pump.

*Pipework*

Combinations of different metals should be avoided since the circulating water may become acidic due to atmospheric pollution via the cooling tower. This greatly accelerates the corrosion due to the interaction of the different metals at contact points. Initial chemical cleaning can obviate many future corrosion and deposition problems in cooling water systems.

Correct grading of pipe runs, and their correct positioning to ensure sufficient head, can assist in eliminating excessive oxygen thereby reducing pitting-type corrosion. It is essential that the circulating water is correctly treated and conditioned.

*Heat Exchangers*

The above comments regarding mixing of metals and correct water treatment also apply to heat exchangers which are more prone to blockage and deposition.

**Fire Services**

The fire services considered in this section are those where materials are in contact with water supplies, i.e. sprinklers, drenchers and hose reel systems. Sprinkler and drencher systems can be considered as having the same requirements. Hose reel systems are treated separately.

*Sprinklers and Drenchers*

After initial filling at mains water temperature the water will stabilise to ambient temperature and is unlikely to significantly affect the rate of corrosion. The local water supply will have its own composition including impurities which will both aggravate and inhibit corrosion. The local water supply analysis should be considered to identify the corrosion potential.

In wet systems the material normally used for pipework downstream of the valve station is black mild steel as systems are often welded. Being a one fill system, corrosion is not normally a problem, but if the local water is particularly aggressive some treatment may be required.

Pipework from the mains or pump set, up to the valve station, is subject to replenishment as regular discharging of water occurs when tests are carried out at the valve station. This section of the system requires protection and this is normally achieved by using galvanised mild steel. Such protection is suitable for most waters but not with acidic or soft waters with a free carbon dioxide content greater than 30 $\mu$g/cm$^3$. Ductile iron pipework with a hot applied bitumen based coating may be a more suitable alternative with some waters.

Alternate wet and dry systems are filled with air in winter months and water for the rest of the year. Therefore, there is a limited turnover of water but black mild steel is still used downstream of the valve set and often gives an acceptable life span. In areas with very aggressive water, galvanised mild steel is used but this prohibits extensive fabrication and on-site welding.

*Hose Reel Systems*

A hose reel is fed by independent risers either under mains pressure or boosted from a break tank. There should be no intercommunication with other cold water services.

After initial filling at mains water temperature the system will stabilise to ambient temperature and is unlikely to significantly affect the rate of corrosion. Whilst the system is basically a static one, there will be some turnover of water when hose reels are tested. The local water supply analysis should be considered to identify the corrosion potential.

Galvanised steel and copper tube are in common use for hose reel systems. Galvanised coatings provide electrolytic protection, the zinc being the sacrificial anode and the steel the cathode. This protection applies to most hard waters and those which can produce inert calcareous scale, but not to acidic waters or soft waters with a free carbon dioxide content greater than 30 $\mu$g/cm$^3$.

The use of copper in some soft water areas with free carbon dioxide may result in copper being dissolved into the water. The use of dissimilar metals in a system may create electrolytic cells, the cathode being displaced. If copper and galvanised steel are to be used the copper should be downstream of the steel with a suitable inert fitting between the two materials.

Valves and certain fittings used on copper tubes are manufactured in brass and in certain waters will be subject to dezincification (see under *Materials Selection*). If water analysis suggests this may be a problem, gunmetal or other suitable material should be used.

## Sanitation and Waste Disposal

Soil and waste pipes are seen as a convenient means of disposal of any unwanted liquids and are exposed to many different solutions. Corrosion from natural waters is not normally a problem and greater consideration should be given to chemical and acidic wastes.

*Urinal Wastes*

Corrosion of copper pipework and brass traps and fittings by uric acid can be a problem and such waste pipes are best installed using uPVC or polypropylene. Lime scale will also adhere more readily to copper than plastic and the bore of a copper urinal waste can be reduced considerably in a few years.

*Photographic Equipment Wastes*

Wastes from processing equipment such as X-ray processors should not be carried in metal pipes as the waste solution contains high levels of silver, and electrolytic action will result with all metals commonly used for waste pipes. Silver is the noble metal in all cases and the pipework will suffer corrosion. Processor wastes should be discharged via plastic tubing directly to a silver recovery unit before being discharged to the drainage system.

*Laboratory Wastes*

Corrosion protection in laboratories is very much a case of material selection. A list of the chemicals, temperatures and solution strengths likely to be used in the laboratory must be obtained. This should be checked against the chemical resistance charts for various waste pipe materials. In many small laboratories, polythene or polypropylene may be adequate but in extreme cases, borosilicate glass with PTFE encapsulated ‘O’ rings will be required.

Dilution traps in sink wastes will reduce the solution strength to acceptable levels in laboratories using only small quantities of chemicals. Where glass waste pipes and/or drainage stacks are required due to the nature of waste, further dilution may be required before disposing of the waste to the drainage system. Apart from the limitation of drainage materials there are statutory limits on the discharge of chemicals to sewers and these should be checked with the drainage authority.

## Solar Systems

In general, solar energy systems include:

(*a*) a method of solar energy collection

(*b*) the working fluid transmitting the collected heat

(*c*) a heat exchanger

(*d*) pipework, controls and pumps

(*e*) heat storage to balance the supply with demand.

The collector will be exposed to all atmospheric conditions. Ambient temperature and moisture, airborne pollutants, heat transfer media and aesthetic considerations all influence the choice of materials. Water, possibly including anti-freeze additives, heat transfer oils, oil/water emulsions and air are all used as heat transfer media. Materials must not suffer deterioration from contact with the fluid under any of the anticipated operational or idle conditions.

In most systems the working fluid is retained and recirculated; therefore it may be adjusted chemically to be non-aggressive to the materials with which it is in contact. Special attention is required if anti-freeze or other chemicals are introduced. The system will include both metallic and non-metallic materials.

Air systems may be prone to condensation in idle periods. Adverse biological conditions must also be identified and precautions taken. The external surfaces of the system will be subject to different conditions and must withstand the extremes of the weather, either in their raw state or with protective finishes. Thermal insulation must be protected from the ingress of moisture.

*Collectors*

Materials used must be compatible with each other and capable of withstanding arduous weather conditions. This and mechanical strength are the two main characteristics influencing choice.

Metals and plastics are both used for casings and flow-paths. Selective coatings may be used for the maximum absorption on the exposed surface. The collector will incorporate insulation to prevent heat loss from the back of the panel which must therefore be watertight. Corrosion of the external surfaces and the fluid flow paths must be prevented.

*Heat Transfer Fluid*

The heat transfer fluid must not introduce conditions likely to result in corrosion or the deposition of surface films if the fluid operates within the specified conditions. If the fluid is known to have a limited life then guidance should be provided to enable the user to check its condition. Water Authority approval may be necessary for heat transfer fluids.

*Heat Exchanger*

Heat collected will usually be passed on to a second fluid via an exchanger which must be constructed to be compatible with both fluids over their working ranges. Special care is required if either fluid will create a hazard in the other circuit if cross contamination occurs. Copper cylinders and pipes are commonly used.

*Pipework and Controls*

Similar considerations apply to solar water pipework as for hot water systems. For other fluids the pipework, controls and pump materials must be appropriate. Non-metallic materials may suffer attack from some additives or oils.

**Fumes**

The term “fumes” is used here to describe unwanted, mainly gaseous, emanations from industrial processes, decaying vegetable matter, combustion products and similar processes. Fumes can be corrosive, foul smelling, hot or cold and may contain particulate matter. They may be considered as having corrosion potential. It is impossible to deal with all sources and only general principles are considered here.

The fume removal system includes any collecting hood, induced or forced draft fan, ductwork/stack carrying the fumes away from the building, any scrubbing or washing equipment prior to discharge to atmosphere, and the discharge pipe itself.

*Hot Fumes*

A typical example is flue gas produced from boiler plant burning fossil fuels. Depending upon the chemical constituents in the fumes they can be corrosive having a dew-point condition below which acids are formed. These will attack any containing ductwork and vessels if the materials of construction are not resistant to the particular acid involved.

It is necessary to consider the chemical composition of the fumes, the acid dew-point and the temperature at which the fumes are to be handled. Consideration of these factors will enable suitable materials to be specified.

*Cold Fumes*

As for hot fumes, but there may be a high moisture content, e.g. washing of hot flue gases in order to remove particulate matter which cannot be ejected to atmosphere directly. In such cases, apart from the correct selection of the materials of construction, adequate drain points must be provided in the ductwork and any induced or forced draft fans must take into account the density of the fumes involved.

The fume collecting and treatment system must be designed for ease of installation, maintenance and cleaning, and for ease of replacement where the fumes are corrosive and/or carry particulate matter.

**Fuel Supply**

This section covers various forms of fuel supply systems to boiler plants and similar applications. The fuel system includes all the necessary storage tanks, pipework, valves and other handling equipment necessary for the correct working of the system. The materials and components which comprise the system must be selected to provide the necessary service for the acceptable life of the plant involved. Refer also to Section B13.

*Gas Systems*

These are systems designed for natural gas taken from British Gas mains, either above ground or buried. Above ground mains can be adequately protected by painting externally. Underground mains may require wrapping and, in some cases, cathodic protection. Natural gas is normally dry and there is no danger of internal corrosion so that no internal pipework protection is necessary, apart from normal cleaning out after erection.

*Oil Systems*

Oil systems include all grades of oil storage and distribution from Class D to Class G, see Section B13. Installation of oil storage tanks and pipework must comply with the relevant British Standards. The following materials should not be used where they may be in contact with oil fuels: yellow brass, including low grade alloys of copper and zinc; lead and zinc; galvanised metals; natural rubber.

In general, thermo-plastic materials are not suitable for use with industrial oil fuels although nylon and polytetrafluoro-ethylene (PTFE) are satisfactory for valve seatings, seals and similar purposes.

Mild steel oil storage tanks are usually supplied with all external surfaces painted with a rust inhibiting primer. Above ground tanks may be finished with a good quality paint to suit the surrounding conditions. Internal tank surfaces do not require further protection except in the case of very large vertical storage tanks with large areas of unwetted surface. These should be given a protective paint coating such as red lead oxide primer. Where tanks are buried directly in the ground precautions must be taken to avoid external corrosion due to the soil.

Buried pipework must be protected by suitable wrappings, together with electrical methods of corrosion prevention where dictated by soil conditions. The internal surfaces of oil pipework do not require protection apart from the normal finishing and cleaning out after erection.

*Coal Systems*

The structural steelwork involved in coal and ash handling systems must be protected from atmospheric corrosion. These installations are often in industrial environments where the atmospheric conditions are conducive to extensive atmospheric corrosion.

*Coal Handling Equipment*

Components must be capable of resisting corrosion from wetted coals and erosion due to abrasive action. Suitable coatings may be formed from plastics, metal platings, or concrete depending upon the particular application.

*Ash Handling Equipment*

The components involved in ash handling systems must be carefully selected and protected in view of the extreme conditions of corrosion and erosion which can occur in these systems.

**Electric Lighting and Power**

This section describes the measures to be taken in the design and installation of power and lighting schemes for minimum corrosion problems, but for more detailed considerations, see CIBSE Applications Guide, Lighting in Hazardous and Hostile Environments[6].

Power and lighting schemes include the cabling/wiring, switchgear, control panels, switches, fittings and luminaires. Where components are likely to be subjected to concentrated wear or attack, the overall design should permit easy access for inspection and component replacement.

Corrosion hazards to the overall electrical system are from atmospheric pollution, soil conditions and the internal ambient conditions where process work etc. can release pollutants into the space.

*Underground Cabling*

The corrosion mechanism for metals in soils is described under *Corrosion of Metals.* Cables must be selected with regard to the type of soil and corrosion hazards.

PVC armoured cables to BS 6346[7] are often used. In particularly contaminated soils, cables to BS 6346 with the addition of a lead sheath are preferred. Where conduit and cables are buried in the building structure, corrosion may occur. Building materials containing magnesium chloride and plaster undercoats contaminated with corrosive salts can attack metal. Aluminium sheets, armour and conduit are prone to attack if laid in contact with damp unpainted walls. Conduits embedded in wet plaster walls should be protected by suitable corrosion resistant paints.

*Internal Cabling and Wiring*

The route chosen for wiring should avoid potential exposure to rain, condensate or any corrosive process vapours. Where there is risk of corrosion, the wiring/conduit should have a corrosion resistant finish, e.g. PVC compounds. When considering plastic compounds, the temperature of operation should be borne in mind since certain compounds become brittle at low temperatures.

The fixings of the cables must also be considered for potential corrosion and corrosion resistant materials must be used.

*Cable Glands*

The use of proper sealing glands for cables, complete with shrouds, is essential. After glanding, any armour which may be exposed must be protected by wrapping with a suitable non-absorbent tape. Heatshrink seals or flexible elastomeric gland shrouds provide excellent resistance against corrosive atmospheres, due to their intimate surface contact with the gland.

*Switchgear and Transformers*

Most switchgear is installed within buildings. Where switchgear is located in an unprotected area, it should be specifically designed for outdoor conditions and have the necessary protective coating. Buildings or enclosures within which switchgear is installed should be dry, adequately ventilated and with provision for maintaining temperatures above freezing in the event of the equipment being out of use for some time. If there is a particular corrosion hazard, the manufacturers should be informed so that suitable protective finishes may be applied during manufacture.

*Luminaires*

The materials of construction must be selected bearing in mind the particular corrosion hazards which may be present. Luminaires should be positioned to ensure good ventilation and should not be mounted in pockets in the ceiling, particularly if steam is present. The manufacturer should be consulted during selection. Special attention should be given to specific aspects of the design of parts, e.g. fixing screws, which must also be corrosion resistant and compatible with the other materials of construction. Some grades of PVC-insulated cable incorporate plasticisers which may attack other plastic components in fully enclosed fixings.

## PREVENTION AND PROTECTION

Corrosion rates can be reduced by correct materials selection, interference with the corrosion mechanism by modifying the corrosion environment and by applying electrical control or material coatings.

The appropriate solution should be based on economic considerations for the particular project concerned. For instance, there is little point in selecting an expensive, corrosion resistant material if this then substantially outlives the other components. The material, and protection applied, must relate both to the life of the equipment and to the overall costs involved.

**Materials Selection**

The simplest way of avoiding corrosion is to use materials of construction which are electro-chemically resistant, such as plastics, wood, ceramics or glass. However, they may not possess the necessary structural properties. Metals differ greatly in their corrosion resistance depending upon their position in the electro-chemical series and the adherence, compactness, and self-healing characteristics of the film on the metal surface in contact with the environment. When a protective oxide film is formed the material assumes the behaviour of more noble metals.

Details of the corrosion properties associated with the most common materials used in building services systems are given below.

*Carbon Steel*

The obvious material for many applications is carbon steel. It has little inherent corrosion resistance but alloying provides a means of combining the cheapness of steel with the high corrosion resistance of expensive metals such as nickel. Adding as little as 0·2% of copper to mild steel increases its atmospheric corrosion resistance by rendering the rust film more compact and adherent. Special alloys can be produced to cope with the most corrosive of conditions. However, such remedies are expensive and may give rise to problems of fabrication and/or installation.

*Stainless Steel*

Martensitic and ferritic grades contain 11 to 18% chromium and the austenitic grades contain 17 to 26% chromium and 8 to 22% nickel; the austenitic grades have the highest corrosion resistance. They are best used under fully aerated or oxidising conditions so as to maintain their protective film. They are subject to pitting, crevice corrosion and stress corrosion in certain environments, particularly those containing chlorides. They are resistant to atmospheric corrosion, nitric acid, some concentrations of sulphuric acid and alkalis. The many available grades must be investigated in order to select that most appropriate for the particular application.

*Cast Irons*

(*a*) *Grey cast iron:* contains carbon, silicon, manganese and iron. Carbon (1·7 to 4·5%) is present as combined carbon and graphite. Grey iron castings are not usually considered corrosion resistant, although they do resist atmospheric corrosion and attack from natural or neutral waters and neutral soils. They are resistant to concentrated acids as well as some alkaline and caustic solutions. They are attacked by dilute acid and acid salt solutions.

(*b*) *White heart cast iron:* made by controlling the composition and rate of solidification of the molten iron so that all the carbon is present in the combined form.

(*c*) *Malleable iron:* made from white cast iron with free carbon as dispersed nodules. Total carbon is about 2·5%.

(*d*) *Ductile cast iron:* contains combined carbon and dispersed nodules of carbon. Composition is approximately the same as that of grey iron, with more carbon than malleable iron. The colloidal graphite produces the knot effect produced by the graphite flakes, making the material more ductile.

(*e*) *Alloy cast irons:* utilised so that the corrosion resistance of cast iron can be improved. High silicon cast irons have excellent corrosion resistance with a silicon content of 13 to 16%.

*Copper and its Alloys*

Copper resists seawater, hot or cold fresh water, de-aerated non-oxidising acids, and atmospheric attack. Copper alloys improve the mechanical and physical properties and corrosion resistance of the pure metal.

(*a*) *Brasses:* copper alloys containing from 22 to 29% zinc and 1% tin. Duplex brasses consisting of alpha and beta phases are subject to dezincification in certain waters due to preferential removal of the zinc from the beta-phase brass. The water conditions necessary for dezincification are certain combinations of chloride content, temporary hardness and pH value.

Fig. B7.2 shows the inter-relationship of chloride content and temporary hardness for synthetic and certain natural waters. Waters within the shaded zone are liable to cause meringue dezincification in the cold if the pH is above 8·3. Below 8·3 it is only likely when heated, since this will raise the pH value.

The boundary between safe and suspect waters cannot be finely drawn since other constituents of the water come into play. For the hotter parts of once-through systems in areas where meringue dezincification is known to occur, materials other than duplex brass should be used; copper, arsenical inhibited alpha brass or gun metal are all satisfactory. Such precautions are not necessary for recirculating systems nor usually for the cooler parts of once through systems.

Dezincification causing leakage, rather than blockage, is seen on water fittings both above and below ground, certain types of soil being particularly aggressive. Alpha brasses include 70 to 30% copper/zinc brasses and aluminium brasses, usually with a small arsenic addition to prevent dezincification. These are used in heat exchanger and pipework but are subject to impingement attack at high water velocity.

Sulphur containing compounds in polluted cooling waters can cause sulphide deposits, which are cathodic to brasses, thus setting up corrosion cells.

(*b*) *Gun metals:* copper alloys containing various amounts of tin, zinc, lead and sometimes nickel. (Typically copper 88%, tin l0%, zinc 2% or copper 85%, tin 5%, zinc 5%, lead 5%). Gun metals are probably the best all-round choice for general valves and fittings in water-based heating systems.

(*c*) *Cupro-nickels:* copper with 10 to 30% nickel, iron and manganese additions. They can be classed as weldable alloys with better impingement resistance than brasses. They are used for heat exchangers, especially in sea-water cooling circuits, the cupro-nickel tubes rolled into naval brass tube plates, and in steam heat exchangers to provide greater resistance to corrosion by acid condensate.

(*d*) *Aluminium bronzes:* contain up to 10% aluminium with up to 5% iron with 5% nickel additions. They are used for tube plates, back boilers, pickling cradles, pump impellers, ships' propellers.

*Aluminium and its Alloys*

Aluminium offers good resistance to atmospheric corrosion. In waters, corrosive behaviour depends on chlorides, carbonate hardness and the copper concentration in the water, see Fig. B7.3. Higher chloride and sulphate content encourages general corrosion whilst higher carbonate hardness reduces the number of pits formed. Traces of copper in the water have a profound effect on pitting corrosion; the amount necessary to initiate pitting varies with hardness and chloride content, which can be as low as 0·02 ppm.

Aluminium magnesium and aluminium manganese alloys generally are more corrosion resistant, followed by aluminium magnesium silicon alloys.

*Nickel*

Nickel is resistant to hot or cold alkalis, dilute non-oxidising inorganic and organic acids, and atmospheric conditions. Copper improves its resistance to reducing conditions and to pitting in sea water; chromium, its resistance to oxidising conditions; molybdenum, its resistance to reducing conditions.

*Titanium Alloys*

These have a high resistance to corrosion in sea water and industrial atmospheres and processes. Alloys are used in chemical plants where high costs can be justified.

*Rubbers and Plastics*

The following rubbers are commonly used as gasket materials and for pipe or vessel linings.

(*a*) *Natural rubber:* resistant to dilute minerals, acids, alkalis and salts but is attacked by oxidising mediums, oils, benzenes and ketones.

(*b*) *Chloroprene and neoprene rubbers:* resistant to attack from ozone, sunlight, oils, gasoline and aromatic or halogenated solvents.

(*c*) *Styrene rubber:* has similar properties to natural rubber.

(*d*) *Nitrile rubber:* resists oils and solvents.

(*e*) *Butyl rubber:* has exceptional resistance to mineral acids and alkalis.

(*f*) *Silicone rubbers:* have good resistance to high and low temperatures, aliphatic solvents, oils and greases.

(*g*) *Fluoro-elastomers:* e.g. Viton, combine excellent chemical and high temperature resistance.

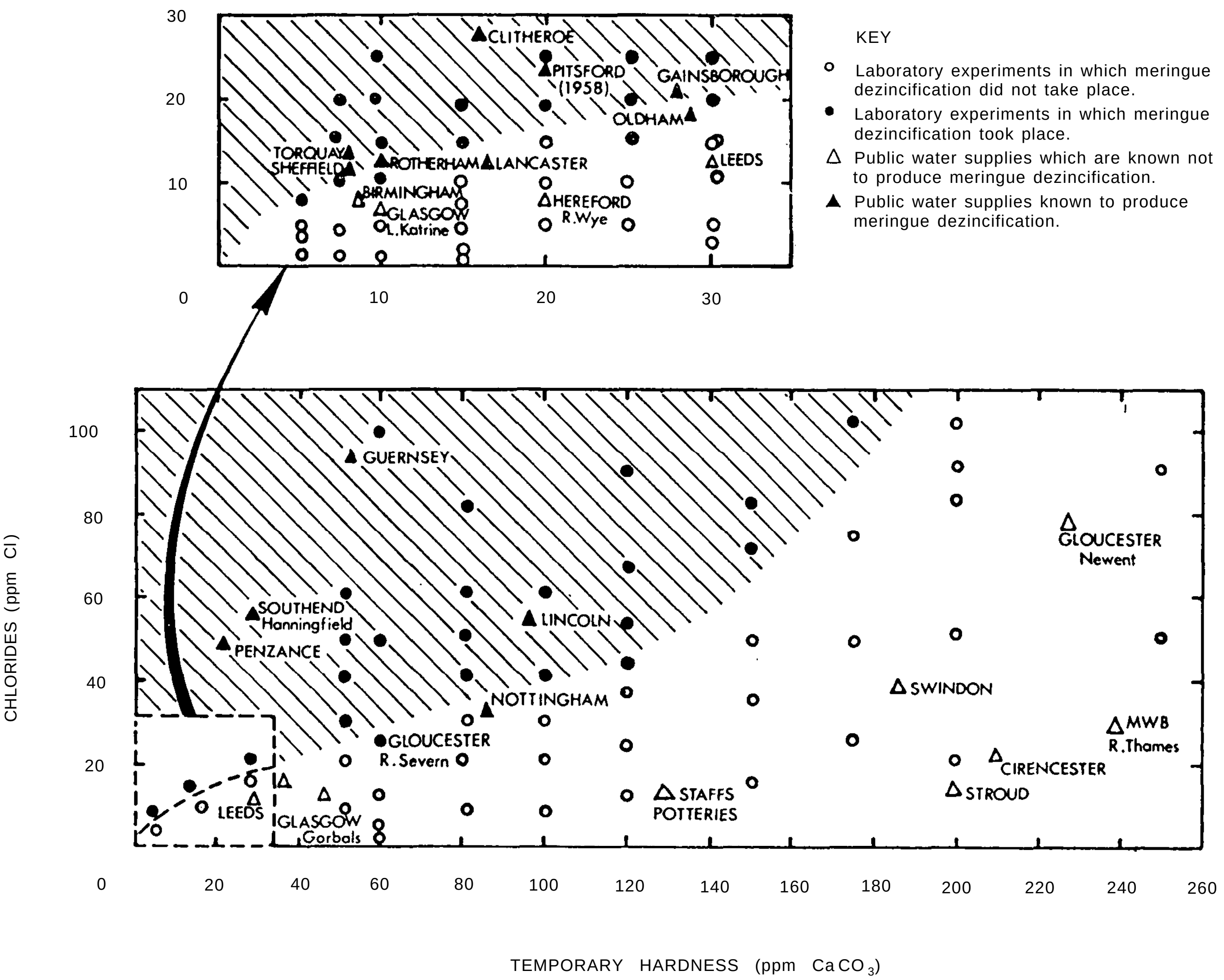


**Fig. B7.2.** Relation between chloride content and temporary hardness.

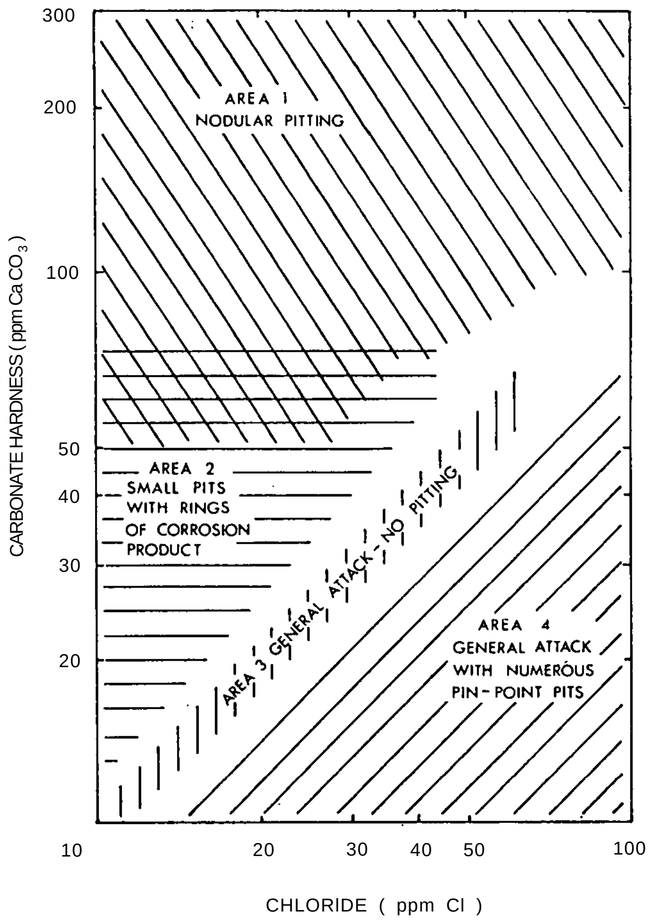


**Fig. B7.3.** Relation between carbonate hardness and chloride content.

The most important plastics used in the building services industry are as follows. The temperatures and related maximum stresses for plastic materials may be critical. If used at elevated temperatures, advice on the suitability of particular materials should be sought from the manufacturers.

(*a*) *Acetal resins:* can replace die cast metals, e.g. for taps or window fittings.

(*b*) *Acrylic resins:* used for sinks and baths.

(*c*) *Acrylonitrile butadiene styrene copolymer (ABS):* used in internal drainage.

(*d*) *Polycarbonate:* used for light fittings and vandal resistant fittings.

(*e*) *Polythene (polyethylene):* used for glazing (poor resistance to sunlight), drains and waste pipes, domestic cold water pipes, buried mains, small bore plumbing.

(*f*) *Polypropylene:* resists higher temperatures.

(*g*) *Plasticised (flexible) PVC:* used for sleeves on electrical cables.

(*h*) *Unplasticised (rigid) PVC:* used for window frames; extrusions; cable ducts; vent ducts; roof lights; rainwater goods; underground drains and fittings; water pipes.

(*i*) *Glass fibre reinforced polyesters (GRP):* used for architectural features; mouldings; panels; tanks; pipes.

(*j*) *Fluorocarbons (PTFE, FEP, PFS):* inert to nearly all chemicals and solvents, have antistick properties, high service temperature up to 200°C. Used as coatings and laminates, tubing and for hygienic uses.

**Modifying the Corrosion Environment**

*Water*

Factors affecting the corrosion rate in water have been described above. The methods of treating water to limit corrosion problems are given under *Chemical Cleaning and Passivation* and *Water Treatment.*

*Atmosphere*

The corrosion potential of the atmosphere depends upon the contaminants present and the moisture content. To minimise corrosion, dust and fumes must be removed and the relative humidity must be reduced. Removal of dust and fumes can be accomplished by static or dynamic filtration or fume scrubbers/chemical filters, depending on the extent and nature of the contamination.

Reduction of the relative humidity is achieved by removal of moisture in the air rather than by heating. This can be achieved by chemical driers using silica gel, amorphous silica and activated alumina. These driers reduce the dew-point by removal of moisture down to 1% r.h. Refrigerated driers are cheaper to operate but may not produce such low levels of r.h. Dehumidification can also be used to protect inside pipe surfaces against corrosion, notably in the case of compressed airlines.

*Soils*

Factors affecting the corrosion rates of metals in soils have been discussed earlier. Acid soil can be made less corrosive if limestone chips are packed around buried metal but it is difficult to modify the environment and resiting the buried structure or pipeline should be considered.

**Surface Protection**

Coatings used to protect metals from corrosion can be divided into metallic, organic and inorganic types, which protect by means of exclusion, inhibition or sacrifice. All coatings exclude the environment to some extent but those that protect by exclusion means alone must completely cover the surface and must be resistant to mechanical damage.

Such coatings include vitreous enamels, lacquers, non-inhibited paints, plastics, and those metal coatings such as nickel and chromium which are more noble than the metal to which they are applied. Soundness is essential in such (cathodic) metal coatings, since they will actively promote corrosion of the metal (anode) at any discontinuities in the coatings.

Sacrificial coatings act as excluders where the coating is sound and also provide cathodic protection at any discontinuity, acting as a sacrificial anode. In these cases the metal coating is less noble than the metal being protected in prevailing corrosion conditions. It is essential that surface cleaning and preparation is carried out thoroughly prior to applying coating.

*Galvanising*

This involves coating the steel surface with zinc, most commonly hot dipped onto the steel. The function of the coating is to delay corrosion of the underlaying steel and this is achieved since zinc protects steel cathodically. When zinc and steel are in contact with each other and with a conducting liquid, a short-circuited electrolytic cell is set up. Current flows from the zinc to the solution, into the iron and back to the zinc, resulting in the dissolution of the zinc leaving the steel unaffected. It is not necessary for the zinc to cover the entire steel surface thus the zinc coating is effective even when scratched. Since zinc is wasting in the corrosion process, the thickness of the coating determines the service lifetime. The corrosion resistance of the zinc is improved by being dipped into a chromate or nitric acid solution, which forms a coherent protective layer on the surfaces.

Care must be taken when selecting galvanised steel for hot water applications. The corrosion rate increases above 50°C to a maximum at approximately 65 to 70°C, and then declines. In soft waters, with appreciable bicarbonate content, a potential reversal of the zinc coating may lead to accelerated attack on the basic metal. In closed hot water systems this may lead to localised pitting. Further factors affecting galvanised components in hot water systems are dealt with later.

Cadmium plate is also used to protect steel, especially in marine environments where the protection of cadmium is cathodic.

*Chromium Plating*

This is used to protect steel and zinc based components. Chromium electro-deposits have a highly protective oxide film that can contain micro cracks. Therefore, the components are first plated with nickel which protects the underlying metal from cracks in the chromium coating. Various combination nickel/chromium coatings offering considerable protection against atmospheric corrosion have been developed.

*Sprayed Coatings*

Many materials can be flame or plasma sprayed to give corrosion resistance coatings. The coatings produced by these processes are thin but the process is very quick. Aluminium and zinc are commonly sprayed to provide protection on steel structures exposed to industrial and marine environments. The best results are obtained by sealing the metal deposit with a paint such as aluminium vinyl.

*Diffusion Coatings*

These are produced by causing elements, usually metals, to diffuse into the surface of steel where it forms a compound having good corrosion resistance. Examples are aluminising, in which aluminium diffuses into the steel to form an iron aluminoid layer which has good corrosion resistance because it is covered by a thin layer of aluminium oxide. Sheradising and aluminising are similar processes. They are normally restricted to low precision components such as pipework grids and condenser plates.

*Paints, Pitch, Tar and Bitumen*

These are common organic coatings applied to metals. With paints, the importance of adequate surface preparation cannot be over-emphasised. The surface being painted must be free from dirt, rust or other corrosion products and, if possible, pre-treated by phosphating, anodising or chromate conversion, to provide adhesion. Structures which are too large for cleaning should be grit blasted. Most paints are used as protection against atmospheric corrosion; protection of buried metal structures is achieved by applying a thick coal tar coating with cathodic protection.

Paints consist of a mixture of pigment and a liquid medium which, after application, dries or reacts to form a coherent coating. Three different coats are usually applied; a priming coat formulated to achieve maximum adhesion and often containing corrosion inhibitors such as red lead or zinc chrome, an undercoat to build up the thickness and a finishing coat designed to provide maximum weathering resistance and the required colour and texture. In general, the final thickness of the three coats should be at least 75$\mu$m. It should be noted that the priming coat is usually porous and will not alone provide protection from corrosion despite the presence of corrosion inhibitors; an impervious finishing coat must always be applied.

*Plastic Coatings*

These are generally available in two types; relatively thin coatings, e.g. epoxy resins and polyurethane, and thicker coatings, e.g. PVC, polyesters, resins, and some fluorocarbons, which are often applied with some kind of reinforcement. The thicker coatings act as chemically resistant linings.

*Vitreous Enamels*

These are applied as a thin layer of glass, bonded to the surface of a metal, usually steel. Grit blasting the surface is essential before enamelling is applied. A low carbon steel is preferred because the presence of carbon in the surface can lead to formation of gas bubbles under the coating. Steels specially made for enamelling contain some titanium and silicon which combine with any carbon present. Enamel coatings are used in chemical plant for their corrosion resistance. They are also used extensively for domestic items because the coatings have good resistance to acids and alkalis and are non-toxic.

*Conversion Coatings*

These coatings are produced by treating the metal surface chemically with an appropriate solution. Phosphate coatings are produced by immersing the metal in a weak phosphoric acid solution of iron, copper or manganese phosphate. They provide only limited protection but make an excellent base for paint or other protective coatings.

Chromate coatings can be produced on aluminium and its alloys, magnesium and its alloys, cadmium and zinc. They form a useful degree of resistance to corrosion and a good preparation for painting. Anodising is an electrolytic process where the metal to be treated is made anodic in a suitable electrolyte to produce a layer of oxide on its surface. This process is applied to various non-ferrous metals but mainly to aluminium and its alloys. It provides some degree of corrosion protection and also is a good pre-treatment for painting.

**Cathodic Protection**

There are two methods of providing cathodic protection for minimising corrosion of metals in use. They are the sacrificial anode method and the impressed current method. Both depend upon making the metal to be protected the cathode in the electrolyte employed.

*Sacrificial Anode Method*

This includes the use of zinc, magnesium or aluminium as anodes in electrical contact with the metal to be protected. Positive direct current flows from the corroding (sacrificial) anode through the soil or electrolyte to any exposed pipe/tank metal, thus preventing ions leaving the metal surface. In buried pipelines, the sacrificial anode is usually connected to the pipe via an insulated wire taken through a link box at ground level. This enables the pipe/soil potentials to be measured together with the current supplied by the anode.

Each anode supplies only a small current (usually 10 to 500 mA for a 10 kg magnesium anode), depending on soil resistivity and area of base metal cathode. The number of anodes required can be calculated from these measurements and the time of operation required. Their use is restricted in practice to soils of less than 3000 ohm/cm capacity. Magnesium anodes have the advantage of the greatest potential difference from iron and a high electro-chemical equivalent (ampere hours per kilogram).

Protection of calorifiers, water tanks etc. can be achieved internally and externally by the use of anodes welded to the outside or suspended centrally. The same principles apply as in the case of pipes in soils. Anodes must be checked regularly and replaced as they are consumed.

*Impressed Current Method*

Similar in principle to the sacrificial method except that the DC power is derived not from a natural difference in potential between anode and metal structure but from a rectified and transformed AC power supply or generator. Anodes of various materials such as graphite, lead, platinum-lead, silicon iron are connected from the source to the metal to be protected, via an insulated cable.

Careful measurements and calculations must be performed to determine the quality and number of anodes used, the operating current and any protection required for other structures in the area. Frequent monitoring of performance is essential since failure of the power supply leaves the metal surfaces unprotected.

**Chemical Cleaning and Passivation**

Chemical cleaning procedures are employed for two main purposes:

(*a*) in the pre-commissioning phases, whereby the pipework system is thoroughly cleaned of oils, greases, mill-scale and other corrosion forming deposits; this follows on from the flushing out of the pipework system to remove loose scale, magnetite and other debris;

(*b*) in order to descale or clean individual items of plant which have built up deposits during operation, e.g. boiler scale, heat exchanger fouling, etc.

The chemicals employed and the method of application, must be agreed between the client and the specialist chemical cleaning contractor. The latter normally provides all the necessary operating labour, chemicals, temporary pipework connections, temporary circulating pumps and test facilities.

Passivation is necessary in order to prevent corrosion of the metal surfaces after the chemical cleaning has been completed.

A guide[8] to solvent applicability and compatibility with various materials is given in Table B7.3 which should be used as a broad guide only. It is not a substitute for detailed discussions with the specialist contractor.

*Procedures for Chemical Cleaning*

A typical pre-commissioning flushing and chemical cleaning procedure might be as follows:

(*a*) Isolate components or materials liable to be damaged by the chemical cleaning fluid. Remember that obstructions such as valves and low points in pipework can trap debris during flushing and chemical cleaning and that isolated components also contain corrosion and debris.

(*b*) Remove as much of the large debris as possible by water flushing. Flushing requires temporary high volume, high head pumps with a volume output ideally at least three times greater than the normal circulating pumps, see CIBSE Commissioning Code W[9].

(*c*) The cleaning solution is circulated via the contractor’s temporary tanks and pumps. Typically the solution may contain citric acid and will be circulated at an elevated temperature. This could entail the use of temporary boiler plant if the installed boiler plant is not commissioned. The chemicals should be circulated until the periodic sampling and analysis indicates that the pipework is clean. The chemicals are then drained down and removed from site. Care must be taken in disposal since certain mineral acids cannot be discharged into normal drainage systems and arrangements must be made to neutralise and dispose of the effluent in an acceptable manner.

(*d*) Further water flushing to remove acid residues must then be undertaken.

(*e*) A suitable passivating chemical solution is then circulated through the pipework system in order to promote the formation of a magnetite layer on the metal surfaces.

(*f*) Finally, the passivating fluid is drained down and the system filled with softened, treated water containing an oxygen scavenger, and pH-corrected to an appropriate value.

The chemical cleaning of individual plant items such as boilers is carried out in a similar manner by circulating chemicals through the plant item. Again, the chemicals must be carefully chosen to be compatible with the metals of construction.

**Table B7.3.** Guide to solvent applicability and compatibility with materials of construction.

| Solvent | Inorganic deposits | | | | | | | | Deposits containing organics and organic deposits | | | | | Materials of construction and maximum cleaning temperature | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Calcium carbonate and magnesium hydroxide | Calcium and magnesium phosphates | Calcium sulphate | Silica and silicates | Iron oxides | Copper oxides | Zinc oxides and salts | Sulphides | Light greases | Light greases and mineral oils | Mineral oils | Animal and vegetable oils | Carbonaceous deposits | Mild steel | Cast iron | Low alloy steel | Stainless steel, martensitic | Stainless steel, austenitic | Copper | Copper alloys | Cupro-nickels | Nickel alloys | Zinc (galvanising) | Aluminium | Other materials on which solvents should *not* be used |
| Hydrochloric acid | ● | ● | | | ● | ● | ● | ● | | | | | | ○ | 49°C | ○ | 60°C | | ○ | ○ | ○ | ○ | | | Enamels |
| Hydrochloric acid+ammonium biflouride | ● | ● | | ● | ● | ● | ● | ● | | | | | | ○ | 49°C | ○ | 60°C | | ○ | ○ | ○ | ○ | | | Enamels, glass silica ware |
| Acetic acid | ● | ● | | | | | | | | | | | | ○ | 49°C | ○ | 60°C | ○ | ○ | ○ | ○ | ○ | † | 66°C | Enamels |
| Formic acid | ● | ● | | | ● | ● | ● | | | | | | | ○ | 49°C | ○ | 60°C | ○ | ○ | ○ | ○ | ○ | | | Enamels |
| Phosphoric acid | | | | | ● | | | | | | | | | ○ | 49°C | ○ | 60°C | ○ | ○ | ○ | ○ | ○ | | | Enamels |
| Sulphamic acid | ● | ● | | | ● | | ● | | | | | | | ○ | 49°C | ○ | 60°C | ○ | ○ | ○ | ○ | ○ | | 66°C | Enamels |
| Suphuric acid | | | | | ● | ● | ● | ● | | | | | | ○ | 49°C | ○ | 60°C | ○ | ○ | ○ | ○ | ○ | | | Enamels |
| Nitric acid | ● | ● | | | ● | ● | ● | | | | | | | | | | ○ | ○ | * | * | * | | | | Enamels |
| Ammoniated citric acid pH 3·5 | | | | | ● | ● | | | | | | | | ○ | ○ | ○ | ○ | ○ | ○ | ○ | ○ | ○ | | | |
| Trisodium phosphate | | | ● | | ● | | | | ● | ● | | | | ○ | ○ | ○ | | | ○ | ○ | ○ | ○ | ○ | ○ | |
| Organic solvents | | | | | | | | | ● | ● | ● | ● | ● | ○ | ○ | ○ | ○ | ○ | ○ | ○ | ○ | ○ | ○ | ○ | Some plastics, etc. |
| Caustic solutions | | | ● | | ● | | | | ● | ● | ● | | ● | ○ | ○ | ○ | | | ○ | ○ | ○ | ○ | | | Glass |

*Key:* ● Solvent may be used on indicated deposit.
○ Solvent may be used on indicated material at all temperatures.
* Rapid dip only.
† Limited attack.

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## MOTHBALLED PROTECTION

Procedures which should be undertaken in order to prevent corrosion taking place in plant and associated systems, where these are out of use, are described below for various systems and components. The aim is to ensure that in "mothball" situations the systems are protected against atmospheric contamination and other forms of corrosion.

*Steam Boilers*

Boilers should be drained down, working out loosely adhering sludge, mud and scale. They should be completely filled to the stop valve level, with treated and deaerated water. Water conditions should be maintained so that pH is 10·5 to 11·0 with an excess of oxygen scavenger, such as sodium sulphite or hydrazine, to ensure no oxygen is present. For extensive shutdown periods, say 6 months or more, the boilers can be drained down and flushed out, dried out with electric heaters, fitted with trays of quicklime and tightly re-sealed and periodically inspected to check conditions.

The flue gas passes should be thoroughly cleaned of deposits, down to the bare metal, dried out, and trays of quicklime placed in the furnace chamber and the gas passes sealed up to eliminate air passage. Alternatively, an alkaline powder (e.g. Dolomite) can be sprayed over the surface prior to sealing.

Ancillary fittings, such as water gauges, float controls and alarms, should be isolated, dismantled, cleaned, lubricated and stored where necessary. This applies particularly to oil burners and stokers. Where it is not desirable to remove the equipment from the boiler, plastic sheeting can be fixed around the equipment forming an air-tight and dust-proof seal.

*Water Boilers*

Water boilers should be dealt with in a similar manner to steam boilers. The water side of these units however forms part of the water system and the water conditions should be as described under *LTHW and HTHW Systems*.

*Chimneys and Flues*

Chimneys and flues should be thoroughly swept and cleaned of all loose deposits internally. Where possible the flues should be sealed to avoid air infiltration, birds nesting, etc. Externally the lagging/external painting should be inspected and repaired where necessary.

*Fuel Storage and Handling Plant*

Oil tanks should be inspected externally and any necessary repairs to lagging or paintwork should be carried out. Oil heaters should be cleaned and protected externally against moisture, possibly by cocooning in plastic sheeting suitably sealed. Coal handling equipment and bunkers should be thoroughly cleaned and lubricated where necessary, with paint renewed as required.

*Steam Condensate System*

Steam mains should be thoroughly drained and sealed to eliminate air and leakage. Condensate mains should be treated in a similar manner with special attention to steam traps and condensate receivers. These should be thoroughly drained and, where necessary, dismantled and checked then reassembled.

*LTHW and HTHW Systems*

All water systems should be left in the fully flooded condition, with the water chemically maintained to give a pH value of 10·5 to 11·0 and with an excess of de-oxygenising chemicals in order to eliminate any dissolved oxygen in the water which would cause pitting corrosion. With this wet system of preservation, anti-frost precautions are essential if the system is to be maintained over the winter months. Where such frost protections cannot be adequately maintained, the system must be thoroughly drained down after being circulated with suitable passivating chemicals injected into the circulating water. This procedure should arrest corrosion to some extent but this method of protection inevitably entails some internal corrosion due to the admission of air to the wet pipe work, thus providing suitable conditions for extensive corrosion.

These procedures also apply to any calorifiers or heat exchangers in the system.

*Pumps and Motors*

Circulating pumps should be thoroughly drained and where necessary dismantled, glands repacked, re-assembled and lubricated as required. Motors should be sealed in plastic sheeting to eliminate dust and moisture.

*Water Storage Tanks*

Water storage tanks should, where possible, be fully drained down, and painted internally with preservative paint. Externally, the protective lagging and external paint finish should be repaired. Where tanks cannot be fully drained they should be filled to overflow level with treated water, making sure they are adequately covered and suitably protected against frost during the winter months.

*Water Treatment Plant*

Regenerate resin beds in the case of base exchange, de-alkalisation and similar ion exchange plant. Drain down all vessels, towers, sumps and pipework. Lubricate pumps, automatic valves etc., to makers' recommendations. Renew protective paintwork where necessary. Remove spare chemicals and store in dry conditions. Seal motors, automatic control gear and instrumentation with plastic covers.

*Fans and Motors*

Inspect impellers and bearings, automatic dampers, belt drives etc. Oil and grease to makers' recommendations. Seal motors with plastic covers. Renew protective paint where necessary.

*Controls and Switchgear*

Panels should be isolated, inspected and cleaned. Silica-gel trays should be placed within the cabinets and the doors closed and sealed as far as practicable. Depending upon the position, location and type, control equipment should be cleaned, lubricated as necessary and sealed with plastic covers.

*Air Conditioning Systems*

Cooling towers should be thoroughly drained down, cleaned and painted where necessary. Any forced draught fans, pumps and motors should be treated as described above. When towers and associated pipework are brought back on line, suitable chlorination and biocide treatments should be applied.

Ductwork should be inspected and openings sealed with plastic sheet. Protective paint and/or insulation should be renewed where necessary. Compressors, evaporators, condensers and the refrigerant system should be dealt with in accordance with the makers' recommendations. Chilled water and cooling water circuits should be treated in a similar manner to LTHW and MTHW systems.

*Instrumentation*

Where installed in panels, it should be treated as controls and switchgear. Individual or wall mounted equipment should be inspected, lubricated or drained as necessary and sealed with plastic sheets. Certain expensive or delicate items should be removed and stored in a safe place.

*Compressed Air Systems*

The air mains must be thoroughly drained down and protective paint renewed as required. Air receivers and vessels should be treated in a similar manner. Compressors should be cleaned, lubricated and treated in accordance with the makers' recommendations.

*Structural Steelwork*

It should be inspected, cleaned down thoroughly and repainted where required. The surface must be thoroughly cleaned and all corrosion and dirt must be removed prior to painting. Particular attention should be paid to joints and areas where steel work is embedded in masonry or the ground.

## WATER TREATMENT

The water quality for a given purpose must meet the particular requirements to ensure optimum operation. Water supplied by the statutory undertakers normally requires further treatment to meet the criteria specified for the purpose. The choice of treatment should be made following consideration of the quality of the water available and the requirements of the equipment into which it will be fed and the system should be monitored to ensure the continued effectiveness of the treatment.

### Sources of Water

The main sources of water are:

(*a*) Municipal supplies.

(*b*) River water.

(*c*) Ground water.

(*d*) Sea water.

The different sources of water have individual water qualities and contain various amounts of impurities. The presence of these impurities is due to the nature of water itself. Water is a solvent, particularly for those substances which dissociate to form ions, thus all natural waters are solutions containing impurities. These impurities fall into three broad categories:

(*a*) Suspended matter.

(*b*) Dissolved mineral matter.

(*c*) Dissolved gases.

The amount of impurities in water at the same location will vary from time to time. Further, the composition of the water in a system will change during the operation of the system. Therefore, it is essential that, in addition to specifying the initial water quality, the supplies should be continually monitored and controlled to maintain that quality.

### Water Induced Problems

Water induced problems become apparent in heating and cooling systems when the following conditions occur:

(*a*) Reduction in heat transfer by formation of an insulating layer reducing efficiency.

(*b*) Reduced water flow resulting from partial or complete blockages of pipelines etc.

(*c*) Damage to fabric from corrosion of metal or deterioration of wood and plastic.

(*d*) Excessive wear of pumps and mechanical seals.

These conditions are caused by scale formation, corrosion, erosion and biological growth.

*Scale Formation*

Scale is caused by the precipitation of dissolved salts in water to form a hard deposit on heating surfaces. The salts which are the prime cause are those of calcium and magnesium, their carbonates and bicarbonates. Consequently scale formation is a function of the hardness of the water.

Three main effects combine to cause scaling:

(*a*) The salts of most metals are to some degree soluble in water and, in most cases, the solubility increases with increasing temperature. The proportion of the salt that will dissolve in water is defined by the solubility coefficient, i.e. a high positive solubility coefficient means that a high proportion of the salt will dissolve in water. The salts of calcium and magnesium not only have low solubility coefficients, they are negative. This means that their solubility in hot water is less than in cold water. At high temperatures the solubility coefficient may be so reduced that the calcium and magnesium salts present constitute a saturated solution resulting in the precipitation the salts as scale.

(*b*) The most common calcium salt found in water supplies is calcium bicarbonate. At temperatures approaching boiling point the salt tends to decompose to form calcium carbonate, carbon dioxide and water:

$$Ca(HCO_3)_2 \rightarrow CaCO_3 + CO_2 + H_2O$$

The carbon dioxide gas escapes from the water leaving enhanced concentrations of calcium carbonate. This is the most insoluble of the calcium salts which invariably precipitates as a white soft adherent scale.

(*c*) In open systems water leaves the system in the form of vapour. As this vapour is pure water the remaining salts become more concentrated. If calcium and magnesium salts are present their concentrations will reach saturation point and scale deposition ensues.

*Corrosion*

Corrosion is the loss of metal from the structure. This can happen in several ways:

(*a*) Some waters contain high concentrations of dissolved carbon dioxide, e.g. certain groundwaters. This dissolved carbon dioxide is in equilibrium with carbonic acid:

$$H_2O + CO_2 \rightleftarrows H_2CO_3 \rightleftarrows H^+ + HCO_3^-$$

Normally this equilibrium lies to the left of the equation. However, if the pressure is high enough to prevent the escape of carbon dioxide from the solution, the equilibrium moves towards the right and the solution becomes weakly acidic. In acid solutions, metals dissolve, displacing free hydrogen ions to form hydrogen gas, thus:

$$2HCl + Fe^{2+} \rightarrow FeCl_2 + H_2$$

At high temperature and pressure, water itself acts as acid, releasing hydrogen:

$$4H_2O + 3Fe \rightarrow Fe_3O_4 + 4H_2$$

The $Fe_3O_4$ produced is a black magnetic iron oxide called magnetite and is a common feature of condensate systems.

(*b*) Dissolution of metals can also occur when two points on a metal surface have different electrical potentials. Electrons move from one to the other; at the anode the metal dissolves producing electrons and, at the cathode, these electrons are used to form hydroxyl ions. The hydroxide and ferrous ions combine to form ferrous hydroxide. If there is sufficient oxygen present, this hydroxide is further oxidised to form red ferric oxide, i.e. rust:

$$Fe \rightarrow Fe^{2+} + 2e^-$$

$$H_2O + {}^1/_2O_2 + 2e^- \rightarrow 2OH^-$$

$$Fe^{2+} + 2OH^- + Fe(OH)_2$$

These aqueous corrosion cells occur when the metals or the water are non-homogenous. This can be exacerbated when parts of the metal are covered with scale or bacteriological growths. Blemishes and crevices in the metal surfaces cause a similar result.

(*c*) Galvanic corrosion is similar to aqueous corrosion but is caused by joining together materials of different electrochemical potentials. Although this is not primarily a water induced problem, control of the water to reduce its electrolytic properties will contribute to reducing the effect.

*Erosion*

Erosion is the loss of metal by physical removal through impingement either of water alone, or the gases and metals suspended in it.

*Bacteriological Growth*

Algae, bacterial slimes and fungi can cause problems mainly in open systems. When water is exposed to sunlight algal growth is rapid and causes blockages and fouling of surfaces. The excessive growth can eventually cause de-oxygenation of the water. Where biological growth occurs, particularly in open systems, there is an additional problem of the proliferation of bacteria which, although they do not interfere with the operation of the system, do pose a hazard to health. An example of this is the bacterium *Legionella pneumophila*.

**Treatment Methods**

In order to control the water induced problems discussed above a programme of water treatment will be required. Generally, the methods of treatment fall into two groups, as follows.

(*a*) *External methods:* the removal of impurities from the water before it is fed to the equipment. The processes concerned are the removal of suspended and particulate matter, the removal of dissolved salts and the removal of high concentrations of mineral matter.

(*b*) *Internal methods:* the addition of chemicals to the water to inhibit corrosion and the formation of the scale. The processes concerned include scale control, corrosion control and pH control.

These processes are described in detail in the following sections. Usually, a combination of internal and external methods is used to achieve the required result. Disinfection and/or sterilisation of the water may be regarded as either external or internal depending on the circumstances and is dealt with separately.

**Removal of Suspended and Particulate Material**

This may consist of one or more of the following processes.

*Sedimentation*

Removal of coarse suspended material. Raw water is passed through a sedimentation tank sized to allow the particles to settle out.

*Coagulation and Flocculation*

Removal of fine colloidal particles. A coagulating agent is added to the water to encourage the colloidal particles to coalesce to form larger particles which may then be removed by sedimentation. Common coagulants used include aluminium sulphate, ferric sulphate and ferric chloride. Synthetic polymeric compounds such as polyacrylic acid are also used, often as a secondary coagulant. pH adjustment is sometimes necessary to obtain maximum benefit from the coagulant.

*Filtration–Removal of Suspended Solids*

The raw water is passed through a filtration bed of granular material such as sand or anthracite. The suspended material is retained on the surface of the bed or in the voids.

*Oxidation–Removal of Dissolved Metals*

Metals such as iron and manganese are held in solution in the presence of carbon dioxide. Oxidation, either by atmospheric aeration or the addition of oxidising agents, converts the dissolved iron and manganese to the insoluble oxides. These oxides can then be removed by one of the methods described above.

**Removal of Dissolved Salts**

The soluble salts of particular concern are those derived from calcium and magnesium carbonates and bicarbonates.

The main parameters associated with water analysis, along with their terminology and use, are listed in Table B7.4. The various methods of removal are described below.

**Table B7.4.** Water analysis–main parameters.

| Name | Terminology | Use |
|---|---|---|
| Dissolved cations | | To determine the ionic chemical composition of the water |
| Calcium | $Ca^{2+}$ | |
| Magnesium | $Mg^{2+}$ | |
| Potassium | $K^{+}$ | |
| Sodium | $Na^{+}$ | |
| Silica | $Si^{4+}$ | |
| Iron | $Fe^{2+}$ | |
| Manganese | $Mn^{2+}$ | |
| Dissolved anions | | |
| Bicarbonate | $HCO_3^-$ | |
| Carbonate | $CO_3^{2-}$ | |
| Chloride | $Cl^-$ | |
| Hydroxide | $OH^-$ | |
| Nitrate | $NO_3^-$ | |
| Sulphate | $SO_4^{2-}$ | |
| Total dissolved solids | S (Anions + Cations) | |
| Conductivity | | Estimate of TDS and electrolyte properties |
| pH | | To measure the acidity or alkalinity of an aqueous solution |
| Alkalinity<br>Bicarbonate alkalinity<br>Carbonate alkalinity | S ($HCO_3$ + $CO_3^{2-}$ + $OH^-$) | |
| Hardness<br>Alkaline hardness<br>Non-alkaline hardness | S (Multivalent Cations) | Scale forming tendency of the water |
| Carbon dioxide | | To measure the aggressive state of water |

*Removal of Hardness*

The lime-soda ash process is used for the removal of calcium and magnesium. This is achieved by the following sequence of chemical processes.

Conversion of carbon dioxide: before any softening can be carried out $CO_2$ must be converted. Lime is used to convert the $CO_2$ to $CaCO_3$, which can then be removed by sedimentation. (Sediments are underlined in the equations.)

$$CO_2 + Ca(OH)_2 \rightarrow \underline{CaCO_3} + H_2O$$

Removal of calcium and magnesium carbonate hardness:

$$Ca(HCO_3)_2 + Ca(OH)_2 \rightarrow \underline{2\ CaCO_3} + 2H_2O$$

$$Mg(HCO_3)_2 + 2Ca(OH)_2 \rightarrow \underline{2\ CaCO_3} + \underline{Mg(OH)_2} + 2H_2O$$

Removal of calcium and magnesium non-carbonate hardness:

$$Ca^{2+} + Na_2CO_3 \rightarrow \underline{CaCO_3} + 2Na^+$$

$$Mg^{2+} + Ca(OH)_2 \rightarrow \underline{Mg(OH)_2} + Ca^{2+}$$

Re-carbonation for the removal of excess lime and pH control:

$$Ca(OH)_2 + CO_2 \rightarrow \underline{CaCO_3} + H_2O$$

$$Mg(OH)_2 + CO_2 \rightarrow MgCO_3 + H_2O$$

Removal of soluble silica: it is possible to achieve partial removal of soluble silica by adsorption onto the precipitated magnesium hydroxide. The degree of removal depends to a large extent on the hardness of the precipitate.

*Ion Exchange*

Ion exchange is the displacement of one ion by another. As applied to water treatment, it may also be described as a reversible exchange of ions between a liquid and a solid. The solid part is an insoluble material containing fixed active groups and mobile exchangeable ions. When a liquid containing ions is brought into contact with this solid the mobile ions are exchanged with the solute ions having a stronger affinity for the fixed active group. For example, $Ca^{2+}$ replaces $Na^+$, or $SO_4^{2+}$ replaces $Cl^-$. By selecting the appropriate solid material, and providing the optimum conditions, it is possible to remove selected dissolved ions from water.

Natural ion exchange materials, known as Zeolites, were used for water softening but there are now many synthetic materials, primarily resins. Four main types of ion exchange resins are used:

(*a*) Strong acid cation exchangers having a strong acid functional group.

(*b*) Weak acid cation exchangers having a weak acid functional group.

(*c*) Strong base anion exchangers having strong base functional groups.

(*d*) Weak base anion exchangers having weak base functional group.

These resins may be used in combination to remove the appropriate hardness salts or to achieve demineralised water by removing other dissolved materials such as sulphate and chloride. Typical ion exchange reactions are given below.

For natural Zeolites (Z):

$$Na_2Z + \begin{pmatrix} Ca^{2+} \\ Mg^{2+} \\ Fe^{2+} \end{pmatrix} \rightleftarrows \begin{pmatrix} Ca^{2+} \\ Mg^{2+} \\ Fe^{2+} \end{pmatrix} Z + 2Na^{+}$$

For synthetic resins (R):

Strong acid cation exchange:

$$RSO_3H + Na^{+} \rightleftarrows RSO_3\ Na + H^{+}$$

$$2RSO_3\ Na^{+} + Ca^{2+} \rightleftarrows (RSO)_2\ Ca + 2Na^{+}$$

Weak acid cation exchange:

$$RCOOH + Na^{+} \rightarrow RCOONa + H^{+}$$

$$2RCOONa + Ca^{2+} \rightarrow (RCOO)_2\ Ca + 2Na^{+}$$

Strong base anion exchange:

$$RR'_3NOH + C1 \rightleftarrows RR'_3NCl + OH^{-}$$

$$2RR'_3NCl + SO_4^{2-} \rightleftarrows (RR'_3N)_2\ SO_4 + 2Cl^{-}$$

Weak base anion exchange:

$$RNH_3OH + Cl^{-} \rightleftarrows RNH_3Cl + OH^{-}$$

$$2RNH_3Cl + SO_4^{2-} \rightleftarrows (RNH_3)_2\ SO_4 + 2Cl^{-}$$

After a period of use the resins become saturated, i.e. the above reactions move to the right and they approach equilibrium with the feed solution. The resin is regenerated by bringing a brine containing the original mobile ion into contact with the resin and the equilibrium is shifted to the left. In softening and most other cation exchange processes, a NaCl brine is used. If all cations are to be removed, a strong acid such as $H_2SO_4$ is used.

**Removal of High Concentrations of Mineral Matter**

In some cases the concentration of mineral matter is too high for the preceding processes to be effective, e.g. when sea water or brackish ground water is the raw source. In other situations it is necessary to have process water with almost no dissolved solids. In these cases two further processes are available. These are often described as membrane processes in which dissolved solids can be removed from water through the use of a semi-permeable membrane having very small pore sizes.

*Reverse Osmosis*

If two solutions having different concentrations of dissolved salts are separated by a semi-permeable membrane, molecules of the solvent, e.g. water, will pass through the membrane by the process of osmotic diffusion. The solvent molecules will pass from the side with the lower concentration of salts to that having the higher, thereby creating a pressure difference known as osmotic pressure. This pressure depends on the difference in concentrations, the characteristics of the solute and the temperature. If a pressure greater than the osmotic pressure is applied to the side of the membrane having the higher concentration, molecules of the solvent will be forced through the membrane in the opposite direction, thus achieving a reversal of the natural osmotic process known as reverse osmosis.

This process removes all undissolved material, including bacteria and viruses, all dissolved organic material of high molecular weight and 90 to 95% of dissolved inorganic solids. However, high pressures are required, e.g. the pressure at which water may be extracted from a salt solution is in the order of 2760 kPa.

*Electro-dialysis*

If a water rich in ions is subjected to an electrical field by means of two electrodes with a continuous potential difference applied between them, the cations will be attracted to the negative electrode and the anions will be attracted to the positive electrode.

If specially selected membranes are placed between the electrodes, some positive and permeable only to anions, some negative and permeable only to cations, the migration of ions is restricted. A membrane stack is constructed using alternating anion-permeable and cation-permeable membranes separated by spacers and a current passed across it.

As water is passed through the stack, partially demineralised water is removed from each cell pair, and brine is discharged from the space between the cells. This process is not likely to be encountered in building services applications.

**Scale Control**

To reduce the possibility of scale formation, treatment can be applied either to prevent salts coming out of solution or to prevent the solids coming out of solution becoming adherent.

Impurities such as calcium and magnesium can be kept in solution by the addition of a chelating agent, which combines chemically with the impurity to form a soluble salt. These agents form complexes with metals which are very soluble and always have the characteristics of sodium salts so that the effects of calcium and magnesium are masked. Such salts are ethylene diamine tetra-acetate (EDTA) and nitrilo tri-acetic acid (NTA).

Treatment to prevent scale becoming adherent can be divided into two main classes; chemical and physical conditioning.

*Chemical Conditioning*

This is used to precipitate scale forming impurities throughout the bulk of the water. Thus localised scaling, e.g. on heat surfaces, can be avoided. Phosphates or carbonates are two chemical groups used in conjunction with appropriate pH control.

*Physical Conditioning*

This is employed to effect the precipitation of microcrystals and then to disperse or flocculate them so that a free flowing sludge is obtained. This is non-adherent and can be removed from the system. Naturally derived organic substances such as starches, tannin, lignin derivatives and synthetic organic compounds are used.

**Corrosion Control**

Corrosion control can be effected by the use of protective coatings, removing dissolved oxygen from the water, or adding chemicals which inhibit the electro-chemical cell reactions. These may be used in combination.

Film forming amines are long chain fatty acids where one end of the chain is hydrophilic and the other hydrophobic. The molecules form a continuous water-proof film over the metal surface hence preventing contact between the water and the metal. This film is made up of molecules running parallel to each other and perpendicular to the metal surface. These are known as physical inhibitors.

Chemical inhibitors make use of the electrical potential difference in the metal to produce a protective layer. Certain inhibitors act on the anodes, such as in nitrate and chromate based inhibitors. Others, such as phosphates, act on the anodes and/or cathodes depending how they are applied.

The cathodic reaction can be suppressed by limiting the amount of dissolved oxygen in the water. This is effected by the addition of oxygen scavengers, i.e. substances which react with the dissolved oxygen and incorporate it into a stable chemical substance. Examples of these are sodium sulphite, hydrazine and certain tannins.

**pH Control**

The control of pH by the addition of acids or alkalis is used for many different purposes in internal treatment. Some inhibitors, for example, are most effective in a limited pH range. At high pressures carbon dioxide can be prevented from coming out of solution and hence the water becomes acidic. Alkalis are added to combat this acidification.

**Disinfection and Sterilisation**

The presence of biological and bacteriological matter can lead to the growth of algae, fungi and slimes and to the multiplication of harmful bacteria such as *legionella pneumophila*. Systems which are particularly at risk are open evaporative cooling systems, air washers and humidifiers. Treatment by disinfection and sterilisation is necessary to minimise these problems.

Sterilisation involves the complete destruction of all organisms whereas disinfection is the selective destruction of those organisms that cause disease. Mains supplies are routinely disinfected by chlorination but other sources, such as river water, require treatment. The most widely used method of disinfection is chlorination using chlorine gas, chlorine dioxide, sodium hypochlorite or calcium hypochlorite. Proprietary biocides and compounds, such as peracetic acid, are available but specialist advice should be sought concerning their use. Other methods, such as exposure to ultra-violet light are sometimes used.

Biocides should be added to the system in shock doses. Continuous dosing at low levels is neither economical nor effective and can lead to a proliferation of organisms. The frequency of application will vary from once every two or three days in summer to once every two weeks in winter. In addition, the system should be drained, cleaned and dosed with chlorine for a period of 24 hours, twice a year. Detailed recommendations for the control of *Legionella pneumophila* are given elsewhere[10, 11].

## APPLICATIONS

There are certain standard procedures which can be applied to the water treatment requirements of any element of a process system, provided that the following information is available.

*Quality of Raw Water Sources*

If the source is a mains supply it is possible to obtain a full chemical analysis from the supplier. The variations in the quality should be ascertained and assessed for acceptability. An independent analysis of the water is advisable. It should also be established if the quality of the water is going to remain the same for the foreseeable future and that the required volume will be available, particularly if a phased program is envisaged.

Where a mains supply is not available an alternative source of raw water, such as river, ground or sea water should be established. Extensive chemical analysis may be required to define the quality of the water. Again, fluctuations in quality and volume should be assessed.

*Required Initial Quality of Process Water*

Specialised information can be obtained from suppliers of equipment on the optimum water quality required. There are normally several different standards specified in a complete system and thus sequential or phased treatment may be required.

*Physical Characteristics of System Operation*

The operating temperatures and pressures are the major physical constraints which have an effect on water quality. Rates of flow, time of use and other operational information on the water use should be obtained. The entire system should be assessed for possible dead areas.

*Materials*

All materials used in each component should be identified. Details of the different metals involved are needed to determine whether electrolytic action will occur. Other materials such as rubbers and plastics in valves, etc., should also be identified.

*Water Quality and Quantity*

The volumes of water for operational needs such as blowdown should be assessed. The amount and quality of water removed from the system and the sludges should be quantified for disposal purposes. Changes in the water quality within elements should be identified.

**Selection of Treatment Methods**

The selection of a treatment programme to satisfy the requirements of all component elements is a complex, and iterative, process. The requirements for each element should be defined and a programme designed to achieve the best practical treatment with regard to the following:

(*a*) efficiency

(*b*) cost

(*c*) control of process

(*d*) make-up and dosing equipment for chemicals

(*e*) monitoring the concentration and effect of chemicals

(*f*) available maintenance

(*g*) chemical handling and safety

(*h*) disposal of waste products.

**Monitoring and Control**

A comprehensive and efficient monitoring and control programme must be designed at an early stage. Sampling points and access should be provided at locations convenient for the operators. A comprehensive operational manual should be provided. Training for operators is essential and it is often beneficial to consult operators early in the design process, both so that their practical requirements are incorporated and that they may obtain knowledge on the construction of the plant for which they will eventually have responsibility.

**Individual Elements**

The range and complex nature of current equipment precludes a full description of water treatment combinations. Brief summaries of the requirements and possible external and internal treatments are given below. However, the relevant specialists in equipment and water treatment should be consulted. In particular, many internal treatments are combined in proprietary products and these should be selected where appropriate.

*Hot Water Heating System*

These are normally closed re-circulating systems and should not need internal treatment. However, if operating experience indicates that large volumes of make-up water are required some treatment may be advisable.

A stable, non-corrosive, non-scaling water should be used for filling the system, softened or demineralised if necessary. Mains supplies would not normally require further suspended solid removal or disinfection but for raw water sources, such as rivers, such treatment would be necessary. On start-up, debris introduced during construction such as mill scale, weld spatter, protective oil coatings and iron oxide should be removed by flushing with low foaming detergents, with alkali and dispersant to prevent re-soiling. Tables B7.5 and 6 suggest suitable chemicals for hot water heating systems.

*Steam Boilers*

There is a wide range of treatment options available to cover the various applications which may be required. A brief description of the possible treatment programmes is given here but specialist advice should be sought for specific boiler applications.

Priming, foaming and carryover are problems which affect operational efficiency. Prevention of these problems requires blowdown and water treatment.

**Table B7.5.** Internal treatment methods for hot water heating systems with flow temperatures greater than 120°C.

| Treatment | Chemicals |
|---|---|
| Oxygen scavenging | Sodium sulphite<br>Hydrazine |
| pH control (8·0 to 10·0) | Caustic soda<br>Neutralising amines |
| Dispersal | |

*Notes:* The above information is for guidance only. Proprietary blends fulfilling one or more of the above functions are available.

**Table B7.6.** Internal treatment methods for hot water heating systems with flow temperatures less than 120°C.

| Treatment | Chemicals |
|---|---|
| Corrosion inhibiting | Borax<br>Sodium nitrite<br>Sodium benzoate<br>Sodium phosphate<br>Sodium silicate<br>Tannins<br>Brass inhibitor<br>Copper inhibitor |
| pH control (8·0 to 10·0) | Alkali |
| Scale inhibiting | Synthetic organic polymers<br>Lignins<br>Phosphonates |

*Notes:* The above information is for guidance only. Proprietary blends fulfilling one or more of the above functions are available.

Blowdown is the removal of water with a high concentration of dissolved material from the boiler under pressure. The steam produced during operation is pure water vapour containing no dissolved solids. Hence, the concentration of dissolved material in the remaining water increases. The use of make-up water to replace the evaporated water results in additional dissolved solids being added to the boiler water. Finally, the residual concentration rises to the point where the dissolved solids precipitate out as scale, or give rise to priming, foaming and/or carryover.

The concentration of the dissolved matter must be limited to a maximum value by blowdown. These limits are stated by the manufacturer. Blowdown removes some of the water having a high concentration of solids which is replaced by feed water. The sequence by which solids accumulate in the boiler water is called a cycle of concentration. A boiler operating at 10 cycles of concentration will contain boiler water with concentration of solids ten times greater than that of the feed water.

The cycle of concentration may be determined from a comparison of total dissolved solids, chlorides or conductivity. The required blowdown is that necessary to maintain the lowest acceptable cycles of concentration.

General guidance on operating parameters at different pressures is given in BS 2486[12].

(*a*) *Low pressure steam healing boilers:* Operating conditions are unique in that steam is normally produced for heating purposes only. Steam is passed through closed heat exchangers, condensed and returned to the boilers. Hence, the system is a closed circuit.

The fill and make-up water should be softened or demineralised. Treatment chemicals are likely to include those shown in Table B7.7.

The suggested treatment applies only when all the steam is returned to the boiler as condensate. If losses are incurred through use, the following methods should be considered.

(*b*) *High pressure boilers and low pressure process boilers:* High pressure boilers require a different approach due to the higher temperatures employed. Low pressure boilers used for process require treatment to compensate for water losses.

Feed water requires softening, demineralisation and de-aeration. Treatment methods are indicated in Table B7.8.

*Chilled Water Systems*

The main problem is corrosion, especially if dissolved oxygen or chloride levels are high. In addition, the use of mixed metal systems containing aluminium, brass, copper, iron and steel requires balanced blends to achieve effective protection. Some treatment chemicals are indicated in Table B7.9.

*Refrigeration Brines*

Coolants most commonly used include calcium chloride brine, sodium chloride brine and glycol solution. Normally, glycol solutions do not require additional chemicals as they are prepared from inhibited ethylene glycol or propylene glycol. Treatment chemicals are likely to include those indicated in Table B7.10.

**Table B7.7.** Internal treatment methods for low pressure steam heating boilers.

| Treatment | Chemicals |
|---|---|
| Corrosion inhibiting for boiler | Borax<br>Organic polyphosphates<br>Sodium nitrite<br>Sodium benzoate<br>Sodium phosphate<br>Sodium silicate<br>Tannins |
| Neutralising steam and condensate lines | Neutralising or filming amine |
| pH control (7·0 to 10·0) | Caustic soda |
| Oxygen scavenging | Sodium sulphite |
| Scale inhibiting | Organic polyphosphates |
| Dispersal | |

*Notes:* These treatments apply only when all the steam is returned to the boiler as condensate. Table B7.8 should be consulted if losses are incurred during use. The above information is for guidance only. Proprietary blends fulfilling one or more of the above functions are available.

**Table B7.8.** Internal treatment methods for high pressure heating and low pressure process boilers.

| Treatment | Chemicals |
|---|---|
| pH control (10·5 to 11·5) | Caustic soda |
| Alkalinity adjustment | Caustic soda |
| Oxygen scavenging | Sodium sulphite<br>Hydazine |
| Scale inhibiting | Phosphates or phosphonates<br>Chelating agents<br>Polymers and/or dispersants<br>Phosphonates |
| Prevention of foaming and/or carryover | Anti-foaming agent |
| Corrosion inhibiting and pH control (7·0 to 9·0) for steam and condensate lines | Neutralising amine or filming amine if steam losses are high |

*Notes:* The above information is for guidance only. Proprietary blends fulfilling one or more of the above functions are available.

**Table B7.9.** Internal treatment methods for chilled water systems.

| Treatment | Chemicals |
|---|---|
| Corrosion inhibiting | Borax<br>Sodium nitrite<br>Sodium benzoate<br>Phosphates<br>Copper inhibitors<br>Silicates<br>Organic phosphonates<br>Amines |
| pH control (8·5 to 9·5) | Alkali |
| Dispersal | Polyacrylates<br>Lignins |

*Notes:* The above information is for guidance only. Proprietary blends fulfilling one or more of the above functions are available.

**Table B7.10.** Internal treatment methods for refrigeration brine systems.

| Treatment | Chemicals |
|---|---|
| Corrosion inhibiting | Sodium nitrite |
| pH control (7·5 to 8·0) | Alkali |
| Dispersal | |
| Deposit inhibiting and dispersal | |

*Notes:* The above information is for guidance only. Proprietary blends fulfilling one or more of the above functions are available.

*Solar Heat Exchanger Systems*

Solar systems employ closed recirculating systems containing water or brine. The materials include metals, concrete, plastics, etc. Corrosion and scale deposition are the main problems.

Generally, the treatment methods required are those used for chilled water systems and hot water heating circuits. Where frost protection is required, the system should be filled with inhibited glycol antifreeze.

*Open Recirculating Water Systems*

These systems include water-cooled cooling towers, and spray humidifiers. The loss of water vapour leading to concentration of dissolved solids in the remaining water is a major factor to be controlled by water treatment. Microbiological fouling is also likely as the water may be in contact with the air and therefore acts as an air scrubber.

The feed water should be softened and alkalinity removed. Treatment chemicals may include those indicated in Table B7.11.

A system of bleed-off and make-up with fresh water to control is required to the concentration of dissolved solids to a limiting maximum. Cycles of concentration, similar to boiler water make-up, can be calculated, usually based on the choride concentration.

**Table B7.11.** Internal treatment methods for open recirculating water systems.

| Treatment | Chemicals |
|---|---|
| Scale inhibiting | Polyphosphates<br>Starch<br>Lignins<br>Tannins<br>Synthetic organic polymers |
| pH control (about 7·0) | Acid |
| Dispersal | |
| Corrosion inhibiting | Chromate or zinc compounds with pH control<br>Phosphate<br>Lignins<br>Gluconate |
| Biocide dosing | Frequent shock dosing with suitable biocide |

*Notes:* The above information is for guidance only. Proprietary blends fulfilling one or more of the above functions are available.

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## APPENDIX

### Langelier Saturation Index

A nomograph for the determination of the Langelier Saturation Index is shown in Fig. B7.4. The use is illustrated by means of an example, as follows:

(*i*) For the known value of total dissolved salts, a vertical line (a) is drawn to intersect the appropriate temperature curve.

(*ii*) From the intersection, a horizontal line (b) is drawn to determine the corresponding value of the temperature/total salinity constant (column 1).

(*iii*) This value is then connected by a straight line (c) to the known calcium concentration (column 3). The point at which this line intersects the pivot line (column 2) is noted.

(*iv*) A straight line (d) is drawn to connect this point on the pivot line to the known alkalinity value (column 5).The saturation pH is determined from the intersection of this line with the saturation pH scale (column 4).

(*v*) The Langelier Saturation Index($L$) is the algebraic difference between the measured pH and the saturation pH, $(pH)_{sat}$, determined from the nomograph:

$$L = pH - (pH)_{sat}$$

Negative values of $L$ indicate corrosive conditions. Positive values indicate scale formation.

*Example*

For the example shown on the nomograph:

| | | |
|---|---|---|
| Total dissolved salts | = | 210 ppm |
| Alkalinity (as calcium carbonate) | = | 30 ppm |
| Calcium concentration | = | 120 ppm |

For a water temperature of 50°C, the temperature/total salinity constant is 1·77. Hence, from the nomograph, the saturation pH is 7·5.

Comparing with typical measured pH values gives:

(*i*) For a measured pH value of 7·1:

$$L = 7{\cdot}1 - 7{\cdot}5 = -0.4$$

The negative value indicates corrosive conditions.

(*ii*) For a measured pH value of 8·0:

$$L = 8{\cdot}0 - 7{\cdot}5 = +0.5$$

The positive value obtained indicates scale formation.

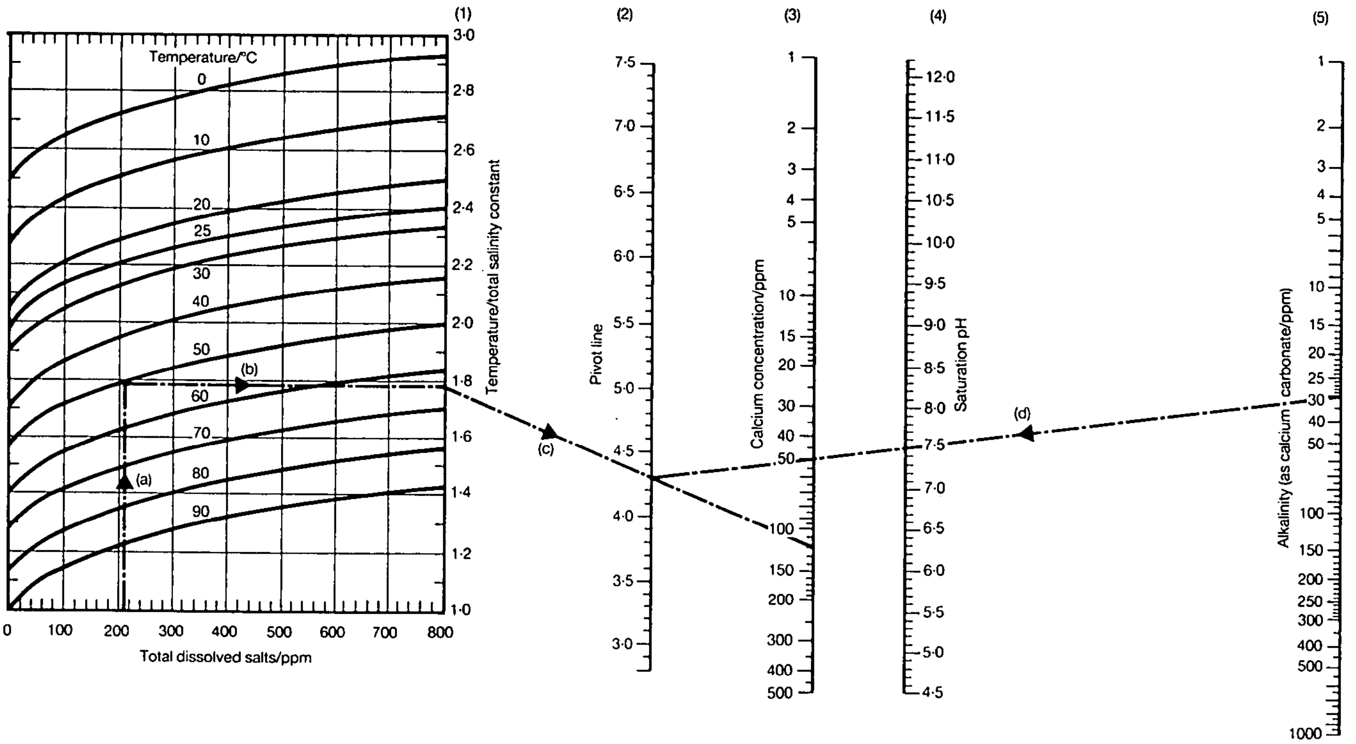



**Fig. B7.4.** Nomograph for the determination of saturation pH. (Correct for pH values between 7 and 9·5.) Reproduced with permission from *The Water Treatment Handbook*[1].

# SECTION B8 SANITATION AND WASTE DISPOSAL




*This edition of Section B8 first published: 1986*

# SECTION B8 SANITATION AND WASTE DISPOSAL

## INTRODUCTION

With regard to sanitation this Section is restricted to those aspects of foul and surface water disposal which are the immediate concern of the building services engineer. However, some mention is made of aspects of design related to cesspits and small sewage disposal plants. Waste disposal data are included.

The design of sanitary intallations is closely regulated by the Building Regulations and various British Standards. Detailed design needs thorough reference to these documents, which are referenced throughout this Section.

Sanitation and waste disposal systems for buildings in the United Kingdom are controlled by legislation and, although Local Authorities may have appending or additional requirements, the general statutory requirements are contained in the following:

**England and Wales**

Building Regulations 1985
Clean Rivers (Estuaries and Tidal Waters) Act 1960
Factories Act 1961
Food Hygiene Regulations, Food & Drugs Act 1965
Health and Safety at Work Act 1974
Highways Act 1959
Land Drainage Act 1961
Offices, Shops and Railway Premises Act 1963
Public Health Act 1936
Public Health Act 1961
Public Health Act 1977 (Draining of Trade Premises)
Radioactive Substances Act 1960
Rivers (Prevention of Pollution) Act 1951
Road Traffic Act 1960

**Additional for Scotland**

Building Standards (Scotland) Regulations 1971 to 1975
Flood Prevention (Scotland) Act 1961
Rivers (Prevention of Pollution) (Scotland) Act 1951
Sewage (Scotland) Act 1968

**Additional for Northern Ireland**

Building Regulations (Northern Ireland) 1973
Public Health (Ireland) Act 1878
Public Health Amendment Act 1890
Public Health (Ireland) Act 1896
Public Health Amendment Act 1907

## DEFINITIONS

**Bedding**
Natural or synthetic material introduced around and under a drainage pipe to improve its load resistance.

**Benching**
A surface added at the base of manholes to improve working safety; constructed so as to contain the channel and minimise the accumulation of deposits.

**Combined System**
A system of drainage in which foul and surface water are conveyed in the same pipes to a common or combined sewer.

**Crown**
The highest point on the internal surface of a pipe or channel at any cross section.

**Foul Drain**
A pipe conveying soil and waste discharges from sanitary appliances within a single curtilage.

**Foul Sewer**
As foul drain, but conveying soil and waste from a number of curtilages; normally constructed and/or maintained by the Local Authority.

**Invert**
The lowest point on the internal surface of a pipe or channel at any cross section.

**Manhole**
A chamber facilitating access to a drain or sewer.

**Pumped (Pressure) Drain**
A system in which the effluent is elevated by pump or ejector.

**Separate System**
A system in which foul and surface water are segregated and discharged into separate sewers or other places of disposal.

**Sub-soil Drains**
A system of porous or unjointed pipes laid below ground to collect ground water and convey it to a convenient discharge point. Also termed field or agricultural drains.

**Surface Water Drain**
A pipe containing rainwater from roofs and paved areas within a single curtilage.

**Surface Water Sewer**
As a surface water drain, but serving a number of curtilages; usually the responsibility of the Local Authority.

**Trade Waste/Chemical Drain**
A pipe conveying the contaminated discharges from industrial processes; subject to rigorous Local Authority control.

## DESIGN OF EXTERNAL SYSTEMS

All design for underground drainage must comply with statutory requirements and bye-laws. Refer to the relevant British Standard Code of Practice. The choice of disposal method depends on the nature of the effluent, the Local Authority requirements and the availability of sewers or watercourses etc.

### Disposal of Foul Drainage

Combined or foul drain discharge may be disposed of in three ways:

*Sewers*

The Local Authority may require that drains be connected to a public sewer if the sewer is not more than 30 m from the site boundary. Where the distance is greater the Local Authority may only require this if they bear the extra cost.

*Conservancy*

The drains are discharged into a cesspool which is periodically emptied by tanker lorry. The siting and sizing of cesspools are restricted; consult the Local Authority.

*Treatment Plants*

There are many methods of disposal by treatment; brief details are included later in this Section.

### Disposal of Surface Water Drainage

Surface water drainage may be dealt with by four recognised methods:

*Sewers*

Where surface water sewers are available and the Local Authority agree that they have adequate capacity.

*Storage*

Normally used in conjunction with connections to sewers or watercourses where the run-off from the site is greater than the capacity of the sewer or watercourse.

*Watercourse*

Before discharging into any natural watercourse, the recommendations and approval of the Local Authority and River Board or Local Water Conservancy Authority must be sought. The capacity and normal and flood levels must be established.

*Soakaway*

The permeability of the ground and the water table must be established.

### Special Requirements

*Garage Drainage*

The Public Health Act 1936 makes it illegal to discharge any petrol or oil which gives off a flammable vapour at a temperature of less than 23°C. To prevent petrol entering sewers, bye-laws require that garage drainage be separated from all other systems and pass through a petrol interceptor before entering the public sewer. BS 8301: 1985 Figures 7 and 8 show the commonest arrangements for petrol interceptors. Most Local Authorities have a petroleum office whose guidance should be sought as to the provision of one, two, or three chambers. The ventilation pipe must connect with each chamber, but can be connected into a common vertical pipe above ground level. The open end of the vent must discharge into free air at a height not less than 2.5 m above paving height. Ventilated manhole covers can be provided in 'soft' and isolated areas.

*Grease Traps*

Grease is a usual constituent of water-borne waste from kitchens, where it arises from washing crockery and utensils. In a correctly designed drainage system free from obstruction, grease and fats are unlikely to cause blockages or separate, except in pump sumps.

Bulk collection and removal of grease, fat and oil is recommended to prevent their discharge to the drainage system.

Where the Local Authority requires the use of a grease trap it should be of adequate size, be situated so as to receive sewage-free waste from the kitchen, be convenient for maintenance, and not cause a nuisance through health and hygiene hazards.

Regular and frequent cleaning is essential and should include removal of settled solids to avoid putrefaction. The efficiency of traps is impaired by detergents.

There are proprietary appliances to break down grease biochemically.

Grease traps should not be used with food and waste macerator disposal units.

*Anti-flooding*

Where a drainage system is liable to flooding caused by sewer surcharge, the building must be protected from back flooding.

The system should be designed so that open drainage inlets below surcharge level are provided with an anti-flooding gulley or an anti-flooding valve.

Anti-flooding devices shoud be maintained regularly and the occupier of the protected accommodation should be advised of the limitations of the device.

### Design Procedure

The following information must be to hand before any calculation:

(*a*) ground surface levels.

(*b*) positions of all entries to drains, e.g. rainwater pipes, gullies (surface water and foul), soil and vent stacks, ground floor WCs.

(*c*) number and type of sanitary appliances contributing to each drain entry.

(*d*) layout of drains including manhole positions and sewer connections.

The site layout drawing should indicate the invert levels for each manhole to give minimum drain cover. Roofed or paved areas contributing to rainwater pipes or gullies should also be shown.

The minimum sizes for underground drains are:

(*i*) 100 mm internal diameter for foul drain collecting WC appliances.

(*ii*) 75 mm internal diameter for a waste water drain and surface water drain.

*Foul Water Drains*

Approved Document H1 to the Building Regulations 1985 states: 'To reduce the risks to the health and safety of persons in buildings the foul water drainage system should:

(*a*) convey the flow of foul water to a foul outfall, and

(*b*) minimise the risk of blockage or leakage, and

(*c*) prevent foul air from the drainage system from entering the building under working conditions, and

(*d*) be ventilated, and

(*e*) be accessible for clearing blockages.'

The system should convey and discharge its contents without causing a nuisance, or danger to health arising from leaks, blockages, surcharging and flooding.

The system should be as simple as possible, with minimal changes of direction and gradient. Access should be provided at every change of direction or gradient so that every section of the system can be rodded to clear blockages.

Ventilation should be adequate, with branch pipes longer than 10 m ventilated individually.

Refer to BS 8301: 1985 and The Building Regulations 1985 for details of access requirements such as rodding eyes, access points and manholes. The minimum size of manhole cover recommended for inspection chambers and shallow manholes is 900 × 600 mm.

The material and construction for access points should be compatible with the material for the drain pipe.

Flow rates based upon the number and type of appliances served, together with the probable simultaneous discharge rates are given in BS 8301: 1985; the tables and graphs in which enable drain sizes and gradients to be determined.

Foul water drains should be sized so that at estimated peak flow the drain has a proportional depth not greater than 0·75 times the pipe diameter, and preferably 0·66 times pipe diameter. The gradient should be such as to maintain a self-cleansing velocity (usually taken as 0·76 m/s).

*Surface Water Drains*

Approved Document H3 to the Building Regulations 1985 states: 'To reduce the risks to the health and safety of persons in buildings the rainwater discharge system should:

(*a*) carry the flow of rainwater to an outfall, and

(*b*) minimise the risk of blockage or leakage, and

(*c*) be accessible for cleaning blockages.'

Surface water drains should be sized such that at estimated peak flow the drain has a proportional depth of 1 (full bore).

*Combined Drains*

For combined drainage systems, allow capacity for foul water discharges together with surface water discharges included in BS 6367: 1983. Large surface water areas may require detailed analysis.

*Pumping*

It may be necessary to pump building drainage to make a connection to a sewer where the sewer invert is higher than the lowest discharge point, or where the sewer is subject to surcharge or flooding, and it is necessary to install the discharge pipe above such surcharge or flood level to avoid back flooding.

Depending on the quantity to be handled, the type of equipment used to lift the sewage may be an unchokeable centrifugal pump, a submersible pump or a sewage ejector. As a guide the pump capacity should be six times the 24 hour average dry weather flow. Pumps should always be duplicated to offer a standby in the event of mechanical failure, with automatic changeover. Sewage pump sumps should be sized to prevent long periods of storage, when sewage may become septic.

Separate ventilating pipes should be provided from the pressure drain and also for the exhaust compressed air from pneumatic ejectors.

## DESIGN OF INTERNAL SYSTEMS

Drainage systems should comprise the minimum of pipework necessary to carry away foul water from a building quickly, quietly, free from nuisance or risk of injury to health and without escape of foul air into the building.

To prevent air from the drainage system entering the building, there should be a trap having an adequate water seal on each sanitary appliance and on all points of discharge into the system. The discharge from appliances produces pressure fluctuations and the system should be designed to maintain the integrity of water seals in all traps under normal working conditions.

Possible pressure effects are due to:

(*a*) self-siphonage, which is the suction due to full bore flow in horizontal drainage pipework

(*b*) induced siphonage, which is the suction normally associated with water flow down the drainage stack

(*c*) back pressure, which is also normally associated with water flow down the drainage stack. Conditions near the base of the stack, e.g. bends and offsets, influence the back pressure.

In certain circumstances a vent pipe will be required to limit these pressure fluctuations.

### Discharge Systems

These are illustrated in BS 5572: 1978.

*Single Stack System*

This is used where grouping of appliances makes it practicable to provide branch discharge pipes without the need for branch ventilating pipes, and where the discharge stack is large enough to limit the pressure fluctuations without the need for a ventilating stack.

*Modified Single-stack System*

As single-stack system, but providing ventilating pipework to appliances having extended lengths of discharge pipework. The ventilating pipework can be extended to the atmosphere directly or by connecting into the discharge stack above the spill-over level of the highest appliance served.

*Ventilated System*

A ventilated system is used where there are many appliances in ranges or where they are widely dispersed, and it is impracticable to provide additional discharge stacks close to them. Trap seals are safeguarded by extending the discharge and ventilating stacks to atmosphere and providing individual branch ventilating pipes.

*Ventilated Stack System*

A ventilated stack system is used where close grouping of appliances makes it practicable to provide branch discharge pipes without the need for branch ventilation pipes. Trap seals are safeguarded by extending the discharge stack to atmosphere and by cross-connecting the ventilating stack to the discharge stack.

*Two-Pipe System*

This traditional system, in which soil and waste appliances are connected into separate stacks, is not now commonly used.

*Stub Stack*

A stub stack is used for single-storey buildings to connect one set of appliances directly to the drain. This method can also be used for ground-floor appliances in multi-storey buildings where it may be considered undesirable to connect these to the main discharge stack because of positive pressure at the base of the stack.

**Design Principles**

It is convenient here to consider separately the effects of the flow of water in the branch connecting the appliance to the stack and the flow of water down the stack.

Seal losses caused by flow in a branch depend on:

(*a*) the design of the appliance: funnel-shaped appliances increase the chance of self-siphonage.

(*b*) the length, fall and diameter of the branch pipe.

Seal losses resulting from flow down the stack depend on:

(*a*) the flow load (which depends on the number of appliances connected to the stack and the frequency with which they are used).

(*b*) the height and diameter of the stack.

Excessive seal losses can be prevented by choosing the size of stack appropriate to the height of the building and to the number of appliances connected.

*Traps*

A trap which is not an integral part of an appliance should be attached to it immediately beneath its outlet. If a trap forms part of an appliance, Building Regulations require that the appliance be removable. All traps should be accessible and provided with an adequate facility for cleaning. The sizes of tubular traps and seals are given in BS 5572: 1978 Section 8.

*Branch Discharge Pipes*

Branch discharge pipes should not be reduced in diameter in the direction of flow. Data for the sizing of branch discharge pipes are given in BS 5572: 1978 Sections 9 and 10 and in the Building Regulations 1985.

All branch discharge pipes should be connected into a discharge stack except where appliances on the lowest floor are within 1·5 m of the invert level of the drain. Where branch discharge pipes connect into a discharge stack, they must be arranged to avoid cross flow from one branch into the other. Branch discharge connections at the base of a discharge stack or offsets in the wet portion of the stack should not be closer than shown in Table B8.1.

**Table B8.1.** Minimum height for lowest discharge pipes.

| Number of storeys in building | Minimum height from invert level to centre of lowest branch discharge pipe |
|---|---|
| Up to three | 450 mm |
| Up to five | 750 mm |
| Six and over | One storey height |

*Branch Ventilating Pipes*

BS 5572: 1978 Section 9 gives the data required to size branch ventilating pipes and also provides design arrangements. When connecting to an appliance the ventilating pipe should be within 300 mm of the trap and not connect to the discharge stack below the ‘spill-over’ level of the highest appliance. The minimum diameter of a branch ventilating pipe serving one appliance is 25 mm, unless the branch is longer than 15 m and has more than five bends, in which case a 32 mm diameter pipe should be used.

*Discharge Stacks*

All discharge stacks should have an internal diameter not less than the largest trap or branch discharge pipe connected. The discharge stack above the topmost appliance should be continued with no reduction in diameter and discharge to atmosphere. Offsets in the wet portion of a discharge stack should be avoided where possible, but where they must be fitted, large radius bends should be used. However, a ventilating stack may still be necessary with connections to the discharge stack above and below the offset. Bends at the base of a discharge stack should be of large radius, or preferably two 45° large radius bends should be used.

Data for the sizing of discharge stacks are given in BS 5572: 1978 Section 10.2 and the Building Regulations 1985, Approved Document H1, Table 4.

Access points for clearing blockages should be provided at each floor level, every change of direction and at the base of the discharge stack.

*Ventilation of Discharge Stacks*

Where discharge stacks serve appliances on different floors, induced siphonage and back-pressure must be limited. These effects depend on the number and distribution of appliances, pattern of use, height of building and dimensions of pipes. Where many appliances are connected to the discharge stack, a vent stack is necessary; data for sizing ventilating stacks are given in BS 5572: 1978, Table 6.

All ventilating pipes terminating to atmosphere, whether for branch discharge pipes or discharge stacks, should end not less than 900 mm above any window or opening into the building within 3 m of that opening. The outlet should be protected by a cage or other cover not restricting the flow of air.

Discharge and ventilating stacks or branch ventilating pipes may terminate inside a building if fitted with an air admittance valve. The use of this arrangement should be considered carefully, however, since failure of the valve could cause a health hazard. Where used, these valves should not hinder the ventilation required for below-ground drainage by traditional open ended discharge ventilation systems. Any room or area within the building where the pipe terminates must be adequately ventilated. Only valves with a current British Board of Agrement Certificate should be used.

*Admission of Rainwater Outlets into Discharge Stacks*

Discharge stacks which also collect rainwater from roof areas can develop severe pressure fluctuations during heavy rainfall, and flooding may occur. It is recommended that the practice be limited to roof areas of not more than 40 $m^2$ per stack for buildings of less than ten storeys. See BS 5572: 1978.

## ROOF DRAINAGE

### Areas to be Drained

The volume of water to be drained from a roof is determined from the roof area and the intensity of rainfall. The method of calculation of areas to be drained and outlet gutter sizes is detailed in Building Regulations 1985, Approved Document H3, Tables 1 and 2.

An overflowing gutter or flat roof can cause extensive damage. Where necessary, weir overflows at the end of a gutter or overflow pipes from flat roofs should be provided. Valley gutters should have a minimum width of 450 mm.

### Rainfall Intensity

Rainfall intensity varies with geographical location. For the British Isles refer to BS 6367: 1983 to estimate the return period of rainfall intensity. A design rate of rainfall of 75 mm/h is generally satisfactory for eaves roof gutters where the overflow is unlikely to occur inside a building. For valley gutters use a minimum of 150 mm/h. For detailed data and calculations refer to BS 6367: 1983 Appendix A.

### Capacity of Gutters

The positioning of rainwater outlets is important. The Building Regulations 1985: Approved Document H3, Table 2 calculations use an 8 m interval; an adjustment is given for 16 m between outlets using half-round eaves gutters laid level with a sharp-edged outlet. For other sections and gutters laid to a gradient, refer to BS 6367: 1983.

### Rainwater Pipe Inlets

The inlet capacity depends on the permissible water depth and approach velocity. Also, as inlets are effectively weirs, the capacity is greater if the inlet is central. For roofs, flat-bottomed gutters and box-type receivers, see BS 6367: 1983 Section 8.0 concerning gutter outlets and rainwater pipes.

The capacity of a vertical rainwater pipe is greater than the inlet from a gutter or roof. Outlets with grilles (domes) need attention since the outlet from the gutter to the rainwater pipes can only pass through the sum of the area of the slot openings. Gutter capacity can be increased by discharging into a sump receiver before the rainwater pipe. The diameter of the rainwater pipe is a function of the type of gutter, the type of outlet and the receiving underground drainage system.

## INCINERATORS AND MACERATOR SYSTEMS

Modern heating boilers cannot be fed with combustible rubbish. This has greatly increased the demand for incinerators and macerators, particularly those for the disposal of sanitary towels and dressings.

Gas and electric incinerators are now less common because of problems in discharging flue gases. Macerators are now favoured; the waste is shredded and flushed into the drainage system.

### Macerators

*Individual Macerator Units*

These may be wall or floor mounted or recessed. Each unit requires water, waste and electricity connections. The water is introduced through a solenoid valve; an air break is fitted inside the machine to prevent back-siphonage.

A tubular trap should always be fitted; this and the discharge pipe should be not less than 40 mm in diameter. The discharge pipe should be as short as practicable, connecting directly to a main discharge pipe or stack. The discharge pipe gradient should be at least 4°, although a steeper gradient is advisable.

*Central Macerator System*

This system is suitable for multi-storey buildings where the female toilets are aligned vertically and a chute may be installed ending at a central disposal unit at the base. Fig. B8.1 shows a typical configuration; a vertical pipe rises the full height of the building and is generally concealed in ducts carrying the services supplying the toilet areas.

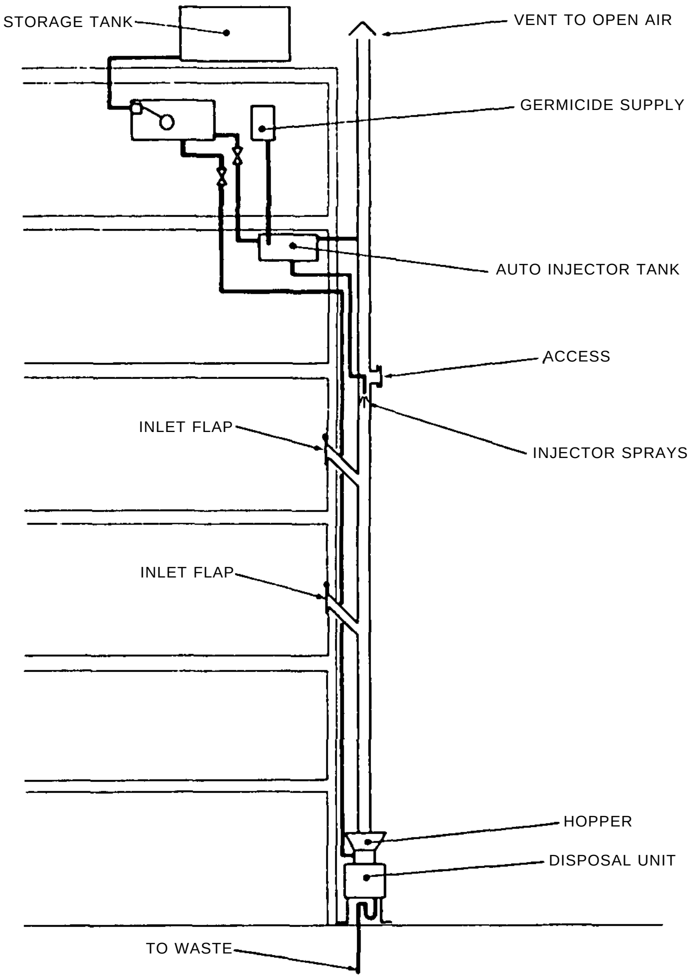



**Fig. B8.1.** Central macerator system.

Sanitary towels are inserted into the chute through access flaps flush in the toilet area wall. The disposal unit with receiver hopper is located at the base of the chute. At periodic intervals the disposal unit is activated automatically, pulverising the towels which have accumulated in the hopper and passing them in suspension into the soil drainage system. To maintain hygienic conditions in the chute it is flushed regularly and automatically with disinfectant including a wetting agent.

The chute is constructed from soil grade PVC pipe of 150 mm internal diameter. Each floor has a branch pipe of 100 mm diameter set at an angle of 135°, terminating with a lift-up flap door on the toilet area wall.

The system is served by a storage tank of about 1000 litre capacity which is normally supplied from the rising main within the building or from the building's main cold water storage tanks. The tank supplies the disposal unit at the base of the stack and the automatic flushing system at the top.

The flushing system comprises an automatic flushing tank of the urinal flushing pattern which discharges about 5 litres of water every 20 minutes into the stack through a rose similar to a shower head. Disinfectant is stored in an inverted 5 litre can gripped in a bracket above the automatic flushing cistern. The concentrated disinfectant is drip fed into the flushing water by a device set to pass 5 litres every six days.

Where the installation serves more than ten storeys, there should be an intermediate flushing system about halfway up, or at nine storey intervals if the height is greater than twenty storeys. The disposal unit has a 50 mm outlet fitted with a 75 mm deep seal which should be a tubular, not a bottle, trap connected to the nearest available soil drain by the shortest possible run. Where required, bends should be of the largest possible radius, and the waste pipe branch should be completely independent of all other fittings up to its connection to the drain.

*Kitchen Waste Macerator Units*

These range in size from the domestic to units for canteens and large kitchens; the latter are either attached to sinks or free standing, and may incorporate a stainless steel hopper to receive refuse. The function of the macerator is to reduce waste to small particles with an accompanying flow of water. The units have a water and waste attachment discharging direct to the soil and waste pipe system. Connection methods, materials, construction and pipework ventilation follow the same general principles as for the discharge of ordinary waste. However, there are special requirements if the possibility of blockages in the waste pipe system is to be minimised.

The waste water is passed through a tubular trap of not less than 40 mm diameter for domestic and 50 mm diameter for industrial applications. The trap should be located to allow easy access for cleaning. The discharge pipe should be as short as possible, and any bends of large radius. The gradient of the discharge pipe should be not less than 7½° or as recommended by the manufacturer. It is an advantage, particularly in industrial installations, if other waste appliances are connected to the discharge pipe upstream of the macerator. The discharge pipe should connect either directly into a vertical soil and waste stack or into a drain without an intervening gulley trap.

*Commercial and Hospital Macerator Units*

Commercial waste macerators and a number of special units for hospital use, for example the papier mâché bedpan disposer, are also available. The latter is fully automatic and can take up to three bedpans or four urine bottles. Interlocks ensure that the lid remains shut during operation. Bedpans are reduced to pulp and flushed away through the drainage system. The cycle lasts about 75 seconds.

Water supply to these machines should be from an independent feed tank of nominal capacity about 23 litres, located 1·5 to 2 m above above the machine. The plumbing connections follow traditional practice in accordance with requirements for connections of soil fittings to soil pipes.

**Gas Incinerators**

*Ventilation*

Air must be introduced to a toilet compartment to meet the Public Health requirements, and this will normally also furnish enough combustion air for an incinerator.

*Natural Draught Conditions*

A gravity or natural draught flue is the simplest means of venting an incinerator, but this method can be used only where the appliance is in a naturally ventilated toilet. Discharge of combustion products will be influenced by the configuration of the internal flue piping, and the position of the external flue in relation to adjacent structures and average wind conditions.

The following general requirements apply to the installation of single incinerators with separate flues:

(*a*) The independent flue for each incinerator should have a constant internal diameter not less than 100 mm; where the total length exceeds 15 m this should be increased to 150 mm.

(*b*) Any horizontal flue run should be kept to the absolute minimum and not exceed 3 m or one third of the vertical height, whichever is less.

(*c*) The length of the external flue should be kept to a minimum and terminate at a height of not less than 0·9 m above the eaves.

*Mechanical Draught Conditions*

For only one appliance a small low-powered fan in the flue duct will suffice. A well-designed incinerator can work over a range of extractor flow rates of 9 to 30 litre/s, allowing considerable latitude in assessing fan size.

A common flue may be provided for multi-appliance applications, the size depending on the number of incinerators discharging into it. In general, for gas-fired incinerators the flue branches from individual incinerators should be of 100 mm diameter. On the upper floors it may be necessary to reduce this to 75 mm diameter to reduce the air flow through those incinerators nearest to the extract fan.

With correct flue design, dampers on the branches or elsewhere are unnecessary; their use can cause operational and maintenance problems. A damper may be needed to the fan to regulate overall air flow rates. Both the fan and the damper must be easily accessible.

*Flue Sizing Procedure*

The principles underlying the sizing of incinerator flues are the same as those for boiler flues; the method is outlined in Section B13. The quantity of air in excess of that needed for combustion is greater in an incinerator than in a boiler, and the incinerator must be capable of operating over a wide range of air supply rates. A flow rate of combustion products from typical small gas-fired incinerators may be taken as 20 litre/s, and incinerators must be capable of operating over the range 9 to 30 litre/s. If a number of incinerators are connected to a main flue, this is sized on the basis of allowing an air and combustion product flow rate of 20 litre/s from each incinerator and a velocity of 7·5 m/s in the main flue. Table B8.2 gives the total combustion and air product volume as a function of number of incinerators. The nearest common flue size is also noted.

The data in Table B8.2 apply to incinerators:

(*a*) having a gas flow rate of about 0·1 litre/s.

(*b*) producing an average of 20 litre/s of air and combustion products.

(*c*) having a maximum gas velocity of 7·5 m/s in the flue.

**Table B8.2.** Volume flow rate of combustion products for sizing a common flue.

| Number of incinerators | Volume flow rate of combustion products (litre/s) | Nearest commercial flue size (mm) |
|---|---|---|
| 1 | 20 | 100 |
| 2 | 40 | 100 |
| 3 | 65 | 100 |
| 4 | 85 | 125 |
| 5 | 105 | 150 |
| 6 | 125 | 150 |
| 7 | 150 | 175 |
| 8 | 170 | 175 |
| 9 | 190 | 200 |
| 10 | 210 | 200 |
| 11 | 235 | 200 |
| 12 | 255 | 225 |
| 13 | 275 | 225 |
| 14 | 295 | 225 |
| 15 | 315 | 225 |

In fixing the diameter of the main flue it must be remembered that the same size is used throughout, and that this must relate to the maximum flow of air and combustion products. 100 mm is regarded as a minimum diameter to prevent blockage, although this may have to be reduced to 75 mm diameter on some installations to balance individual branch flow rates within the specified limits. To allow for variation in fan suction between incinerators up to the common flue, a step-by-step calculation procedure is recommended.

The calculation starts at the branch of the index incinerator, i.e. that farthest from the fan. The assumed flow of combustion products through the branch of the index incinerator is given in Table B8.3. The pressure loss through the branch and the common flue between the index branch junction and the junction of the branch of the next incinerator may be calculated using the assumed flow. This pressure loss is the same as that across the branch of the next incinerator; it is then used to establish the flow through this branch. Added to the previous flow this enables the pressure loss in the next section of the common flue to be determined.

**Table B8.3.** Volume flow rate of combustion products.

| Number of incinerators | Volume flow rate of combustion products (litre/s) |
|---|---|
| Up to 4 | 20 |
| 5 to 7 | 15 |
| 8 or more | 10 |

**Electric Incinerators**

Electric incinerators generally have a 65 mm or 80 mm flue and are intended for single installations. The first 600 mm of flue pipe should always be fixed vertically.

When discharging multiple electric (fan exhaust type) incinerators into a common flue, a suitable relay and wiring must be provided to start all incinerator fans automatically when any one is operated. Where multiple electric incinerators without built-in fans are connected to a common flue, mechanical extraction should be provided as for gas incinerators.

**Manual Collection Methods**

Where mechanical destruction systems are not used, some form of manual collection method is normally adopted. Individual floor standing units are provided in toilet areas on the basis of one unit per 40 females. The units are supplied sterilised and charged with concentrated germicidal fluid. They are collected monthly, cleaned and sterilised before reissue, and need no service connection. An alternative method employs an electrically heated sealing unit mounted on the wall in the toilet area. This comprises an opaque polythene bag supported in a housing with a hinged access lid. When the bag has been filled a clamp is pulled foward and a control button is depressed to hermetically seal the contents. The sealed bag is then placed with the other building refuse ready for collection.

**Industrial Incinerators**

Industrial type incinerators must be installed so as not to emit smoke, smell or flyash. Smoke and smell are caused mainly by incomplete combustion of the refuse. Given a temperature high enough to ensure ignition, enough air for complete combustion, and enough time for the process to be completed, they can be avoided. Separate air inlets, special grates and fuel-fired after-burners provide ignition conditions and large secondary combustion zones allow time for the whole process.

Industrial incinerators are usually oil or gas-fired. Where refuse of low calorific value needs additional fuel for ignition, the gas burner is usually of the high pressure type using compressed air rather than the normal aerated burner. Some types have a recuperator producing hot water, steam or hot air.

Refuse disposed of by industrial incinerators can be divided into the following categories:

(*a*) Household refuse

(*b*) Paper, cardboard, wood scraps, floor sweepings

(*c*) Hospital wastes; dressings, papers, flowers, sweepings, human tissue

(*d*) Plastics, oily wastes, rubber, bituminised paper tea chests

(*e*) Particulate wastes; wood chips, sawdust, flax shives, cotton waste

(*f*) Animal bodies and bedding

(*g*) Liquids; flammable and non-flammable

(*h*) Sludges; flammable and non-flammable

(*i*) Gases.

The calorific value and density of various types of municipal refuse vary widely; the situation is fluid as new packaging techniques are introduced and social habits change. Table B8.4 gives data typical of 1970.

Density varies from 100 to 300 kg/m$^3$, 150 kg/m$^3$ being a typical figure for 1970.

*Hospital Waste*

Hospital waste is incinerated in industrial type oil or gas-fired appliances. Their design varies according to the materials to be incinerated. One model uses reverberatory after-burning following the first drying on the hearth. This enables the unit to deal effectively with wet wastes.

## WASTE COLLECTION AND STORAGE

The provision of on-site storage facilities for waste before its removal must be considered at the beginning of building design, as must the method of waste collection to be used by the Local Authority or private contractor.

Waste from dwellings is removed by the Local Authority without charge, usually once per week. For most other premises the waste falls into the trade classification for which collection is charged by volume.

**Calculation of Waste Output**

The amount of waste generated from a building can be estimated by reference to BS 5906: 1980 which indicates the output in kilograms together with the bulk density figures.

**Site Collection and Storage**

For single dwellings, waste is stored in a dustbin or plastic sack and left in a convenient location for the weekly collection. Dwellings arranged in blocks up to four storeys high should be provided with waste chutes where possible, depending on the building layout. Where this is not possible, communal waste storage containers should be provided. Where dwellings are constructed over four storeys in height, waste chutes should be provided at nominally 60 m intervals to ensure that no occupant is required to walk further than 30 m to the

**Table B8.4.** Calorific value and analysis of municipal refuse.

| Constituent | % by mass | Analysis (% by mass) | | | | | | | Gross Calorific Value |
|---|---|---|---|---|---|---|---|---|---|
| | | $H_2O$ | C | H | O | N | S | Ash | (kJ/kg) |
| Dust and cinder | 22·9 | 16·6 | 14·4 | 0·2 | 0·2 | – | 1·6 | 67·0 | 4 070 |
| Large cinder | 4·5 | 19·9 | 41·8 | 1·3 | 5·1 | – | 2·9 | 29·0 | 16 200 |
| Paper | 32·5 | 5·5 | 35·0 | 5·1 | 39·1 | – | 0·2 | 15·1 | 15 550 |
| Vegetable | 19·3 | 76·5 | 11·1 | 1·4 | 8·1 | 0·7 | 0·1 | 2·1 | 4 650 |
| Metal | 7·1 | – | – | – | – | – | – | 100 | – |
| Glass | 7·9 | – | – | – | – | – | – | 100 | – |
| Rag | 2·2 | 14·0 | 31·0 | 5·4 | 38·7 | 2·3 | 0·5 | 8·1 | 15 800 |
| Unclassified | 3·5 | 2·9 | 67·7 | 8·6 | 7·8 | – | 0·9 | 12·1 | 34 300 |
| Composite Sample | | 21·5 | 21·7 | 2·5 | 15·6 | 0·2 | 0·6 | 37·9 | 9 150 |

chute. The installation of sink waste macerators should be considered to dispose of organic waste into the drainage system. If these are to be installed, Local Authority approval must be obtained.

For large projects containing residential or a mixture of residential and commercial premises, other methods of transferring collected waste to a single storage facility may be necessary. Pneumatic or hydraulic conveyors may be used. To date, pneumatic conveyors have found limited application in the UK. A single large-bore pipeline transports crude waste material. Although a small-bore system has been developed by the BRE for transporting shredded waste, no such system has yet been installed in this country.

A few hydraulic conveyors have been installed in residential buildings; they can remove about 40% of household waste. An alternative collection method is needed for the rest.

Waste is normally collected manually by cleaning staff from commercial buildings in the United Kingdom, regardless of height. Some exceptions are, however, emerging.

Practice abroad shows wider use of waste chutes installed in a cleaner's room on each floor, to transfer the waste vertically to the basement storage area.

Waste chutes should conform to BS 1703: 1977, have a minimum diameter of 450 mm, be provided with cleaning and fire protection facilities and be adequately sound insulated.

### Reduction in Waste Volume

Since the charge for removing trade waste is based on volume, there is economic motivation for installing equipment to reduce its volume. The following methods can be considered:

(*a*) *Compaction*
There is a large selection of equipment available capable of volume reduction ratios varying from 1·5 to 15, which can be installed at the base of waste chutes or coupled to a sealed waste container.

(*b*) *Shredding and Crushing*
Where this method is adopted the waste can be reduced to approximately 50% of its volume. This is a necessary component of a small bore pipeline transporting system.

(*c*) *Baling*
Where large quantities of paper or cardboard waste are involved, significant reduction in volume can be obtained by the installation of baling equipment.

(*d*) *Incineration*
Due to the stringent requirements contained in the Clean Air Acts 1956 and 1968, there has been a marked reduction in the use of incinerator equipment as a means of reducing the volume of waste in buildings. Where this equipment is used, a reduction in the region of 90% can be achieved.

## EXTERNAL LOADS ON BURIED PIPELINES

The structural design of pipelines is based on the work of Marston, Spanger and Schlick of the University of Iowa, supplemented by data obtained from British tests. This work is summarised in Section 3, Clause 11 of BS 8301: 1985. This clause gives illustrated examples of beddings for rigid and flexible pipes together with the relationship between pipe crushing strength, bedding class, range of cover and surface conditions.

## CESSPOOLS, SEPTIC AND SETTLEMENT TANKS

Cesspools, septic tanks etc. should generally be used only temporarily for dealing with sewage from domestic or small buildings until a system of sewers is provided. If this will not be in the foreseeable future then it is preferable to use treatment and disposal unless exceptional conditions render this impracticable.

### Cesspools

A cesspool is a tank, normally in the ground, designed to store the whole sewage discharge from premises between disposals by tank vehicle or otherwise.

There may be a conflict of priorities in siting a cesspool between ease of emptying and adopting a drainage layout to suit a future sewerage system. Approved Document H2 to the Building Regulations 1985 includes the requirements that a cesspool shall be:

(*a*) of not less than 18 $m^3$ capacity measured below the level of the inlet. (This generally gives 45 days capacity).

(*b*) covered and impervious to rainwater, groundwater and leakage.

(*c*) ventilated.

(*d*) sited so as not to pollute water sources or cause public nuisance.

(*e*) sited so that the contents may be removed other than through a building, with reasonable access for tanker vehicles where required.

For sizing calculations and constructional details, refer to BS 6297: 1983. Cesspools are not permitted in Scotland.

### Septic Tanks

A septic tank is a purification installation designed to accept the whole sewage discharge from premises and treat it so as to render the final effluent acceptable, by prior agreement with the Water Authority, for discharge to a watercourse or the land.

Settled sludge is retained for three to twelve months; this changes bacterial action from aerobic to anaerobic, producing methane gas. The septic effluent from the tank needs more aeration than that from a straightforward settlement tank and, depending on size and conditions, the aeration and final purification may be by percolation filter or by sub-surface irrigation, or both.

For details of design of septic tanks refer to BS 6297: 1983.

### Disposal of Effluent

This is usually to a watercourse or to underground strata. However, before discharge can be made to a watercourse, the appropriate water, river or other controlling authority must be consulted. Where it is proposed to discharge final effluent to underground strata, the appropriate authority must also be consulted.

Care must always be taken to prevent pollution of potable water supplies and where underground water is used as a source of supply, the Water Authority must be consulted before any effluent is discharged to the land.

## MATERIALS AND FITMENTS

### Piping and Jointing

British Standards cover all materials from which above and below-ground drainage pipes and fittings are manufactured. Each material has specific characteristics making it suitable for particular applications. For above-ground drainage, the choice of material usually depends on:

(*a*) the nature and temperature of the discharge

(*b*) the building fabric fire protection requirements

(*c*) the physical strength, weight and ease of assembly of the pipework.

The physical properties and usual applications of common pipework materials are given in BS 5572: 1978, Table 5. Approved Document H1 to the Building Regulations 1985 gives further information.

For below-ground drainage applications, the principal properties and applications are summarised in BS 8301: 1985, Table 3.

### Materials for Above-ground Drainage Systems

Materials fall into the two groups of metal and plastic, although glass and pitch impregnated fibre may be used in special circumstances.

*Metals*

Cast iron is available with standard manufactured fittings; jointing being in the form of plain ended pipe with bolted two-piece clamps and gaskets; or spigot and socketed pipes caulked with molten lead, fibrous lead yarn, cold compound or push-fit rubber gasket. See also BS 416 and BS 437.

Copper is available with standard manufactured fittings. Jointing may be of either the compression or capillary types, although welded, brazed or bolted flanges may be used. See also BS 864, BS 2871 and BS 1184.

Galvanised steel pipework incorporates prefabricated sections to suit individual applications. Each section is galvanised after fabrication. The jointing between sections may be in the form of plain ended pipe with bolted two-piece clamps and gaskets or spigot and socketed pipes caulked with a cold-compound or rubber gasket. Cutting and welding of pipework already galvanised should be avoided. See also BS 3868.

Lead was once the traditional material, but is now little used for discharge pipework. It is very adaptable and easy to repair but undergoes thermal movement (creep) at ambient temperatures and requires closely spaced or even continuous support. It can be attacked by portland cement, lime, plaster, brickwork and magnesite. Normal jointing methods are wiping with plumbers solder and lead welding. See also BS 602 and BS 1085.

Stainless steel is generally only used for rainwater systems to suit decorative environments. Jointing is normally by spigot and socket pipes with rubber gaskets. See BS 4127.

*Plastics*

Several thermoplastics are suitable for discharge pipes. They fall into two groups:

(*a*) employing solvent cemented or synthetic ring type joints.

| | |
|---|---|
| ABS: | Acrylonitryl butadiene styrene; BS 5255 |
| muPVC: | Modified unplasticised polyvinyl chloride; BS 4514 |
| uPVC: | Unplasticised polyvinyl chloride: BS 4514. |

(*b*) employing fusion welded or synthetic ring type joints.

| | |
|---|---|
| HDPE: | High density polythene; BS 1973 and BS 3796 |
| PP: | Polypropylene; BS 5254. |

These pipes are available with standard manufactured fittings, are light in weight and easy to handle. They are highly resistant to normal corrosion but particular solvents and organic compounds can damage them. The relevant British Standard and pipe manufacturer should be consulted if these conditions are likely to be encountered. The coefficients of thermal expansion of these plastics are much higher than those of metal.

uPVC is the most commonly used plastic for large-diameter discharge and ventilating pipes, but due to softening is unsuitable where large volumes of water are discharged at temperatures exceeding 60°C. The short term full bore discharge from some appliances may be at a much higher temperature, e.g. some types of washing machine discharge water at 80°C or even higher, and the risk of distortion is correspondingly greater.

ABS pipes can be used in much the same way as uPVC but have the advantage of suitability for use at up to 70°C.

HDPE pipes are used mainly for smaller diameter waste pipes and laboratory waste installations. They are more flexible than uPVC, less liable to impact damage and resist damage by freezing. The pipework requires closely spaced or even continuous support.

Polypropylene pipes can be used in much the same way as HDPE pipes but have the advantage of suitability for use at up to 100°C.

*Glass*

Borosilicate glass is used mainly for laboratory waste installations, but may also be used for waste systems from specialised processing or environmental equipment, e.g. food factories and air conditioning systems. The pipework is available with standard manufactured fittings. Jointing is in the form of machined spigots with bolted clamps and gaskets.

*Pitch Impregnated Fibre*

This material is little used today and is normally restricted to main discharge stacks. The pipework is available with standard manufactured fittings. Synthetic push-fit ring joints are used. See BS 2760.

**Joints between Pipes of Different Materials**

Electrolytic corrosion may occur where dissimilar materials are in contact in the presence of moisture. For details of suitable joints between pipes of different materials, refer to BS 5572: 1978.

**Provision for Thermal Expansion**

The extent to which this is necessary varies with the materials used. Copper, glass, pitch fibre and particularly plastic pipes fail if thermal expansion is not provided for. Cast iron and galvanised steel are less vulnerable but extremes of usage, such as canteen kitchens on upper floors, need expansion provision.

BS 5572: 1978 and Section B16 of the *Guide* give further information.

**Support and Fixing of Pipes above Ground**

Cast iron, galvanised steel and pitch fibre pipework may be supported by cast iron, malleable iron or steel hangers, or by purpose-made brackets and hangers fixed to the structure.

Copper pipework may be supported as cast iron, but the fixings should be of copper alloy.

Lead pipes may be supported using lead tacks soldered or welded to the pipe and fixed to the structure.

Plastic pipework may be supported by metal or plastic coated metal holder bats or suitable plastic brackets, but particular allowance should be made for thermal expansion of the pipework.

**Table B8.5.** Types of flexible/telescopic pipe joints.

| Type of pipe | Material | Type of pipe ends | Joint description | Pipe size range (mm) | Trade name(s) of joint | British Standard |
|---|---|---|---|---|---|---|
| Rigid pressure | Asbestos cement | Double spigot | Push-in sliding composite rubber rings fixed in socket groove | 75-750 | Everite Widnes | 486, 3856 |
| | Asbestos cement | Double spigot | Push-in sliding rubber rings with single grooved fin fixed in grooves in asbestos cement | 75-750 | Eternit Comet | 486, 3656 |
| | Asbestos cement | Double spigot | Sliding rubber 'O' rings axially compressed in cast iron collar by bolted glands | 75-250 | Eternit and Everite detachable | 486, 3656 |
| | Asbestos cement | Double spigot | Wedge shaped rubber ring axially compressed in steel collar by bolted steel glands | 50-900 | Viking Johnson | 486, 3656 |
| | Cast or spun iorn | Double spigot | Wedge shaped rubber rings axially compressed in steel collar by bolted steel glands | 50-1000 | Viking Johnson | 78, 1211 |
| Rigid non-pressure | Cast or spun iron | Double spigot | Synthetic rubber gasket with central register fixed with bolted external metal clamp | 50-150 | Stanflex Timesaver | 437, 6087 |
| Flexible pressure | Ductile iron | Spigot and socket | Push-in sliding composite rubber ring fixed in socket groove | 80-1000 | Tyton | 4772 |
| | Ductile iron | Double spigot | Wedge shaped rubber rings axially compressed in steel collar by bolted steel glands | 80-1000 | Viking Johnson | |
| | Unplasticised PVC | Double spigot | Wedge shaped rubber rings axially compressed in steel collar by bolted steel | 50-630 | Viking Johnson | |
| | Unplasticised PVC | Spigot and socket | Push-in sliding 'O' ring permanently fixed in socket groove | 75-300 | Wavin Safe Joint | 3506 |
| Flexible non-pressure | Unplasticised PVC | Double spigot | Push-in sliding rubber 'O' ring fixed in moulded PVC sleeve with no central register | 80-300 | | 4660, 5481 |
| | Unplasticised PVC | Double spigot | Push-in sliding rubber 'O' ring fixed in grooves in moulded PVC sleeve with central register | 100-250 | | 4660, 5481 |
| | Unplasticised PVC | Double spigot | Solvent-welded PVC sleeve with central register and push-in sliding rubber 'O' ring fixed in grooves in sleeve | 80-200 | | 4660, 5481 |

The maximum distances between pipe supports for various materials should not exceed those given in BS 5572: 1978, Table 13, and are further discussed in Section B16.

**Access to Above-ground Drainage Systems**

Access to discharge and ventilating pipework is required for testing the system as well as for clearing stoppages. For multi-storey systems sectional testing may be necessary. Access openings should themselves be accessible. It is advantageous to ensure that WC pans and similar appliances are easily removable. Obvious positions for access are at branch connections, at all changes of direction and at the foot of all discharge stacks. Where pipework is installed in ducts with restricted accessibility the access should be positioned above the spill-over level of the sanitary fitments served by the pipework. All access openings should have airtight and watertight covers firmly secured.

**Flexible Joints in Below-ground Drainage**

Below ground joints should preferably be flexible. Proprietary flexible joints are manufactured to suit all pipe materials; a typical range is given in Table B8.5.

**Access to Below-ground Drainage Systems**

Access to below-ground drainage systems is required for testing, inspection and maintenance as well as for clearing stoppages. Inspection chambers and manholes allow for rodding to both upstream and downstream drain sections. Access fittings are a limited provision for rodding in both directions, and rodding eyes permit rodding downstream only.

For dimensions and spacing of access points, refer to Building Regulations Approved Document H1, Tables 9 and 10.

Rodding eyes provide access at the surface level and are usually of the same diameter as the drain, with a sealed cover.

Access fittings provide access either at the surface level or within an inspection chamber manhole. They are constructed as an opening in the top of the drain, with a raising piece as necessary to suit the required depth. A sealed cover is also provided. Inspection chambers provide access from the surface and are used on shallow drains. Manholes allow personnel entry for direct access to deeper drains. Inspection chambers and manholes are normally constructed after drain laying. Dimensions depend on the size of the main drain and the number and position of the branch drains connecting into it. Where there are several branches, inspection chambers and manhole sizes can be calculated as follows:

(*a*) Length – allow 300 mm for each 100 mm or 150 mm branch on the side where the greater number of branches occur, plus an allowance at the downstream end for the angle of entry of the branches.

(*b*) Width – the width should be the sum of the widths of the benching plus 150 mm or the diameter of the main drain, whichever is greater. The benching width should be 300 mm where there are branches or 150 mm where there is no branch.

Inspection chambers and manholes are generally constructed from:

(*a*) brickwork

(*b*) *in situ* and precast concrete

(*c*) uPVC, polypropylene and GRP

(*d*) Vitrified clay.

Typical constructional details for brickwork inspection chambers and manholes are shown in Figs. B8.2 to B8.6 and Table B8.6. Deep manholes should be provided with landing platforms at intervals of 6 m.

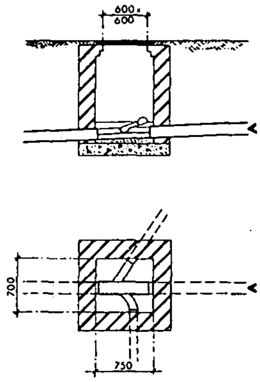



**Fig. B8.2.** Inspection chamber (invert – surface < 0·9 m).

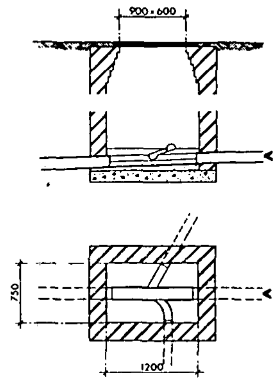



**Fig. B8.3.** Shallow manhole (0·9 m < invert – surface < 2·7 m).

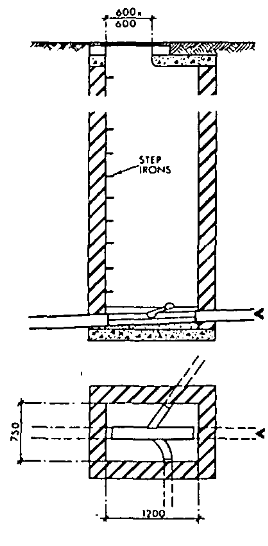



**Fig. B8.4.** Shallow manhole (2·7 m < invert – surface < 3·3 m).

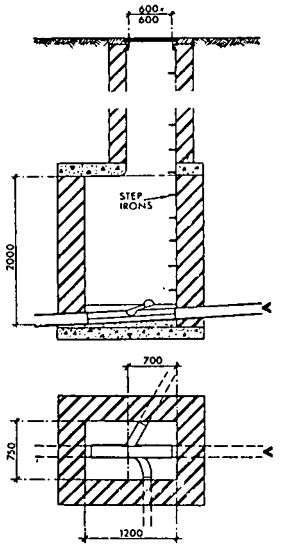



**Fig. B8.5.** Deep manhole (invert – surface > 3·3 m).

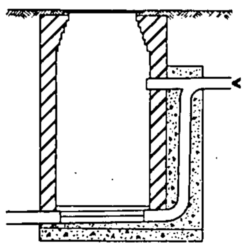


**Fig. B8.6.** Back-drop manhole for pipes up to 225 mm.

**Table B8.6.** Manhole dimensions.

| Depth to invert (mm) | Minimum internal dimensions (mm) | Cover size (mm) |
|---|---|---|
| 900 or less | 750 × 700 | 600 × 600 |
| 900-2700 | 1200 × 750 | 900 × 600 |
| 900-2700 | 1100 diameter | 550 diameter |
| 2700-3300 | 1200 × 750 | 600 × 600 |
| 2700-3300 | 1100 diameter | 550 diameter |
| Over 3300 | 1200 × 750 × 2000 high, with 700 × 700 access shaft | 600 × 600 |
| Over 3300 | 1200 diameter × 2000 high, with 700 diameter access shaft | 550 diameter |

Inspection chamber and manhole covers and frames are available to British Standards in ductile and grey cast iron, cast steel and precast concrete in a variety of styles and grades to suit particular conditions. Covers within buildings should be airtight and mechanically sealed.

Step irons and access ladders should comply with the relevant British Standards. Bricks used in inspection chamber and manhole construction should be at least to Class B engineering standard complying with BS 3921. Concrete blocks used in surface water drainage works should comply with BS 6073. Granular bedding materials should comply with the relevant clauses of BS 882.

### Gullies

Gullies for drainage work are available in materials compatible with the drainpipe. Gullies may be used with trapped or untrapped outlets depending on their function. The provision of rodding access in gullies can assist maintenance operations where no direct connection is made to an inspection chamber or manhole, or where the gulley is positioned on a long branch drain.

Where gullies are sited in areas that are likely to have loose grit, mud etc. drained off with the surface water or effluent, they should be provided with a silt or sediment pan. A variety of grids, covers and frames are available to suit particular conditions.

### Inspection and Testing

Full details for the inspection and testing of drainage and sanitary pipework are given in BS 8301: 1985 Section 5 and BS 5572 Section 12.

## BIBLIOGRAPHY

The Building Regulations 1985

Approved Documents H1, H2, H3, H4; The Building Regulations 1985

BS 416: 1973, Cast iron spigot and socket waste and ventilating pipes (sand cast and spun) and fittings

BS 437: 1978, Specification for cast iron spigot and socket drain pipes and fittings

BS 602 & 1085: 1970, Lead and lead alloy pipes for other than chemical purposes

BS 864, Capillary and compression tube fittings of copper and copper alloy

BS 882: 1983, Specification for aggregates from natural sources for concrete

BS 1184: 1976 (1981), Copper and copper alloy traps

BS 1703: 1977, Specification for refuse chutes and hoppers

BS 1973: 1970 (1982), Polythene pipe (Type 32) for general purposes including chemical and food industry uses

BS 2760: 1973, Pitch impregnated fibre pipes and fittings for below and above ground drainage

BS 2871, Copper and copper alloys: tubes

BS 3796: 1970, Polythene pipe (Type 50) for general purposes including chemical and food industry uses

BS 3868: 1973 (1980), Prefabricated drainage stack units: galvanised steel

BS 3921: 1974, Clay bricks and blocks

BS 4127, Light gauge stainless steel tubes

BS 4514: 1983, Specification for unplasticised PVC soil and ventilating pipes, fittings and accessories

BS 5254: 1976, Polypropylene waste pipe and fittings (external diameter 34·6 mm, 41·0 mm and 51·4 mm)

BS 5255: 1976, Plastics waste pipes and fittings

BS 5572: 1978, Code of practice for sanitary pipework

BS 5906: 1980, Code of practice for the storage and on-site treatment of solid waste from buildings

BS 6073, Precast concrete masonry units

BS 6297: 1983, Code of practice for the design and installation of small sewage treatment works and cesspools

BS 6367: 1983, Code of practice for drainage of roofs and paved areas

BS 6465: Part 1: 1984, Code of practice for scale of provision, selection and installation of sanitary appliances

BS 8301: 1985, Code of practice on building drainage

BS CP 2005: 1968, British Standard Code of Practice on Sewerage

# SECTION B10 ELECTRICAL POWER




*This edition of Section B10 first published: 1976*

*Reprinted with amendments: 1986*

# SECTION B10 ELECTRICAL POWER

### Introduction

This section deals with those aspects of electrical power using equipment most commonly encountered by the Building Services Engineer and which are not covered elsewhere in the *Guide*. Motors are discussed at some length, together with their associated equipment, e.g. circuit-breakers etc. Transformers and stand-by generators are mentioned as are meters and radio and TV interference suppression. Details of the differing types of wiring diagram are given along with three-phase system relationships.

## ELECTRIC MOTORS

### Motor Classification

The degree of motor protection is designated by the letters IP followed by two characteristic numerals whose classifications are given in Table B10.1. This designation may be followed by a letter indicating whether the protection against water was tested with the machine not running (S) or in operation (M). The absence of any letter means that the protection was tested with the machine running and with it not running. A machine is weather protected when its design reduces the ingress of rain, snow and airborne particles under specified conditions to an amount consistent with correct operation. This degree of protection is designated by the letter W placed between IP and the numerals. BS 4999: Part 20 gives these classifications in greater detail.

The designation used for the method of cooling depends on the nature of the coolant used. In the case of air-cooled machines, a simplified code is used which consists of the letters IC followed by the two characteristic numerals shown in Table B10.1. For other methods of cooling, the designation consists of the letters IC followed by a third letter which specifies the nature of the coolant and the two characteristic numerals shown in Table B10.1. The designations for other arrangements, such as cases where more than one coolant circuit is required, are detailed in BS 4999: Part 21, which should also be consulted for further explanation of the above classification.

**Table B10.1.** Classification of motor enclosures.

| Protection | | | | Cooling | | | |
|---|---|---|---|---|---|---|---|
| **First numeral** | **Degree of protection** | **Second numeral** | **Degree of protection** | **First numeral** | **Cooling circuit arrangement** | **Second numeral** | **Coolant circulation arrangement** |
| 0 | None | 0 | None | 0 | Free circulation | 0 | Free convection |
| 1 | Protected against solid bodies greater than 50mm | 1 | Protected against dripping water | 1 | Inlet duct or pipe circulated | 1 | Self circulation |
| 2 | Protected against solid bodies greater than 12mm | 2 | Protected against drops of water falling up to 15° from the vertical | 2 | Outlet duct or pipe circulated | 2 | Integral dependent component mounted on separate shaft (not motor shaft) |
| | | 3 | Protected against spraying water | 3 | Inlet and outlet duct or pipe circulated | 3 | Dependent component mounted on the machine |
| 4 | Protected against solid bodies greater than 1mm | 4 | Protected against splashing water | 4 | Frame surface cooled | | |
| 5 | Protected against dust | 5 | Protected against water jets | 5 | Integral heat exchanger using surrounding medium | 5 | Integral independent component |
| | | 6 | Protected against conditions on ship's deck | 6 | Machine mounted heat exchanger using surrounding medium | 6 | Independent component mounted on machine |
| | | 7 | Protected against the effects of immersion | 7 | Integral heat exchanger not using surrounding medium | 7 | Independent and separate device or coolant system pressure |
| | | 8 | Submersible | 8 | Machine mounted heat exchanger not using surrounding medium | 8 | Relative displacement |
| | | | | 9 | Separately mounted heat exchanger | 9 | Circulation by any other component |

*Note:* For example, the old weatherproof TEFC classification would be IPW54 IC41.

Insulation materials are classed on the basis of their thermal stability in accordance with BS 2757 which is summarised in Table B10.2. The ageing of insulation is proportional to temperature, an approximate guide being that for every 10°C rise above operating temperature, the life of the insulation is halved.

For normal duties the classes of insulation used are E, B and F to BS 2757. Class F is the most popular.

**Table B10.2.** Classification of insulation materials.

| Material class | Temperature /°C |
|---|---|
| Y | 90 |
| A | 105 |
| E | 120 |
| B | 130 |
| F | 155 |
| H | 180 |
| C | above 180 |

**Motor Selection**

BS 2757, BS 4999 and BS 5000 together provide a background specification for the majority of motors used on mechanical services installations. As a rule, when motors are being specified, the following interchange of information between engineer and manufacturer is necessary.

The manufacturer will need to know:

(*a*) type of motor and enclosure;
(*b*) class of insulation;
(*c*) nominal speed and variation;
(*d*) power output;
(*e*) method of starting;
(*f*) method of electrical protection;
(*g*) ambient temperature and conditions of installation;
(*h*) supply voltage and frequency;
(*i*) single or three phase windings;
(*j*) function of motor;
(*k*) reference to relevant British Standards etc;
(*l*) type and size of cable to be connected;
(*m*) approximate starting frequency and running hours per annum;
(*n*) type of bearings and mountings.

In return, the manufacturer should supply:

(*a*) starting current and duration for duty required;
(*b*) power factor at rated load;
(*c*) efficiency at rated load;
(*d*) energy returned to supply under fault conditions (for motors larger than 100 kW);
(*e*) maintenance instructions and life.

The types of motor enclosures are given in Table B10.1 and the characteristics of motors are detailed in Table B10.3.

It should be noted that the use of mineral insulated cables for boiler house and plant room motor and control wiring is common practice due to the cables' ability to withstand high temperatures. This type of cable, however, is not capable of withstanding high voltage surges caused by switching some inductive and solid state circuits. Equipment manufacturers should be informed when mineral insulated cable is to be used in order that they can advise the use of surge divertors where necessary.

**Methods of Starting Motors**

The choice of starter is normally affected by:

(*a*) the mechanical considerations such as inertia;
(*b*) the frequency of starting;
(*c*) the effect on the electrical supply distribution network such as voltage and frequency fluctuations which may affect other loads, such as computers, requiring supply stability;
(*d*) the local electricity supply authority restrictions on the methods of starting or on the starting currents and the associated voltage drops.

Local electricity supply authority regulations frequently classify starting methods related to output power rather than current limitations. Direct on-line starting limitations between 4 and 8 kW are common with the higher figure becoming increasingly acceptable. For buildings provided with their own independent electricity supply transformers, high direct on-line starting values can normally be accepted. The values selected will depend upon the effects of starting on the building electrical distribution system and stand-by alternator system. Direct on-line starting limits of up to 40 kW are not unusual for such installations. At this value, mechanical limitations of equipment and drives become paramount.

Reduced voltage starting methods using star-delta, auto transformers or primary resistance are generally acceptable up to about 25 kW depending on the requirements of the local electricity board, unless the building has its own supply transformer. When transformers are included, the electricity board are interested in the voltage drop and frequency of the motor start as registered on the high voltage or primary side of the transformer. This voltage drop is proportional to the impedance of the transformer and the motor starting current.

Severe limitations upon starting current or the need to provide a continuous high torque during starting may necessitate the use of slip-ring motors using rotor resistances during the starting cycle. The starter suppliers should be aware of which condition is important in order to select the correct values of starting resistance.

For large motors of several hundred kW the method of starting is frequently a matter of negotiation with the local electricity supply authority. Large centrifugal or screw refrigeration compressors form the principal type of load requiring such powers. Open type machines can be provided with a squirrel cage induction motor using any of the normal starting methods, alternatively, a slip-ring machine with associated rotor resistance starting can be used. Hermetic machines frequently require auto transformers with a tapping in the order of 70% to ensure adequate starting torque. A few of these machines are capable of being started by part winding methods, closed transition star-delta starters or primary resistance starters. When considering the switch gear for motors rated over 400 kW, it is possible that high voltage motors, direct on-line started, may offer economic advantages. Any economic comparison should, in such cases, include the supply transformer, cabling, the starter and the motor cost.

**Table B10.3.** Characteristics of motors.

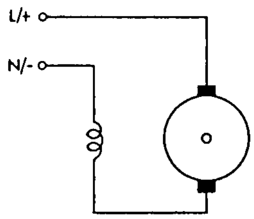
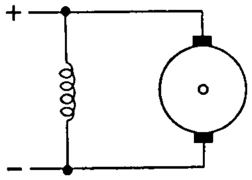
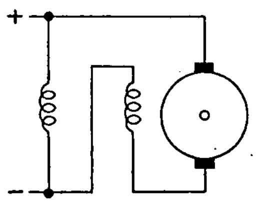
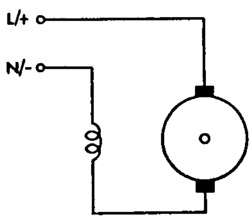
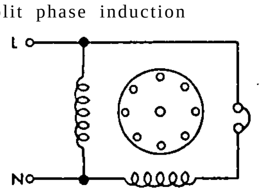
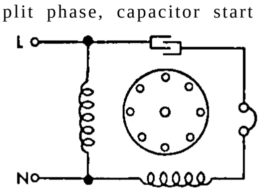
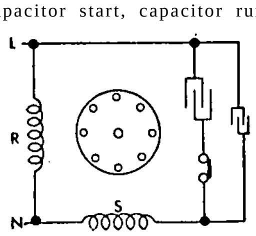
| Type of motor | Starting method | Speed control | Characteristic | Application comments |
|---|---|---|---|---|
| DC MOTORS<br>Series wound<br> | Small sizes with direct on-line switch. Large machines started with external series resistances. | External series resistances. Semi-conductor devices may be used for precise speed control from AC supplies (at higher cost). | High torque at low speeds. Speed will increase with falling load. Must be always used with a limiting load. | Used on small fans and portable tools. |
| Shunt wound<br> | Full voltage applied to field winding. Series resistance in armature circuit progressively removed. | Speed increase by added resistance in field circuit. Speed decrease by added resistance in armature circuit.<br>Semi conductor devices may be used for precise speed control (at higher cost). | Torque and speed fall slightly with load. | Suitable for pump drives and where constant speed is required. |
| Compound wound<br> | Full voltage applied to field winding. Series resistance in armature circuit progressively removed. | As above. | Characteristic of series and shunt machine according to balance of winding and polarity of connections. | Flexible speed/torque relationship possible. |
| DC or AC MOTORS<br>SINGLE-PHASE<br>Series wound<br> | Direct on-line switch. | (*a*) Series resistance.<br>(*b*) Variable voltage.<br>(*c*) Silicon controlled rectifier unit. | High torque at low speeds. Must always be used with a limiting load. | Extensively used for small portable tools. |
| AC MOTORS<br>SINGLE-PHASE<br>Split phase induction<br> | Direct on-line switch. Starting winding may be:<br>(*a*) Permanently in circuit if designed for this application.<br>(*b*) Switched out by manual, centrifugal or current control switch. | Limited speed control possible with voltage variation only when motor has been suitably designed. Wide speed range possible with auto transformer or semi-conductor control of voltage to main winding for fan motors where (load torque)$^2$ is proportional to fan speeds. | Starting torque generally poor. | Noise level of single-phase motor is generally higher than equivalent three-phase machine. |
| Split phase, capacitor start<br> | Direct on-line switch with capacitor in starting winding switched out by centrifugal or current control switch. | As above. | Improved starting torque. Quieter running. | |
| Capacitor start, capacitor run<br> | Direct on-line switch with start and run capacitors in starting circuit. Starting capacitor removed by centrifugal switch or other switching device when motor is up to speed. | As above. | Improved starting torque compared with split phase motor. Improved power factor. | Considered to be the most quiet running single-phase machine. |

**Table B10.3.** Characteristics of motors—*continued.*

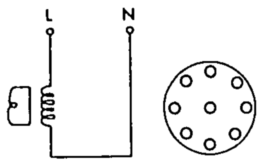
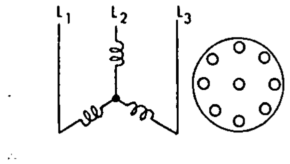
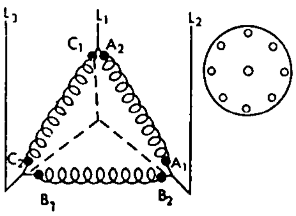
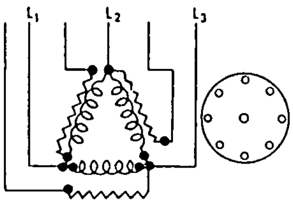
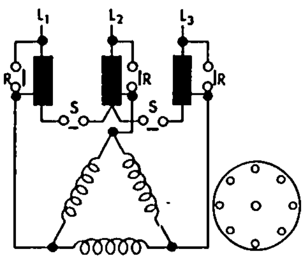
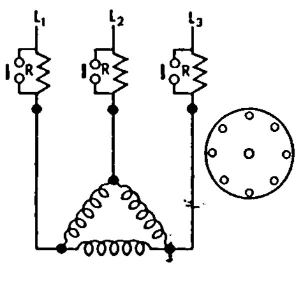
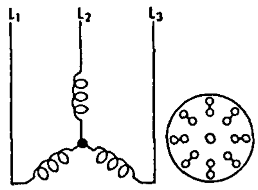
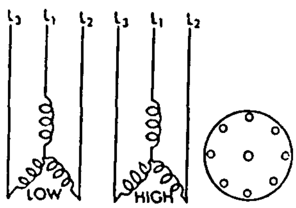
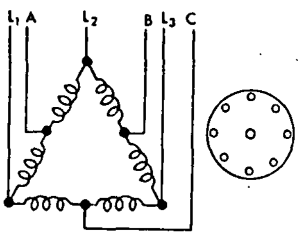
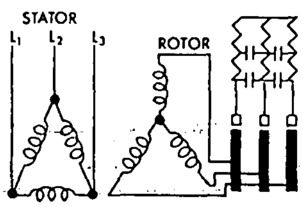
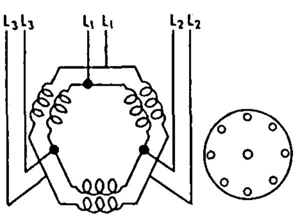
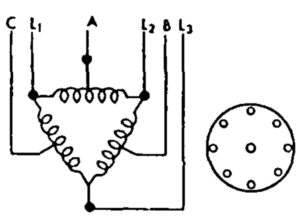
| Type of motor | Starting method | Speed control | Characteristic | Application comments |
|---|---|---|---|---|
| Shaded Pole Motor<br>Single phase induction motor shaded pole type<br> | Direct on-line switch. | (*a*) Pole changing.<br>(*b*) Voltage variation with R or X in stator winding. | Starting torque poor.<br>Low efficiency.<br>Low power factor. | Used for small FHP fan motors. Simple and cheap to manufacture. |
| POLYPHASE MOTORS<br>Squirrel cage induction (SCI) (single winding)<br> | Direct on-line switch (DOL).<br>Windings may be star or delta connected. | Essentially constant speed motors but voltage and frequency changing by semi-conductor devices allows wide range of speed control. | High starting torque with high starting currents. | Normal starting currents 5 to 8 × full load current, special cases up to 11 × FLC.<br>*Note:* The term locked rotor current is frequently used to describe value of initial switching current. |
|  | Star-delta.<br>Motor windings connected in star for initial start. After a predetermined time delay, windings are connected in delta. | As above. | 33% DOL starting torque.<br>33% DOL starting current.<br>Peaks of high current can occur during motor open circuit during change. | Method of starting must be specified to motor supplier to ensure correct connections are available. |
|  | Closed transition, star-delta.<br>Resistances are provided within the starter to enable the star-delta transition to take place without disconnection of the motor windings from the electrical supply. | As above. | Applies continuous torque during starting period.<br>Eliminates high peaks of current which may occur on open transition changeover. | Expensive. |
|  | Auto-transformer.<br>The auto-transformer is selected to provide a reduced voltage to the motor for initial starting. Open and closed voltage tappings are specified to ensure sufficient starting torque with reduced current. After starting, the supply is connected directly to the motor and the transformer connections are opened. | As above. | Starting torque and line current reduced to the square of the fraction of full voltage. | Expensive.<br>Used frequently with large, three-terminal hermetic motors on refrigeration compressors. If damaged, replacements may be on long delivery. |
|  | Primary resistance.<br>Starter contains resistors to be. connected in series with each phase to the motor. On starting the heavy current causes a large voltage drop across the resistors: thus limiting the starting current to a lower value. As motor accelerates current and voltage drop fall until resistors are short-circuited. | As above. | Reduced starting current and torque. | Used where three motor terminals only are available.<br>Restricted normally to direct coupled loads. |

**Table B10.3.** Characteristics of motors—*continued.*

| Type of motor | Starting method | Speed control | Characteristic | Application comments |
|---|---|---|---|---|
| Squirrel cage induction (high torque)<br> | Direct on-line, with special windings. The motor is designed with effectively two or more rotor cages, often the outer cage being of higher resistance. The outer cage is more effective on starting and the inner on running. | As above. | May be designed for high starting torque *or* low starting current. Power factor slightly lower compared with straight SCI motor. | Motor manufacturers must be fully informed about load characteristic and starting requirements. *Note:* On drives with long run-up times, rotor bars may overheat. |
| Dual wound squirrel cage induction<br> | Each winding treated as a normal SCI motor. Interlocks provided between contactors to prevent both windings being energised simultaneously. | Two windings provided on stator with differing numbers of pole pairs to give two speeds. | Power factor and efficiency slightly lower due to space requirements for two stator windings. | Overloads required to match each winding. |
| Pole change squirrel cage induction<br> | Special three contactor starter. Direct on-line normal. | Coil windings designed to provide 2:1 speed ratio when inter-connections are changed. | As SCI motor. | Overloads required to match connections for each speed. |
| Wound rotor or slip ring<br> | Stator connected direct on-line. Resistors externally connected in rotor circuit and progressively shorted out. Normally on a time basis. | Normal stator design. Rotor wound with the same number of poles as the stator. One set of end connectors star connected. The others taken to slip rings to which external connectors may be taken via brushes. | Power factor lower than SCI motor. | |
| Part winding squirrel cage induction<br> | Starter with two contactors and time delay. Larger winding energised first and second winding added after delay. | Two windings on stator connected in parallel for running. Ratio of windings vary from 50:50 to 30:70. Windings often asymmetrical. | Asymmetrical currents in stator windings during starting. Often noisy at start-up due to magnetic stress. | One set of overloads possible; two separate sets are normal. |
| Pole amplitude modulated<br> | Starter requirements differ with individual manufacturer's motor designs. Usually direct on-line started with contactors to provide required speed. | Special winding with short span coils which may be connected to give a number of effective pole pairs and speeds. | As SCI motor. | Economic comparisons must be made with combined starters and motors. |

*Star-delta Starting*
Some of the advantage of using this method of starting may be lost if care is not taken to maintain the correct phase sequence when changing from the star to the delta connection, The phase sequence in the UK is Red (R) Yellow (Y) Blue (B) and the correct star-delta connections are as follows (where S is the star point):

(*a*) Phase 1 winding connected RS in star changes to RB in delta.
(*b*) Phase 2 winding connected YS in star changes to YR in delta.
(*c*) Phase 3 winding connected BS in star changes to BY in delta.

To reverse the direction of rotation of the motor two of the phases should be interchanged (but not reversed) at the motor terminals leaving the star-delta relationships unchanged. It is not necessary to connect the star point to the supply neutral.

*Starting of Large Centrifugal Fans*
The inertia of centrifugal fans is higher than most other mechanical loads normally found in building services systems. With the high torque provided by a direct on-line started motor, severe stress is imposed on the drive and upon the fan shaft making alternative methods of starting desirable.

**Full Load Running Current for Motors**
Motor manufacturers publish tables showing typical running currents of motors at differing speeds and outputs. Additional information may also be provided, indicating the efficiency and power factors at differing loads. Fig. B10.l recommends terminology to be used in order to avoid misunderstanding.

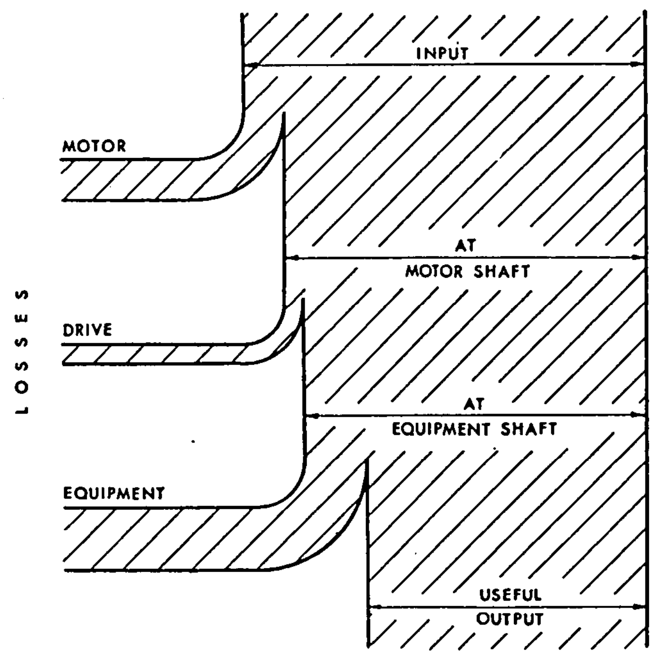



**Fig. B10.1.** Motor terminology

Values of motor full load running current for a typical squirrel cage induction motor operating at about 24 rev/s on 415 V, 50-Hz, three-phase supply are given in Table B10.4. This information is sufficient for system and equipment selection, but for overload trip selection, manufacturers' data should be consulted for each machine. Values of input current are liable to vary due to speed performance and genera1 quality.

Motors are most efficient when running at full load, i.e. a 10 kW motor being run at 5 kW is less efficient than a 5 kW motor being run at 5 kW.

**Table B10.4.** Typical values of full load running current for a three-phase squirrel cage motor (based on 415V, 50 Hz supply).

| Output at shaft/kW | Full load current/A |
|---|---|
| 1* | 2 |
| 2 | 4 |
| 3 | 6 |
| 4 | 8 |
| 5 | 10 |
| 6 | 12 |
| 8 | 16 |
| 10 | 20 |
| 15 | 29 |
| 20 | 38 |
| 25 | 47 |
| 30 | 57 |
| 40 | 73 |
| 50 | 89 |
| 100 | 175 |
| 150 | 250 |
| 200 | 330 |
| 300 | 525 |
| 400 | 700 |
| 500 | 900 |

*For motor powers of less than 1kW, manufacturers' data should be consulted owing to wide variation in full load running current.

**Protection against Overcurrent**
In accordance with IEE Regulation 552-3, every electric motor having a rating exceeding 0.37 kW is provided with control equipment incorporating overload devices to protect it against excess current. Overcurrent may be caused by mechanical overloading of the motor or by electrical failure of the winding. To protect cables, starters and motors against the effects of earth fault or short-circuit conditions, a fuse or circuit breaker is provided at the origin of the circuit, the basic arrangement being shown in Fig. B10.2. The fuse or circuit breaker must be capable of withstanding the prospective fault current of the mains. When a star-delta starter is used, there will be six cables between starter and motor and each of these cables may be rated at $1/\sqrt{3}$ of the motor full load current.

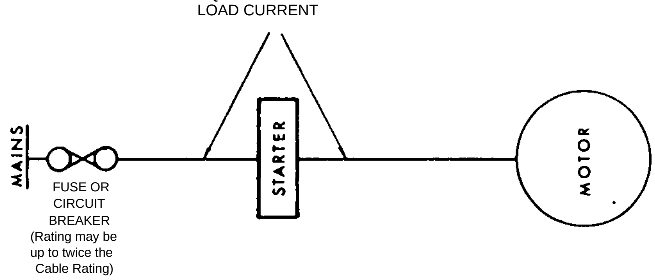



**Fig. B10.2.** Typical fuse circuit.

*Prospective Fault Current*
The prospective fault current can be regarded as the maximum root mean square (RMS) steady state current flowing if all the active conductors were solidly bolted together at points of fault. In practice, such factors as conductor and joint resistance reduce this current to a lower figure, as does the cable resistance if there is any distance between the supply and point of fault.

The prospective fault current at the secondary terminals of the supply transformer is given by:

$$I_{sc} = \frac{A \times 100}{B} \quad \text{B10.1}$$

$$= \frac{C \times 10^5}{V \times B \times \sqrt{3}} \quad \text{B10.2}$$

where:

- $I_{sc}$ = prospective fault current at secondary terminal A
- $A$ = transformer full load current .. .. A
- $B$ = percentage impedance .. .. .. %
- $C$ = transformer rating .. .. .. kVA
- $V$ = rated line voltage .. .. .. V

The short circuit current increases with the size and number of transformers feeding into a fault and Table B10.5 shows typical figures for various sizes and arrangements of transformers. In practice, impedance in cable runs, terminals etc., substantially reduces the fault currents although large rotating machines increase them. If cartridge fuses are employed as a first line of defence the symmetrical short circuit values given in Table B10.5 can be regarded as maxima.

The high rupturing capacity cartridge fuse is quick acting under short circuit conditions, and fault conditions are usually cut off before they can reach the values given in Table B10.5. This does not apply if circuit-breakers or re-wirable fuses are used exclusively, as these devices are much slower acting.

**Table B10.5.** Typical prospective fault currents for 11 kV/415 V transformers.

| Transformer rating /kVA | Prospective fault current /A |
|---|---|
| 250 | 7 800 |
| 500 | 14 600 |
| 800 | 23 400 |
| 1000 or 2 × 500 in parallel | 29 200 |
| 1 500 | 38 000 |
| 2 × 1 000 in parallel | 58 400 |
| 3 × 1 000 in parallel | 87 600 |

*Fuses*

Fuses are selected on the basis of category of duty and class. The category of duty depends on the prospective fault current and the class is determined by the fusing factor defined as:

$$F = \frac{I_{fm}}{R_a} \quad \text{B10.3}$$

where:

- $F$ = fusing factor
- $I_{fm}$ = minimum fusing current in a specified time (usually 4 hours) .. .. .. A
- $R_a$ = current rating .. .. .. .. A

Table B10.6 shows the categories of duty and Table B10.7 the classes for fuses used in AC circuits.

Classes P and $Q_1$ fuses come into the IEE category of "close" protection while classes $Q_2$ and R offer "coarse" protection necessitating the use of larger cables.

**Table B10.6.** Categories of fuse duty.

| Category of duty | Prospective current/A |
|---|---|
| AC 16 | 16 500 |
| AC 33 | 33 000 |
| AC 46 | 46 000 |
| AC 80 | 80 000 |

**Table B10.7.** Classes of fuses.

| Class | Fusing factor | |
|---|---|---|
| | exceeding | not exceeding |
| P | 1.00 | 1.25 |
| $Q_1$ | 1.25 | 1.5 |
| $Q_2$ | 1.5 | 1.75 |
| R | 1.75 | 2.5 |

For motor protection applications, cartridge fuses are preferred to re-wirable types as their characteristics can be closely controlled and they have a high resistance to deterioration in use. The fusing factor of re-wirable fuses is about two but the amount of fault energy let through is many times greater than with cartridge fuses. The class of fuse selected depends on the type of protection required and on the characteristics of the load. To protect a steady load such as a heater battery, the fuse rating can be equal to the load but for motor protection, the fuse must be able to carry the transient starting current and the rating must be increased by an amount depending on the time/current characteristics of the fuse and of the motor.

When replacing a blown fuse, it is important to ensure that the replacement fuse has the same category of duty, the same class and the same current rating as the original fuse.

*Recommended Fuse Ratings for Three-phase Induction Motors*

Table B10.8 has been compiled using the following recommendations: Direct on-line start: 7 × Motor full load current for 10 seconds. Assisted start: 3.5 x Motor full load current for 20 seconds. Class $Q_1$ fuses with fusing factor of 1.5.

Suitable modifications to the recommended ratings should be made to cover the following special conditions, which may occur either singly or in combination:

(*a*) Heavy accelerating torque which will cause an excessively long starting period.

(*b*) A large number of successive starts without intermediate cooling periods.

**Table B10.8.** Fuses for three phase induction motors (based on 415V; 50 Hz supply).

| Fuselink rating/A | Motor full load current/A | |
|---|---|---|
| | Direct on-line start | Assisted start |
| 2 | 0.9 | 1.8 |
| 4 | 1.3 | 2.5 |
| 6 | 1.9 | 3.7 |
| 10 | 2.6 | 5.1 |
| 15 | 5.2 | 10.3 |
| 20 | 7.8 | 14.6 |
| 25 | 10 | 18.6 |
| 30 | 11.8 | 22.6 |
| 35 | 15.7 | 29.2 |
| 40 | 19.3 | 35 |
| 50 | 22.9 | 42.8 |
| 60 | 28.6 | 55 |
| 80 | 41.4 | 74.1 |
| 100 | 54.3 | 97.3 |
| 125 | 71.5 | 125 |
| 150 | 94.5 | 150 |
| 200 | 127 | 180 |
| 250 | 164 | 225 |
| 300 | 186 | 250 |
| 350 | 257 | 325 |
| 400 | 307 | 350 |
| 450 | 371 | 425 |
| 500 | 450 | 450 |
| 550 | 472 | 525 |
| 600 | 535 | 550 |

(*c*) Where the peak ambient temperature exceeds 40°C or the average temperature over 24 h exceeds 35°C.

Fuse and motor manufacturers should be consulted on the modifications to be made.

*Circuit-Breakers*

The use of circuit-breakers for high voltage systems and heavy current installations is a well established practice. Air break “moulded case circuit-breakers” and “miniature circuit-breakers” have extended the application of circuit-breakers to all current ratings, down to domestic needs.

Circuit-breakers offer a number of advantages over alternative systems using high breaking capacity fuses namely: a circuit-breaker can easily be reset after clearing a fault; on three-phase systems all phases are opened simultaneously when failure occurs on one phase; remote tripping of a circuit-breaker is possible by application of over current devices, under voltage coils, earth leakage or shunt trip coils.

Circuit-breakers are recognised as being more expensive than fuses in initial cost for most applications. It is also necessary to have a full working knowledge of the distribution systems of which the circuit-breaker is to be part before selecting the correct unit. Generally, the prospective through-fault current of the distribution system should be examined before selecting a breaker, especially when they are used throughout a system. In such cases, it is usual to use high rupturing capacity fuses before the circuit-breaker to enable a smaller unit to be applied.

Circuit-breakers can be manufactured as either “close” or “coarse” protective devices and this information should be specified by the designer. As the characteristics of different makes of circuit-breakers vary, manufacturers should be consulted before selecting a circuit-breaker for a specific system.

**Protection against Mechanical Overload**

Fuses and circuit-breakers protect motor circuits against heavy fault currents; they are unable to provide adequate protection against mechanical overloading.

A motor must be protected against overheating of windings caused by machine overloading. Practical devices for achieving this include magnetic and thermal over-current units and direct thermal protection of the motor windings. It is not practical to achieve complete motor protection by this means as short circuits, and rotor faults can occur when the only benefit of fuses and trips will be to restrict the amount of damage caused.

BS 587 and BS 5000: Part 99 do not provide an exact basis for matching the overload protection requirements of the motor with the starter performance. The corresponding German standards VDE 0530 and VDE 0660 do provide a common basis to enable trip performance to be matched to motor overload capacity, basic requirements being that the trip should operate within two minutes at 1.5 full load current (FLC), within two hours at 1.2 FLC and not trip at 1.05 FLC. The approppropriate standard should be consulted for a more detailed explanation.

The method of setting trips differs between manufacturers. In general, magnetic trips and some thermal trips tend towards direct “current to trip” calibration requiring the additional excess allowable current to be added (normally 15%) before setting. Most continental designs of thermal trips and many British designs are calibrated for setting at the motor current. The correct method of setting must be determined in every case, bearing in mind that continuous maximum rating (CMR) motors are rated at full load and have no allowable overload capacity.

*Magnetic Trips*

Magnetic trips can be directly connected in the electrical supply to the motor for almost any duty. It is also possible to feed smaller trips from current transformers having a high primary current when large motors are used. They can be closely adjusted to the motor design by varying the operating characteristics of the trips. Operation times may be changed by varying the leak port on the solenoid plunger piston or by varying the viscosity of the oil in the dashpot. Time variations due to changes in oil viscosity with temperature are overcome by using a silicone oil.

Magnetic trips require regular maintenance; they require more space than thermal types and with a few exceptions tend to be expensive in the smaller sizes. The characteristics of magnetic trips are of value when dealing with high inertia loads requiring long run-up times.

*Thermal Trips*

Thermal trips are connected directly in the motor line for small motors. Current transformers are used when current values above 50 A are involved.

Thermal trips may be ambient temperature compensated. They can be designed with mechanical linkages between the trip elements in order to provide early operation under single phasing conditions. Thermal trips require little maintenance. It is also relatively easy to change the rating of most of this type by changing the elements. However, thermal trips cannot be adjusted directly to accommodate very long motor run-up times. The use of saturating type current transformers to feed the elements provides the usual solution to this problem.

A number of design variations occur between manufacturers. The more expensive trips can provide very close matching of the overload to motor protection needs. It should be noted that where a melting alloy type is used, this must be obtained for the exact motor current because no provision for adjustment is available.

*Direct Thermal Winding Protection*

Series-connected temperature-sensitive elements embedded in each phase of the motor windings can, in conjunction with an amplifier relay, provide better protection of motors against operating conditions. Examples are where frequent starting is called for or where short period overloads frequently occur. In such cases the motor windings can over-heat without excess current flowing for a sufficient time to operate most overload trips. Thermal detectors can prevent this.

Thermal protection must be determined when the motors are ordered to ensure that the thermal elements are embedded in the windings during manufacture. The use of thermal protection also involves extra wiring between the motor and the starter.

All types of trips should be selected on the manufacturer’s declared figures of motor running currents. The use of generalised tables, particularly on small motor sizes, can lead to the incorrect range of overload trips being fitted.

**Power Factor Correction**

Induction motors have an inherently poor power factor, particularly when lightly loaded, and in larger installations using motor sizes of 1 kW or more it is well worth considering the use of power factor correction capacitors.

For example, a typical 12.5 kW three phase 415 V 4-pole motor will draw a full load current of 25 A at 0.88 pf and a no-load current of 7A at 0.15 pf both power factors lagging, see Fig. B10.7.

Power factor is an indication of the relative magnitudes of the ‘in phase’ and ‘out of phase’ components of current, the latter doing no useful work and often referred to as the ‘wattless current’. As all cables, switchgear, etc., must be sized to carry the total current and as the ‘wattless current’ produces energy losses in the electrical system, Electricity Board tariffs such as the kVA maximum demand tariff are arranged so that installations with poor power factors are penalised. As a general rule, power factor correction capacitors are desirable if induction motors make up more than half the total electrical load of an installation.

Ideally, each motor should be fitted with its own correctly sized capacitor per phase connected directly to the motor terminals and mounted as close as possible to the motor. The size of capacitors should be such that the capacitive current is not more than 85% of the no-load phase current of the motor, otherwise there is a danger of resonance.

Where a large number of motors is concerned a more economical approach is the installation of bulk capacitors connected to the main electrical intake. These are usually controlled by automatic switching of banks of capacitors in and out of circuit to match varying load conditions.

Manufacturers of motors and capacitors should be consulted for each application and the following are some of the factors to be considered:

(*a*) capacitors should not be fitted where inching, plugging or direct reversal takes place;

(*b*) for multi-speed motors the capacitor should be sized for high speed running;

(*c*) momentary interruptions of supply may result in voltage peaks unless provision is made to ensure that motor speed drops below 80% of normal before the motor is switched back into circuit.

**Motor Control Switchboards and Control Panels**

The IEE Regulations for electrical installations do not cover in detail the internal design specification for motor control switchboards and control panel design. The design principles laid down for separately mounted equipment should, however, be regarded as a sound guide to good engineering practice.

BS 5486: Part 1 provides a performance specification for AC motor control switchboards for systems rated up to 660 volts. Equipment for use on high fault level systems should be designed to this standard. For most building services installations, class IFF arrangement suitable for 10 MVA fault level will be adequate. This fault level is likely to be exceeded only where switchboards have short cable connections from large transformers or where very heavy cables are used to supply a very heavy load. In such cases a proper fault level calculation should be carried out and the appropriate equipment selected.

Motor control switchboards complying with BS 5486: Part 1 may be of cubicle or multi-motor designs.

*Cubicle Panels*

Cubicle panels have individual cubicles for every electrical load, each provided with an interlocking isolator, starter and control accessories. In most instances the protective fuses will also be located within the same compartment. Cubicle panels are suitable for the large electrical loads, particularly where duty and standby equipment is employed or where it is preferable that one item of switchgear be maintained without disturbance to the remaining plant.

*Multi-motor Switchboards*

Multi-motor switchboards are provided with one main door interlock isolator, starter accessories and fuses for electrical loads together with one enclosure. This arrangement is employed where a group of motors have to operate together and where stopping all machines to attend to any one component is acceptable. This arrangement is generally unsuitable where duty and standby equipment is installed. It should be noted that economies can sometimes be achieved by using multi-motor sections within a cubicle panel.

*Combined Instrument and Control Panels*

Protected equipment control panels are designed on the basis of one or more large cubicles containing electrical and control equipment. Starters and controls are mounted in individual protected cases, generally of the component manufacturer’s standard design, and all equipment within the panel is shrouded or protected to enable control re-setting and individual starter attention with safety. The main isolating switch for panels of this type is not interlocked with the cubicle doors.

This design of panel ensures a higher degree of electrical safety and protects all equipment to a very large extent during installation, commissioning and maintenance against the harmful dust and abrasive found on building sites.

Open equipment control panels are constructed on the basis of mounting all equipment within one or more cubicles, all of which are interlocked with the main panel isolating switch. Starters and control components may be mounted within the panel without a further protection. or enclosure. This arrangement provides panels at lower cost and occupying smaller space than other forms of panel construction.

Open equipment type control panels have the disadvantage that during installation electrical and control equipment is exposed to site dust and abrasive frequently causing increased commissioning difficulties. Further, it is necessary to switch off all equipment associated with the panel during the adjustment or maintenance of any component inside. The disturbances caused to control equipment by switching off the panel can sometimes take a considerable time to readjust.

*Grouping of Switchgear and Control Equipment*
In most installations it is desirable that the control equipment and motor control switchgear should be grouped in close proximity to the control plant. Where electrical loadings are high or where a large number of electrical drives are involved a panel should be made for electrical switchgear and the instruments and control equipment mounted in separate panels of appropriate design.

## STANDBY GENERATORS

Standby generators supply electricity to the whole or part of a load in the event of mains failure.
The standby unit may also be used for:
(*a*) "Peak lopping" (reduces the maximum electrical demand hence electricity charges).
(*b*) "Base loading" (supplies the continuous part of the load).
These systems increase the maintenance required and additional sets are needed for continuity of supply during maintenance periods.

### Parallel Operation
"Peak lopping" and "base loading" will probably involve parallel operation of the alternators and in some cases of the alternator and the Electricity Board supply. The advantages of parallel operation are:
(*a*) optimum loading of the sets;
(*b*) flexibility;
(*c*) utilisation of the load diversity.
However, the following additional protection and control equipment is required:
(*a*) reverse power relays;
(*b*) synchronising equipment;
(*c*) equipment to ensure sharing of power and reactive load components.

### Types of Prime Mover
The prime mover may be one of several types as given in Table B10.9. Usually the most economical engine for unit size < 1MW is the reciprocating engine.

**Table B10.9.** Types of prime mover.

| Engine Type | Fuel | Approx. Range |
|---|---|---|
| Reciprocating | Petrol<br>Diesel<br>Diesel or Gas* | 0.5-15 kW<br>0.5- 5 MW |
| Gas Turbine | Oil<br>Kerosene<br>Gas* | 50 kW-40 MW |
| Steam Turbine | Boiler fuel<br>Waste process heat | Over 200 kW |

*British Gas will not allow the use of gas for standby only.

*Heat Emission from Reciprocating Engine*
Typical heat emissions of reciprocating engines are:
(*a*) From exhaust: 1 kW per kW of engine output.
(*b*) From coolant: 0.7 kW per kW of engine output.
(*c*) Radiated: 0.15 kW per kW of engine output.
Part of the heat emitted can be recovered from the exhaust gases (650°C max) and the jacket water (80°C max) increasing the overall efficiency from 30/40% to possibly 70%.

*Rating*
BS 5514 gives standard operating conditions with the appropriate derating factors.

*Governor*
The speed of the engine and hence the frequency of electricity supplied by the set is controlled by the engine governor. Special conditions may exist i.e. computer load, heavy motor starting duties, where the governor/generator performance has to be particularly selected for the load requirement; this is class Al governing. The classes of governing are given in BS 5514: Part 4.

### Sizing of Generators
The sizing of generators should take into account the transient as well as the steady conditions.

The power required to start a motor can be as much as 4× (full load kW) therefore where motors are a large part of the load the sizing of the generator depends on the motor/s starting characteristics.

Large motors should preferably be sequenced to start first, off load when possible.

### Air Requirement
Air is required for aspiration and cooling, the amounts required vary considerably with the various sizes and types of prime mover. That is, for water radiator cooled diesel engines sizes 15 to 500 kW the amount of air required varies between 20 and 45 $dm^3/s$ per kW of engine rating.

### Noise
Noise is emitted from:
(*a*) the engine moving parts;
(*b*) the exhaust;
(*c*) the radiator.
The engine is frequently installed in a separate housing (prefabricated housings are available) to limit the noise transference. The walls should be substantial (200 mm) and may be fitted with sound absorbing tiles.

The exhaust is fitted with a silencer and standard low noise ("residential") silencers are available for most sets.

Airborne noise from the engine moving parts and radiator can be attenuated by placing baffles of sound absorbing material in the intake and extract airflow.

The methods used for noise reduction also reduce heat transference from the surroundings of the generator set and may result in derating the set.

## TRANSFORMER CONNECTIONS

Power transformers are covered by BS 171.

Transformers are used to reduce the high voltage of power distribution systems to voltages suitable for use by the consumer. The most common consumer voltages are 415 volts three phase and 240 volts single phase.

High voltage power distribution systems are normally three phase, i.e. there is no neutral, so that the high voltage or primary winding of the transformer is normally connected in Delta. The low voltage or secondary winding could be connected in delta if only three phase supplies are required but if single phase or three phase and neutral supplies are needed then the secondary windings would be connected in star in order to establish a neutral point.

The most common transformer connection is Delta/Star and this is shown diagrammatically in Fig. B10.3. This connection is designated by the code reference Dy.

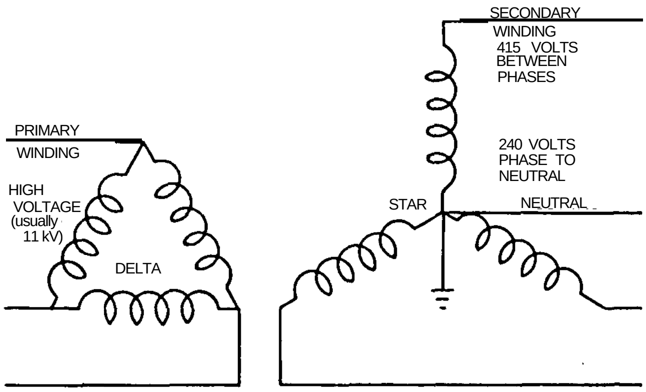



**Fig. B10.3.** Diagrammatic of Delta/Star connection.

There are two different ways of arranging Dy connections relative to the phase rotation of the supply, one giving a 30° lag phase displacement between primary and secondary voltages and the other giving a 30° lead. The first is coded Dy 1 and the second Dy 11.

In general the possible transformer connections are classified into 4 groups as follows:

- Group 1 Zero phase displacement (Codes Dd0, Yy0, Dz0)
- Group 2 180° phase displacement (Dd6, Yy6, Dz6)
- Group 3 30° lag phase displacement (Dy1, Yd1, Yz1)
- Group 4 30° lead phase displacement (Dy11, Ydl1, Yz11).

The zig-zag connection, designated Z, is designed to reduce third harmonic voltages caused by unbalanced loading but it requires 15% more turns than the normal phase connection and for that reason is not very common.

## METERS

In order to measure the power consumed by an electrical device operating on alternating current it is necessary to determine volts, amps and power factor and then to apply the relationship:

$$\text{True Power} = \text{Volts} \times \text{Amps} \times \text{Power Factor} \qquad \text{B10.4}$$

The power factor compensates for the fact that volts and amps are not in phase with each other. It is the product of the in-phase components which gives true power.

A wattmeter is an instrument which employs two electromagnetic coils operating on a disc or drum to produce a torque. One coil carries a current proportional to the voltage while the other carries the load current, and the torque they produce on the disc or drum is proportional to the product of the in-phase components of voltage and current, i.e. the wattmeter measures true power.

To measure the power consumed by a single phase load one wattmeter is used. For 3 phase loads with no neutral connection two wattmeters are required and for unbalanced loads with neutral connections three wattmeters are necessary.

The three methods of connection are shown in Fig. B10.4 (a), (b) and (c).

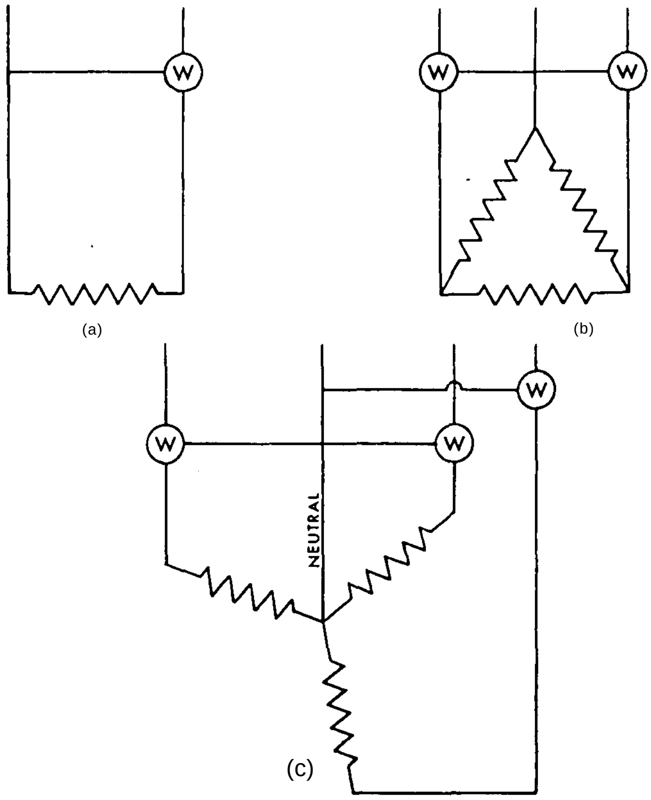



**Fig. B10.4.** (a) one wattmeter measuring single phase power
(b) two wattmeters measuring 3 phase power
(c) three wattmeters measuring unbalanced 3-phase power with neutral

If voltage, current and power consumption are measured separately then power factor can be easily calculated although this method does not indicate whether it is lagging or leading. Alternatively, use may be made of a power factor meter which does show whether the power factor is lagging or leading.

## RADIO AND TV INTERFERENCE AND SUPPRESSION

Electrical interference can be of three kinds:

(*a*) radio frequency airborne;
(*b*) radio frequency mains borne;
(*c*) audio frequency.

(*a*) and (*b*) are due to the fact that electric oscillations are established in every circuit of less than critical damping when electrons receive acceleration: the frequency of the oscillation is determined by the circuit constants.

Radio frequency oscillations of a heavily damped nature are established by the current variations caused by such things as: switches, thyrister controls, motor commutation, rectifiers, telecommunication equipment, welding plant and spark ignition systems. Owing to the heavy damping the interference is not confined to a single frequency, but is distributed to an extent over the entire spectrum, as with atmospheric interference which is caused by electro magnetic waves of natural origin.

Some of the radio frequency energy is carried to the vicinity of the receiving aerial by true radiation and where mains cables connect the place of origin with the receiving station, the latter is also subject to mains-borne interference. This is picked up by the aerial circuit of the receiver in the ordinary way.

The third type of interference (audio frequency), is caused by direct radiation from the mains into the audio-frequency circuits of the receiver. Varying types of suppression devices are used to stop interference. A simple example is shown in Fig. B10.5 where the suppression devices are the 2µF and 4µF capacitors and the 600 µH inductor.

It is usual for items of equipment which can cause interference to be suppressed at source.

In some cases where sensitive medical or computer equipment is being used which is subject to external interference, the only solution is complete electromagnetic screening of the room or the sensitive equipment within the room.

BS 613 specifies the components and filter units for electromagnetic interference suppression. BS 800 specifies radio interference limits and measurements for household appliances, portable tools and other electrical equipment causing similar types of interference.

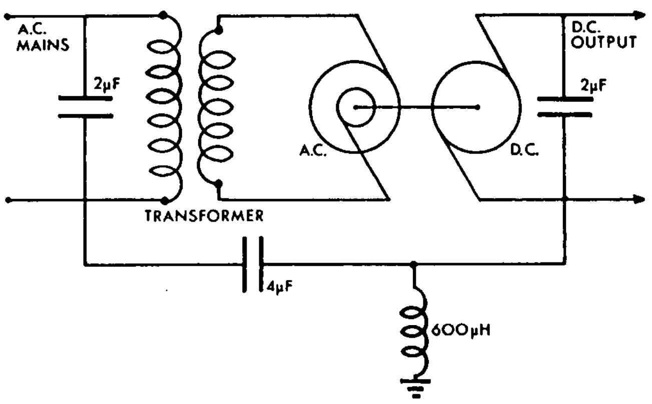



**Fig. B10.5.** Interference suppression circuit.

## THREE-PHASE SYSTEM RELATIONSHIPS

These relationships are given by Fig. B10.6 and by:

$$P = I_L \times E_L \times \cos\phi \times \sqrt{3} \quad \ldots \quad \ldots \quad \text{B10.5}$$

where :

| | | | |
|---|---|---|---|
| $P$ | = | power | W |
| $I_L$ | = | line current | A |
| $E_L$ | = | line voltage | V |
| $\cos\phi$ | = | power factor | |

### Vector and Circle Diagram

Vector diagrams are an extremely useful aid to understanding the relationships between voltage, current and power factor in alternating current circuits. By drawing the vector diagram accurately to scale, voltages and currents can be added or subtracted graphically and power factors can be measured.

The locus of the tip of the current vector for an induction motor follows a circular path as the load is varied from zero to maximum and gives rise to the circle diagram. Conventionally, the applied voltage vector is drawn vertically and the circle diagram then takes the form shown in Fig. B10.7.

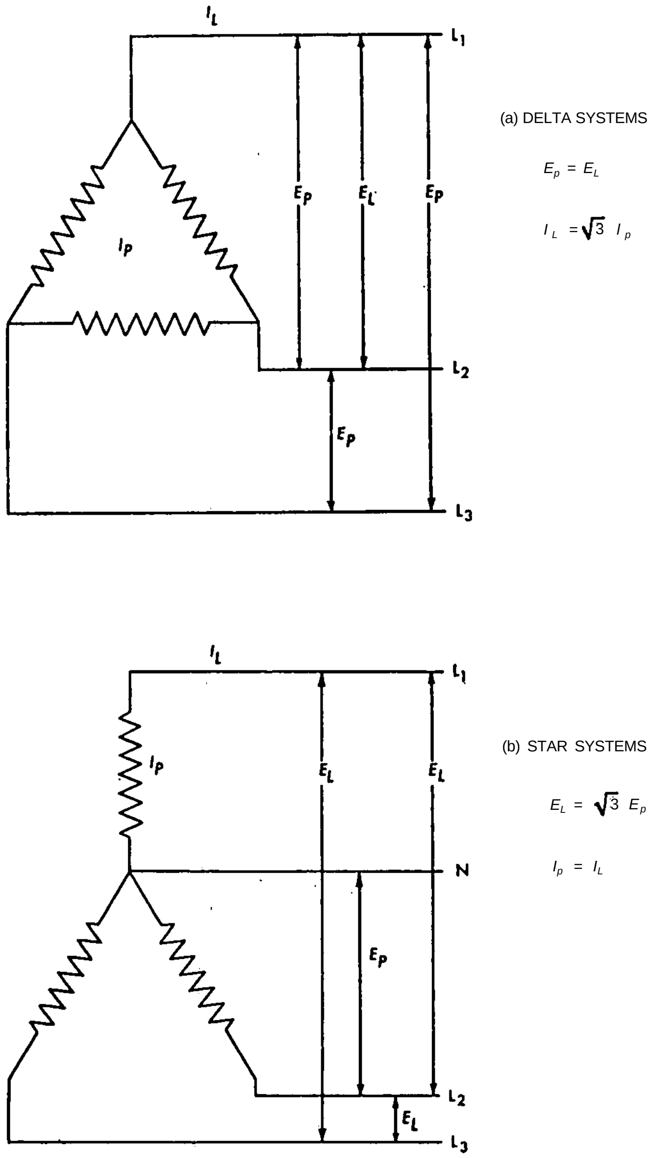



**Fig. B10.6.** Voltage and current relationships.

Developing the circle diagram a little further provides additional information as shown in Fig. B10.7 which is drawn to a scale of 1 mm = 1 Amp and is typical for a three phase 415 volt 25 rev/s 12.5 kW induction motor. The phase voltage lies on the axis OE and the current vectors are drawn relative to this axis. OF is the full load current, OS the starting current and ON the no load current. These currents all lag the voltage by different angles and in each case the power factor is the cosine of the angle of lag, e.g. at full load the power factor is $\cos\phi_f$.

If the diagram is drawn to scale, then all lines parallel to OE represent the "true" component of the associated current and as the phase voltage, number of phases and synchronous speed of any particular motor are all constant, the following information is immediately obtainable from the circle diagram.

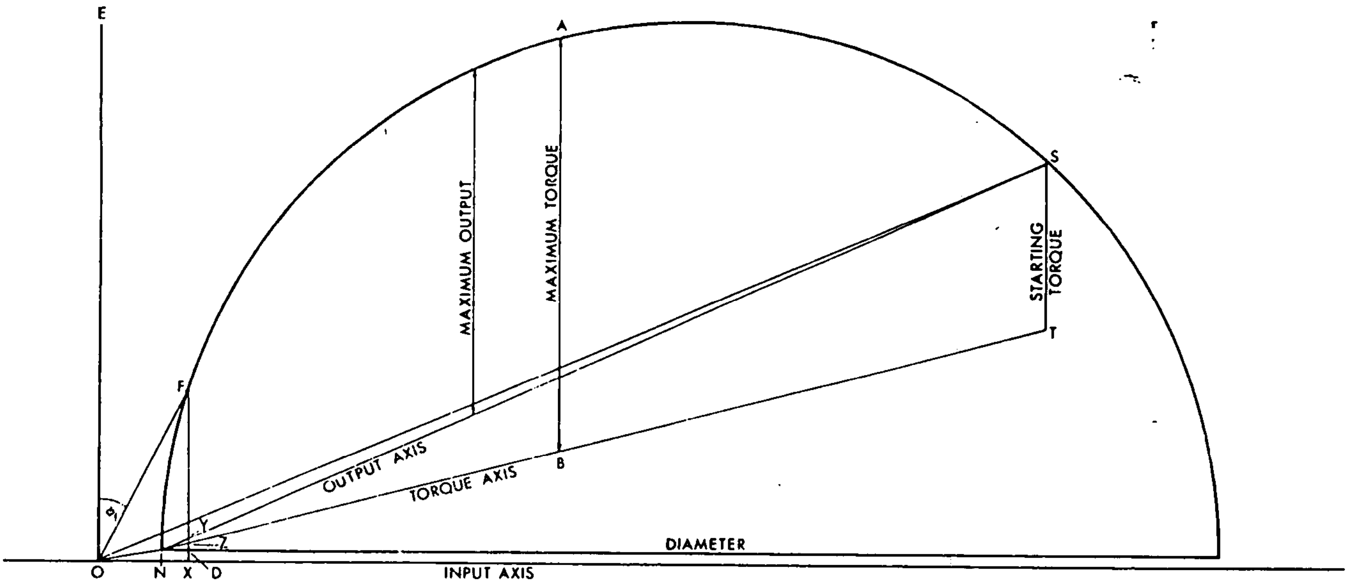



**Fig. B10.7.** Circle diagram.

$$P_{tf} = FX \times E_p \times N \qquad \text{B10.6}$$

$$P_{of} = FY \times E_p \times N \qquad \text{B10.7}$$

$$T_f = \frac{FZ \times E_p \times N}{2\pi \times R} \qquad \text{B10.8}$$

$$T_s = \frac{ST \times E_p \times N}{2\pi \times R} \qquad \text{B10.9}$$

$$T_m = \frac{AB \times E_p \times N}{2\pi \times R} \qquad \text{B10.10}$$

$$\cos\phi_f = \frac{FX}{OF} \qquad \text{B10.11}$$

$$\cos\phi_n = \frac{XD}{ON} \qquad \text{B10.12}$$

$$h_f = \frac{FY}{FX} \times 100 \qquad \text{B10.13}$$

where:

| | | |
|---|---|---|
| $P_{tf}$ | = power input at full load | W |
| $E_p$ | = phase voltage | V |
| $N$ | = number of phases | |
| $P_{of}$ | = output power at full load | W |
| $T_f$ | = torque at full load | N m |
| $R$ | = speed | rev/s |
| $T_8$ | = starting torque | N m |
| $T_m$ | = maximum torque | N m |
| $\cos\phi_f$ | = power factor at full load | |
| $\cos\phi_n$ | = power factor at no load | |
| $h_f$ | = efficiency at full load | % |

Increasing the rotor resistance moves the point S further round the circle towards F which increases the starting torque but decreases the efficiency. Reducing the phase voltage reduces the size of the circle in direct proportion to the voltage reduction, i.e. if the phase voltage is reduced by half, the diameter of the circle will be halved also and all the current vectors will be reduced accordingly. This means that, for example, the starting torque will be reduced to one quarter of its previous value, since it is the product of the current vector and the phase voltage, both of which quantities have been halved.

An understanding of the simple circle diagram shown enables the effects of different methods of starting to be easily assessed in terms of torque and maximum power output. It thereby illustrates the dangers of using, for example, a reduced voltage method of starting for a large fan where the torque and output required to accelerate the fan may be more than the motor is capable of producing at reduced voltage, even though the full load rating of the motor is more than adequate.

## WIRING DIAGRAMS

### Need for Wiring Diagrams

Wiring diagrams are essential working drawings.

The discipline of producing them is important for the following reasons:

(1) In order to produce the diagram all relevant information on components and plant must be assembled and matched together. This exercise provides a useful check on the suitability of the equipment ordered and assists the engineer to determine the correct operation of the entire system.

(2) A comprehensive wiring diagram will tell the electrical contractor how to connect the equipment. Separate manufacturers' drawings, often of differing styles with notes for different connections, will not do this and may result in damage to the equipment or a loss of time in finding the source of any malfunction.

(3) A comprehensive diagram is essential as a record for the final installation. A proper drawing will assist the commissioning and maintenance of the electrical installation.

### Forms of Wiring Diagram

The design engineer is likely to meet with many forms of diagram owing to the various requirements of associated industries and of the different countries from which equipment is obtained. British Standard symbols which are available for telecommunication and power installations omit proper reference to other types of circuit requirement. The following notes indicate the conventions and the forms of diagram most suited to the needs of the building services industry.

*Block Diagrams* (see Fig. B10.8)

It is frequently necessary to provide information for the electrical contractor to enable him to estimate wiring costs, but it is unlikely that either the detailed circuit information or time will be available to enable a wiring diagram (Fig. B10.10) to be produced. A block diagram will normally communicate this information in the simplest possible manner. It will show:

(*a*) Electrical equipment and associated loads so that cables and distribution systems may be sized.

(*b*) The number of connections between each item of plant.

(*c*) By cross reference to the physical layout drawings, the location of each item of plant and control equipment.

(*d*) The equipment identification which should indicate duty and standby equipment in order that maximum simultaneous demands can be determined.

The block diagram uses only simple symbols for motors and starters together with control equipment. It will indicate a connection by a line which will carry a reference indicating the numbers of conductors. The load and identification values may be added to the circuit symbols.

Provided the information given on such a drawing is accurate the electrical contractor should not require further detailed wiring diagrams until the equipment is on site ready for wiring.

*Key Diagrams* (see Fig. B10.9)

The key diagram, also referred to as a line or schematic diagram, may be found in two stages of development, viz.:

(*a*) A simple drawing indicating how the equipment is to operate.

(*b*) A similar drawing which forms a record of the final installed circuit. This drawing will show all the intermediate terminal numbers, component identification and possibly times and temperatures set on thermostats and delay relays, etc.

The key diagram is drawn to a convention which allows components to be drawn in the most convenient position without regard to their physical situation.

All components should be shown in the *de-energized* position. Some confusion exists over the illustration of thermostats and pressure contacts. Thermostats are generally shown in the *satisfied* condition whilst pressure switches tend to be shown in the *unsatisfied* operating condition.

*Installation Wiring Diagram* (see Fig. B10.10)

The purpose of the installation diagram is to show the electrical contractor how the plant is to be connected. The drawing must be clear and show all electrical loads, terminal numbers, cross references and notes defining the items of our supply and those which the electrical contractor is to provide. One example would be the provision of local isolating switches. It is not essential to show internal equipment connections provided the makers' drawing number is given in order that the complete circuit can be traced if necessary. Where a number of similar items, such as starters occur, it is good practice to illustrate one starter of each type in full detail. Simplicity should be the basis for layout of this type of drawing, and it is an unnecessary complication to try to show equipment items in any physical relationship.

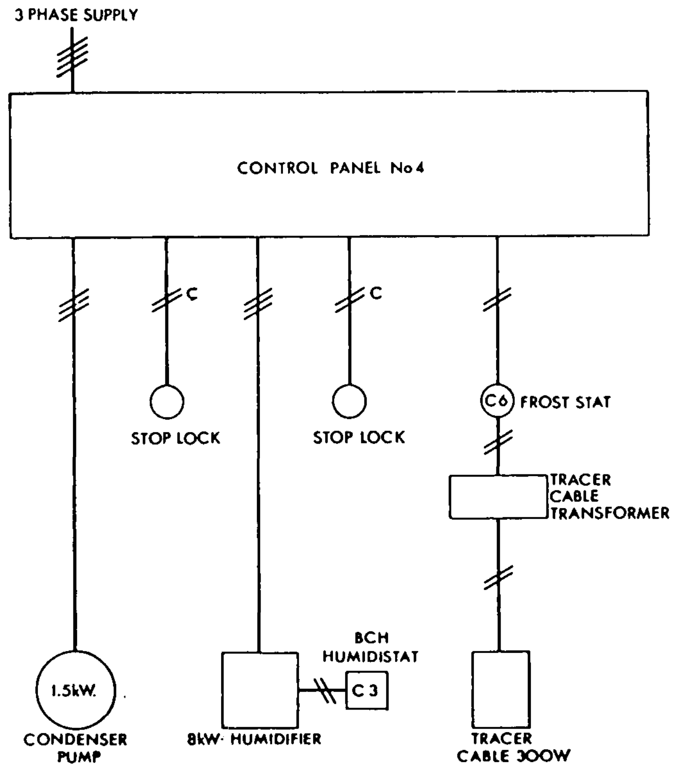



Fig. B10.8. Typical block diagram.

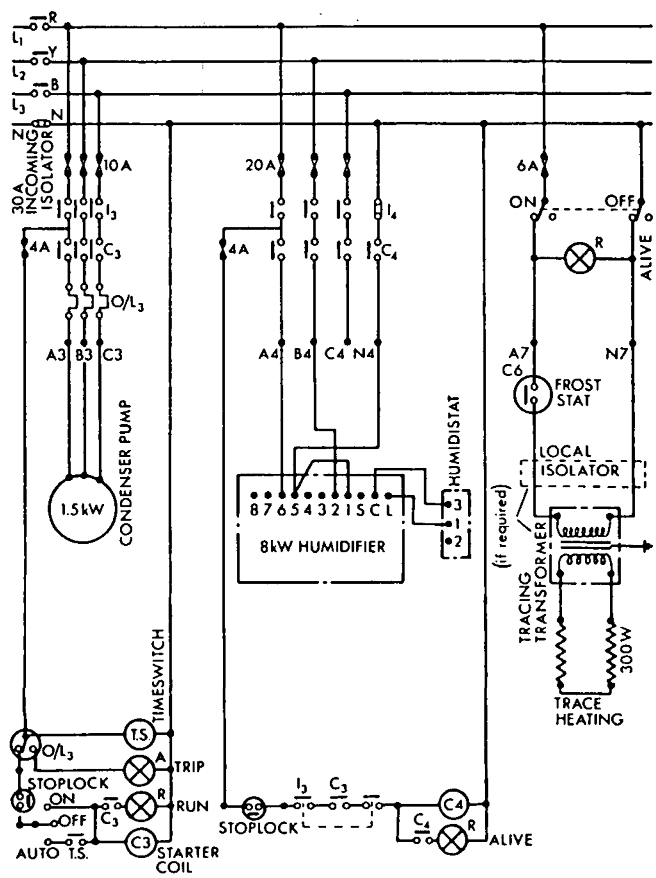



Fig. B10.9. Typical key diagram.

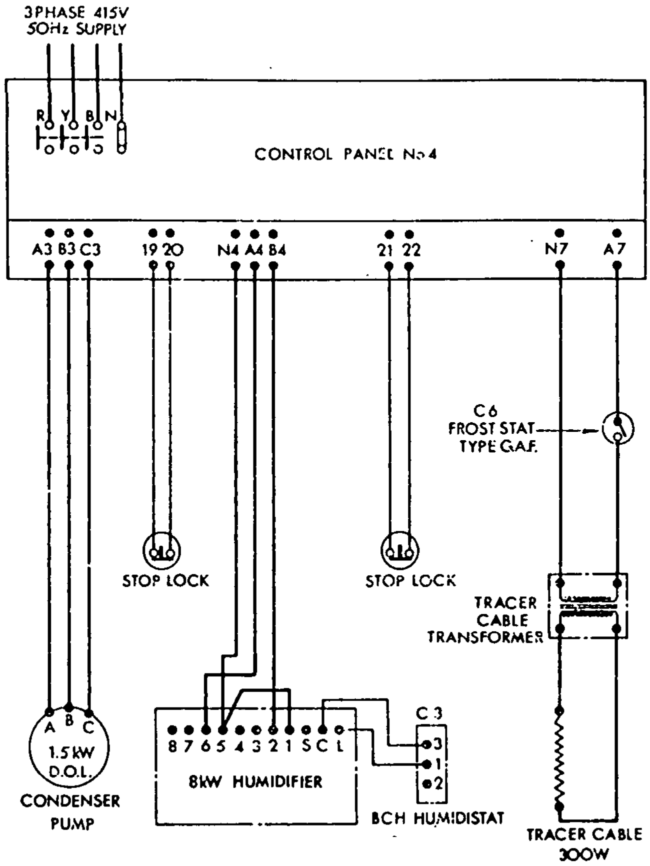



**Fig. B10.10.** Typical installation diagram.

*Panel Wiring Diagrams*

Equipment manufacturers require, for production reasons, drawings which will illustrate components in their physically correct relative positions. Such a drawing is very complicated to read, due to the devious paths of many connections between components. A number of conventions exist which simplify the production of such drawings and these must be understood before the drawings can be of any great value. The conventions tend to make reading of drawings very difficult and when it is required to follow the operation of any particular circuit the key diagram is required. These are generally available because the initial circuit design is determined by means of such a diagram.

**Circuit Design**

Wherever control connections are taken from a starter it is recommended that these should be at phase to neutral potential. For pressurization equipment, boilers or refrigeration control applications this form of circuit arrangement is essential.

It is also worth noting that many thermostats and control devices are only suitable for phase to neutral (240 V) connection.

When producing circuit diagrams the problem of isolating equipment, in order to comply with regulations and to ensure safety of operation, must be regarded as extremely important. All live connections entering a starter or other item of plant must be completely isolated. For small loads, simple isolators may be used. When considering larger loads or equipment having a very large number of connections entering it, the use of control circuit isolators can be a satisfactory solution. An example of this can be seen with the local isolation of a motor connected to a star-delta starter. To isolate the motor six incoming connections must be switched and in addition the control circuits to the starter will have to be passed through a seventh contact. Without this provision of control circuits switching the motor would be connected to start on delta immediately the isolator was closed. The use of a single isolator in the control circuit is capable of ensuring that the machine is dead and at the same time ensures correct starting sequence. Link type terminals are now available for the isolation of relay and other centralized equipment receiving a number of live inputs.

Fusing must be included in control circuits when the size of the cable leaving control equipment is smaller than the preceding fuse will protect. For example, if a starter were fused for a 75 kW motor, the control circuit would require a separate fuse in order to protect the small cables leaving the starter for control purposes. If one fuse is used to protect a large control system, a small fault may render the complete system inoperable.

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# SECTION B11 AUTOMATIC CONTROLS




*This edition of Section B11 first published: 1986*

*(Abridgement of CIBSE Applications Manual, "Automatic Controls and their Implications for Systems Design".)*

# SECTION B11 AUTOMATIC CONTROLS

## INTRODUCTION

Control requirements must be considered from the very inception of the design. Failure to do this may result in an uncontrollable system.

The term 'controls' in the design of building services, and particularly HVAC systems, is often restricted to the automatic thermostatic controls. However, in modern buildings it applies far more widely and the design process must consider the interface of these controls with such elements as lighting, fire/smoke alarms, security, etc.

It is normal practice to adopt dry resultant temperature as the standard parameter for design temperature. However, in terms of automatic controls, temperature sensors provided by the manufacturers are still mainly related to dry-bulb air temperature scales and this Section uses dry-bulb temperature settings, unless otherwise specified, for consistency with the controls industry. Sections A1, A5 and A9 of the *Guide* deal with dry resultant temperature in more detail.

The selection of a control system which maintains the conditions of the measured variables within the specified tolerances can have a very profound and advantageous effect on the energy consumption within a building and this benefit should be included in the economic evaluation of the systems considered. Where energy consumption can be further reduced by optimising the use of major plant, consideration should be given to the inclusion of optimising start controllers and building automation systems.

The design process begins by setting out a brief based on the client's requirements. This results in a series of criteria related to the mechanical services and lighting design. These criteria must include an indication of the required mean conditions, together with the extreme values and permissible tolerances. As soon as the form of system is agreed a schematic diagram should be prepared.

One important area associated with control systems which requires emphasis is that of testing and commissioning (T&C) and it is essential that suitable facilities be designed into the overall system to permit the efficient T&C of the controls. Such facilities include the necessary regulating valves and dampers, ductwork and pipework tappings adjacent to each sensor and suitable permanent and temporary instrumentation for measurement purposes. The positioning of these items and access thereto is of prime importance.

A more detailed approach to the use and application of controls is given in the CIBSE Applications Manual *Automatic Controls and their Implications for Systems Design.*

## DEFINITIONS

BS 1523 : Part 1 : 1967 covers basic definitions for 'Process and Kinetic Control'. It does not however accord with all the terminology normally used in heating, ventilating and air conditioning or the building services control field. The following definitions are those in common usage within both sectors and relate to the content of this *Section.*

*Boost:*
This is a plant start up condition, normally after a prolonged (overnight or longer) shutdown. It is common during the boost cycle to run the plant so as to provide maximum heating (or cooling), consistent with any safety requirements. In this mode maximum permitted temperatures or flows are maintained, by over-riding the normal controls, until the desired internal conditions are achieved. At this point, boost termination occurs and the system is switched to normal control.

*Control Mode:*
See *Controller Action.*

*Controlled Condition:*
The physical quantity or condition of the controlled process, machine, or environment which it is the purpose of the system to control. Also referred to as the *Controlled Variable.*

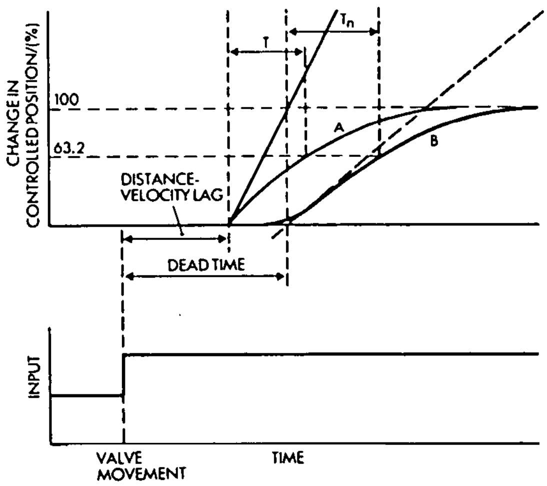
*Controlled Variable:*
See *Controlled Condition.*

*Controller:*
A piece of equipment used in control systems which responds directly or indirectly to deviation from the controlled condition by initiating a control sequence, or actuating a control element or final control element. See also *Correcting Unit.*

*Controller Action:*
A term describing the relationship between the deviation from the controlled condition and the change of output signal from the controller. Also referred to as the *Control Mode.*

*Correcting Unit:*
The single unit combining the actuator element and correcting element in a process control system. Also referred to as the *Final Control Element.*

*Dead Time:*
The time interval between a change in a signal and the initiation of a perceptible response to that change, see Fig. B11.1.




A: Calorifier coil temperature.
B: Calorifier secondary temperature.

**Fig. B11.1.** Transfer functions of a typical plant.

*Dead Zone:*
The zone within which a change of value of an input signal (e.g. controlled condition, set value, etc.) to an element or system, may take place without causing any perceptible change in output signal.

*Derivative Action:*
The action of a controller whose output signal is proportional to the rate at which deviation from the set point occurs.

*Desired Value (D.V.):*
The design value of the controlled condition, which normally determines set point necessary to achieve the condition controlled condition or controlled variable. See also *Set Point* and *Set Value.*

*Deviation:*
The difference between the set point and actual controlled condition. See also *Offset.*

*Differential:*
There are three types of differential:

(*a*) Controlled condition differential is the difference between the upper and lower limits of the controlled condition resulting from some types of control action. See also *Proportional Action.*
(*b*) Manual differential is the difference between the cut in and cut out points on a scale setting determined by raising and lowering the thermostat setting (with no electrical load). (It is useful for a quick check by servicemen).
(*c*) Switch differential is the minimum sustained change in the measured condition to produce a change in the state of the contacts, as determined only by a specified test method.

*Differential Temperature Controller:*
A differential temperature controller is a device which is able to detect a temperature difference between two bodies by processing signals derived from sensors in thermal contact with those bodies. It is able, with reference both to the sign and magnitude of that temperature difference, to operate one or more final control elements.

*Direct Digital Control (DDC):*
The use of digital software programs to provide various *Control Modes.* The term is normally used where thermostatic or similar process loops are directly controlled from a building automation system.

*Discontinuous Action:*
The action of an element, or control system whose output signal is a discontinuous function of its input signal. The discontinuous action may be of a variety of forms such as:

(*a*) On-Off.
(*b*) On-On. Changeover action.
(*c*) On-Off-On. Changeover with centre off.
(*d*) Multi-step, with the steps all in one sense or in both senses and with or without the ‘off’ condition.

*Distance-velocity Lag:*
The time between a signal being sent to an element and the element starting to respond, arising from the finite speed of propagation of the signal, see Fig. B11.1.

*Exponential Lag:*
An exponential lag occurs when the change with time in the output from an element or system, resulting from the application of a step change in the input signal to that element or system is of simple exponential form. In heating and air-conditioning applications a step change in input signal to a plant rarely produces a pure exponential characteristic at the plant output, partly because of distance-velocity lag and partly because the signal is generally passed through a series of stages. For example, if the position of a three-port valve, feeding a calorifier is changed, in order to change the temperature of the water flowing on the secondary side, the following time lags are involved– relating to Fig. B11.1 which illustrates the general shape of the transfer functions of such a plant:

(*a*) Distance-velocity lag, being the time taken for the hotter water leaving the mixing valve to reach the coil.
(*b*) Time lag which occurs in raising the temperature of the coil.
(*c*) Time lag which occurs in raising the temperature of the water in the calorifier.

The time lag described in (*b*) above is associated with a temperature/time relationship which, to all intents and purposes, is a pure exponential (curve A). The temperature/time relationship in (*c*) is not a pure exponential (curve B) because its shape is dependent upon the form of the preceding temperature/time relationship. Curve A has a time constant of $T$. In curve B, the time constant is not valid in the sense of the simple exponential mathematical expression but a notional time constant of $T_n$, as illustrated, is sometimes employed if control equipment and plant are to be analysed.

*Feedback:*
The transmission of a signal from a later to an earlier stage of a system. Where used, the term should be qualified, e.g.

(*a*) Process feedback is a measure of the result of the control action feedback by the detecting element to the controller.
(*b*) Position feedback is a measure of the amount of the control action feedback directly to the controller, e.g. the potentiometer on a valve actuator.
(*c*) Integral feedback is a measure of the effective speed of correcting (control) action feedback from within the controller to the comparing element.

*Final Control Element:*
See *Correcting Unit.*

*Floating Action:*
The action of a control element whose output signal changes at a predetermined rate when the input signal exceeds one threshold value or drops below another, higher, threshold value. Also referred to as *Floating Mode.*

*Floating Mode:*
See *Floating Action.*

*Integral Action:*
The action of a controller whose output signal changes at a rate which is proportional to the deviation from the set point.

*Offset:*
Sustained deviation from the set point.

*On-Off Action:*
A special case of two-step discontinuous action in which one of the output signals is zero.

*Proportional Action:*
The action of a controller whose output signal is proportional to the deviation from the set point.

*Proportional Band:*
That range of values of deviation of the controlled variable corresponding to the full operating range of output signals of the controller resulting from proportional action only. The proportional band can also be expressed as a percentage of the range of values of the controlled condition which the measuring unit of the controller is designed to measure. Also referred to as *Throttling Range.*

*Rate Reset:*
A term used in pneumatic controls terminology which is analogous to *Derivative Action.*

*Regulating Valve (RV):*
An adjustable restrictor (normally hand set) which acts directly on the one medium with which it is associated, e.g. characterised hand valve.

*Reset:*
A term used in pneumatic controls terminology which is analogous to *Integral Action.*

*Set Point:*
The value to which a control device or controller must be pre-set in order to achieve the desired value of the controlled variable.

*Set Value:*
See *Set Point.*

*Systems:*
A system is the co-ordination and integration of components collected together in a way that enables them to perform some specified sequence and/or series of functions. When a system moves from one state to another it does so in a way that is characteristic of the system.

*Throttling Range:*
See *Proportional Band.*

*Time Constant:*
This time constant ($T$ in Fig. B11.1) relates to the transfer functions within a plant subject to load changes. It represents the time taken for exponential change in the controlled variable to reach 63.2% of its final condition.

*Two-position Controller:*
A controller whose output signal changes from one predetermined value to another when the value of the controlled variable crosses predetermined threshold levels, e.g. a thermostat.

## NOTATION

| | | | |
|---|---|---|---|
| $A$ | = | cross sectional area of duct .. .. | $m^2$ |
| $A_v$ | = | valve coefficient (metric units) .. | – |
| $C$ | = | Constant for given pressure conditions and type of orifice .. .. .. | – |
| $C_l$ | = | design capacity (load) .. .. .. | $m^3$ |
| $C_s$ | = | design capacity (source) .. .. | $m^3$ |
| $C_v$ | = | valve coefficient (imperial units) .. | – |
| $F_a$ | = | ratio of actual air flow at partially open position to air flow at fully open position .. .. .. .. .. | – |
| $F_i$ | = | ratio of theoretical air flow at partially open position to air flow at fully open position (inherent characteristic) .. | – |
| $K_m$ | = | valve recovery coefficient .. .. | Pa |
| $M$ | = | mass flow of steam .. .. .. | kg/s |
| $N$ | = | valve/damper authority .. .. | – |
| $P_d$ | = | downstream steam pressure .. .. | Pa |
| $P_e$ | = | pressure on exhaust side of return air damper .. .. .. .. .. | Pa |
| $P_f$ | = | pressure drop across fitting .. .. | Pa |
| $P_i$ | = | pressure drop across valve (imperial units) .. .. .. .. .. | psi |
| $P_{in}$ | = | absolute inlet pressure .. .. .. | kPa |
| $P_m$ | = | pressure drop across valve (metric units) .. .. .. .. .. | Pa |
| $P_{max}$ | = | maximum allowable pressure drop (valve fully open) .. .. .. | kPa |
| $P_s$ | = | pressure on supply side of return air damper .. .. .. .. .. | Pa |
| $P_t$ | = | total pressure drop .. .. .. | Pa |

| | | | |
|---|---|---|---|
| $P_u$ | = | upstream steam pressure | Pa |
| $P_v$ | = | vapour pressure at inlet temperature | kPa |
| $P_1$ | = | pressure drop across valve/damper (fully open) | Pa |
| $P_2$ | = | pressure drop across remainder of circuit | Pa |
| $R$ | = | rangeability | – |
| $T$ | = | time constant | s |
| $T_n$ | = | notional time constant | s |
| $V$ | = | volume flow rate | $m^3/s$ |
| $V_b$ | = | volume flow rate through bypass | $m^3/s$ |
| $V_d$ | = | volume flow rate to load at design load | $m^3/s$ |
| $V_i$ | = | volume flow rate (imperial units) | gal/min |
| $V_m$ | = | volume flow rate (metric units) | $m^3/s$ |
| $V_p$ | = | volume flow rate to load at partial load | $m^3/s$ |
| $V_s$ | = | volume flow rate through source | $m^3/s$ |
| $V_0$ | = | volume flow rate at zero stroke if valve was characterised to this point | $m^3/s$ |
| $V_{100}$ | = | volume flow rate with valve fully open | $m^3/s$ |
| $a$ | = | area of orifice | $m^2$ |
| $d$ | = | diameter of pipe | m |
| $f_{max}$ | = | maximum flow rate | $m^3/s$ |
| $f_{min}$ | = | minimum controllable flow rate | $m^3/s$ |
| $k_d$ | = | derivative control action constant | – |
| $k_f$ | = | floating control action constant | – |
| $k_n$ | = | integral control action constant | – |
| $k_p$ | = | proportional control action constant | – |
| $l_e$ | = | equivalent length of pipe | m |
| $s$ | = | change in valve stroke | % |
| $s_f$ | = | relative density (specific gravity) of fluid | – |
| $t_f$ | = | flow water temperature | °C |
| $t_{fd}$ | = | flow water temperature at design load | °C |
| $t_{fp}$ | = | flow water temperature at partial load | °C |
| $t_m$ | = | mean water temperature | °C |
| $t_{md}$ | = | mean water temperature at design load | °C |
| $t_{mp}$ | = | mean water temperature at partial load | °C |
| $t_r$ | = | return water temperature | °C |
| $t_{rd}$ | = | return water temperature at design load | °C |
| $t_{rt}$ | = | design return water temperature (load) | °C |
| $t_{rp}$ | = | return water temperature at partial load | °C |
| $t_{rs}$ | = | design return water temperature (source) | °C |
| $v$ | = | velocity | m/s |
| $b_i$ | = | input signal | – |
| $b_o$ | = | output signal | – |
| $DP_l$ | = | pressure drop per unit length of pipe | Pa/m |
| $Dt_d$ | = | temperature difference between space condition (20°C) and mean water temperature at design load | °C |
| $Dt_l$ | = | design temperature difference across load | °C |
| $Dt_p$ | = | temperature difference between space condition (20°C) and mean water temperature at partial load | °C |
| $Dt_s$ | = | design temperature difference across source | °C |
| V | = | pressure loss factor | |
| $q$ | = | time | s |
| $q_0$ | = | time for which valve is outside neutral zone | s |
| $r$ | = | density | $kg/m^3$ |
| $r_{f/m}$ | = | density of fluid (metric units) | $kg/m^3$ |
| $f$ | = | heat output | W |
| $f_d$ | = | maximum design heat output | W |
| $f_p$ | = | partial load heat requirement | W |

## CRITERIA FOR CONTROLS

When choosing control equipment, the following should be considered:

(*a*) Why are controls necessary and what will they achieve?
(*b*) Which system of control can meet the requirements?
(*c*) What are the economic constraints for a control system?
(*d*) Can a control system be selected in isolation?

The initial capital cost of the controls may not be the most significant figure in determining the owning and operating costs over the life of the building. Selection must be made on the basis that the system will maintain the desired conditions within the permitted tolerances under all load conditions.

However good the plant, it will be unsatisfactory if the controls cannot operate it correctly. Similarly, however good the controls, the conditions will be unsatisfactory if the plant, hydraulics or air distribution are incapable of providing the desired results.

### Necessity for Controls

Controls are required for one or more of the following reasons:

(*a*) To ensure safe operation of plant.
(*b*) Where manual control is inadequate to maintain desired conditions.
(*c*) To maintain comfort conditions within specified limits.
(*d*) To optimise running costs by means of efficient plant operation and minimum energy consumption.
(*e*) To reduce manpower costs.

**Comparison of Systems**

The choice of control systems available may be broadly classified as direct acting, electric, electronic and pneumatic. Selection of the most suitable system for the application may or may not be self-evident. Direct acting and electric controls, if used alone, are both generally applicable in the domestic sector. Thermostatic radiator valves are the most common direct acting controls–often used outside the domestic sector–and room thermostats and small motorised valves are examples of electric controls. Both types of control are used with systems generically known as electronic or pneumatic.

**System Evaluation Check List**

In addition to the general guidelines, when systems are being compared prior to selection or specification, the evaluation should cover the check list below and that under *Economic Factors* which follows.

(*a*) Will the systems under consideration provide the specified control conditions within the required limits?

(*b*) Do they satisfy any fail-safe criteria? (For example, do primary control valves on MPHW/LPHW calorifiers drive to the closed or bypass condition on plant shut-down or power failure?)

(*c*) Are they safe in respect of the following?
- (*i*) Leakage of the operating fluid
- (*ii*) Maintenance while in operation
- (*iii*) Use in hazardous areas.

(*d*) Are there any special electrical wiring requirements? (For example, screened cable or segregated wiring?)

(*e*) Does the total cost of the control system installation include any or all of the following?
- (*i*) Fitting valve bodies in pipework,
- (*ii*) The fitting of actuators and linkages to valve bodies and the adjustment of linkages.
- (*iii*) Supply and fitting of control dampers and linkages.
- (*iv*) Fitting damper actuators and linkages to dampers and adjusting linkages.
- (*v*) Installing thermostats, etc. in pipework, ductwork and spaces, inclusive of the necessary fittings, e.g. immersion pockets.
- (*vi*) Connections between valves, thermostats and controllers, etc.
- (*vii*) Interlocks between other items of plant and the control equipment.
- (*viii*) Power wiring associated with the mechanical services drives.
- (*ix*) Supply of control and/or motor control panels.

## ECONOMIC FACTORS

The factors affecting the economic evaluation of control systems are as follows:

(*a*) "All-up" cost comprising:
- (*i*) Cost of control equipment, including valves, sensors, controllers, control panels, starters, air compresors, etc.
- (*ii*) Overall installation cost of controls, control panels, starters, electrical wiring and/or pneumatic pipework, particularly the electric wiring associated with the electric or electro-pneumatic elements of a pneumatic control system.
- (*iii*) Cost of supervising the installation.
- (*iv*) Cost of software preparation for computer-based equipment.
- (*v*) Cost of testing and commissioning, including installation of test and recording instrumentation.
- (*vi*) Cost of spares to be held.
- (*vii*) Operating costs of control system over the life of the building or system.
- (*viii*) Servicing and maintenance costs, over the life of the building or system.

(*b*) Availability of spares and servicing facilities from the controls supplier over the building or system life.

(*c*) Evaluation of cost benefits derived from reduced energy consumption provided by the different control systems.

## DIVISION OF RESPONSIBILITY

The responsibility for the correct operation of a plant or control system should not be divided. In modern buildings the control system interfaces with many site installers and plant suppliers including, amongst others, the heating and ventilation sub-contractor, the electrical sub-contractor and the suppliers of boilers, chillers and motor controls. It is essential that:

(*a*) All interface conditions be clearly defined, e.g. see under *Comparison of Systems.*

(*b*) Boundary conditions on both sides of an interface are compatible and accepted as such by the various participants.

(*c*) One of the participants has the overall responsibility for ensuring that the complete system functions correctly.

Where packaged equipment is supplied complete with its own integral control system it is advisable to use these systems where possible. If there are over-riding considerations, such as sequence operation of boilers, ensure that the equipment supplier accepts the need for modification and includes it as part of the equipment submission.

## ENERGY AND CONTROLS

Controls and control systems can have very marked effects on energy consumption and it is essential that consideration be given to these systems at the concept stage of the design.

The major effects of controls on energy use are:

(*a*) Optimisation of plant operating periods.
(*b*) Matching plant output to load.
(*c*) Temperature control.
(*d*) Elimination of simultaneous heating and cooling.
(*e*) Efficiency of energy usage.
(*f*) Monitoring and optimisation of control systems.

## CONTROL THEORY

**Control Modes**

A controller can be made to operate a final control element (FCE), e.g. a valve, damper or luminaire in a number of different ways in response to a signal. The way in which the FCE acts in response to the signal is known as the control mode. There are five principal control modes, sometimes used singly or in combination, while a number of variations also occur. The control mode does not define the means by which control is effected, which may be through mechanical, electro-mechanical, electronic or pneumatic systems.

**On/Off Control**

On/Off control provides only two plant outputs, maximum (On) or zero (Off). The control sensor usually takes the form of an on/off thermostat, pressure switch, humidistat, time switch, etc. and operates such that below the set point the contacts open. The reverse operation can be arranged when the device is said to be reverse acting.

There will always be an interval in between the contacts opening and closing when no signal can be given. This range, where no control occurs, is generally referred to as the 'differential gap'. Since on/off control can normally only produce two plant outputs, cycling is bound to occur, see Fig. B11.2, and such control is more suited to high capacity systems. It should be noted that the temperature swing is wider than the differential due to the thermal inertia of the system being controlled. Simple room thermostats used for heating applications often have a slow response characteristic resulting in wide switching differentials. This problem is often solved by the addition of an accelerator in the form of a very high resistance heating element which is energised only when the thermostat is calling for heat. The effect of this is to heat the sensing element artificially, thus anticipating the effect of the space heating on the thermostat. This increases the speed of response and reduces the switching differential, as shown in Fig. B11.3(a). This shows that in practice the cycling of space temperature is generally sinusoidal.

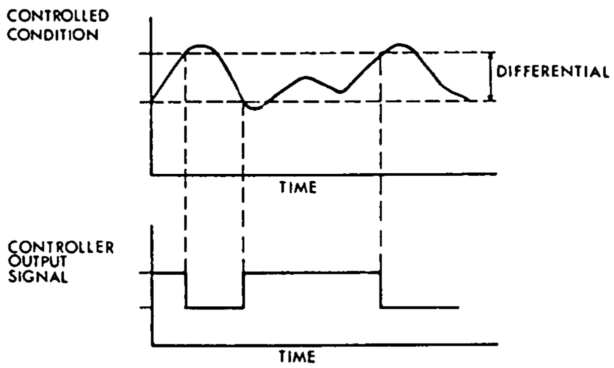
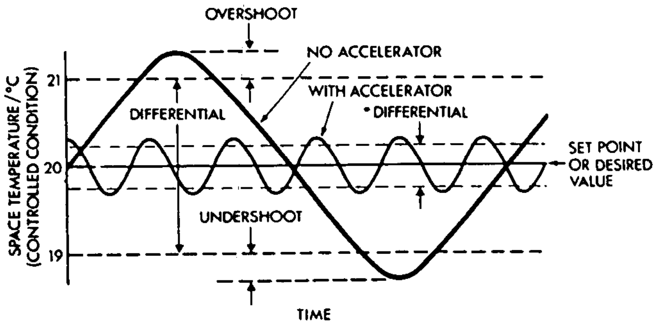



**Fig. B11.2.** Action of an on/off controller.

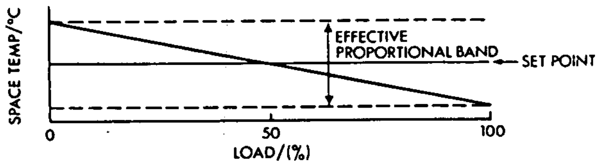
There is a complementary effect associated with this particular arrangement. The ratio of on to off periods of the thermostat with increasing space load varies the internal heat input to the thermostat and alters the controlled condition as indicated in Fig. B11.3(b). This variation is normally acceptable but it does effectively produce a proportional band and the thermostat has some of the characteristice described under *Proportional Control.*

There are certain specialised forms of on/off controller which permit multiple stages of plant capacity. These include multi-step thermostats and step controllers. The former are uncommon except for particular applications, but the latter are frequently used in conjunction with detectors and conventional controllers, see under *Ancillaries.* Multi-step thermostats may also be considered as proportional controllers.




*Corresponds to a small differential in space temperature together with a variation in the thermostat enclosure temperature equivalent to the differential of the non-accelerated thermostat.

(a) Effect of accelerator on differential.




(b) Effect of accelerator on set point.

**Fig. B11.3.** Action of thermostat fitted with accelerator.

**Proportional Control**

Proportional control action refers to a control element having an output signal proportional to its input signal. The proportional band is the deviation necessary to produce the full range of control action. It can be expressed in terms of a physical quantity (e.g. °C, pascal, % humidity,lux, etc.) or as a percentage of the controller scale range. If the scale range of a temperature controller is from 0°C to 80°C and the proportional band setting is such that the controlled variable must change through 100°C to make the valve move from full open to shut, then the proportional band is 100/80 = 1.25 or 125%. If a change of only 20°C is required to bring about this same 100% control action then the proportional band is 20/80 = 0.25 or 25%. Alternatively, in the latter case, the proportional band may be said to be 20° C. Fig. B11.4 illustrates the relationship between the control action and the proportional band width and shows that the controller will give the desired value corresponding to the set point at only one position of the FCE. This occurs at the 50% open position, assuming that the heat output is proportional to the valve opening, see *Valves.*

Under all other load conditions, within the range of stable control, some degree of offset will be present depending on the proportional band setting and the prevailing load. For a controller with proportional only action, the output signal is given by:

$$b_o = -k_p \; b_i \qquad \text{B11.1}$$

where:

$k_p$ = proportional control action (constant equal to the reciprocal of the proportional band)
$b_i$ = input signal (converted deviation)
$b_o$ = output signal

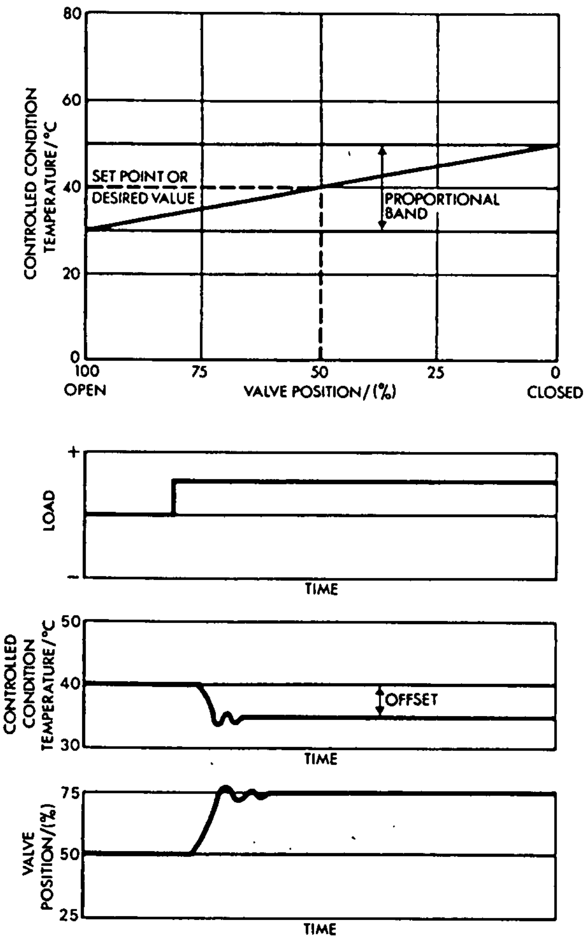



Fig. B11.4. Action of a proportional controller.

**Floating Control**

A correcting element may be arranged to act at a fixed speed by means of a controller which imparts pulses to the element. This mode of operation depends on the use of a dead (neutral) zone in the controlled variable. In this zone the correcting element is not power controlled and remains in its last pulsed position until the controlled variable moves outside the dead zone. This is the basis of floating control as shown in Fig. B11.5.

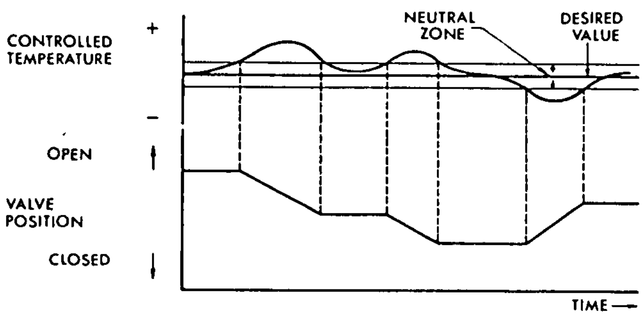



Fig. B11.5. Action of a floating controller.

For a controller with floating action only, the output signal is given by:

$$b_o = -k_f q_o \quad .. \quad .. \quad .. \quad .. \quad .. \qquad \text{B11.2}$$

where:

$k_f$ = floating control action constant
$q_o$ = time for which the value is outside the neutral zone.

**Integral Control**

A correcting element may be arranged to remain stationary when the controlled medium is at the desired value and move in a corrective manner at an increasing speed proportional to the deviation from the desired value. This form of floating action with variable speed is known as integral action. For a controller with integral action only, the output signal is given by:

$$b_o = -k_n \int b_i \, dq \qquad \text{B11.3}$$

where:

$k_n$ = integral control action constant
$q$ = time

**Derivative Control**

A correcting element may be set so that its speed of operation is proportional to the rate of change of the controlled variable. This action is used to eliminate overshoot during a fast load change. Derivative control is mainly applied in special process systems. For a controller with derivative action only, the output signal is given by:

$$b_o = -k_d \frac{db_i}{dq} \quad .. \quad .. \quad .. \quad .. \quad .. \qquad \text{B11.4}$$

where:

$k_d$ = derivative control action constant

**Variations and Combinations of Basic Modes**

While floating and proportional control actions may be used on their own, derivative control must always be used in combination with proportional or floating action. The more common combinations are outlined below.

*Proportional plus Integral (P + I) Control*

This mode combines the stability of proportional control with the accuracy of the integral mode to eliminate offset. The behaviour of P + I control is shown in Fig. B11.6 and the output signal is given by:

$$b_o = -k_p \, b_i - k_n \int b_i \, dq \quad .. \quad .. \quad .. \quad .. \qquad \text{B11.5}$$

*Thermal Feedback*

This is a stabilising action added to single speed floating control by producing a discontinuous action. When corrective action causes the FCE to move, current is passed through an electric heater and the heat derived can be imparted to the necessary element of a Wheatstone bridge or directly to a thermal switch. In either case, it provides a switch action to stop the motor. With corrective movements of short duration, the switch has little or no effect but, when larger deviations call for greater corrective action, the thermal switch interrupts the motor circuit, thus slowing the corrective movement. The greater the corrective action, the greater the total period of interruption. Such action tends to eliminate hunting in the face of load change.

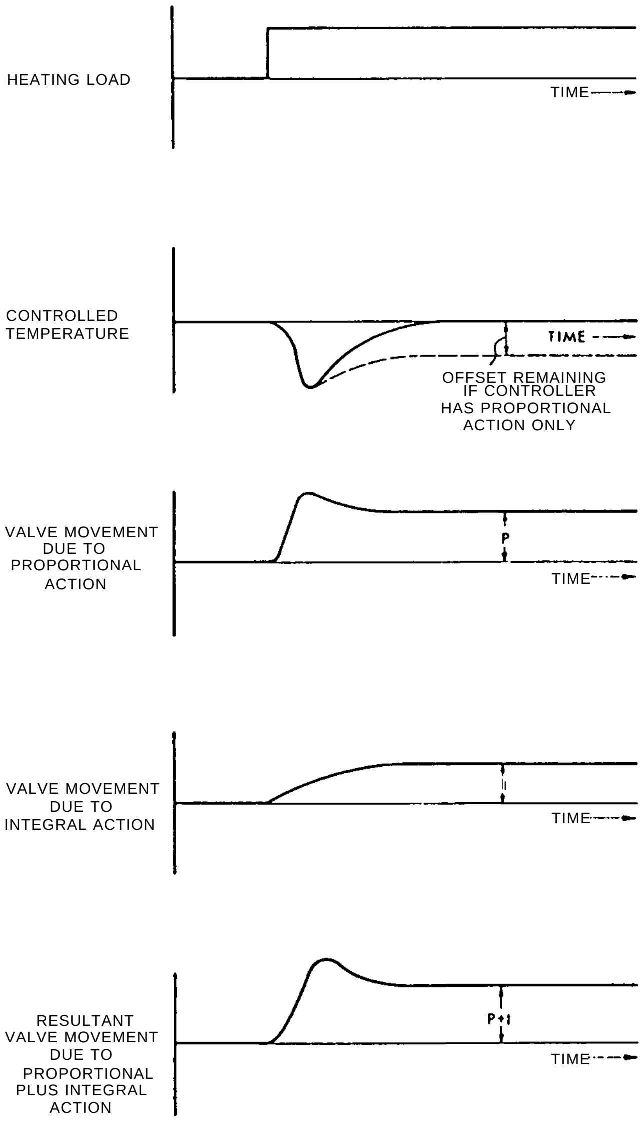



**Fig. B11.6.** Action of proportional plus integral (P + I) controller.

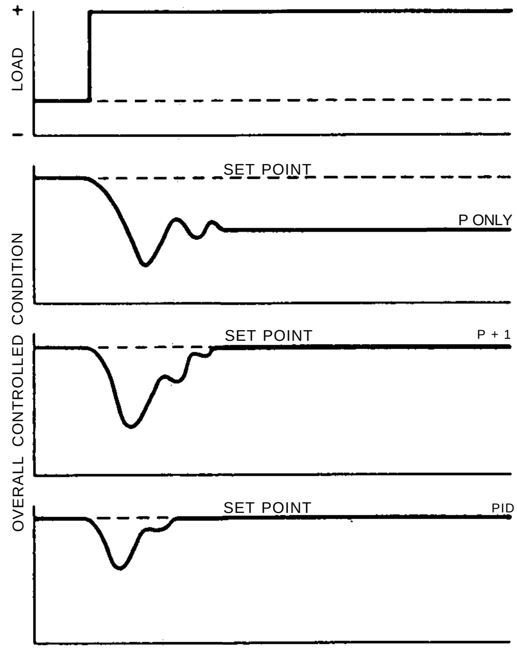



**Fig. B11.7.** Action of proportional plus integral plus derivative (PID) controller.

*Proportional plus Integral plus Derivative (PID) Control*
This form of control combines the advantages of proportional plus integral control with derivative, to combat sudden load changes, whilst maintaining a zero offset under steady state conditions, see Fig. B11.7. The output signal is given by:

$$b_u = -k_p b_i - k_n \int b_i \, dq - k_d \frac{db_i}{dq} \quad \cdot\cdot \quad \cdot\cdot \quad \text{B11.6}$$

**Time Lags**
The full effect of corrective action is not immediately apparent in the controlled medium due to various time lags, which are functions of the plant characteristics, involving units of quantity, potential and time. Some are described under *Definitions* and shown in Fig. B11.1. Another such lag may be defined as a capacity for storing energy which may be on the demand side of the process (e.g. heated water in a tank) or on the supply side (e.g. the hot water in the primary heating coils). Capacity lag will occur in satisfying this demand. In a single capacity process this lag would be exponential, but in practice most processes are multicapacity. Where heat is transferred from one capacity to another transfer lag also occurs.

These lags tend to slow up response and vary considerably according to the process in hand and the magnitude and frequency of the load changes. They may also affect the choice of control mode. A floating mode will provide a temperature with close limits when used to control the air temperature directly downstream of a heater, but if the same heater battery is to be controlled from the extract duct the floating mode is not suitable. The floating controller operates over a very narrow band of temperature and in the first case, there is only a minimal lag between the detector calling for a change in output and sensing the resulting change. In the second case the lag time (comprising capacity, transfer and distance-velocity elements) is large and the change of output will continue until the detector senses the change, which may occur only after the general space temperature has altered very considerably.

**Matching Control Modes and Plant Characteristics**
It is essential, particularly where fine control is needed, that the plant be designed with the required controllability.

*On/Off Control*
Consider a large storage calorifier. The secondary capacity (a mass of water) is large when compared to the primary side capacity; distance-velocity and transfer lags are also relevant. Suppose this heated vessel is controlled by a thermostat in the secondary water operating an on/off valve regulating the heating medium. When the vessel is cold there will be a call for heat, the valve will open and remain open until the water in the calorifier is heated to the required temperature. There will always be a time lag in meeting any load change but since large storage has been provided, this should not matter and an on/off mode of control is quite satisfactory. On/off control is, in general, suitable for processes and applications where there is a large secondary side thermal capacity. e.g. storage calorifiers, space heating of rooms by radiators or convectors.

*Proportional Control*

Consider the case when a non-storage high duty calorifier is served from a high pressure hot water (HPHW) system feeding the primary. This type of calorifier has very little thermal capacity on either the primary or secondary side and consequently has a very short reaction time. If this type of heat exchanger was controlled by on/off control action, a call for heat by the secondary water temperature detector would open the control valve in the HPHW primary flow, but since there is now little secondary capacity, and a fast reaction rate, it would be quickly satisfied and the valve closed. This cycle would occur every few minutes, even with the calorifier under full load, causing rapid wear and possibly early breakdown. In these circumstances the secondary side temperature may exceed boiling point and the water may flash to steam before the valve closes.

If a proportional controller is used a greater change in secondary water temperature must occur in order to cause the control valve to move from fully open to fully shut. If the control valve is properly sized for the maximum duty, a proportional band setting may be found that brings the control system to stability over a given load variation. In general, modulating control action must be used in cases where the secondary capacity is small and the process rate is fast, e.g. non-storage calorifiers and air heater batteries controlled from the discharge air temperature. P + I control or floating controllers employing thermal feedback would also be suitable and are free from the offset inherent in the use of proportional controllers.

*Integral Control and Reset Control*

Air heater batteries must be provided with a modulating control action since they have very little thermal capacity. Where discharge air temperature control is required, floating control, see Fig. B11.5, or proportional control, see Fig. B11.4, may be employed. If the design air temperature change across the battery is large, the proportional controller may need a very wide proportional band to achieve stability and the offset in these circumstances may not be acceptable. Consider the same heater battery with a requirement to control a final space temperature from an extract duct or from the space itself. This results in increased distance-velocity lag, together with an increase in the capacity of the space to be heated. In many cases the capacity lag introduced by the space has the effect of absorbing small heat gains and losses and has a stabilizing influence. However, if load changes in the space are considerable (e.g. intermittent machine operation) and frequent, then a wide proportional band will have to be used to obtain stability and the ensuing offset from the desired value of temperature may be unacceptable. The magnitude of the offset will vary with the load.

Where undesirable offset occurs, due to fluctuating loads in the space, a need to ‘reset’ the desired value or control point occurs. Raising the control point under heavy load and lowering it under light load tends to eliminate offset. The controller itself may be arranged to eliminate offset by adding the integral mode to the proportional mode of control. The proportional corrective action is continued by the integral action of the controller until all deviation is eliminated, i.e. a floating action added to a proportional action, see Fig. B11.6. Electric actuators for valves and dampers are normally constant speed devices and variable speed is brought about by electrical impulses, the frequency and duration of which can be varied. In the case of pneumatic control systems an integral mode of control can be provided by specific reset relays.

P + I control action is desirable where close control is required against wide load fluctuation and process lag.

*Proportional plus Integral plus Derivative (PID) Control*

A PID controller would be used for load matching when there are unavoidable large distance-velocity lags, when there are very rapid load changes, coupled with tight limits on the desired value and when there is a need for fast corrective action as is the case with certain types of multiple chiller control. Normally, this form of control action is not justified in heating, ventilating and air conditioning applications and its use is generally restricted to process control.

*Open and Closed Loop Control*

Control systems are applied to control loops, each loop normally comprising one section of an overall plant, e.g. reheater, calorifier, smoke alarms. Loops are normally defined as being open or closed, most in the HVAC field being closed. When the condition of the controlled variable changes in closed loop control the sensor detects the change and initiates correcting action to the final control element. The effect of this is reflected in the controlled variable and reassessed by the sensor which continuously provides feedback to the controller. Examples of this are space temperature control, control of secondary DHWS temperature and static pressure control of supply fan volume in variable volume systems. All modes of control may be used to provide closed loop control.

Open loop control does not include feedback of the type described above. The simplest form of such a loop would be the detection of smoke where a local and/or remote alarm is initiated by a smoke detector without any feedback to the smoke sensor. A more common version in the HVAC field is the use of an external sensor to control the output from an air curtain in a shop entrance. In this case the output from the heater battery serving the air curtain would be modulated in accordance with the outside temperature so that, for example, at 0°C externally there would be full output and at 15°C, no output. The external sensor is not affected by any variation from the final control element (heater battery control valve) and therefore there is no feedback. Only on/off or proportional modes of control may be used for this type of loop.

## STABILITY ANALYSIS

In the past little attention has been paid to the stability of heating and air conditioning systems at the design stage. Generally speaking, systems have depended for stability upon the large thermal capacity of traditional and heavyweight buildings with large capacity boilers, heat exchangers, etc. However, with the trend towards lightweight and better insulated buildings and the use of air conditioning and low capacity boilers and heat exchangers, it is necessary to consider system stability in greater detail. Stability analysis makes this possible. Such analysis is normally complex and the analytical solutions require numerical data for a wide number of parameters to test the stability criteria.

It should also be added that there may be difficulties in applying these techniques to conventional HVAC systems which frequently have non-linearities due to actuator reactions, heat transfer characteristics and limited plant outputs. At present suitable techniques for the HVAC field are in their infancy, but work is proceeding to provide acceptable standard formats which may be generally applied. Irrespective of which system is adopted there is a need to encourage the production of the numerical data, much of which can only be obtained with the active co-operation of equipment and plant manufacturers. A more detailed discussion of stability analysis is to be found elsewhere.[2]

## SETTING OUT CONTROL REQUIREMENTS

### Introduction

Control systems need to be built into the planning and design stages of a project. Suitable conditions and tolerances are fundamental parameters associated with the planning and acceptable operation of any system. They affect the design brief, detailed design, the specification, commissioning and, ultimately, maintenance of all projects.

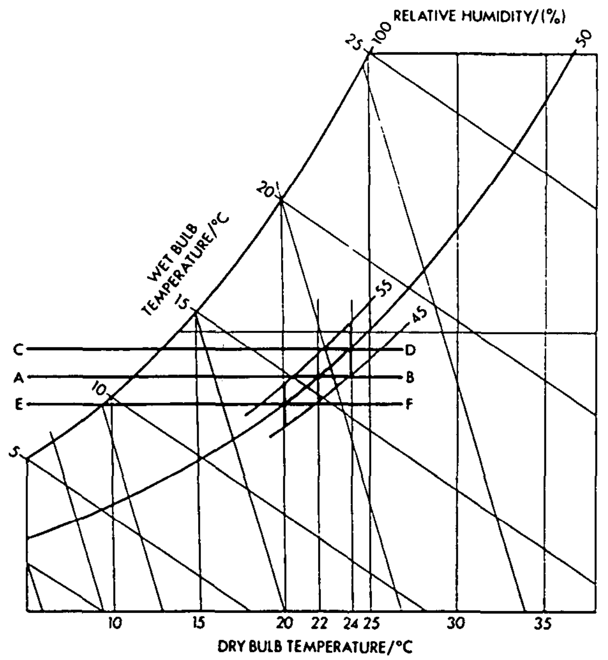



**Fig. B11.8.** Psychrometric diagram of conditions and tolerances.

### Conditions and Tolerances

The design, selection and operation of overall systems are related to the specified conditions and tolerances for both the internal environment and the individual items of plant. These parameters are of particular significance to the control system and care should be taken to ensure that the tolerances are appropriate to the application.

General comfort criteria are covered in Section A1 of the *Guide*. Humidity in the range of 40-70% r.h. and temperatures of 19 to 23° C will normally be acceptable, if the air velocity is low enough to prevent the sensation of draught. There may be situations even for general comfort where these tolerances are too wide. Each case must be considered on its merits, particularly for projects outside the UK. In special areas the tolerances may need to be maintained to ± 5% r.h. and ± 1°C of the specified desired values and in critical cases these figures could be as low as ± 2% r.h. and ± 0.5°C.

Both the general and special cases should be quoted in specific terms, e.g. 50% ± 10% r.h. and 20°C ± 2°C, or 24°C + 4°C - 0°C. Fig. B11.8 indicates this problem on a psychrometric diagram, where there is individual temperature control of a space, with central control of the humidity. If the specified space requirements are 22°C ± 2°C and 50% ± 5% r.h. and the moisture content corresponds to line A-B, then at 22°C, 50% r.h. will be achieved, but at 20°C and 24°C for the same moisture content the relative humidity will fall outside the specified limits. Equally, at 22°C, moisture contents designated by lines C-D and E-F will satisfy the humidity limits, but if the temperature varies from 22°C to the permitted limits, with either of these constant moisture contents the humidity limits may be decisively broken. From this example the relevance of conditions and tolerances to design is clearly illustrated. Either, there should be separate control of humidity in each space if the specified conditions are precisely what is wanted, or the tolerances should be amended so that they can be achieved with normal operation of the plant. In the example this would mean specifying 22° ± 2°C for temperature control and accepting an r.h. of 50% + 12% - .10% for a central plant capable of holding the moisture content between C-D and E-F.

In addition to space conditions, those for heating and cooling source flow and return temperatures (and, at particular locations, within air handling plant) must be specified with suitable tolerances. Select the tolerances for the correct space requirements and then provide the controls to satisfy these parameters. The use of conditions or limits of less than ± 2% r.h. for humidity is not normally practical.

The stability of flow temperatures for boilers and chillers may be very important to the control of individual plants elsewhere in the system. Again, any limits specified must be within the capability of the selected plant and control system. Limits of ± 2°C and ± 0.5°C for heating and cooling respectively are the lowest figures likely to be achieved under all load conditions. A summary of acceptable conditions and tolerances for a wide range of space, equipment and measurement parameters may be found elsewhere.[3] Measuring instruments with a suitable degree of accuracy should be installed to determine whether desired conditions are achieved.

### Planning

Planning is a crucial element for the satisfactory completion of a control system according to programme and can be considered in stages. Compliance with relevant legislation must be borne in mind at all times.

*Design and Staff Brief Stage*

The broad aspects of the control system need to be clarified and agreed by the client with the design team from the inception of the project.

Many of the criteria are equally applicable to other aspects of building services. The staff brief is an essential element for the client to consider. Modern systems are often complex and it is essential that operating and maintenance staff are suitably matched to the technology used.

*Detailed Design Stage and Tender Evaluation*
During this stage the aim is to produce a comprehensive specification which can be priced and evaluated.

*Construction Phase*
During the construction phase modern control systems require a high level of supervision for work both on and off site and this must be incorporated in the cost plan.

In addition, the complexity of design input and agreement of final details are often not recognised. The requirements necessary to obtain a set of agreed wiring diagrams for the control panels are a good example, where the following parties, at least, will be involved:

(*a*) Consulting engineer.
(*b*) Major plant supplier.
(*c*) HVAC sub-contractor.
(*d*) Electrical sub-contractor.
(*e*) Controls supplier.
(*f*) Panel manufacturer.

*Commissioning and Defects Liability Stage*
Controls cannot be commissioned correctly or adequately unless the overall programming for the construction phase incorporates suitable planning, including the functions of handover and routine maintenance during the defects liability period.

The commissioning programme for the automatic control system should include the following:

(*a*) Co-ordinating the commissioning of the control systems with the programmes for the completion and setting-to-work of the associated plant and equipment. Allowance should be made for pre-commissioning cleaning (including chemical cleaning where necessary), air and water balancing, plant trial runs and checks of all associated plant.
(*b*) Planning the control system commissioning for the sequence and content of each activity, numbers of men required (individual skills to be identified) and the need for temporary power, water or other services.
(*c*) Allocation of adequate time for commissioning, making allowance for equipment failures and site damage due to accidents. It is not uncommon on large projects for the commissioning and functional testing of an installation to require a period in excess of six months.

**Control System Brief**
For the purposes of the control system the minimum essential information required for the brief includes:

(*a*) Flow diagram of the plant(s) with basic control schematics, air movement paths and, preferably, flow rates.
(*b*) Available system temperatures, with limits, and pressures, both static and dynamic.
(*c*) Desired values of temperature, humidity and pressure, with permitted tolerances, for each plant.
(*d*) Flow rates and media details for all components requiring control.
(*e*) Permissible maximum and minimum pressure drops across control valves and/or the valve authorities required (the pressure drops across the elements served by the valves will be necessary for this).
(*f*) Turn-down ratio from full load to part load over which the controls are to operate.
(*g*) Locations of detectors, controllers and correcting units.

Other factors which also affect the brief are:

(*a*) Architecture and construction.
(*b*) Zoning and usage patterns.
(*c*) External load variations.
(*d*) Internal conditions and gains.
(*e*) Economics.

**Flow Schematics**
Flow schematics are rarely produced with sufficient detail for control purposes, but they are an essential part of the design process. However, if the necessary information is included, flow schematics greatly reduce the opportunities for errors in the overall control system from the concept stage of the design onwards.

Schematic diagrams are a crucial element for the design of automatic control systems. They are equally important to the overall design process across all the mechanical and electrical disciplines and are of use to all members of the design and construction teams. Known by various names, they show all the functional details of a plant or installation,

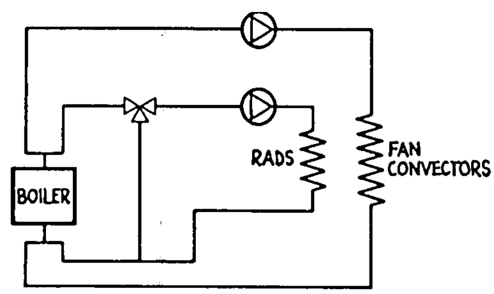



**Fig. B11.9.** Conceptual schematic.

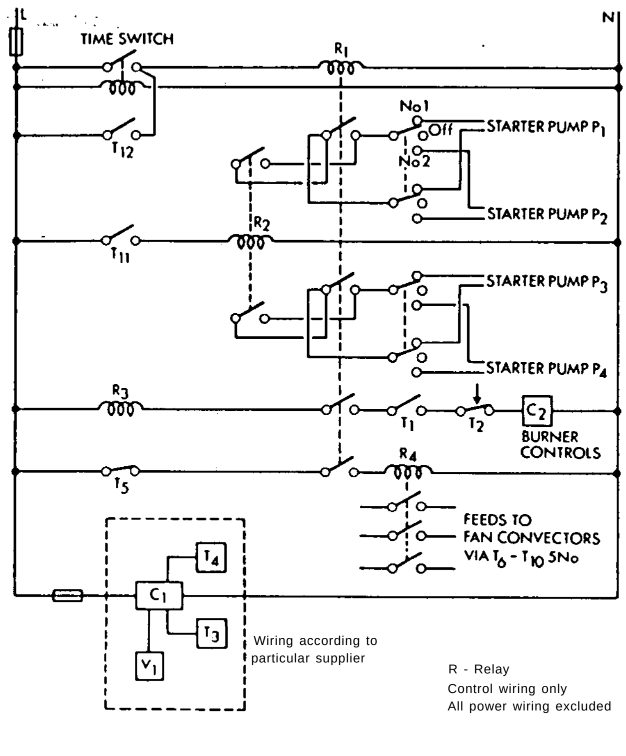



**Fig. B11.10.** Logic schematic for tender.

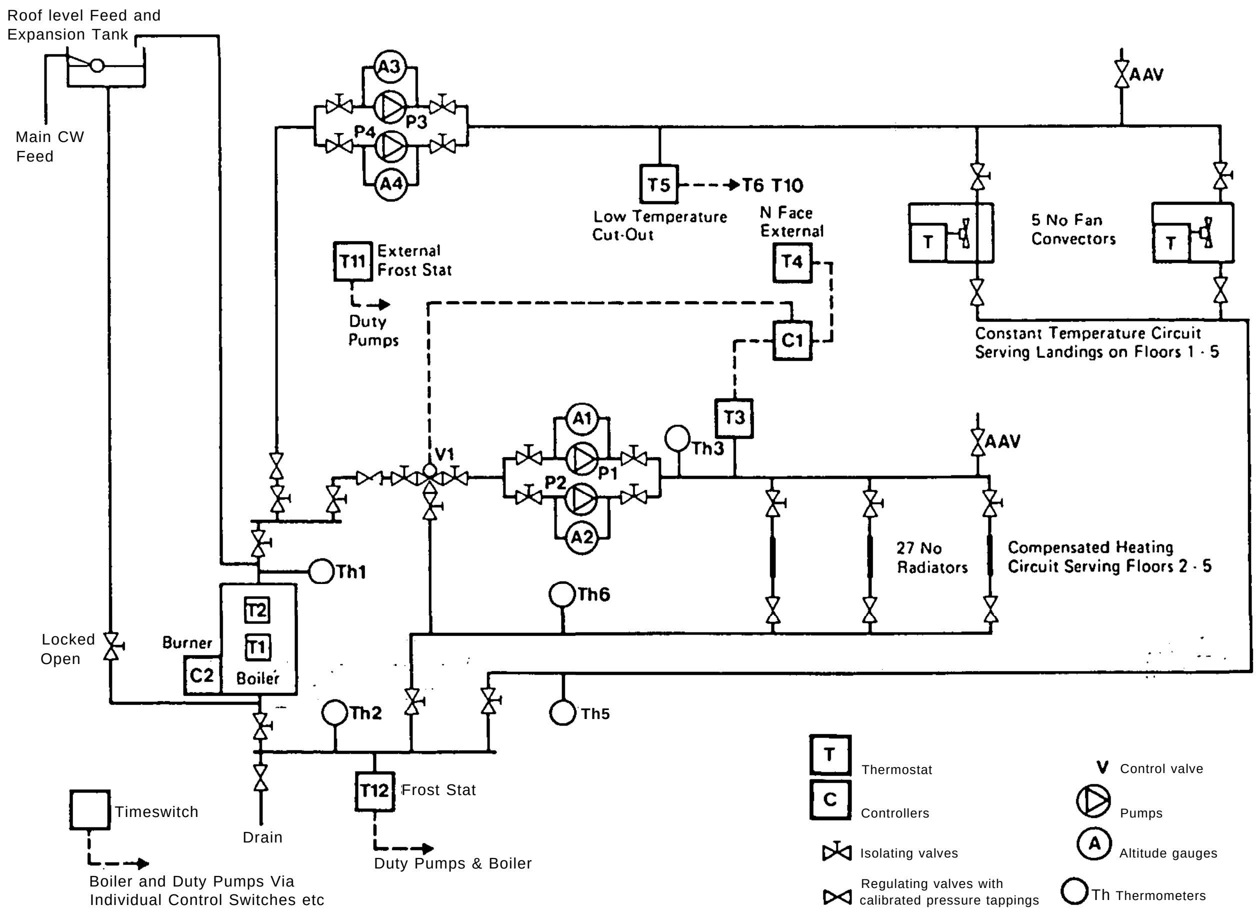



**Fig. B11.11.** Pipework/controls schematic for tender or completion.

including the controls. Schematics should be started at the concept stage in sketch form and gradually expanded so that comprehensive schematics are available by tender stage.

The final form of the schematics at project completion may differ from those at tender stage but they should be used to initiate any other documentation relating to variations in design during the project programme. Changes should be marked and recorded on the schematics first and then on the associated design drawings.

Another important feature of flow schematics is their use for the basic identification, tabulation and cross referencing of specific classes of equipment throughout the project documentation. It is particularly important that each component (e.g. detector, valve, etc.) has a unique reference.

Attention should be given to illustrating important pipework features on schematics, such as reverse returns and the correct boiler and chiller header arrangements. Failure to do this may lead to a working drawing which is not hydraulically correct and thus prevent the correct operation of an otherwise suitable control system.

Figures B11.9 to 11 illustrate the form of flow schematics which have been described. Fig. B11.9 indicates the possible conceptual schematic and Figs. B11.10 and 11 are schematics for interlocking and HVAC systems respectively, at tender or completion stage. The operation of the system shown is probably self-evident but either the schematic and/or project specification should include a description of the system operation. There are several acceptable forms of interlocking or logic diagram and one version is shown in Fig. B11.10. Software versions are being used with increasing frequency and these require different skills for interpretation.

**Specifications and Schedules**

Specifications for control systems have to cover the control systems themselves, testing and commissioning, interfaces between various sub-contractors and the scheduling and cross referencing of equipment and desired conditions.

*Controls Performance*

A plant operation description, complementary to the flow diagrams, should be provided. The description should cover not only the control sequence but the operation of the plant which may be transposed later into the operating and maintenance manuals for the project.

The description must also include the general details for the equipment or the contractual requirements in respect of the capacity of power feeds, the equipment in the control panel, who actually provides it and who is responsible for the installation and commissioning. All these points need to be covered in the specification, some under the control system and others elsewhere.

The controls specification must provide the performance criteria in terms of:

(*a*) Details of the plant switching.
(*b*) The functional operation of the plant.
(*c*) Sequence of events.
(*d*) Individual operation of each control loop.
(*e*) Identification of each control item referred to.
(*f*) Safety and emergency operations and overrides.

(*g*) Operating parameters (e.g. boiler water temperature and pressure).
(*h*) Environmental requirements and permitted tolerances.
(*i*) Interfaces with other equipment.
(*j*) Interfaces between suppliers, sub-contractors and any combination of both.
(*k*) The interaction of the HVAC system with other building services (e.g. lighting, lifts, standby generators, fire alarms, etc.).

The specification should also cover any special requirements such as valve bodies resistant to dezincification, where this is likely to occur, or detector positioning in very large ducts.

*Commissioning and Testing*
The specification of testing and commissioning (T & C) procedures is covered by CIBSE Commissioning Code Series C, *Automatic Controls.* However, this assumes that design and installation up to the stage of T & C has been carried out in the correct manner. In practice this may not be the case and the specification should cover the following, in terms of controls T & C:

(*a*) The exact requirements of the work to be undertaken during commissioning, clearly ascribing responsibility for each activity.
(*b*) The required performance of the control systems.
(*c*) The desired values or set points and the maximum acceptable tolerances.
(*d*) Recognition of approved T & C staff.
(*e*) Comments on potential T & C problems at tender stage.
(*f*) Integration of drawings from various suppliers and sub-contractors.
(*g*) Selection of plant suitable for the proposed control system.
(*h*) Site supervision of the control system and its installation during the site operations.
(*i*) Defining interfaces and areas of responsibility.
(*j*) Preparing a logical T & C programme so that the controls are all available for T & C on the due date.
(*k*) Co-ordination of the various parties involved in T & C.
(*l*) Ensuring that T & C results are recorded and comply with the CIBSE Commissioning Code and other standard tests and procedures considered necessary.

*Boundaries and Interfaces*
The problems of interfaces between the work of the various contractors, sub-contractors and suppliers on site have been mentioned. These interfaces should be identified, listed and cross referenced from one specification to the other at the design stage. Without this the later difficulties and costs escalate out of all proportion by comparison with the time and effort needed to define them at the design stage.

Interfaces which can directly affect the control system in a normal building include lighting, fire, security and lift systems. The number of interfaces possible by permutating from all the possibilities is enormous.

The controls specification must define a large number of interface conditions, often between more than two participants. One obvious simplification is to make one of the participants responsible for a group of interfaces. However, it is incumbent upon the designer to ensure that the boundary conditions match on either side of any interface, regardless of the agreed responsibilities of individual contractors.

*Schedules*
Schedules are very important as a basic element for indexing and cross referencing across the total project documentation. In the controls specification schedules would be required for the following:

(*a*) Plant parameters and tolerances.
(*b*) Environmental requirements and tolerances.
(*c*) Equipment, with unique identification and numbering.
(*d*) Testing and commissioning (for completion on site).

**Commissioning and Performance Testing**
The final stage in the completion of any project is the advancement from static completion to full working order to the client's specified requirements and incorporates the following:

(*a*) Pre-commissioning inspection.
(*b*) Commissioning.
(*c*) Proving.
(*d*) Performance testing.
(*e*) Records of T & C results.

*Maintenance and Defects Liability*
During commissioning and performance testing, the maintenance brief for the automatic control systems should also be checked. In the interests of safety and economy, every automatic control system requires regular maintenance. Therefore it is necessary to consider maintenance procedures and access for maintenance as part of the design brief.

The contract may define a period, after completion, within which the contractor is responsible for any defects which may be revealed. During this period routine maintenance must also be performed and it will be necessary to specify clearly who will be responsible for the normal maintenance and servicing, during the defects liability period.

## CONTROL EQUIPMENT AND ITS PRACTICAL APPLICATION

This section refers to items of control equipment normally utilised in typical control systems. The various forms of element for each class of equipment are identified together with their fields of application. Table B11.1 summarises the ranges of common elements. A detailed examination of the operation of particular items of electric, electronic or pneumatic controls is not included. The ranges and variety of equipment in these fields are now very extensive and the electric/electronic and mechanical functions are described in standard text books and manufacturers' technical literature.

**Sensing Elements**
Irrespective of the form of sensing element or the parameter being controlled, the siting of the elements is vitally important. Methods of sensing are described below for the common parameters in HVAC systems, i.e. temperature, pressure, humidity, flow and enthalpy. Different

sensor mechanisms are available for each parameter and usually manufacturers select particular elements for use in their own ranges of equipment. The selected elements are suitable for general applications in building services.

**Table B11.1.** Summary of common sensing elements

| Controlled variable | Elements | Range | Accuracy |
|---|---|---|---|
| Temperature | Bimetal strip | –180 to 420°C | ±1% up to 300°C |
| | Rod and Tube | –180 to 420°C | |
| | Sealed Bellows | | |
| | Liquid filled | –45 to 650°C | ±0.5% up to 320°C |
| | Liquid/ vapour filled | | ±1% over 320°C |
| | | –35 to 300°C | ±l% |
| | Thermocouple | –260 to 2600°C | ±0.5% to ±3% |
| | Resistance Bulb | –260 to 650°C | ±0.75% of scale |
| | Thermistors | 0 to 300°C | |
| Pressure | Bellows | 0 to 200 kPa | |
| | Diaphragm | 0 to 750 kPa | |
| | Bourdon Tube | 0 to 75 MPa | ±1% |
| Humidity | Filament (hair, nylon, etc.) | 20 to 80% r.h. | 3–4% |
| | Biwood | 20 to 80% r.h. | |
| | Resistance | 5 to 98% r.h. | ±1% |
| Flow | Orifice plate | Any linear flow | ±l% |
| | Venturi tube | Any linear flow | ±2% |
| | Flow nozzle | Any linear flow | ±1% |

*Note:* The range quoted is not for a single device, but for the range over which the type of element can operate. There are also limits on the operating range to which the elements may be subjected without damage. If it is known that the element will be subjected to conditions outside the declared range of operations, this should be checked with the manufacturer.

*Temperature*

The most commonly used temperature detectors rely on either thermal expansion or electrical changes due to temperature variations. The most common sensors are listed in Table B11.1.

In any practical system the response to a change in input will not be instantaneous. This delay in response is called lag and the lag coefficient is defined as the time needed to reach 63.2% of the total change. Fig. B11.12 gives the response curve for various sensors.

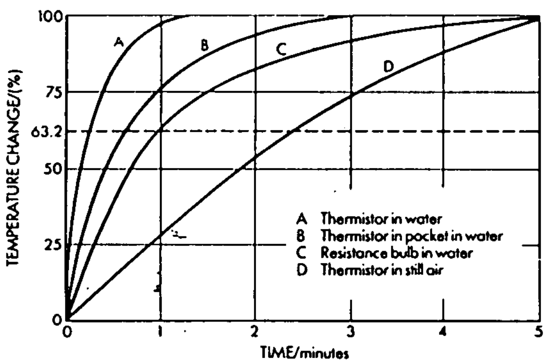



**Fig. B11.12.** Typical temperature sensor response curves.

Another factor influencing response is the type and velocity of the fluid in which the detector is immersed. A low fluid speed, with respect to the sensing element, greatly increases the resistance to heat transfer, see Fig. B11.13. Water should flow past the sensing element at not less than 0.3 m/s to minimise the time lag whereas air should circulate around the sensing element at a minimum of 2 m/s.

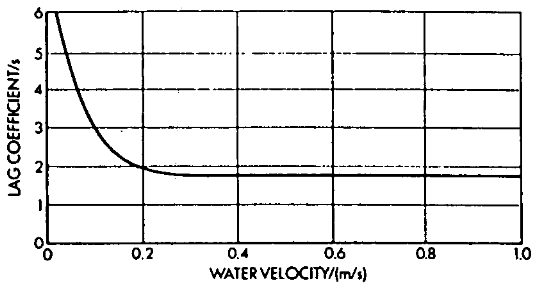



**Fig. B11.13.** Effect of water velocity on heat transfer.

*Pressure*

For pressure sensors, bellows, diaphragms, Bourdon tubes, or similar devices may be used. The medium under pressure may be transmitted directly to the device used to operate the mechanism of a pneumatic or electric controller. Rate of flow, quantity of flow, liquid level, and static pressure may all be controlled by pressure sensors. The most common types of pressure sensing devices utilise a bellows or diaphragm acting against a spring. Elements of this type are very sensitive, being able to detect pressure changes of about 10 Pa and can be used over a range of absolute pressures from 0.15 Pa to 150 Pa.

*Humidity*

The primary element of a humidity sensor can be human hair, nylon, wood or other substance that responds to humidity change. As the moisture content of the surrounding air changes, moisture is absorbed or released and the element expands or contracts, operating the mechanism of the controller.

A biwood element operates on a similar principle to the bimetal element for temperature measurement, but in this case two woods with differing moisture absorbancies are used. Humidity sensing elements suitable for electronic application and control may also consist of metallic contacts fused to a piece of glass or plastic. Electrical contact is maintained between the two strips by hygroscopic salt, e.g. Lithium Chloride, painted over the surface of the material. The resistance of the salt varies with the amount of moisture absorbed.

*Flow*

Existing flow measurement instruments operate on a variety of principles but the most widely used are based on the differential pressure across a restriction. This readily lends itself to either electric or pneumatic transmission and control. The restriction is installed in a duct or pipe to create a pressure differential which bears a square law relationship to the rate of flow. This restriction may be an orifice plate, a Venturi tube or a flow nozzle. The orifice plate is widely used because of its convenience and economy and reliable data are available for its calibration. Extensive use is now also made of calibrated regulating and control valves for the same purpose. Positive displacement and turbine instruments are commonly used and magnetic flux flow meters are now available.

*Enthalpy (Total Heat)*
A measure of enthalpy is obtained using temperature and relative humidity sensors (often mounted in a common housing) which feed their signals into a controller, the output of which is characterised to give an approximately proportional signal.

**Actuators**
An actuator is a device which responds to the signals from a controller and enables power to be transmitted to a final control element. A variety of types of actuator exist and the two characteristics which most affect actuator selection are the available torque (and its efficient transmission to the final element) and stroke time, which may be significant in terms of overall system response. The most common application of actuators is to the operation of valves and dampers.

The most common types of actuator are:

(*a*) Electric motors.
(*b*) Solenoids.
(*c*) Pneumatic actuators.
(*d*) Hydraulic actuators.
(*e*) Self operating actuators.
(*f*) Manual actuators.

The following points should be checked for actuators used with:

(*a*) *All valves*
Are there any requirements for rapid closure or opening of particular valves, e.g. HPHW/LPHW calorifiers or condenser water?
(*b*) *Three-port valves*
Maximum differential pressure rating of actuator compared to the pressure drop at design flow.
(*c*) *Two-port valves*
Maximum differential pressure rating of actuator compared to the pressure drop at design flow. Does the pressure difference across the valve, when closed, exceed the maximum differential pressure rating, e.g. pump head or steam pressure operating against closed valve?
(*d*) *Dampers*
Check that actuator torque is adequate for the proposed damper size, coupled with the maximum air velocity through the damper.
(*e*) *Step Controllers*
Is the response time consistent with its application?
(*f*) *Any final control element*
Ensure that the crank/linkage is correctly designed to move the final element in direct proportion (or to some other characteristic in special cases) to the actuator movement over the full stroke, without over-run.

**Valves**
A valve which is automatically controlled to regulate fluid flow from zero to full quantity is generally a control valve. Manually operated valves which can be made to perform the same function also have a control function but they are more commonly referred to as regulating or isolating valves, or by their means of construction, e.g. gate or globe valves. The type of control valve for a specific duty depends on the service required, conditions of temperature and pressure of the fluid involved and other circumstances, including valve characteristics.

Terminology in respect of valves varies according to the field of application. The use of the alternative terms ‘two-way’ and ‘three-way’ or ‘two-port’ and ‘three-port’ respectively, is normally acceptable?

Fig. B11.14 shows some common types of valve. The selection of the correct control valve for a system is an integral part of overall design and an essential prerequisite for a stable system which can be properly commissioned. Valve selection is based on the use of parameters such as valve characteristic, authority and flow coefficient.

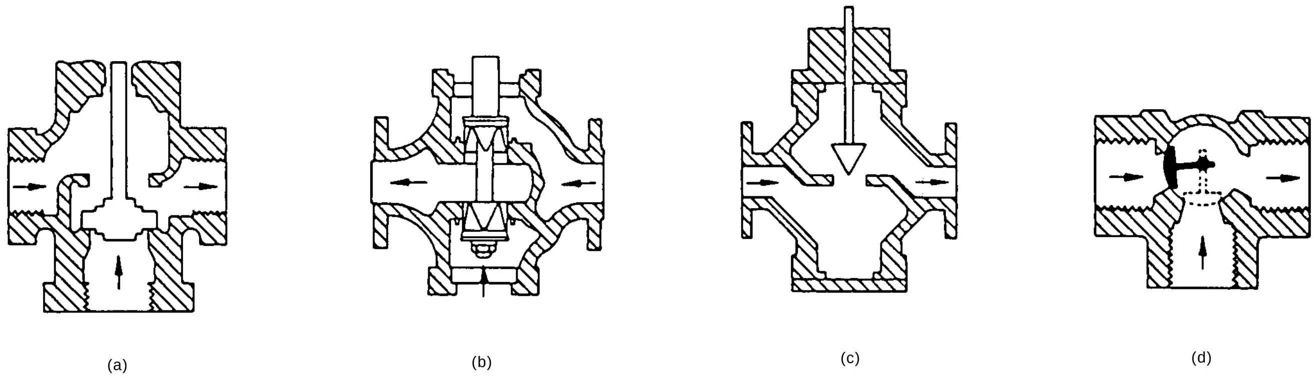
Whereas two position or on/off valves need only to be selected for specific throughputs under clearly defined conditions, the requirements of modulating valves need much closer attention. In the modulating mode the aim is normally to maintain a linear relationship between load requirements and power output over the full travel of the valve. A valve which provides such an output is described as power linear. It should be noted that power output is not directly proportional to flow and varies with both valve and emitter characteristics. It is important to ensure that the valve may be shut off or opened against the prevailing pump and/or static pressures: this is particularly true of large valves but some terminal unit control valves have a very limited tolerance in this respect.

*Control Valve Bodies*
The valve body contains the orifices to the external connections and between the internal seating arrangements. Various types of body are used for control valves, globe valves being the most common. Double-seated globe valves largely eliminate the out of balance forces which occur with single-seated versions. However, when used for mixing applications, i.e. two inlets and one outlet, the out of balance forces rarely create a problem and single-seated valves are usually suitable in conventional HVAC systems. Where a diverting application is essential, i.e. one inlet and two outlets, either a double-seated or shoe type valve should be used.

The construction of double-seated valves virtually precludes a tight shut off condition. Therefore, if used with steam or HPHW the controlled temperature may rise to dangerous levels, even with the valve fully closed. A back-up single-seated tight shut-off control valve is then essential.

Valve bodies for low pressure applications up to approximately 80 kPa are normally screwed except in sizes of 100 mm and above. Above these LPHW pressures and for HPHW and steam above 70 kPa, flanged valves should be used.

Common body materials are bronze, brass and cast iron for less arduous service conditions and cast steel and similar materials for more severe duties. Special valve bodies are available for process applications involving fluids containing solids, abrasives or corrosives.




(a) Three-port single seat valve.

(b) Three-port double seat valve.

(c) Two-port single seat valve.

(d) Three-port shoe type valve.

(e) Butterfly valve.

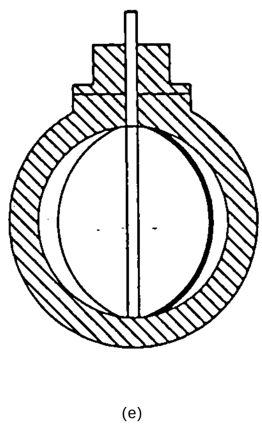



**Fig. B11.14.** Some common valve types.

Within the valve body various items comprising the trim are fitted. These items are mainly associated with seating the valve plug and comprise guide bushes, guides and seat rings. Normally these items can be replaced *in situ* as they are subject to greater wear than the valve body as a whole, but there are some instances where the seats are integral with the valve body. Butterfly valves are sometimes used for control purposes but the characteristics of such valves limit their use to specific applications.

Shoe or slipper type valves have a relatively simple body construction and characterised internal ports over which the shoe slides. Such valves are normally restricted to three way operation and may generally be used for mixing or diverting applications without the drawback of unbalanced forces from which single-seated plug valves suffer. They normally have a higher let-by rate than plug type and frequent maintenance is required to ensure consistent performance under certain operating conditions.

*Bonnet and Packing Box*

The bonnet, screwed or flanged to the valve body, contains a gland or packing box which provides a tight seal for the fluid, little resistance to movement of the valve plug stem and easy maintenance. The bonnet is frequently of the same material as the valve body. The materials for the packing box or gland include moulded V-rings of PTFE, neoprene O-rings and other forms of PTFE. Where no leakage can be tolerated through the gland, e.g. refrigerant control valves, a bellows seal is employed, normally of stainless steel, which is integrally formed with the valve plug stem. High and low temperature applications require special bonnet arrangements. For temperatures above approximately 200°C, finned bonnets are used to dissipate conducted heat. Below 0°C extended bonnets are used to prevent the valve plug stem being affected by icing condensation.

*Plug Design*

The design of both valve plug and internal ports affect the valve characteristic. The characteristics are vitally important in the correct selection and application of valves. Several basic plug types are normally available and each provides a different orifice area for comparable positions of the valve plug stem and, hence, varying characteristics.

*Characteristics*

The relationship between valve spindle lift and area of the valve opening is known as the valve characteristic. In the case of shoe type, or rotary, valves a similar relationship exists between shoe rotation and area. There is an enormous range of possible valve characteristics but those most commonly encountered in control systems are:

(*a*) Linear, where the orifice area is directly proportional to the valve spindle movement and the flow varies linearly with spindle lift.
(*b*) Characterised V-port, with a characteristic falling between linear and equal percentage.
(*c*) Equal percentage, or similar modified parabolic, where equal increments of valve spindle lift provide an equal percentage change of the previous area. The characteristic is represented by:

$$s \quad = \log\left(\frac{V_{100}}{V_0}\right) \qquad .. \quad .. \quad .. \quad \text{B11.7}$$

where:

| | | |
|---|---|---|
| $s$ | = change in valve stroke .. .. | % |
| $V_0$ | = volume flow rate at zero stroke if valve was characterised to this point .. .. .. .. | $m^3/s$ |
| $V_{100}$ | = volume flow rate with valve fully open .. .. .. .. | $m^3/s$ |

(*d*) Quick opening, where the flow increases very rapidly from zero for small valve spindle movements, with a fairly linear relationship between flow and spindle movement. Beyond this initial movement the flow rate varies more slowly with increased spindle movement. Such valves are primarily used for on/off service.

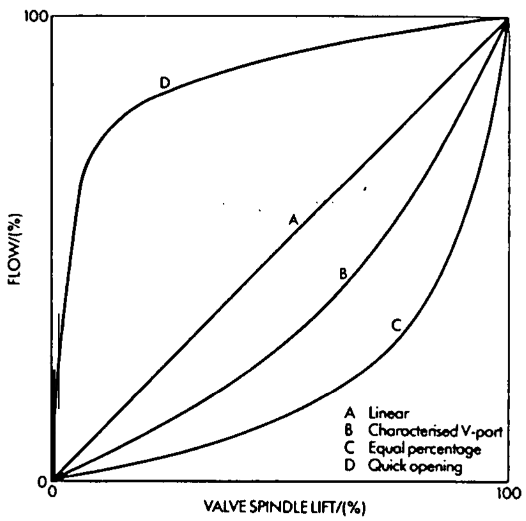



**Fig. B11.15.** Stylised basic valve characteristics.

Fig. B11.15 shows in stylised form the basic characteristic curves for valves described in (a) to (d) above. The basic characteristics are obtained by measuring flow against a constant pressure drop across the valve for all positions of the valve. In these circumstances the valve is said to have an "authority" of unity. In practice these curves are modified as the valve authority decreases. Normally, the plug design determines the characteristics on the assumption that the plug lift will be directly proportional to the signal from the control loop. However, it is possible to achieve a particular characteristic by making the plug lift follow a particular curve not proportional to the signal.

*Valve Authority*

Normally, a control valve is employed to regulate the flow in a system where the total pressure drop in the circuit concerned is derived from the circulating head of a pump, which should be reasonably constant. Figs. B11.16 to 23 show examples of systems where the valve authority is defined by:

$$N = \frac{P_1}{P_1 + P_2} \qquad .. \quad .. \quad .. \quad .. \quad .. \qquad \text{B11.8}$$

where:

| | | |
|---|---|---|
| $N$ | = valve authority | |
| $P_1$ | = pressure drop across the valve (fully open) | Pa |
| $P_2$ | = pressure drop across the remainder of the circuit .. .. .. .. .. .. | Pa |

The definition is often given in terms of *DP* but the above notation has been adopted to simplify explanation of the schematics.

For clarity, isolating valves are omitted from Figs. B11.16 to 23, as are regulating valves in some cases.

Suitable valve authorities in practical terms should be kept as high as possible for reasons explained under *Combination of Criteria.* The authority should not be less than 0.5 for diverting and throttling applications and not less than 0.3 for mixing applications.

Three-port valves used in the applications of Figs. B11.16 and 17 should be designed for constant total volume throughout, which ensures reasonably constant pressure. In mixing applications $P_2$ is measured round the source circuit and for diverting applications, round the load circuit. Therefore it is possible to produce a set of curves for varying authorities to show how the basic characteristic varies in practice under constant pressure conditions, see Fig. B11.24 upper right hand quadrant.

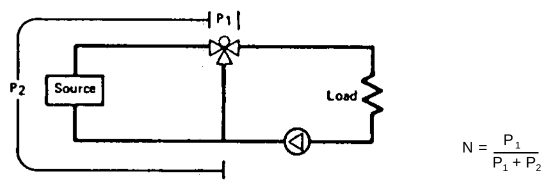



**Fig. B11.16.** Valve authority–three-port mixing application.

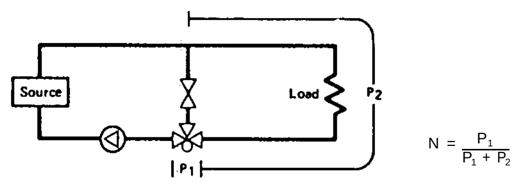



**Fig. B11.17.** Valve authority–three-port diverting application.

In the case of two-port valves the situation is more complicated, as illustrated in Figs. B11.19 to 23. The authority is defined as before from the circuit shown in Fig. B11.18, which is an unlikely practical situation, as the system pressure does not normally remain constant when the volume is reduced as the control valve is throttled. Obviously a constant head pump would provide at least a partial answer to the second point, but consider the more likely situations shown in Figs. B11.19 and 20 each of which shows multiple two-port control valves, Fig. B11.19 with a ladder connection arrangement and Fig. B11.20 with a reverse return system. For simplicity, assume that the design loads in each of the legs are equal so that each requires the same flow rate and presumably the same control valve. The authority for each valve in both figures is defined by equation B11.8.

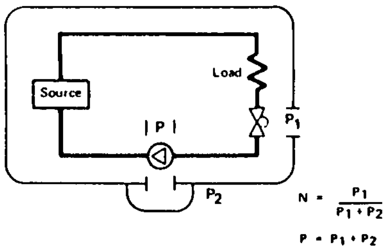



**Fig. B11.18.** Valve authority–basic two-port application.

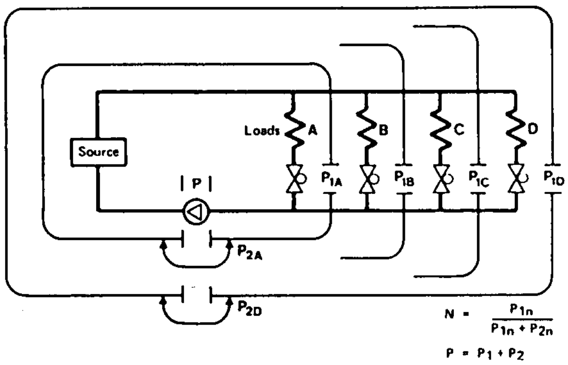



**Fig. B11.19.** Valve authority–two-port application (ladder arrangement).

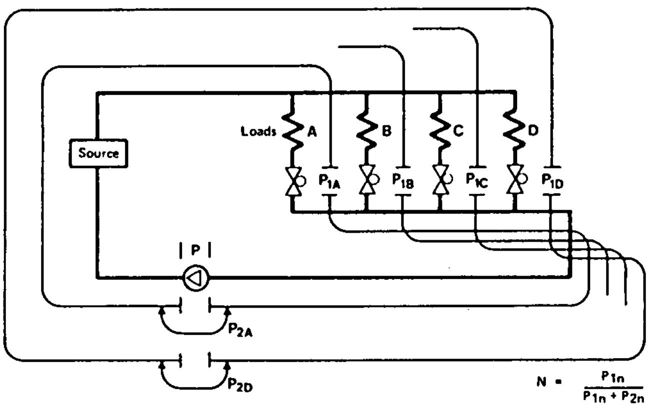



**Fig. B11.20.** Valve authority-two-port application (reverse return arrangement).

The total pressure drop developed is given by:

$$P_1 = P_1 + P_2 \qquad \text{.. .. .. .. ..} \qquad \text{B11.9}$$

where:

$P_1$ = total pressure drop .. .. .. Pa

Specimen figures, including valve authorities, are added to the diagrams as shown in Figs. B11.21 and 22. These figures are based on a pump developing 100 units of pressure with each load requiring 10 units at design flow.

In Fig. B11.22, each of the control valves would be open by a different amount to provide the pressure drops shown. This would also mean that that valves would have to control the load over a restricted portion of the overall valve lift, whereas for the arrangement shown in Fig. B11.23 the authorities are equal but low.

These figures indicate some of the difficulties in selecting suitable valves in the two-port control systems. In Figs. B11.20, 22 and 23 consistent valve authorities for similar loads are achieved, and the valves may all operate over their full range, but they are all low and to reach suitable levels the pressure drop across the control valves would need to be increased considerably with a consequential rise in pump head and energy consumption.

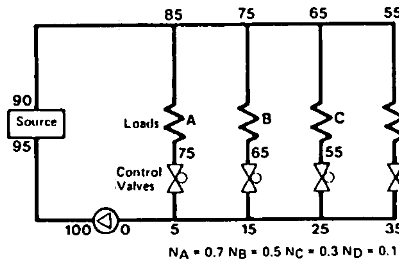
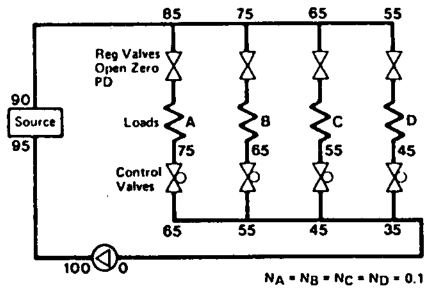



**Fig. B11.21.** Ladder arrangement–typical pressure drops indicated.




Regulating valves assumed to be fully open with zero pressure drop.

**Fig. B11.22.** Reverse return arrangement with regulating valves–typical pressure drops indicated.

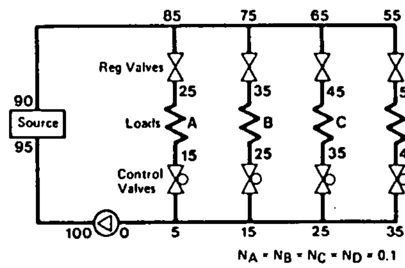



**Fig. B11.23.** Ladder arrangement with regulating valves–typical pressure drops indicated.

Fig. B11.24 illustrates the importance of suitable valve authority. The upper right hand quadrant of the diagram shows a stylised set of valve authority curves for both linear and equal percentage valves while the lower right hand quadrant depicts a typical heater or cooler battery characteristic. The relationship between valve lift and power (heating or cooling output) is shown in the bottom left hand quadrant and this is a function generated from a valve authority curve and the battery output characteristic. Using curve 2 (equal percentage characteristic $N$ = 0.7) the points on the new curve are defined by drawing a series of lines $a_1$ $b_1$, $a_1$ $c_1$, $a_2$ $b_2$, $a_2$ $c_2$, etc. and then drawing lines vertically through $b_1$, $b_2$, etc. and horizontally through $c_1$, $c_2$, etc. The points of intersection $d_1$, $d_2$, etc. define the power output/valve lift curve. This particular curve shows an almost linear relationship between lift and output, but for comparison the power/lift curve is also shown based on curve E (linear characteristic $N$ = 0.8). The latter illustrates why linear valves are not recommended for this type of application.

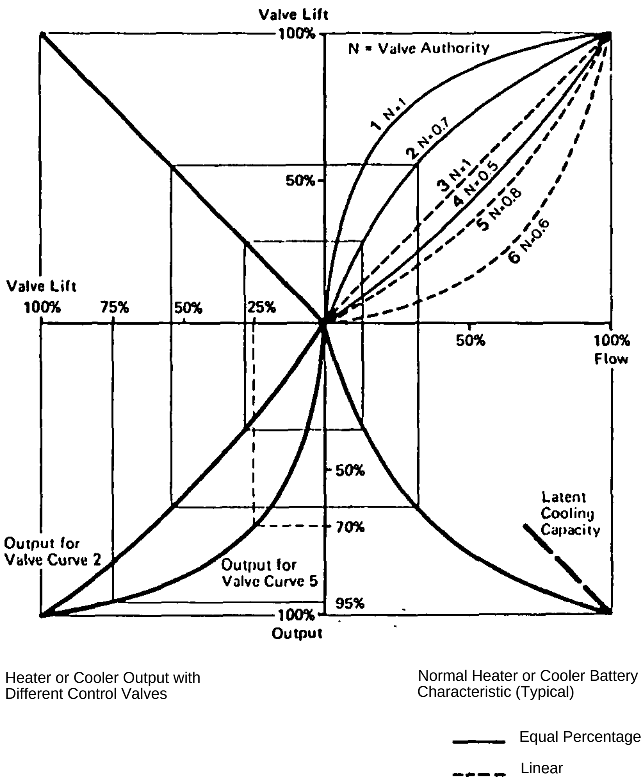



**Fig. B11.24.** Stylised valve authority and power output characteristics.

Note that the latent cooling characteristic of the cooler battery does not follow the same curve as the total output as this may be relevant to applications where latent cooling is paramount.

*Flow Coefficients (Capacity Indices)*

The pressure drop across given control valves is a function of the flow coefficients, or capacity indices, which are published by individual manufacturers. These coefficients represent the quantity of fluid which passes through a fully open control valve at unit pressure drop.

In SI units a flow coefficient $A_v$, based on the quantity of fluid in $m^3/s$ for a pressure drop of 1 Pa across the valve, has been defined in BS4740, Part 1, 1971, to replace the existing coefficients. In practice manufacturers' figures are only published as $C_v$ and $K_v$ values. The traditional imperial valve capacity coefficient, $C_v$, is defined from:

$$V_i = C_v \left( \frac{P_i}{S_f} \right)^{0.5} \quad .. \quad .. \quad .. \quad .. \quad \text{B11.10}$$

where:

$V_i$ = volume flow rate (imperial units) .. gal/min
$C_v$ = valve coefficient (imperial units)
$P_i$ = pressure drop across valve (imperial units) .. .. .. .. .. psi
$S_f$ = relative density (specific gravity) of fluid

However, BS4740 defines a metric valve capacity coefficient in different terms:

$$V_m = A_v \left( \frac{P_m}{r_{fm}} \right)^{0.5} \quad .. \quad .. \quad .. \quad .. \quad \text{B1l.11}$$

where:

$V_m$ = volume flow rate (metric units) .. $m^3/s$
$A_v$ = valve coefficient (metric units)
$P_m$ = pressure drop across valve (metric units) .. .. .. .. .. Pa
$r_{fm}$ = density of fluid (metric units).. .. $kg/m^3$

Dimensional analysis shows that the two coefficients are not comparable. However, an effective conversion factor can be derived thus:

$$A_v = 28.8 \times 10^{-6}\ C_v \text{ (UK)}$$

Other conversions can be calculated similarly and give:

$$A_v = 24.0 \times 10^{-6}\ C_v \text{ (USA)}$$

$$A_v = 28.0 \times 10^{-6}\ K_v$$

The use of valve capacity coefficients is generally of use only in the calculation of pressure drops across valves and they may in future be replaced by velocity head factors, making resistance calculations for valves consistent with those for pipework and fittings. The relationship can be simply calculated from equation B11.11 using data given in Section C4 of the *Guide.*

Using pressure loss factors, the pressure drop across a fitting is:

$$P_f = \zeta \times l_e \times DP_l \quad .. \quad .. \quad .. \quad .. \quad \text{B11.12}$$

where:

$P_f$ = pressure drop across fitting .. .. Pa
$z$ = pressure loss factor
$l_e$ = equivalent length of pipework .. m
$DP_l$ = pressure drop per unit length .. Pa/m

The equivalent length is given by:

$$l_e = \frac{0.81\ r_{fm}\ V_m^2}{D\ P_l\ d^4} \quad .. \quad .. \quad .. \quad .. \quad \text{B11.13}$$

where:

$d$ = diameter of pipe .. .. .. m

Hence, from equations B11.11, 12 and 13:

$$z = \left( \frac{d^2}{0.9\ A_v} \right)^2 \quad .. \quad .. \quad .. \quad .. \quad \text{B11.14}$$

On this basis Table B11.2 compares velocity head factors with published $C_v$ factors for two types of valves.

**Table B11.2.** Comparison of velocity head factors and $C_v$ values

| Type | Diameter*/mm | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | 25 | | 40 | | 50 | | 80 | | 100 | |
| | $C_v$ | [illegible] | $C_v$ | [illegible] | $C_v$ | [illegible] | $C_v$ | [illegible] | $C_v$ | [illegible] |
| Globe | 8 | 13 | 22 | 9 | 33 | 11 | 82 | 9 | 120 | 13 |
| Butterfly | – | – | 100 | 0.46 | 180 | 0.36 | 470 | 0.29 | 820 | 0.27 |

*The diameters taken are the internal diameters for BS 1387 medium steel pipe.

*Rangeability*

This valve parameter is defined as the ratio of the maximum controllable flow to the minimum controllable flow as shown in Fig. B11.25 with the maximum figure taken at the full open position. Thus:

$$R = \frac{f_{max}}{f_{min}} \qquad \text{B11.15}$$

where:

$R$ = rangeability
$f_{max}$ = maximum flow rate .. m$^3$/s
$f_{min}$ = minimum controllable flow rate .. m$^3$/s

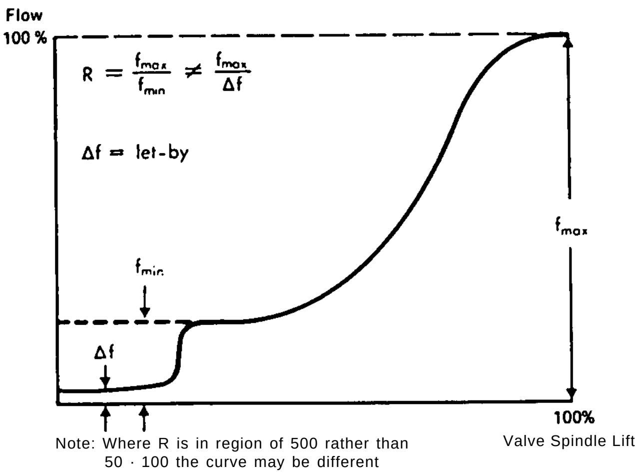



**Fig. B11.25.** Valve rangeability, R.

A rangeability of 50 indicates a valve which will control to its defined characteristic down to 2% of its maximum flow. Care should be taken to ensure that the rangeability is suitable for control at low load conditions, particularly in view of the heat transfer characteristics of some heat exchangers at low flow rates. Valves with rangeabilities above 40 should be the aim and some are available with published figures in excess of 100.

*Combination of Criteria*

The use of a suitable flow coefficient determines the pressure drop and provides a valve authority which, when combined with the basic valve characteristic and heat exchanger output curve, should produce a power linear characteristic, i.e. linear variation of power output with valve spindle movement. An inappropriate combination of these criteria may result in a large change in power output for a small variation in spindle lift, see Fig. B11.24, curve E.

This can lead to a situation where stability is achieved during commissioning, because the valve is operating near the open position, whereas the system may be unstable when the valve is nearly closed. It is possible, given the basic valve characteristic equation, the valve flow coefficients and the heat exchanger output curve and pressure drop, to calculate which of the available valves will provide a suitable power output characteristic. However, it is usually sufficient to obey the simple rules of the *Valve Selection Check List* given later.

*Three-Port Valves*

These valves may be used with two inlet ports and one outlet port, described as mixing valves or, in some cases, with one inlet port and two outlet ports, described as diverting valves. In both cases they may be installed for mixing applications (constant volume variable temperature, e.g. compensator circuits), or diverting applications (variable volume constant temperature circuits, e.g. heater batteries). Figs. B11.26 and 27 show mixing valves used for mixing and diverting applications respectively while Figs. B11.28 and 29 show diverting valves for the same applications respectively. It is recommended that valves are installed as shown in Figs. B11.26 and 27

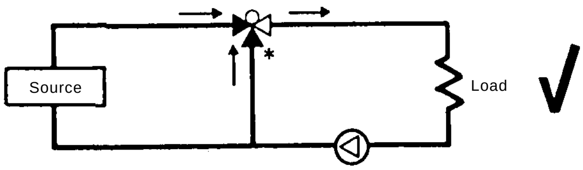



**Fig. B11.26.** Three-port mixing valve–mixing applications. (Isolating valves not shown)

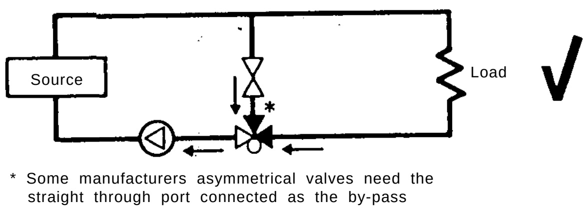



**Fig. B11.27.** Three-port mixing valve–diverting applications. (Isolating valves not shown)

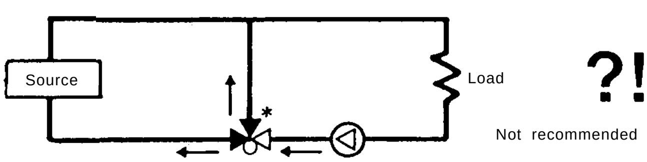



**Fig. B11.28.** Three-port diverting valve–mixing applications. (Isolating valves not shown)

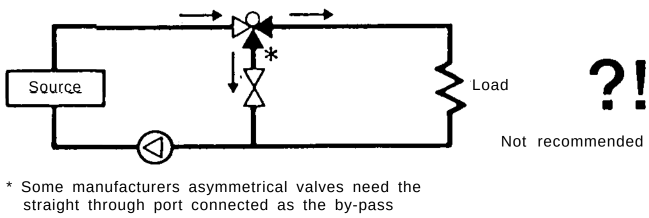



**Fig. B11.29.** Three-port diverting valve–diverting applications. (Isolating valves not shown) because the majority of valves available in the building services industry can be used as mixing valves but not as diverting valves. To ensure the correct operation of control valves, regulating valves should be installed at appropriate points in the pipework circuits as shown in the figures, but for simplicity the usual isolating valves are not shown.

In the application of three-port valves there are two basic requirements–operation under constant pressure conditions and maintenance of constant total volume through the valve. The former allows the authority to be defined as shown in Fig. B11.17, where ($P_1$+$P_2$) remains constant as viewed from the pump, to provide the power linear output from the heat exchanger. Where total constant flow is required, a further design parameter associated with three-port valves must be considered, i.e. the symmetry of the internal ports. Symmetrical design means that both valve inlet ports, for mixing valves, have the same characteristic and either inlet may be connected to the load. However, with this form of construction it is not possible to produce a valve with a total constant volume characteristic. An asymmetric valve has its inner ports designed such that the port connected to the load provides the correct operating characteristic while the port connected to the bypass is designed to maintain a constant total flow through the valve independent of valve opening. This means that the ports are not interchangeable as is the case with symmetrical valves. The basic curves for three port valves with symmetrical and asymmetrical characteristics are shown in Figs. B11.30 and 31. The curves clearly show the different bypass and total flow characteristics.

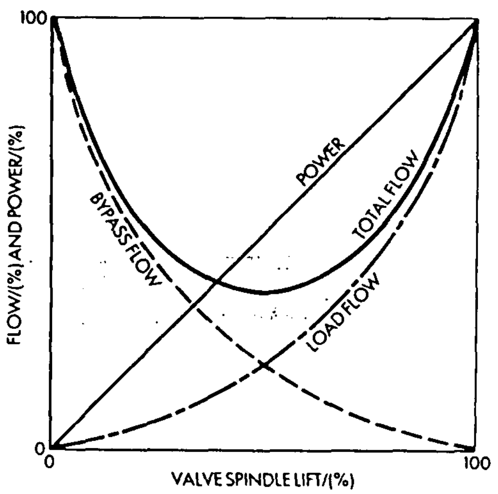



**Fig. B11.30.** Curves for symmetrical three-port valve selected for power linear output.

The twin requirements of power linearity and constant total flow are to some extent incompatible and there has to be some compromise in valve design to meet the requirements of the specific application. The *Valve Selection Check List* will assist in achieving this compromise solution.

*Two-Port Valves*
These valves may appear simpler to use than three-port valves. Their selection follows the same rules and they can be used in suitable circumstances to provide identical power linear characteristics. The major advantages claimed for two-port modulating valves as compared to three-port alternatives, are cheaper capital cost and

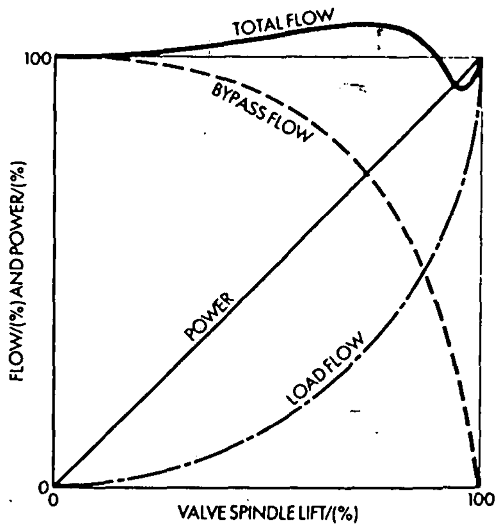



**Fig. B11.31.** Curves for asymmetrical three-port valve selected for power linear output.

reduced pumping costs due to reduced flow under part load conditions. However, there are additional capital costs implicit in their use, related to the pumping system required to satisfy the variable flow requirements.

It is necessary to provide a variable flow pumping system and pressure control facilities to maintain approximately constant pressure conditions for each loop. This requires either variable flow pumps or controlled bypasses around the constant volume pumps or across various parts of the circuit.

One further point in considering these valves is the maximum differential pressure (MDP) rating. Although both three-port and two-port valves are sized on the basis of their pressure drop at maximum design flow, three-port valves nominally have the same pressure drop for all positions of the valve and, provided that the pressure drop at maximum design flow is less than the MDP rating, no further consideration need to be given to this criterion. However, for two-port valves, the maximum pump head occurs across the valve when it is closed and both this head and the pressure drop at maximum design flow must be less than the MDP rating. If not, the valve will be incapable of opening (or closing), depending on the application. These comments also refer to on/off valves except that power linearity is not a requirement.

Two-port valves should only be used in systems where these qualifications have been recognised otherwise system stability will be uncertain and commissioning may be virtually impossible.

*Steam Valves*
Valves used for steam are a specialist application of two-port valves for compressible fluid flow. The general formula for the flow of steam through an orifice is:

$$M = Ca\,(P_u - P_d)^{\frac{1}{2}} \quad \ldots \quad \text{B11.16}$$

where:

| | | | |
|---|---|---|---|
| $M$ | = mass flow of steam | .. | kg/s |
| $a$ | = area of orifice | .. | $m^2$ |
| $P_u$ | = upstream steam pressure | .. | Pa |
| $P_d$ | = downstream steam pressure | .. | Pa |
| $C$ | = a constant for given pressure conditions and type of orifice. | | |

The function of an automatic control valve is to vary the area of the orifice in order to vary the quantity of steam passing through. However, reduction of the area increases steam velocity which counteracts the decrease in volume of steam through the orifice. A maximum 'critical' steam velocity is reached when, for dry steam, the downstream absolute pressure $P_d$ is about 60% of the inlet pressure $P_u$. Thus, in sizing steam valves, the valve pressure drop may be assumed to be approximately 40% of the absolute pressure at full load, immediately upstream of the valve and, using this figure and the given duty, the valve may be sized from manufacturers' tables. If the inlet pressure is below 100 kPa, applying the above 40% rule gives impracticable conditions downstream and, in such cases only, it is necessary to assume a smaller pressure drop, ignoring any possible loss in degree of control. Guidance on these reduced pressure drops is normally given in manufacturers' literature. Additional pressure drops due to pipe runs, isolating valves, etc. mean that control valves must be sized on their inlet pressure at full load and not on the boiler pressure.

It is also important to ensure that the heat exchange surface is adequate for the full load intended when the pressure at the inlet of the heat exchanger is 60% of the absolute pressure at the control valve inlet (less any further losses in pipework and fittings).

Care must be taken to ensure that suitable steam traps are incorporated (see Section B16 of the *Guide*). Failure to do so will result in unacceptably poor control under light load conditions.

*Butterfly Valves*
Although basically a special type of two-port valve, their applications are markedly different from those of conventional two-port valves. Because of their construction they offer little restriction to fluid flow when in the fully open position, but the basic characteristic severely restricts their use in modulating applications.

Normally they are used for on/off or quick opening applications and are often applied as boiler or chiller isolating valves where automatic sequencing of multiple sources is employed. They are suitable for any on/off control application where tight shut-off is not an absolute requirement, as they usually have a let-by of approximately 1% of full flow rate. Special versions can be obtained with liners to permit tight shut-off, but the costs increase appreciably.

*Noise and Cavitation*
The problems of noise and cavitation in control valves may be caused by three basic means:

(*a*) Mechanical vibration, causing noise from plug and guide or resonating trim parts.
(*b*) Hydrodynamic noise, caused by cavitation in the liquid.
(*c*) Aerodynamic noise, caused by turbulent gas or vapour flow.

The most likely source of noise in valves used for incompressible fluids is cavitation. This is caused by the increased velocity of the fluid through the restricted area orifice in the internal porting of the valve. The velocity increase causes the static pressure of the fluid to decrease to a point where the pressure at the orifice may fall below the vapour pressure of the liquid. This causes bubbles to form, which is the onset of cavitation. As the velocity decreases downstream of the valve port, the static pressure rises above the vapour pressure of the fluid and the bubbles or voids will collapse, which is the end of the cavitation phase. Cavitation restricts flow through the valve and, in extreme cases, the implosion of these voids can damage the valve. Minor cavitation effects may be acceptable but an upper limit may be defined in genera1 terms as:

$$P_{max} = K_m (P_{in}-P_V) \quad \text{B11.17}$$

where:

$P_{max}$ = maximum allowable pressure drop (valve fully open) .. kPa
$K_m$ = valve recovery coefficient
$P_{in}$ = absolute inlet pressure .. kPa
$P_v$ = vapour pressure at fluid inlet temperature .. kPa

Values for $P_v$ for water are given in Table B11.3. Values of $K_m$ vary according to valve construction, spindle position and flow coefficient but for valves commonly used, $K_m$ is between 0.6 and 0.885.

For example, take a valve operating with water at 90°C, an inlet pressure of 250 kPa and $K_m$ of 0.6. From Table B11.3, $P_v$ is 70.1 kPa, whence, from equation B11.17:

$$P_{max} = 109 \text{ kPa}$$

For pressure drops across valves of less than 70 kPa no problems are likely to be encountered but valves used for pressure relief and bypass temperature control applications are most likely to suffer from the effects of cavitation. Where there is any doubt, specific figures for $K_m$ should be obtained from the valve manufacturer.

**Table B11.3.** Vapour pressure of water, $P_v$

| Water temperature/°C | Vapour pressure/kPa | Water temperature/°C | Vapour pressure/kPa |
|---|---|---|---|
| 10 | 1.2 | 110 | 143 |
| 20 | 2.3 | 120 | 199 |
| 30 | 4.2 | 130 | 270 |
| 40 | 7.4 | 140 | 361 |
| 50 | 12.3 | 150 | 476 |
| 60 | 19.9 | 160 | 618 |
| 70 | 31.1 | 170 | 792 |
| 80 | 47.3 | 180 | 1003 |
| 90 | 70.1 | 190 | 1255 |
| 100 | 101 | 200 | 1555 |

**Valve Selection Check List**
On water circuits decide whether two- or three-port valves are the most suitable. When selecting valves for on/off operation the following should be considered:

(*a*) Is the valve body rating suitable for the temperature and pressure of the fluid system? (Remember that the pressure may be the sum of static and dynamic (pump) head).

(*b*) Ensure that the valve passes the quantity specified at a pressure drop within the maximum differential pressure rating of the valve. (Remember that two-port valves may also have to open or close against the full pump head).
(*c*) Check that there is sufficient pump head capacity available to provide the pressure drop across the valve at the specific duty.
(*d*) Specify any special conditions regarding leakage limits or tight shut-off.

When selecting the valves for modulating modes of heat transfer control:

(*a*) Apply the above rule for on/off valves.
(*b*) Select an equal percentage valve characteristic in preference to linear or other characteristics.
(*c*) Ensure that the pressure drops through heat exchangers and associated pipework are known, or stipulated, before control valves are selected.
(*d*) Keep the valve authority as high as possible, not less than 0.5 for diverting applications and not less than 0.3 for mixing applications. (Requires the use of flow coefficients for calculating the pressure drop through the valve in the fully open position).
(*e*) Where possible use the heat transfer curves of flow against output for a heat exchanger to check possible anomalies.
(*f*) Ensure that the rangeability of the selected valves permits stable control under low load conditions.
(*g*) Three-port valves with asymmetric port characteristics should be used to maintain a constant total flow condition.

For steam circuits:

(*a*) Ensure that the valve body is suitable for the temperature and pressure of the steam.
(*b*) Size the valve bearing in mind that the quantity of steam will not vary until the pressure drop across the valve is approximately 40% of the absolute pressure available.
(*c*) Ensure that the heat exchanger downstream of the valve is sized using the outlet pressure from the valve.
(*d*) Check that the valve will open (or close) against the available steam pressure (largely a function of actuator size).
(*e*) Note any special requirements regarding leakage limits or tight shut off. (Some form of tight shut off will normally be required, e.g. separate back-up valve).
(*f*) Valve characteristics of any type may be used for on/off applications but valves with linear characteristics should be used for modulating applications since heat transfer is closely related to mass flow.
(*g*) Ensure that adequate steam trapping is provided.

**Control Dampers**

The following details and data on dampers and their basic application are intended to illustrate the effect of suitable selection on the stability of air flow for control purposes. Certain assumptions have been used to simplify the techniques. Where detailed selection becomes necessary a more rigorous procedure should be adopted. The flow of air in an air handling system is regulated by control dampers in a similar fashion to the regulation of water by valves. Control may be either two-position, as in the case of an isolating damper, or modulating, where two air streams are to be mixed, as in face and bypass control. Regulating or balancing dampers are not dealt with here.

*Characteristics*

There are two arrangements of rotating blade dampers: parallel and opposed. The relationships between damper blade rotation and flow through the damper, for a constant pressure drop across the damper, are shown in Fig. B11.32.

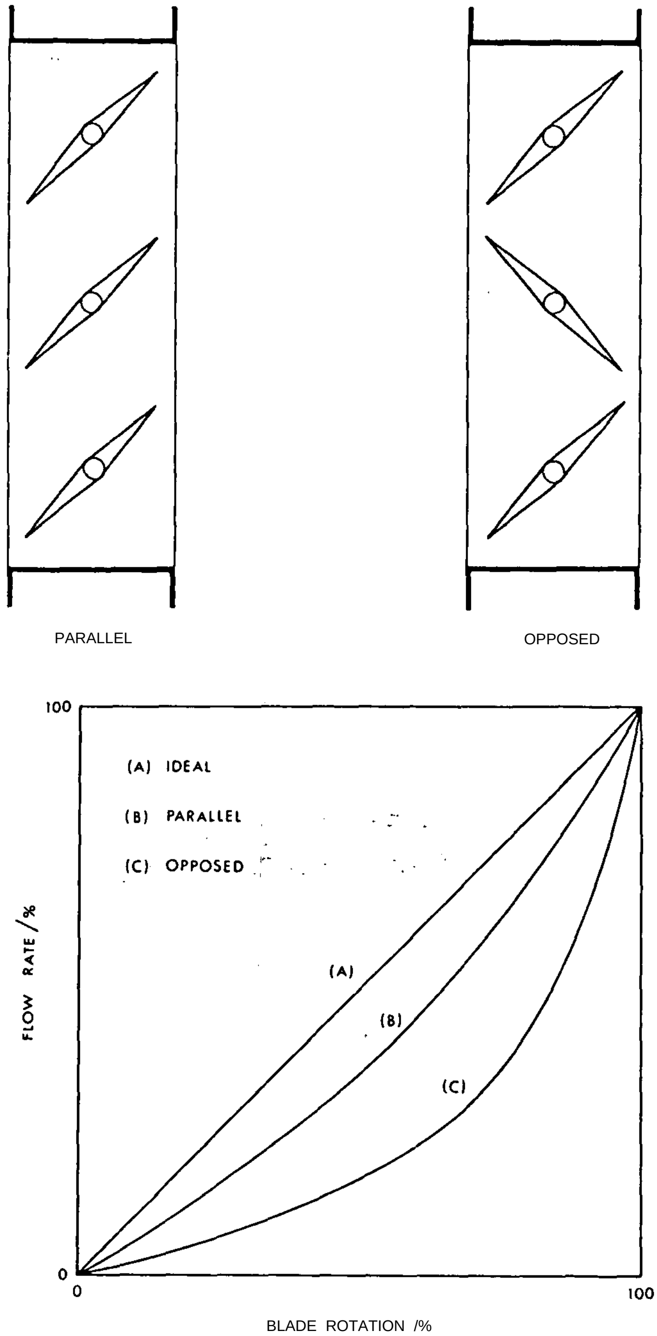


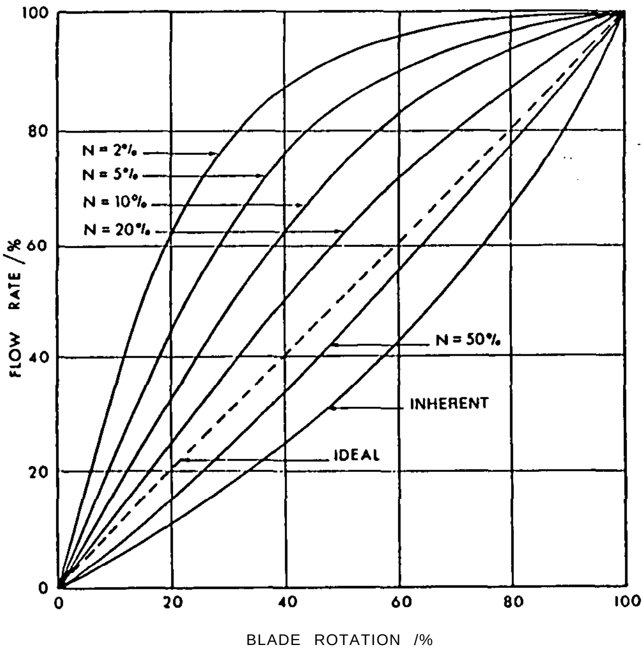

**Fig. B11.32.** Damper types and inherent characteristics.

These are called the inherent characteristics of the damper. Ideally these would be straight lines but in practice they take the form shown. In practice also, the pressure drop across the damper is not constant but increases as the damper is closed. Consequently the inherent characteristics are modified to the installed characteristics, see Fig. B11.33. This modification is a function of the damper authority which is defined in the same way as valve authority:

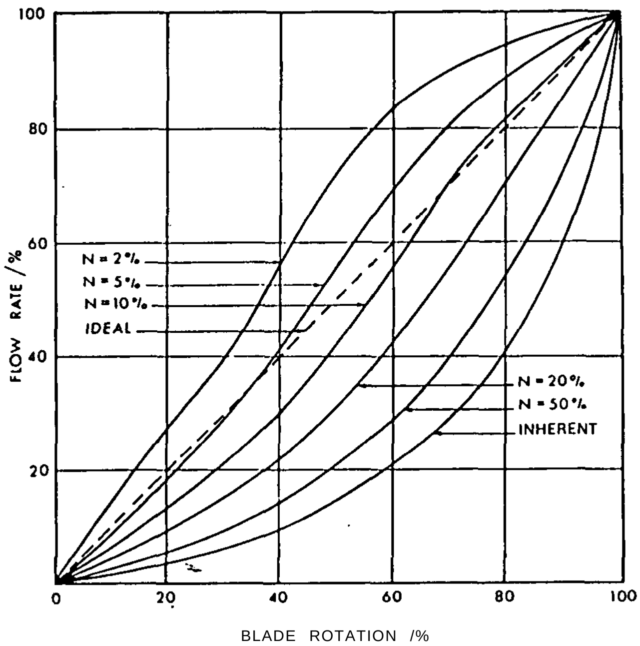
$$N = \frac{P_1}{P_1 + P_2} \quad \text{.. .. .. .. ..} \quad \text{B11.18}$$

where:

$N$ = damper authority
$P_1$ = pressure drop across damper .. .. Pa
$P_2$ = pressure drop across rest of system .. Pa




(a) Parallel blade damper.




(b) Opposed blade damper.

**Fig. B11.33.** Installed damper characteristics.

It should be remembered that ($P_1 + P_2 = P_1$ is constant, see Fig. B11.34.

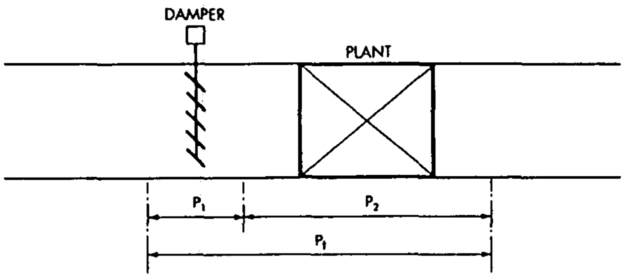



**Fig. B11.34.** Example system.

From Fig. B11.33 it can be seen that for an opposed blade damper the closest approach to linearity occurs with $N$ = 5% while for a parallel blade damper it occurs when $N$ = 20%. This means that for the same total pressure drop across damper and system, the pressure drop across a parallel blade damper will be four times that across an opposed blade damper. It is recommended, therefore, that opposed blade dampers are used whenever possible.

The pressure drop across a fitting can be expressed in terms of a pressure loss factor for the fitting:

$$P_f = \tfrac{1}{2}\, r v^2 \varsigma \quad \text{.. .. .. .. ..} \quad \text{B11.19}$$

where:

$P_f$ = pressure drop across fitting .. .. Pa
$r$ = density .. .. .. .. .. kg/m$^3$
$v$ = velocity .. .. .. .. .. m/s
$\varsigma$ = pressure loss factor

Where the velocity pressure stays approximately constant along the section under consideration, then the velocity pressure loss factors can be used to calculate the damper authority. The air volume flow rate is derived from the velocity and cross-sectional area of the duct:

$$V = Av \quad \text{.. .. .. .. ..} \quad \text{B11.20}$$

where:

$V$ = volume flow rate .. .. .. m$^3$/s
$A$ = cross-sectional area of duct .. .. m$^2$

Combining equations B11.19 and 20 gives:

$$V = K(Pf)^{1/2} \quad \text{.. .. .. .. ..} \quad \text{B11.21}$$

where:

$$K = \left(\frac{2A}{r\varsigma}\right)^{1/2} \quad \text{.. .. .. .. ..} \quad \text{B11.22}$$

Equations B11.18 and 21 can be evaluated at a partially open position of the damper and compared with the inherent characteristics at the same position. This gives an expression for the authority in terms of the installed and corresponding inherent characteristics:

$$N = \frac{(1/F_a)^2 - 1}{(1/F_i)^2 - 1} \quad \text{.. .. .. ..} \quad \text{B11.23}$$

where:

$F_a$ = ratio of actual air flow at partially open position to air flow at fully open position
$F_i$ = ratio of theoretical air flow at partially open position (inherent characteristic) to air flow at fully open position.

Since the idea of selecting the damper authority is to linearise the installed damper characteristics from the inherent characteristic a good first approximation is to look for 50% air flow at 50% actuator travel. Fig. B11.35 illustrates a typical inherent characteristic which may be used as an example. At the 50% air flow rate, the inherent characteristic gives $F_i$ = 0.19 (flow rate of 19%) at 50% travel, hence substituting in equation B11.23 gives:

$$N = \frac{\left(\frac{1}{0.5}\right)^2 - 1}{\left(\frac{1}{0.19}\right)^2 - 1}$$

$N$ = 0.112, or 11.2%

Rearranging equation B11.23 gives:

$$F_a = \frac{(1/N)^{½}}{[(1/N) - 1 - (1/F_i)^2]^{½}}$$

This enables the installed characteristics to be plotted from the authority and the inherent characteristics.

*Applications - Mixing Control*
Control dampers are most commonly used in mixing applications as shown in Fig. B11.35. Each damper in the system should be considered in its own sub-system and in conjunction with the characteristics of the other dampers. The fresh air, exhaust air and return air dampers should be considered individually. The fresh air dampers should be sized with the following considerations:

(*a*) The velocity of the air through the damper should prevent the entrainment of rain and foreign bodies.
(*b*) The velocity should be such that noise levels are acceptable.

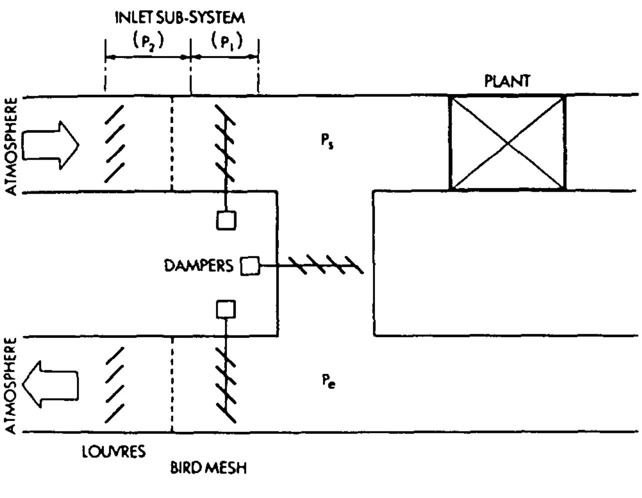



**Fig. B11.35.** Mixing damper system.

(*c*) The authority and characteristic should be suitable for the application.

From Fig. B11.33(b) it can be seen that an opposed blade damper should have an authority of from 5% to 8% for linearity. In order to consider the relative merits of aerofoil or pressed blade dampers, a quick calculation can be performed using equation B11.18 and some pressure loss factors from Section C4 of the *Guide* are given below. It is assumed that the dampers are the same size as the duct.

| | |
|---|---|
| Aerofoil damper: | $\zeta$ = 0.2 |
| Pressed blade dampers: | $\zeta$ = 0.5 |
| Outdoor louvre: | $\zeta$ = 1.3 |
| Bird mesh: | $\zeta$ = 0.4 |
| Duct (assumed): | $\zeta$ = 0.6 |

Equations B11.18 and 19 may be transposed to give:

$$N = \frac{\zeta_d}{\zeta_d + \zeta_s}$$

where:

$\zeta_d$ = pressure loss factor for dampers
$\zeta_s$ = pressure loss factor for remainder of system

Hence, the pressed blade damper has an authority:

$$N = \frac{0.5}{0.5 + 1.3 + 0.4 + 0.6} = 0.18, \text{ or } 18\%$$

and the aerofoil damper has an authority:

$$N = \frac{0.2}{0.2 + 1.3 + 0.4 + 0.6} = 0.08, \text{ or } 8\%$$

This indicates that the fresh air damper of duct size should have an aerofoil section where possible. However, care should be taken in all cases to check the authority and not to assume that duct sizing will be satisfactory in every case.

When the exhaust damper of a system is wide open, the pressure drop across the exhaust duct and damper is $P_e$ as shown in Fig. B11.35. Part of this pressure is lost across the louvre, bird mesh and ductwork and the exhaust damper should be sized to absorb the rest of the available pressure. This will probably result in an opposed blade damper of duct size which should ideally have a linear characteristic.

The return air damper should be sized to produce an absolute pressure drop of ($P_e$ – $P_s$) minus the duct loss. It should also have a characteristic (not necessarily linear) that complements the fresh air damper characteristic to produce a constant total flow condition through the fan.

The return air damper will probably have a high authority, ranging from, say, 40% to 100% when the damper is fitted into the wall of a plenum chamber. The characteristic curves for opposed and parallel bladed dampers show that the parallel blade damper has a better characteristic at high authority and, as the damper plus ductwork has to drop a pressure of ($P_e$ - $P_s$), there is no penalty incurred in energy terms by using a parallel blade damper. The authority of the damper can be reduced by adding resistance in the form of a grid or mesh in series with the return air damper or by arranging the ductwork to have a specific resistance.

**Silicon Controlled Rectifiers**

The use of silicon controlled rectifiers (SCR) for control purposes in building services systems is most commonly applied to:

(*a*) Electric heaters.
(*b*) Variable speed motors.
(*c*) Lighting systems.

The two types of SCR which are presently used for power control are thyristors and triacs.

*Thyristor*

This is effectively a semi-conductor diode with an additional connection called the gate. Applying an appropriate momentary signal to the gate allows the diode to conduct in its normal direction until the current fails to almost zero. This occurs 100 times per second on a normal 50 Hz electrical supply and therefore the thyristor has to be re-triggered at the same rate to achieve control. The point in the cycle at which triggering occurs determines the amount of current allowed to pass. The thyristor can conduct in one direction only and therefore it has to be duplicated and connected in inverse parallel for control of alternating current devices.

*Triac*

The triac is a dual polarity device with a single control gate which operates in the same way as the inverse parallel connected thyristors. Triacs are generally used for currents of less than 30A.

*Applications*

Control of power is achieved by phase angle firing, full cycle control or synchronised burst firing. Phase angle firing, see Fig. B11.36, is the most versatile method of power control but it can cause radio and TV interference which restricts its application. As the trigger point is advanced to an earlier part of the half cycle, the mean current, and therefore power, delivered to the load is increased. When used for speed control, it is important to ensure that the motor is suitable. Arcing may be troublesome on commutator motors and rotors may overheat on induction motors.

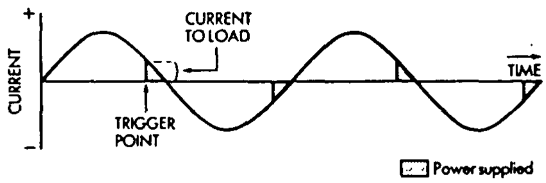



**Fig. B11.36.** Phase angle firing.

Full cycle control, see Fig. B11.37, does not cause interference since switching occurs when the voltage is at zero. This system should only be used on heaters and tungsten filament lighting. At half power, as shown in Fig. B11.37, the thyristors are triggered for one cycle in two, for quarter power they are triggered for one cycle in four. This gives a form of step control more than adequate for most heating systems but which may cause flicker when used for lighting control. Synchronised burst firing also relies on switching when at zero voltage. In this case, the load is switched on a low repetition rate with the output averaged over a short period. This method of control is suitable for use on electrical heating appliances.

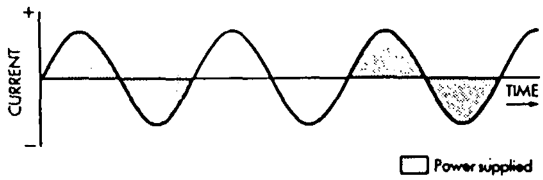



**Fig. B11.37.** Full cycle firing.

*Installation*

In operation, an SCR produces heat proportional to the connected load current, about 2 W dissipated for each amp of connected load. This heat must be dissipated by a heat sink which will generally be designed to operate in air at up to 40°C. Large units may need to be force cooled or water cooled. High rupturing capacity fuse links will not protect an SCR device unless it is very conservatively rated and manufacturers should be asked to fit fast-acting semi-conductor fuses. It will always be necessary to use contactors to isolate the load in the event of a fault as an SCR normally fails to maximum power.

**Time Controls**

There are three basic methods of time control; on/off time switch, fixed time/boosted start, and optimum start control.

*On/Off Time Switch*

Simple time switch control is usually limited to small installations such as small fan convector and unit heater installations. This method is simple, low in cost, and easy to install. The method has the disadvantage that the start time is selected to ensure that the space is at the desired temperature at the start of occupation under the design temperature conditions. This means that in less severe conditions energy is wasted.

*Fixed Time Boosted Start*

Fixed time boosted start is designed to reduce preheat time by allowing the central plant to provide maximum preheat capacity. Facilities for an earlier start time following a prolonged shut-down (e.g. week-end) should also be provided on the time switch to supply the additional heat required to bring the building up to the correct temperature for occupation. The system should incorporate a means of terminating the boost by either time or temperature. This method is now largely superseded by optimum start control which reduces the energy used for preheat in the intermediate seasons.

*Optimum Start and Stop Control*

This is a time controller for intermittently occupied buildings which has shown substantial energy savings when compared to an on/off time switch or fixed time boosted start. It provides optimum plant start time by continuously monitoring both inside and outside temperatures to provide sufficient preheat to give the desired conditions at the beginning of the occupancy period. Early start can also be provided following prolonged shut-down, such as at week-ends. Early-on and day omission devices are normally provided, and a day extension timing facility with manual setting, can be added.

The controller takes into account the thermal response of the system, structure and duration of the off periods over a daily cycle. For maximum economy, the plant should be started at the latest possible time, allowing for the minimum preheat period at full capacity. Termination of boost may be provided by an internal detector in advance of occupation time if the internal conditions are satisfactory. This detector may also be used for frost protection.

Modern versions use adaptive software which simplifies initial setting and permits the unit to monitor and correct its own performance. During unoccupied periods, control should be based on minimum inside air temperature, to prevent freezing or condensation and separate frost protection override can be provided where required.

Some optimum start controllers include the facility for optimum stop. The early stop time is again determined by the comparison of inside and outside temperatures.

*Optimiser Characteristics*
Assume that at a fixed time near the end of the occupation period the optimiser switches off the plant, see Fig. B11.38, until either preheat starting time, or the internal temperature has fallen to a low limit value (normally 10°C), whichever occurs sooner. In the latter event, the heating plant is cycled on and off to maintain the internal temperature at the low limit value, until the next normal starting time.

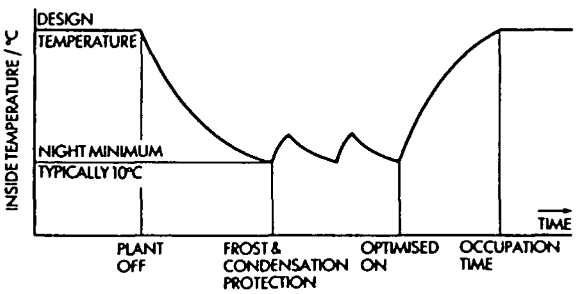



**Fig. B11.38.** Optimum start control characteristic with frost protection.

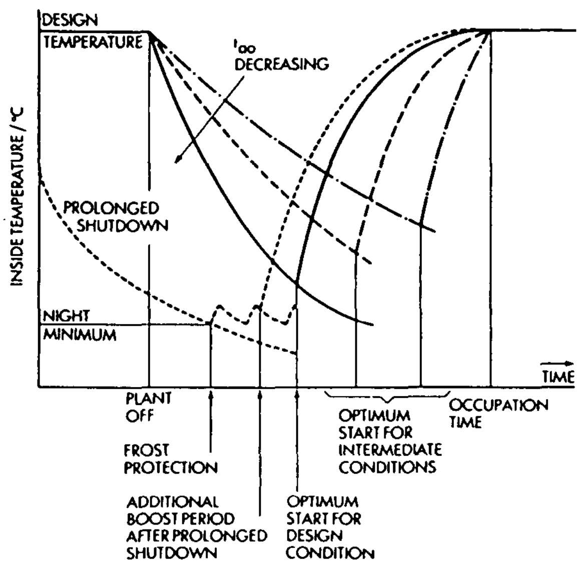



**Fig. B11.39.** Optimum start control characteristics for varying conditions.

Fig. B11.39 illustrates how the switch on time for the boiler and pumps is determined by the fall in room temperature, and how earlier switch on takes place with lower inside temperatures. It also shows that the outside temperature influences the rate at which the inside temperature drops during the off period. The outside temperature also affects the ability of the heating system to put back this heat loss and to supply the heat which is lost during the preheat period. It is therefore possible to reduce the preheat period, utilising the full effect of the additional boost capacity available at the higher outside temperatures. Fig. B11.39 also shows that after a prolonged shutdown, e.g. week-end, the building structure and contents will have dropped to a lower temperature than that following a normal night shutdown and the additional heat is replaced by an advance preheat.

**Building Automation Systems**

*General*
Most thermostatic control manufacturers now market building automation systems to monitor and control other building services such as energy, security and fire protection systems. Other manufacturers also offer such systems, particularly in the fire and security sector and there is a rapidly increasing number of large and small firms with basic processor and software capabilities who are entering the market. It is always advisable to question the level and validity of the software offered to perform the tasks required in the context of building services automation. Much of the software has still not been proven in practice and it is unwise to accept these programs until they have been demonstrated on a real project. There is often a deep gulf between software capability and the necessary knowledge of building usage patterns and building services operation to produce adequate programs.

One other area which requires emphasis is the possible susceptibility of all processor based systems to electrical interference, however caused. Interference phenomena and their suppression are not fully understood and many building automation systems suffer intermittent faults as a result of this[4,5]. Question this aspect of any proposal or specification.

The use of the term building automation system (BAS) is a generic title for all forms of data centres used in building system data collection and control of services. An energy management system (EMS) is one element of the BAS, albeit very important, and some forms of BAS operate wholly as an EMS. This is one simple example of the problems associated with the terminology used in this area of technology and great care is needed to ensure that any literature or oral communication is unambiguous. The term building management system (BMS) is also in common use and these systems usually provide additional management functions such as maintenance programming and fuel billing in addition to the EMS and/or BAS features.

The problems of communication and ambiguity lead to one other generalised statement. It is essential that the specification for any BAS is precisely detailed in terms of all performance requirements, functional sequences and clearly defined interface conditions for every facility required. Minor facets or requirements, generally accepted amongst engineers as basic to any operational sequence, will not be provided by the BAS software unless they are identified in the specification. A detailed specification of this nature will be more conducive to a satisfactory conclusion than one which concentrates on detailing the precise technology which is to be used for each element of the BAS.

*Basic Considerations*

The basis of operation of these systems is to continuously scan the connected input data points and report unusual occurrences whilst carrying out control functions automatically. The centralisation of the logic equipment allows greater flexibility of operations than conventional hard wired systems and also frees maintenance staff from watchkeeping duties, enabling more effective use of manpower.

If such a system is considered on any project there are essential prerequisites if it is to provide its optimum performance. These are:

(*a*) It must be considered very early on in the building/plant design concept. If it is added at a later stage it will duplicate a number of other elements which are complete in themselves.

(*b*) The client should be prepared to employ suitable staff to operate the centre and utilise all its facilities.

(*c*) The cost effectiveness should be properly evaluated and include the following:

- (*i*) capital cost of system, including outstations, detectors, instrumentation, modified plant, standard software and wiring, less any savings for equipment which it replaces,
- (*ii*) interest on capital cost,
- (*iii*) value of energy savings which the system itself will provide,
- (*iv*) cost of additional staff employed to operate the system, less the saving on those maintenance staff who will be replaced by the system,
- (*v*) annual cost of maintaining the building automation system,
- (*vi*) a substantial capital sum to cover the cost of collecting data from the building during the first one to three years of operation and producing software for optimising the energy use in the building, since existing software does not cover the full range of optimisation possibilities for large buildings or complexes,
- (*vii*) costs saved by using programmed maintenance co-ordinated by the system (which should include value of increased plant life) less than the cost of preparing such a programme,
- (*viii*) savings in using the software capability of the system to replace the interlocking relays, etc. normally used in motor control centres and control panels–set against the cost of the project oriented software.

This basic list of factors leads on to direct cost details and interface problems which must be considered when a BAS is contemplated. Some examples are given below:

(*a*) The functions of a building automation system could be achieved by hard wired control systems and the economic justification for a sophisticated system includes the fact that the engineering, site wiring and commissioning costs can be a fraction of the hard wired system.

(*b*) If process control is to be achieved by direct digital control (DDC) then savings will occur in the elimination of local loop controllers and in the avoidance of duplicating sensors for control and monitoring purposes.

(*c*) Energy management programs will optimise energy usage but this does not relieve the designer from his responsibilities for the correct selection of building services systems.

(*d*) Failure of power or plant may result in a direct loss to the customer, e.g. computer rooms, specified pathogen free animal buildings. If it can be shown that a building automation system minimises the number of failures and reduces the down time, this is a strong factor in the economic argument.

Building automation systems with full facilities will be more easily justified on larger projects, and offer the most savings where services are particularly complex. Options include fully owned systems and those where third parties own the central building automation equipment and sell a ‘remote’ supervision and control service.

It should be remembered that if a building automation system is not fully utilised or understood the building services control and data collection will not be adequate and it is possible to increase rather than decrease energy used, due to the system falling into disrepute.

The following notes describe the systems which most are commonly available.

*Computer Based Systems*

Transmission of information from the data collecting outstations to the central monitoring panel is via a single or two wire trunk using pulse coded messages. In some systems the scan initiation is entirely from the central processor which addresses each individual point in turn and then processes the data received. The processor may include software providing PID control of loops or it may be part of the outstation facilities. The use of software for this purpose is currently described as direct digital control (DDC) and individual conventional controllers are made redundant. Alternatively, intelligent outstations are provided, whereby the outstations address the processor on alarm occurrence. A variation is the use of microprocessors at various locations which contain all the intelligence needed for local supervision and control, including DDC, hosted by a central computer having overall control. This is particularly suitable for larger projects possibly involving several buildings where more conventional transmission systems become overloaded. Data transmission rates are, typically, 300 to 9600 baud.

Alarms for analogue devices may be set and programmed at the central processor and/or the outstations, for each of the separate sensors. These alarms can normally be inhibited by means of software when the particular plant is off, for the period following start-up and when they sequentially follow on from an initial alarm. Many computer based systems allow various control and energy management programs, as detailed later, thus dispensing with the need for conventional items of equipment, instrumentation or controls. Ultimately, they will reduce control panels to little more than starter boards, where control relays are superseded by software. Such a system permits the logic to be changed without extensive wiring modifications but this is not intended to encourage delayed design decisions.

Planned and programmed maintenance schedules can also be produced and monitored by the computer based systems but this normally requires a large memory storage capability and considerable project orientated input data.

*Hardware*

(*a*) *Data Inputs*
These fall into four basic groups:
- (*i*) *Digital:* two position switching inputs for indication of status or alarm. The input device may be thermostat or pressure switch contacts, or clean relay contacts.
- (*ii*) *Analogue:* measuring input for indication of values (temperature, pressure, etc.). Signals may be variations in voltage or current or in the form of a pulsed signal which is ‘cleaned up’ and translated at the data collecting outstation for onward transmission.
- (*iii*) *Control:* switching of drives through low voltage interposing relays sited either at the data collecting outstations or at the starter panel.
- (*iv*) *CPA:* control point adjustment of local controllers via potentiometer, gradual switch, or digital indicator. Feedback of the controlled variable is desirable.

(*b*) *Outstations*
These may be relatively simple data processors or intelligent with sufficient software capability to maintain control of remote systems. The question of terminology is again relevant because outstations are given different titles, dependent on the supplier. The data processing systems assemble the various data input signals and translate them into a suitable language for transmission and for acceptance by the central processor.

(*c*) *Transmission System*
This is the system which connects the data collecting outstations to the central panel. It is a common requirement for fire and security systems that the transmission system is duplicated via different routes and alternately scanned. Telephone lines may be leased for use for data transmission and are normally the only permitted form of transmission where outstations are located on different sites.

(*d*) *Central Processor*
This contains the memory and software for automation and alarm functions and provides an interface with peripheral equipment. Batteries should be provided for standby facilities and possible memory protection in the event of power failure. The individual user needs to decide if standby power should be provided for complete protection of the system (essential where fire and security monitoring are provided). The level of centralised software will depend on the overall distribution of intelligence throughout the system.

(*e*) *Operators Keyboard and Display*
This unit provides the means for the operator to request data, carry out control functions and add or modify programmes. The display indicates the instantaneous value or status of a single point or group of points.

(*f*) *Printers*
Frequently there is a case for using multiple printers, for example, one for alarms and another for manual logs.

(*g*) *Visual Display Units (VDU)*
A desirable feature is to display a schematic or schedule of plant to enable rapid interpretation and analysis of a plant in trouble. The display is achieved with graphics software, which can be programmed to provide dynamic up-dating during the display periods. Simple instructions are included on the display to assist the operator. Closed circuit television cameras transmitting to a VDU may also be required as a facility within the BAS.

(*h*) *Permanent Displays*
There is still a frequent requirement in the building services industry for permanent displays, taking the form of mimic diagrams or annunciator panels for the more critical services, such as power production and distribution equipment, fire, security, lift control, etc. This can be achieved through the central processor which constantly updates the displayed data.

(*i*) *Intercom*
Two way communication between the operator and data collecting outstations is a feature often required. The communication cable is run alongside or integral with the main transmission trunk. The system may also be integrated into a personnel call system.

(*j*) *Miscellaneous*
There are sometimes requirements to install secondary keyboards, printers, VDUs, etc. at positions other than at the central control room. The secondary positions may be required to monitor or separate specific functions, e.g. services in a separate building, or security services only.

*Software*
Software (programs) available can be defined in two categories: operational and management. The software currently available is listed below but this is a rapidly developing technology and as knowledge of designers’ requirements increases so will the availability of programs.

(*a*) Alarm priorities.
(*b*) Alarm inhibition.
(*c*) Analogue alarms at limit conditions.
(*d*) Integration of measured values.
(*e*) Totalisation of plant running times.
(*f*) Timeswitching.
(*g*) Event initiated sequences.
(*h*) Load shedding.
(*i*) Restart after power failure.
(*j*) Optimised start/stop.
(*k*) Cycling of plant for energy conservation.
(*l*) Process control (DDC).
(*m*) Reheat reduction from increased supply plant temperature.
(*n*) Optimum damper control.
(*o*) Security.
(*p*) Fire.
(*q*) Programmed maintenance.

*Specification and Installation*

In order to obtain a building automation system suitable for a particular project it is essential that the installer is provided with a clear specification. Schedules, used in conjunction with schematic diagrams of the systems, are a convenient method of conveying details of requirements, connections and interfaces to the suppliers and installers.

The following identifies some areas to be considered:

(*a*) Who is to supply and install data input devices (analogue and digital)? Are pockets provided with pipe line mounting equipment? What special contact features are required on the digital devices?

(*b*) Where are outstations to be located in order to minimise wiring?

(*c*) Who is to supply and/or install input, transmission and intercom cables? Who is to be responsible for checking and terminating these cables? Is special screening or segregation required?

(*d*) How is switching of 240V start/stop circuits to be handled, i.e. will relays be located at the outstation or the starter panel? How will back-feeds between systems be avoided?

(*e*) Who will provide the hand(test)/off/auto switches which are essential both to enable basic commissioning and for emergency operation of connected plant?

(*f*) Can power and monitoring cable terminals at the starter panels be segregated to minimise damage to outstations due to cross connection?

(*g*) If sequence/control logic is to be executed through the automation system, will safety interlocks be required and how will they be incorporated?

(*h*) Can all the necessary control and energy management programs be supplied as standard software? Have the programs been validated in actual use?

(*i*) What special requirements are there for duplication of peripherals? Are mimic diagrams and annunciator panels necessary?

(*j*) What language and units are to be used for display on VDUs and printers and who will bear the cost of translation?

(*k*) What are the power supply requirements? Are batteries necessary for software/memory protection of back-up facilities?

(*l*) What spare capacity is to be allowed for later additions?

(*m*) Are spare parts provided? Is there a local service organisation? What is the maximum call-out time for attendance to correct faults?

(*n*) Who will carry out the testing and commissioning of the system hardware and software? What system of servicing and maintenance is to be employed during and after the defects liability period?

(*o*) How are the interfaces between all the different systems and sub-contracts to be defined in an unambiguous manner? Who will be responsible for specific interfaces?

**Control Panels and Motor Control Centres**

Control and starter panels are mentioned in Section B10 of the *Guide*. However, in view of their importance in comprehensive control systems, some features are amplified below.

*Type and Suitability*

(*a*) *Floor or Wall Mounting*
Wall types are normally used only for small plants with few controls.

(*b*) *Wardrobe/Cupboard*
Such panels fall into two categories. In the first type, the opening mechanism is interlocked with the main incoming supply to the panel. Where the predominant equipment is starters, this type of panel is often referred to as a multi-motor panel. The second type is used where access to specific items of equipment may be necessary without a complete shutdown of the panel.

(*c*) *Cubicle*
Each major item is mounted in its own isolatable section. Favoured when maintenance or repair is required during periods of plant operation.

*Construction*

Points which are important include metal thickness and bracing, finishes suitable for the site environment and methods of equipment mounting.

*Equipment and Wiring*

(*a*) *Electric or pneumatic controls*
The choice may affect who does what in the panel prior to, and after, delivery.

(*b*) *Starters*
It is essential to identify power ratings at which changeover from direct on-line to star-delta, etc. takes place. Ensure that any special features are also stipulated, e.g. back contacts on overloads for alarm purposes, closed transition on star-delta starters. Ensure suitability of overloads for run up times on large motors.

(*c*) *Fuses*
Specify characteristics to suit applications, e.g. motors, immersion heaters, SCR controllers all require different types.

(*d*) *Isolators*
Stipulate whether suitable for on load isolation.

(*e*) *Switches and lamps*
Types and method of operation are important for control, e.g. three position switch for selection of run and standby plant should have a positive off position between the two operational positions.

(*f*) *Ammeters*
These are necessary for all but the smallest motor drives. They permit checks on performance to be carried out on all fans and motors against their published data and aid testing and commissioning.

(*g*) *Relays and Contactors*
Suitable circuitry and performance criteria should be detailed, e.g. coil voltages, in-rush currents and acceptable number and types of contacts.

(*h*) *Wiring Standards*
Is the wiring to be loomed or carried in trunking? Detail the methods of identifying each connection to match drawings. Specify the type of wiring to be used.

(*i*) *Terminations*
Detail the type of terminations to be used both for the incoming and outgoing cables and for interconnections in the panel.

*Mounting Requirements*

These cover the following:

(*a*) Separation of equipment types.
(*b*) Access and door interlocks.
(*c*) Shrouding.
(*d*) Space and spares.

*Interlocking*

(*a*) *Logic*
Functional sequences should include all the requirements for control loops and interfaces to ensure that incorrect sequences cannot occur.
(*b*) *Safety and Alarms*
Ensure that all safety and limit devices to protect plant and satisfy any statutory or other regulations are included in the logic sequences.
(*c*) *Equipment for Logic Operations*
Decide whether the interlocking sequences are to be carried out using conventional relays or programmable controllers (processor software).

*Responsibilities*

(*a*) *Panel Supplier*
Who will supply the control panels? Interface problems are minimised if panels are provided by the controls supplier.
(*b*) *Wiring and Pneumatic Pipework*
The wiring in the panel is always carried out by the panel manufacturer. The external wiring to the panel may be by the electrical sub-contractor, the HVAC sub-contractor, the controls supplier or a specialist sub-contractor. It is often advisable to make the HVAC sub-contractor responsible for the co-ordination even if the work is carried out by others.
(*c*) *Drawings*
The information for obtaining and co-ordinating the information required from the various suppliers should be vested in one of the sub-contractors alone. The HVAC sub-contractor is usually best suited to this function. Drawings finally incorporated into panel diagrams will include those from:
Controls supplier
Boiler supplier
Chiller supplier
Cooling tower supplier
Starter supplier
Electrical sub-contractor
HVAC sub-contractor

*Testing*

(*a*) *At Works*
All functional, operational and safety sequences should be simulated and checked and compliance with any statutory or other requirements, such as the IEE Wiring Regulations, should be ensured.
(*b*) *On Site*
All connections in to the panel should be checked before power is switched on and the functional operational and safety sequences should be rechecked before the plant is operated.

**Compressed Air**

Pneumatic controls require clean, dry, oil free air and care must be taken in the design and selection of the compressed air production plant and distribution system. Failure to ensure the correct quality of air supply may result in the necessity to clean pipework and replace controls components.

*Compressors*

Reciprocating compressors are generally preferred to the diaphragm pump of centrifugal machines, and wherever possible carbon ring (oil free) compressors are used. This does not eliminate the need for oil removing filters as backup against possible oil carry over. It is advisable to provide standby facilities, with duplicate compressors, each selected for the maximum load. The selection is made to allow a run time of between 20 minutes per hour (smaller projects–less than 5 l/s) and 30 minutes per hour (large projects). Control of the duty compressor is normally on/off by pressure switch (600 to 700 kPa) in the air receiver, with a further pressure switch to start the standby unit on low pressure.

*Air Receivers*

Air for pneumatic control systems is normally stored at gauge pressures of 600 to 700 kPa. Therefore, receivers must be constructed in accordance with BS 5169 and be provided with suitable control and protection devices, i.e.
Control pressure switch
Stand-by pressure switch
Pressure gauge
Safety valve
Automatic drain (some condensation is inevitable)
Manual drain
Access port/manhole
Inlet and outlet isolating valves

The receiver should be sized to restrict the number of compressor starts to 10 or 12 per hour, which is governed by the differential of the control pressure switch (100 to 150 kPa) and the lowest permissible operating pressure.

It is advisable to duplicate the air receivers (and compressors) as a full duplex unit in order to maintain supplies at all times. It would also appear that whereas previous regulations only covered pressure vessels on industrial premises, the Health and Safety at Work Act demands closer control of standards in non-industrial applications. Duplicate receivers will, therefore, be desirable on most applications.

*Air Driers*

Pneumatic controls operate using very small bleed orifices which may easily become blocked or restricted with water if moisture is not removed at source. Refrigerated driers reduce the high pressure air dewpoint temperature to 10 to 15°C. In normal situations where pipework is run within conditioned spaces this is sufficiently low to eliminate the risk of condensation occurring. Driers will normally be duplicated and be complete with high temperature cut-out thermostats and solenoid valves to stop air being supplied to the system on failure. This protection circuit may also be utilised to start the stand-by unit.

Adsorber driers, are used for dewpoint temperatures of 0°C or less. Each drier comprises dual adsorber vessels, one drying the air while the other is being regenerated using a proportion of the air from the drying vessel. The quantity of air for regeneration is additional to that required for the system load. In order to minimise this additional air requirement, units are available with electric heaters in each vessel to improve regeneration efficiency.

Chilled water driers may be the cheapest method of drying air if chilled water is available continuously at a sufficiently low temperature. The use of mains water as the cooling medium is unlikely to be permitted.

*Filters*
Filters are required for the removal of particles in the air (at least 98% of particles down to 0.4$\mu$m) and elimination of oil. Oil filters should have some positive means of indication that cleaning or changing is required.

*Distribution System*
Air is normally distributed at 600 to 700 kPa via steel or copper pipes to pressure reducing stations sited at strategic positions to service groups of control equipment with air at 140 kPa. Each pressure reducing station normally comprises the following, housed in a protective enclosure:

- Pressure reducing valve
- Filter (may be combined with the pressure reducing valve)
- Isolating valve
- Upstream and downstream pressure gauges

Distribution of low pressure air from pressure reducing stations may be through polythene or nylon pipe in metal or plastic trunking or conduit, with either copper or metal (spring) sheathed plastic final connections to sensors and regulators where liable to damage. Alternatively all-copper pipe clipped to cable tray or direct to walls and ceilings can be used but this is probably more vulnerable to damage during construction.

**Lighting**
The simplest form of lighting control is the on/off switch. When an on/off switch is combined with a time clock, a fundamental method of automatic control is achieved which is basically suitable for all types of lighting circuit. In applications requiring a smooth transition of light level, a proportional controller such as a dimmer is used. However, as will be discussed later, the proportional controller is not suitable for all types of lighting circuits and sources. Control may be on/off or proportional and may be of the open-loop or closed-loop type. If a closed-loop system is used with an on/off control, there must be a large differential between switching levels to prevent frequent switching of the lamps.

On/off controllers are usually electro-mechanical (conventional switch gear, time or solar switches), or electronic (photo-electric switches). Electro-mechanical contactors may be replaced, to advantage, in a number of circumstances, by solid-state relays which use thyristor or triac semiconductor devices as on/off switches. Where it is necessary to control the rate-of-change of light, or to maintain it at an intermediate level, thyristor or triac dimmers are usually used. However, consideration of the type and rating of load is necessary before selecting a dimmer for a particular application. The continuing progress in advanced technologies has produced and will continue to produce, additional controllers and control systems. Systems are already installed which allow considerable flexibility for modifying the grouping of luminaires for switching purposes, with a minimal amount of wiring alterations and they may be coupled to a variety of initiating devices. The trend will be towards a programmed system where each luminaire has a unique address enabling it to be switched remotely, either individually or as part of a group, with local switching as either the master or slave system. Microprocessors will be programmed for the desired lighting control and to provide digital switching to each address, or the switching might be accomplished by acoustic, ultrasonic, infra-red or radio signalling devices, or by signal transmission over the power wiring.

*Tungsten and Tungsten-Halogen Lamps*
Tungsten and most tungsten-halogen lamps can be smoothly dimmed to zero light output but as they are dimmed the light output drops more rapidly than the power consumption, see Fig. B11.40. Tungsten lamps rated at 220/240 volts may be controlled directly from any type of lighting dimmer. Low voltage lamps, for example, 120 volt type, should not be operated from a 220/240 volt dimmer without a step down transformer, otherwise the peak voltage may damage the lamp. Tungsten halogen lamps which are designed to operate at high colour temperatures might not have their life extended by dimming.

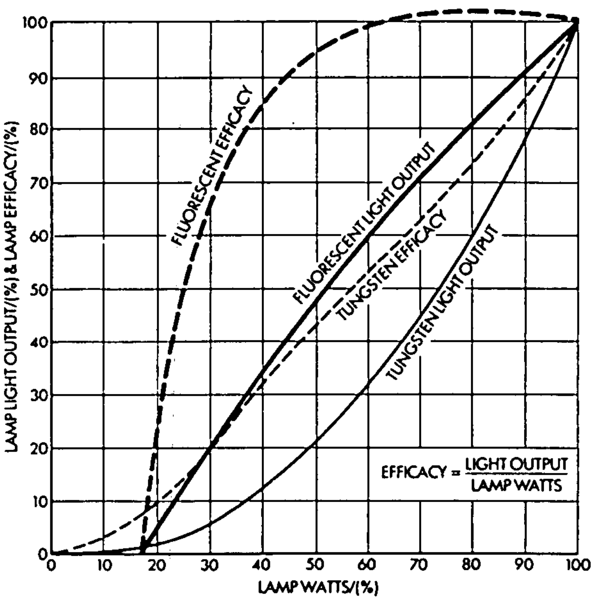



**Fig. B11.40.** Characteristics for typical tungsten and fluorescent lamps during dimming.

*Tubular Fluorescent Lamps*
Fluorescent tubes operating from standard switch-start or ignitor circuits can now be dimmed to approximately 40% of full light output. Only 'lagging' circuits with parallel power factor correction capacitors should be used. The power factor capacitor must be separated from the lamp circuit and connected to the mains input of the dimmer (or as recommended by the manufacturer), whilst the remaining circuitry is fed from the dimmer output. In this situation the load on the dimmer is the tube current which is

considerably higher than the mains current. The chopped waveform produced by the dimmer may cause additional heating of the control gear which may affect the luminaires.

Other circuits, such as the ‘semi-resonant’ type, may damage thyristor or triac dimmers and should not be used, unless specified for use with dimmers. The mains current depends on the dimmer setting and may be higher than the normal power factor corrected value.

Dimmers designed for tungsten lamps are usually unsuitable for fully dimming tubular fluorescent lamp circuits. By using circuits incorporating specially designed ballasts and dimmers, most standard fluorescent tubes can be smoothly dimmed to a few per cent of full light output, at which point the discharge becomes unstable and striations occur although high frequency circuits minimise this effect. Throughout most of the dimming range, the efficacy remains reasonably high and the colour change is small. It is advisable to contact the lamp manufacturer if it is required to dim tubes other than the standard hot cathode argon or neon/argon filled tubes of 38 mm diameter, non amalgam, ‘white type’, because not all of the range of fluorescent tubes dim equally well. Whilst good dimming of fluorescent tubes depends on using the correct ballasts and dimmers, stable dimming can only be achieved if the mains supply is free from disturbance, particularly at the zero-crossover point. A common source of disturbance is the use of high power motors on the same supply where the slip-frequency produces a small cyclic variation of voltage which can result in flicker. Disturbances to the mains are more likely to occur in rural areas where the supplies have higher impedance than in towns.

*High-Intensity Discharge Lamps*
The operation of high-intensity discharge (including low pressure sodium) lamps relies on temperature of the gases within the arc tube. If the lamp power is reduced this condition is altered and the lamp fails to operate correctly. Consequently, these lamps are generally considered unsuitable for dimming.

*Transformers*
When transformer loads are driven by a dimmer, it is necessary to ensure that both the transformer and dimmer are suitable for the application. The primary of each transformer should be protected by a suitably rated fuse as a precaution against the dimmer having d.c. in its output. Transformers used to drive lamps from the output of a dimmer must be suitably designed for the application with allowance made for the harmonics generated by the chopped waveform. When it is required to change components, particularly if this involves access to the wiring, the mains input must be turned off because dimmer circuits and solid state relays have a leakage of several milliamps in their off state. The feeds between the mains supply, dimmer and lamps have a high harmonic content and therefore the line and neutral should be run together but should be separated from low-voltage control-leads and audio signals such as public address systems.

**Ancillaries**

*Step Controllers*
Step controllers comprise a series of switches operated sequentially, normally by means of a rotating shaft fitted with adjustable cams and driven by a conventional electric or pneumatic actuator. It should be remembered that the drive to the actuator will be provided through another detector/controller, selected for the application. The most common applications are the control of multiple boilers or chillers, cooling towers and electric heater batteries.

Depending on the application, step controllers should have the following features:

(*a*) Adjustment of each switch operation for any position of shaft rotation, which is necessary for all applications.
(*b*) Adjustable differential switching, i.e. each switch independently makes and breaks at different positions of shaft rotation.
(*c*) Switch rating capable of coping with the load, particularly for electric heater batteries operating with line-neutral steps. There should be at least one switch with change-over contacts for use as a recycling device in the case of plant shut-down or power failure.
(*d*) The facility for fitting potentiometers which operate over the complete angular rotation of the shaft which can be used for modulating boiler and chiller control.
(*e*) An indication of position, e.g. 0-100% scale.
(*f*) Facility for fitting a spring return or auxiliary power device for fail-safe drive back to the no load position.

For electric heater battery switching, a master load contactor, or series of contactors, will be required to operate in conjunction with the recycling switch to avoid restart on full load. Semi-conductor versions of step controllers are available and in these variants it is possible to program the switches so that a large range of steps can be accomplished using a small number of switches, e.g. from basic loads of 1, 2, 4 and 8 kW, 1 kW steps from zero to 15 kW may be provided. There are situations, particularly with the loading and unloading of chiller solenoid valves, where virtually all switches may require changeover contacts.

*Self Acting and Self Operating Devices*
These are devices which exercise a controlling function without using any auxiliary energy, the power needed to operate them being derived from signals of temperature, pressure, etc. They operate as proportional controllers. Since these devices operate by direct action, it is not normally possible to remove the driving force in the event of abnormal operating conditions arising. Therefore they cannot easily be made fail-safe.

Sensors can be remote or integral and thermostatic radiator valves take both forms, but outside the domestic sector the majority of devices use remote sensors. In addition, there is a limited range of devices where the motive power for operating the final control element is the pressure of the primary air. In the case of remote sensor temperature controlled devices, the power is derived from the expansion of a liquid or solid, melting wax, or increase in vapour pressure. As the temperature changes, a capillary tube transmits the signal to the valve. Both two- and three-port valves can be controlled in this way, allowing mixing and diverting applications. These devices have proportional bands of the order of 3°C and they should be checked to ensure that they will operate within acceptable tolerances.

Valves with sensors integral to their body, or connected by capillary, can be used to control the flow of fluid through them, when used as a thermostatic return temperature limiter which sets a maximum to the temperature of the fluid flowing in the return pipe of a system.

In the case of remote sensor pressure controlled devices, valves with a sensor/capillary tube connected to each side of a diaphragm, together with a balancing spring, are available to:

(*a*) Maintain a constant differential pressure across a circuit.
(*b*) Limit the flow rate through a circuit when used in conjunction with a fixed orifice.
(*c*) Maintain a constant flow in a circuit.

The functions (*a*) and (*b*) can be achieved by placing the valve, which closes in response to increases in differential pressure, in series with the circuit. The function (*c*) can be achieved by placing the valve, which opens as the differential pressure increases, in parallel with the circuit.

## SYSTEM APPLICATIONS

The CIBSE Applications Manual, *Automatic Controls and their Implications for Systems Design,* contains a comprehensive range of applications of controls in practice. The following section gives guidance on basic HVAC systems only.

The choice of detectors, controllers and final control elements (FCE) is based on the theoretical considerations dealt with earlier and a knowledge of the essential engineering requirements of the system being controlled.

A proper understanding of these parameters should enable the correct initial choice to be made between on/off or modulating control. Selection of equipment for the former mode is relatively straightforward with the qualification that limiting and safety devices may also be needed. The actual control can be accomplished with a detector and FCE. The latter mode is more complex in that a suitable controller must be selected. Controllers to operate valves or valves and dampers in sequence are available for all modulating modes (P, P+I, PID, etc.). Sequential operation is normally confined to a maximum of three stages, any system requiring more stages should be closely examined.

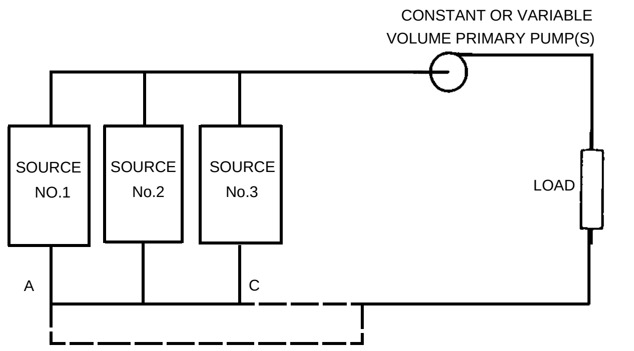
The adjustment and setting of individual controllers is not covered in this section. While the correct selection of a controller is the most important factor, other important considerations include simplicity of setting up, calibrating, testing and commissioning and the ability of the final user to understand the adjustment facilities.

### Boilers and Chillers

The control of heating and cooling sources resolves itself into individual control and multiple control. The hydraulic design, particularly with multiple sources, is very important for the correct operation of control systems and is, therefore, treated in some depth.

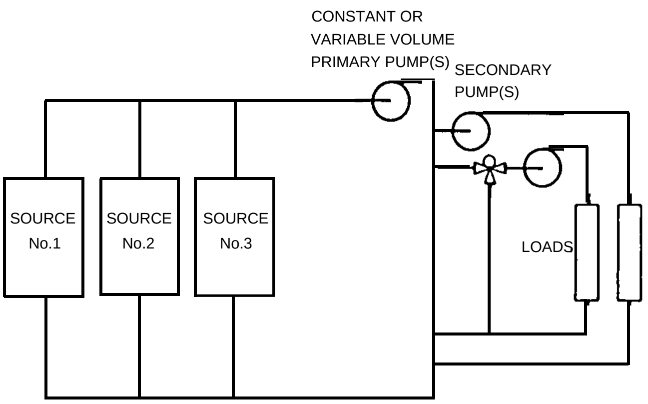
*Individual Control*
The vast majority of both boilers and chillers are supplied with their own packaged control system. Where only one source unit is required at any time, use of the packaged controls is the best solution. The provision of a standby unit means that some manual or automatic changeover system has to be devised in addition to the packaged controls.

*Multiple Control*
With multiple boiler or chiller combinations the first choice to be made is whether to run them for series or parallel operation and control–chillers may occasionally be installed for series/parallel operation. The answers to a number of checklist questions are essential for the selection of a suitable system; the different basic arrangements are shown in Figs. B11.41 to 44.



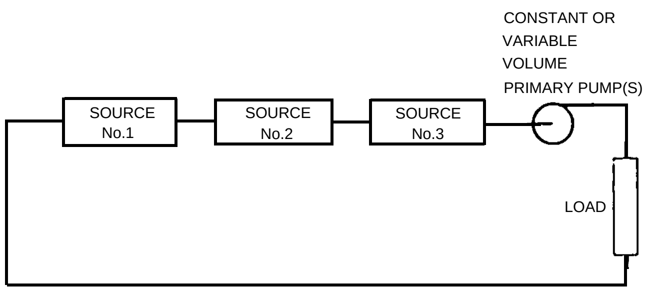

For variable volume systems use connection A-B.
For constant volume systems use connection B-C.
Excludes all regulating, isolating and control valves and bypasses from diagram.

**Fig. B11.41.** Parallel sources with primary pumping.



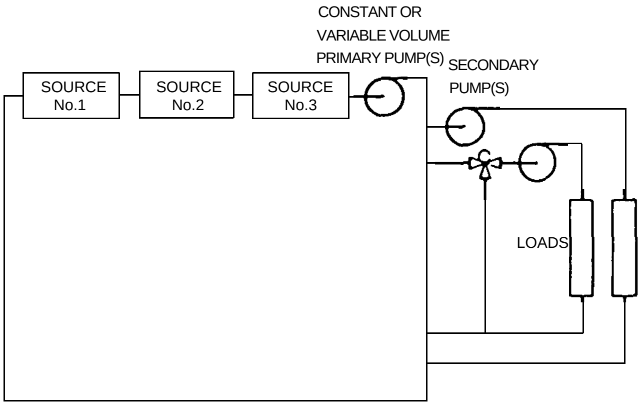

Excludes all regulating and isolating valves from diagram.

**Fig. B11.42.** Parallel sources with primary/secondary pumping.




Excludes all regulating, isolating and control valves and bypasses from diagram.

**Fig. B11.43.** Series sources with primary pumping.




Excludes all regulating and isolating valves from diagram.

**Fig. B11.44.** Series sources with primary/secondary pumping.

In the parallel mode points to be considered are:

(*a*) The amount of over-temperature (boilers) and sub-cooling (chillers) which can be tolerated during sequence operation over the total load span. Will the return water temperature on boiler systems drop below an acceptable level?

(*b*) The choice of primary plus secondary pumps or primary only, in conjunction with the valving-off, of nonoperational units. This is related to both hydraulic stability and (*a*) above.

(*c*) The permitted variation in flow through the units–flows through boilers and, particularly, chillers should not be reduced much below design rating.

(*d*) Flow balance between individual units–short circuiting may occur for flows below maximum, unless the system is correctly designed on, say, a reverse return principle.

(*e*) The maximum number of units which can reasonably be controlled in sequence. This is related to over-temperature and sub-cooling and the availability of a suitable control system.

In the series mode the points are:

(*a*) The problems of over temperature (boilers) and sub-cooling (chillers) do not occur. What is the minimum temperature rise (boilers) or fall (chillers) acceptable across each unit? Is there a likelihood of flow between the cold feed and common boiler vent?

(*b*) The choice of primary plus secondary pumping, or primary only.

(*c*) Does the selected sequence of operation for the sources affect their output capability? For example, for three chillers in series on a system designed for 5°C flow and 11°C return, any one chiller must be capable of providing its maximum 2°C drop, for inlet conditions varying from 5°C to 9°C.

(*d*) Will the advantages of a series system outweigh the additional pumping costs for units in series, particularly chillers?

For both series and parallel operation:

(*a*) The choice between modulating or stepped mode of temperature control–modulating control is preferable for system stability.

(*b*) Is temperature control from flow or return? Control from the return is not acceptable except with constant total flow through the units and control from the flow is not advisable with the stepped mode.

(*c*) Is there a suitable sequence control system available for the proposed arrangement? There may be difficulties for more than three units.

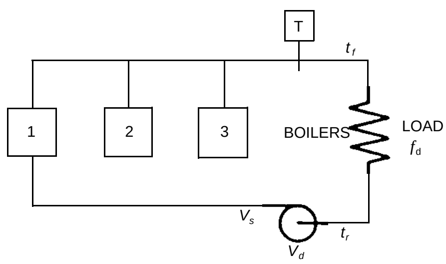
(*d*) What is the effect of superimposing an overall control system over that of the individual units? Manufacturers' guarantees should not be affected but modifications to the normal single unit operating functions may be required.

(*e*) Are the unit sizes selected so that operation at low partial loads does not create additional problems? On multiple unit installations one unit normally operates at minimum load more frequently than the overall system operates at maximum load. Large machines are only permitted a limited number of starts in a given period. What is the maximum turndown ratio? At least one machine must be selected so that its minimum output is less than the lowest normal operating load.

(*f*) Do the limit devices have the span to cope with the unit operation over the full range of loads. This is illustrated by the example in Figs. B11.45 and 46.

It should be emphasised that the analysis of the conditions at the heating or cooling source cannot be treated in isolation from the remainder of the hydraulic circuit. The converse is also true.

The example shown in Fig. B11.45 illustrates some of the problems listed. They are relevant to a boiler sequence control system with continuous constant flow through all the boilers for both low pressure hot water (LPHW) and high pressure hot water (HPHW) systems. This clearly shows the magnitude of the over-temperature problem with sequence control under partial load conditions. While the over-temperature may be acceptable for the LPHW system, other than for roof top boiler houses, the HPHW system reaches over-temperature conditions which may be outside the limits of the designed system pressure. Both systems illustrate that any boiler-mounted thermostats




T: Boiler control detector. Isolating and regulating valves not shown.

| Load $f$ (%) | Case 1 (LPHW) $tfd$ = 90°C; $tfd$ = 78°C | | | | Case 2 (HPHW) $tfd$ = 130°C; $tfd$ = 85°C | | | |
|---|---|---|---|---|---|---|---|---|
| | Return temp. $t_r$/°C | Boiler flow temp. $t_f$/°C | | | Return temp. $t_r$/°C | Boiler flow temp. $t_f$/°C | | |
| | | 1 | 2 | 3 | | 1 | 2 | 3 |
| 100 | 78 | 90 | 90 | 90 | 85 | 130 | 130 | 130 |
| 66.7 | 82 | 94 | 94 | 82 | 100 | 145 | 145 | 100 |
| 33.3 | 86 | 98 | 86 | 86 | 115 | 160 | 115 | 115 |

**Fig. B11.45.** Constant volume system.

must be set much higher than normal, impossibly so in some cases, if the sequence control is to operate correctly and efficiently. The tabulated temperatures are simply calculated from the load percentage with a fixed flow temperature. There is also the necessary consideration of the safety requirements and devices under the Health and Safety at Work etc. Act.

In the equivalent chiller system similar calculations may be carried out for the sub-cooling case. There is however, little latitude for sub-cooling before the operating chiller is locked out by its low temperature cut-out thermostat. This must always be a primary consideration for selecting a suitable multiple chiller configuration.

Fig. B11.46 shows a system with LPHW variable volume flow through the boilers with a compensated mixing circuit serving radiators and a sequence control (firing order 1, 2, 3) is proposed. The flow and return temperature data are based on:

$$\frac{f_p}{f_d} = \left(\frac{D t_p}{D t_d}\right)^{1.3} \quad .. \quad .. \quad .. \quad .. \quad .. \quad \text{B11.24}$$

or

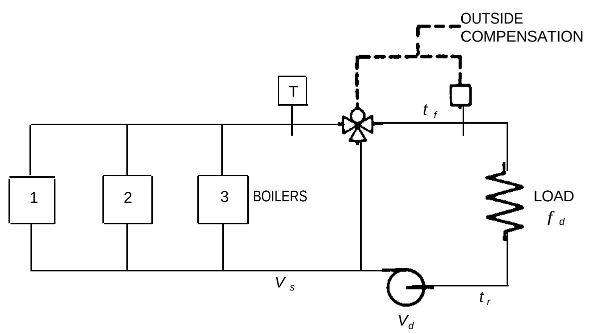
$$D t_p = \sqrt[1.3]{\frac{f_p}{f_d}} \times D t_d \quad .. \quad .. \quad .. \quad .. \quad \text{B11.25}$$

$$f = V(t_f - t_r) \quad .. \quad .. \quad .. \quad .. \quad .. \quad \text{B11.26}$$

$$t_m = ½(t_f + t_r) \quad .. \quad .. \quad .. \quad .. \quad .. \quad \text{B11.27}$$

where:

$f$ = output .. .. .. .. .. W
$f_d$ = maximum design output heat (assume 100% of all boilers) .. .. W
$f_p$ = partial load heat requirement .. W




T: Boiler control detector. Isolating and regulating valves not shown.

| Load $f$ (%) | Case 1 (LPHW) $t_{fd}$= 90°C; $t_{rd}$ = 78°C | | | | | |
|---|---|---|---|---|---|---|
| | Return temp. | Mean temp. | $V_s$ / $V_d$ | Boiler flow temp. $t_f$/°C | | |
| | $t_r$/°C | $t_m$/°C | $V_s$ / $V_d$ | 1 | 2 | 3 |
| 100 | 78 | 84 | 1.0 | 90 | 90 | 90 |
| 66.7 | 63 | 67 | 0.3 | 103.5 | 103.5 | 63 |
| 33.3 | 45.5 | 47.5 | 0.09 | 179.0 | 45.5 | 45.5 |

**Fig. B11.46.** Variable volume system serving compensated circuit.

$D t_d$ = differential temperature between space condition (20°C) and mean water temperature at design load .. °C
$D t_p$ = differential temperature between space condition (20°C) and mean water temperature at partial load .. °C
$V$ = flow rate .. .. .. .. .. $m^3/s$
$V_d$ = flow rate to load at design load.. .. $m^3/s$
$V_P$ = flow rate to load at partial load.. .. $m^3/s$
$V_s$ = flow rate through source .. .. $m^3/s$
$t_m$ = mean water temperature .. .. °C
$t_{md}$ = mean water temperature at design load .. .. .. .. .. °C
$t_{mp}$ = mean water temperature at partial load .. .. .. .. .. °C
$t_f$ = flow water temperature.. .. .. °C
$t_{fd}$ = flow water temperature at design load °C
$t_{fp}$ = flow water temperature at partial load °C
$t_r$ = return water temperature .. .. °C
$t_{rd}$ = return water temperature at design load .. .. .. .. .. °C
$t_{rp}$ = return water temperature at partial load .. .. .. .. .. °C
$t_{rs}$ = source return water temperature .. °C

From these equations the tabulated data given in Fig. B11.46 may be derived, taking $t_{fd}$ as 90°C and $t_{rd}$ as 78°C.

The calculations indicate that the choice of this sequence control system may be wrong although in theory it would be perfectly capable of maintaining the required flow temperature. In practice, if the boiler thermostats were left set in the normal range of 80°C to 95°C, the boilers would operate on these thermostats thus making the sequence system virtually redundant. The system is acceptable if it is possible to set the limit thermostats above the calculated maximum temperatures and still maintain the necessary flash margin.

When modular boiler systems are used, i.e. systems with multiple part load on/off units, each providing 10 to 20% of the design load, this problem is accentuated. Unless the limit thermostat settings can be suitably adjusted, the high part load efficiency claimed for this type of system will not be achieved.

*Design and Control Implications*

Some general points can be made about all multiple source systems in respect of hydraulics, excessive pump head and energy management. A careful analysis of the hydraulics under varying load conditions is necessary to ensure that there are no situations where the characteristics of one pumped circuit inadvertently affects another. The selection of suitable pumps for each different application is also important, as is the positioning of the pumps to avoid cavitation or over-pressurisation.

For sources having stepped outputs, control of multiple sources from the return is a common arrangement. Only proportional controllers are suitable and this method should only be used when the volume through all the sources is constant for all load conditions. Such systems suffer from mismatch of design load with source capacity on the loss of one of the sources in the sequence and this may be overcome by controlling the sources from the flow (but not with a proportional controller). However, care should be taken to limit the effect of the step outputs on the flow control detector to avoid instability.

*Boiler Return Temperature*
Under system design conditions the return water temperature in a boiler system is a primary design parameter and should always be high enough to prevent corrosion of the boiler by condensation in the flue. However, certain flow conditions and systems (e.g. compensated circuits on part load) can give much lower return temperatures, below the acid dewpoint of the flue, causing corrosive conditions. The same situation occurs during the warm up period of boilers after a plant shutdown. The minimum operating return temperatures should always be determined and checked against the flue conditions and the boiler manufacturer's recommendations. Where there is any conflict between these criteria special arrangements can be made to maintain the return water temperature or minimise the warm up period.

*Hot Gas Bypasses*
Whatever the choice of chillers(s) to satisfy the part load situation, there are occasions when the lowest possible chiller output is in excess of the cooling load. On some machines a hot gas bypass valve may be fitted which allows the chiller to be modulated down to virtually zero output to match the load. The fitting should be considered as an integral part of the machine and on-site installation should be avoided. The function of these valves is to impose an artificial load so that the machine can operate below the normal cut-off output. The valves are specifically designed to carry refrigerant gas and can normally be fitted with an actuator which is compatible with the chiller control system.

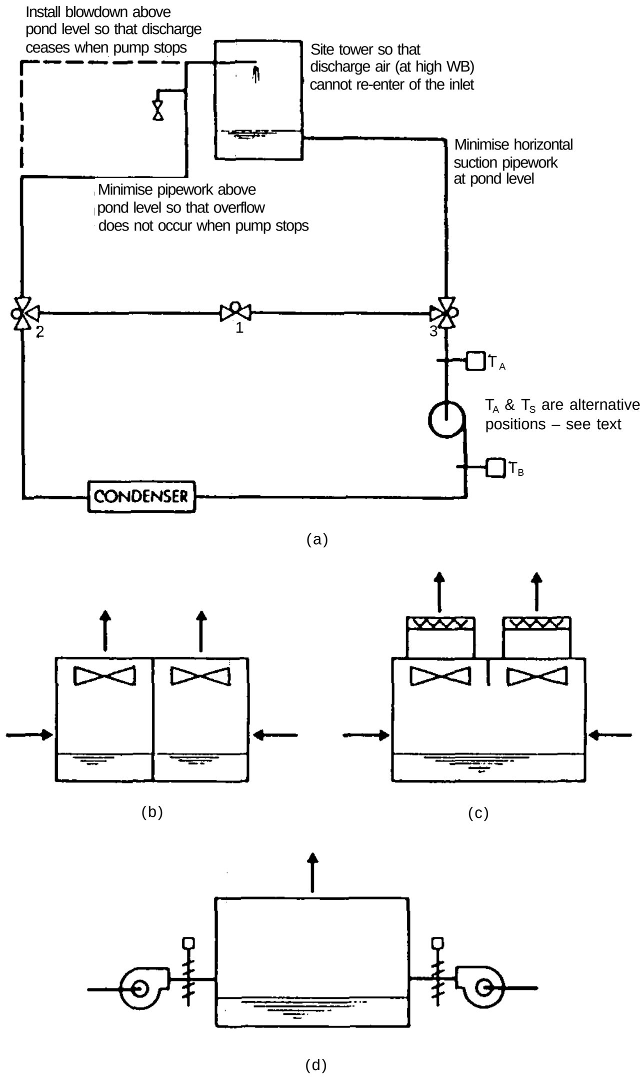
**Cooling Towers**
There are basically two types of cooling tower in common use; open circuit and closed circuit. Air is again either induced or forced over the cooling pack in contra-flow to the secondary water.

A general schematic for cooling tower control is shown in Fig. B11.47 which is also annotated with design notes for good practice. The system is described for a single chiller coupled to a cooling tower. With multiple machines the principles remain the same but there are a multiplicity of pipework alternatives between the machine and the tower(s) and there may be bypass valves for each condenser.

*Control Methods*
The variable to be controlled and the mode of control will depend on the type and number of chillers, the configuration of the condenser water system and any heat reclaim requirements. There are three methods of achieving adequate control of cooling tower plant with sufficient accuracy to ensure reliable operation of chiller plant:

(*a*) Air side, by fan and/or damper control.
(*b*) Water side, normally using bypass control.
(*c*) A combination of (*a*) and (*b*).

These options apply equally to open or closed circuit towers, although the actual methods of achieving control may differ. The combination system is normally preferred on larger projects. The controlled variable can be flow or return water temperature, or refrigerant head pressure.




(a) General schematic.

(b) Induced draught close coupled single fan units.

(c) Induced draught multi-fan units.

(d) Forced draught multi-fan units.

**Fig. B11.47.** Cooling tower control.

Air side control is achieved by varying the volume of air through the cooling tower by damper control, fan speed control or sequenced switching of fans. Where fan control is allied to water side control the water bypass valve should be fitted with fully adjustable auxiliary switches as part of the fan switching operations.

Water side control is achieved by varying the volume of water through the tower by a bypass across it, see Fig. B11.47(a), to ensure a nominally constant volume to the condenser(s). The valves should always be sited below the minimum pond level to facilitate initial priming during start up. The possible ways of achieving water volume control are:

(*a*) *Single two-port valve:* see Fig. B11.47(a) position 1. It is important to ensure that the valve-plus-bypass pressure drop is less than the tower static lift so that when the valve is fully open (no load) water is not discharged over the tower. It is still necessary to aim at a valve authority of 0.3 to 0.4 for good control. A suitably selected butterfly valve will operate perfectly adequately and is more cost effective than other two-port valves, or three-port valves.

(*b*) *Diverting valve:* see Fig. B11.47(a) position 2, which may be preferred when there is little difference in height between the tower and condenser as it minimises the risk of water discharging over the tower at no load. It also avoids possible cavitation at the pump. The valve authority should be 0.3 to 0.5.

(*c*) *Mixing valve:* see Fig. B11.47(a) position 3, should only be used on open circuit towers when the hydraulics have been checked to ensure that cavitation will not occur. The valve authority should be 0.3 to 0.5.

Combined air side and water side control offers the advantages of close control coupled with maximum economies in fan power consumption, particularly where flow temperature control is required.

Three common methods of combined air side and water side control are by flow temperature, return temperature and head pressure.

*Heat Reclaim*

This topic relates to a whole range of options covered elsewhere but there are particular facets which relate to condensers. Reclaiming heat from condensers demands a higher condensing pressure than normal so that condenser water at a usable temperature (up to 45°C at present without appreciable loss of COP) can be obtained.

*Protection Systems*

Flow switches should be provided downstream of the condenser to ensure cooling water is available before starting the chiller and it is advisable to back-up the interlock by a series connection through the duty pump starter to guard against a sticking flow switch. Differential pressure switches may be used instead of flow switches. Cooling water pumps or closed circuit tower recirculation pumps should be protected against running without water, by a level switch in the tower pond. Frost protection must always be considered.

**Air Cooled Condensers**

Air cooled condensers are generally associated with smaller plants, but they are now available up to at least 1000kW of refrigeration. Both condensing pressure and temperature must be controlled. High condensing temperatures can lead to overloading of the compressor motor and a safety limit cut-off normally protects against such conditions. Low condensing pressures will cause insufficient pressure for liquid feed devices thus starving the evaporator, with resultant loss of capacity, and the resultant low suction pressures will cause the compressor to trip on the low pressure switch. Methods of head pressure control vary widely, depending on circumstances. The two types of system suitable for head pressure control are refrigerant side control or air side control.

*Refrigerant Side Control*

This is accomplished by reducing the active amount of condenser surface, by flooding the coil with liquid refrigerant. This type of control requires the use of a liquid receiver and an excessive charge of refrigerant and should not be used where sub-cooling coils are used on condensers.

**Water and Steam**

In HVAC systems, the relationship between volume and temperature is frequently governed by the use of control valves. Their application affects the flow and temperature conditions in both the load and source sections of the hydraulic system. The designation for the types of circuit described below relate to the flow/temperature relationship in the load. The choice of suitable characteristics for valves has already been considered. In summary:

(*a*) For on/off applications any valve characteristic may be used subject to limitations on pressure drop and shut off.

(*b*) For modulating applications use equal percentage characteristics for water circuits and linear characteristics for steam systems.

*Normal Three-Port Valve Applications*

The uses of three-port valves have been covered under *Valves,* but Figs. B11.48 and 49 show their practical use in mixing and diverting applications. The circuits designated as constant volume variable temperature are normally associated with compensator control. However, in some cases the valves maintain a constant flow temperature with varying load, e.g. as part of an injection circuit or in chilled water secondary circuit applications.

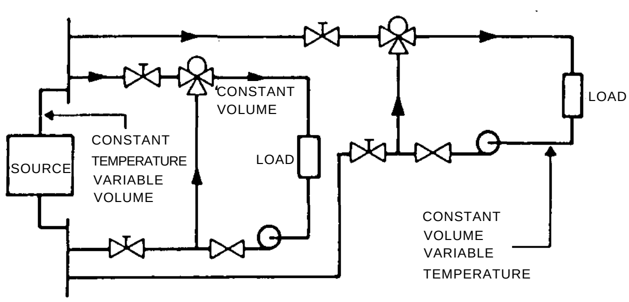



**Fig. B11.48.** Constant volume variable temperature-mixing applications.

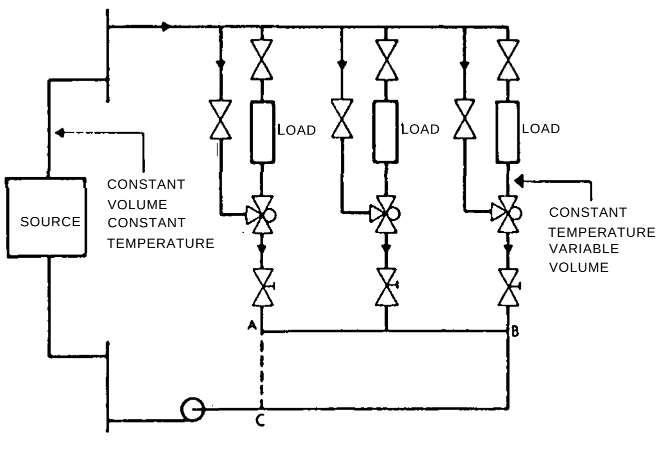



**Fig. B11.49.** Variable volume constant temperature-diverting applications.

*Compensator Control*

The use of three-port valves as shown in Fig. B11.48 is normally associated with circuits serving radiator or natural convector systems. A large proportion of existing heated (non-domestic) buildings use outside compensators as a means of overall temperature control for the internal environment of the building. By controlling the position of the valves it is possible to obtain a flow temperature to the load which can be held at, or vary between, the full boiler outlet temperature (valve fully open) and the space temperature (valve on full bypass). It is normal to control the valve position by means of a detector situated in the flow downstream of the valve as shown in Fig. B11.11 in order to achieve a predetermined temperature. The temperature level required at the flow detector is determined by an external detector which varies the requirements according to the outside temperature. In normal operation the system would provide a typical flow temperature characteristic against outside temperature of the form shown as curve 1 in Fig. B11.50.

The actual characteristic is dependent on the system design parameters and the heat losses from the building. Some compensators permit the characteristic to be generated from point A (curve 2) others from point B (curve 3) and some may be set at both A and B (curve 4). The 'curves' are drawn as straight lines although a non-linear curve would more nearly approach the ideal. Irrespective of the characteristic the upper flow limit is determined by the boiler outlet temperature (line AC) and the lower limit, in theory, by the internal space temperature.

Other variations to the characteristics may be achieved by detectors which are sensitive to wind and solar effects. When a compensator is used with natural convectors rather than radiators, a different characteristic is required and, in general, the two types of emitter should not be served from the same compensator system.

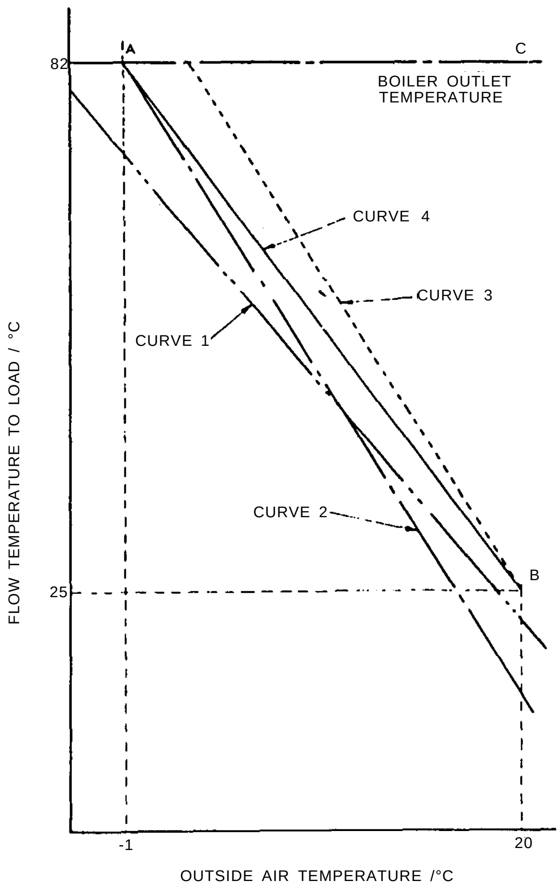



**Fig. B11.50.** Typical compensator characteristics for a system serving a radiator circuit.

*Multi-zone Pumped Systems*

On large systems where primary, secondary and, possibly, tertiary pumped circuits occur, it is necessary to ensure that the pump heads available do not interfere with the correct operation of the valves. Fig. B11.51 shows a primary pumped circuit designed for constant volume, feeding a number of secondary circuits, where the pressure available across each decreases as the distance from the pump increases. Generally, if there is sufficient head available for circuit *n* then circuit 1 will have excess head available at the take off from the distribution mains.

Circuits 1 and *n* show mixing applications in which $RV_{P1}$ and $RV_{Pn}$ are adjusted to drop the total head across $A_{P1}$-$B_{P1}$ and $A_{Pn}$-$B_{Pn}$ respectively. The total volumes in circuits $A_{P1}$-$A_{S1}$ and $B_{P1}$-$B_{S1}$ (for circuit 1) and $A_{Pn}$-$A_{Sn}$ and $B_{Pn}$-$B_{Sn}$ (for circuit *n*) remain constant, as do the pressure drops across $RV_{P1}$ and $RV_{Pn}$. Therefore the pressure differences across $A_{S1}$-$B_{S1}$ and $A_{Sn}$-$B_{Sn}$ are virtually zero and the circuits operate with zero head applied across the secondary, thus permitting a stable circuit characteristic governed by the secondary circuit constant volume pumps. Circuit 2 shows a similar circuit without the use of $RV_P$ and $A_S$-$B_S$. In this case the pressure $A_{P2}$-$B_{P2}$ is developed across the control valve and in many cases this will greatly exceed the design head of the secondary pump, which will render the secondary circuit unstable.

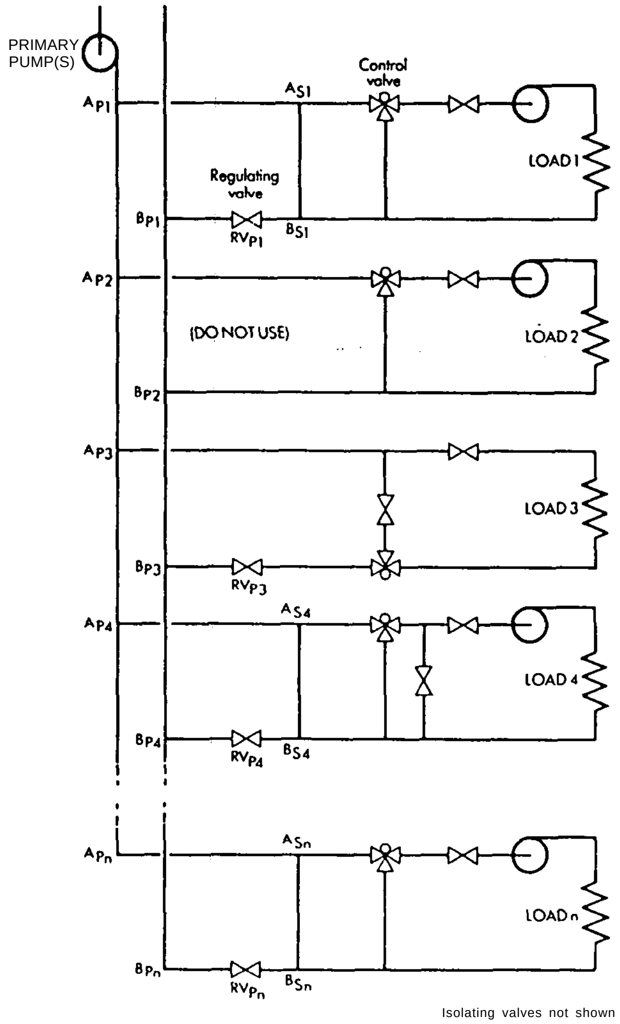



**Fig. B11.51.** Primary/secondary multi-system circuits.

Circuit 3 shows a conventional diverting application where $RV_{P3}$ is used to absorb any excess head not required by the secondary circuit. As in circuits 1 and *n*, the volume and pressure drop through $RV_{P3}$ are constant, as is the head across $A_{P3}$-$B_{P3}$ and secondary circuit operates under constant applied pressure conditions. Circuit 4 is an injection circuit operating under the same stable conditions as circuits 1 and *n*.

**Air Systems**

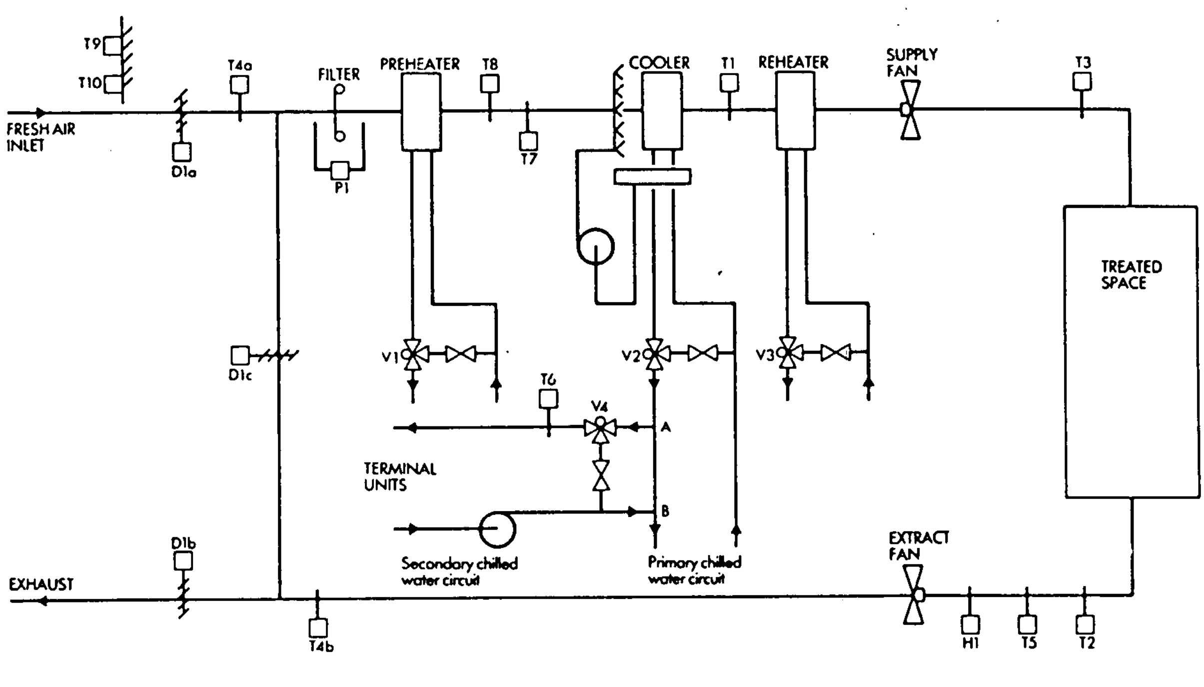
The control of air systems may be considered in two parts, the first related to the main air plant and the second to the terminal units. The basic controls for the main plants are generally the same regardless of system type. However, some systems may require additional controls. The dewpoint control system with its various options and special features, represents a large proportion of the problems encountered in the control of all air systems. The psychrometrics for air conditioning and air systems generally are covered in detail in Section B3 of the *Guide* and are an essential part of understanding the operation of the control system. Only the basic psychrometrics necessary for particular elements of the schemes described below are shown here.

In energy conservation terms, overall dewpoint systems may not be the most efficient. Sequential heating and cooling with separate humidification and a cooling coil override feature for dehumidification, or dewpoint control only for the fresh air supply, should be considered as alternatives. However, each system should be considered both in terms of control accuracy and the 'free cooling' potential of the dewpoint plant.

*Dewpoint Control and Auxiliary Features*

In the system shown in Fig. B11.52 the dewpoint is controlled by T1 which sequentially modulates a preheater battery control valve V1, dampers D1a, D1b and D1c, which operate in parallel, and a sprayed cooler battery valve V2 to maintain a constant saturated temperature condition. The reheater, which is part of many systems, is controlled by the extract temperature detector T2, modulating the control valve V3 to maintain a constant space temperature. A low limit detector (T3) is sometimes employed in the discharge duct to override T2 and maintain the discharge temperature above a predetermined limit. Floating thermal feedback, or proportional plus integral controllers should be used, because, for stability, proportional control alone requires a wide band which results in unacceptable deviation in dewpoint.

The psychrometrics of the system are shown on Fig. B11.53 where the dewpoint plant is set to provide a moisture content of H so that air is supplied to the space after reheat in a condition S. This permits the design room condition R to be achieved when the latent gains (R-X) are added and the sensible losses ($S_W$-X) or gains ($S_S$-X) are taken into account. In winter when the incoming air is at $O_W$, the dampers can be modulated to achieve a mixed air condition corresponding to condition M, with the cooler and preheater both off (any point on the line $O_W$-R, represents a mixed air condition with $O_W$ defining full fresh air and R, total air recirculation). Normally there is a requirement that the fresh air quantity shall not fall below a minimum quantity represented by $M_F$, which means that the dampers cannot take up a position between $M_F$ and R. From M the air is then adiabatically saturated




T1 Dewpoint detector
T2 Temperature detector
T3 Low limit detector
T4 Enthalpy detectors/comparators
T5 Boost limit thermostat
T6 Secondary water control detector
T7 Modulating low limit
T8 Frost protection thermostat
T9 Free cooling limit thermostat
T10 Free cooling enthalpy detector
H1 Return air humidity detector
P1 Differential pressure switch
V1 Preheater valve
V2 Cooler valve
V3 Reheater valve
V4 Secondary chilled water valve

**Fig. B11.52.** Dewpoint Schematic.

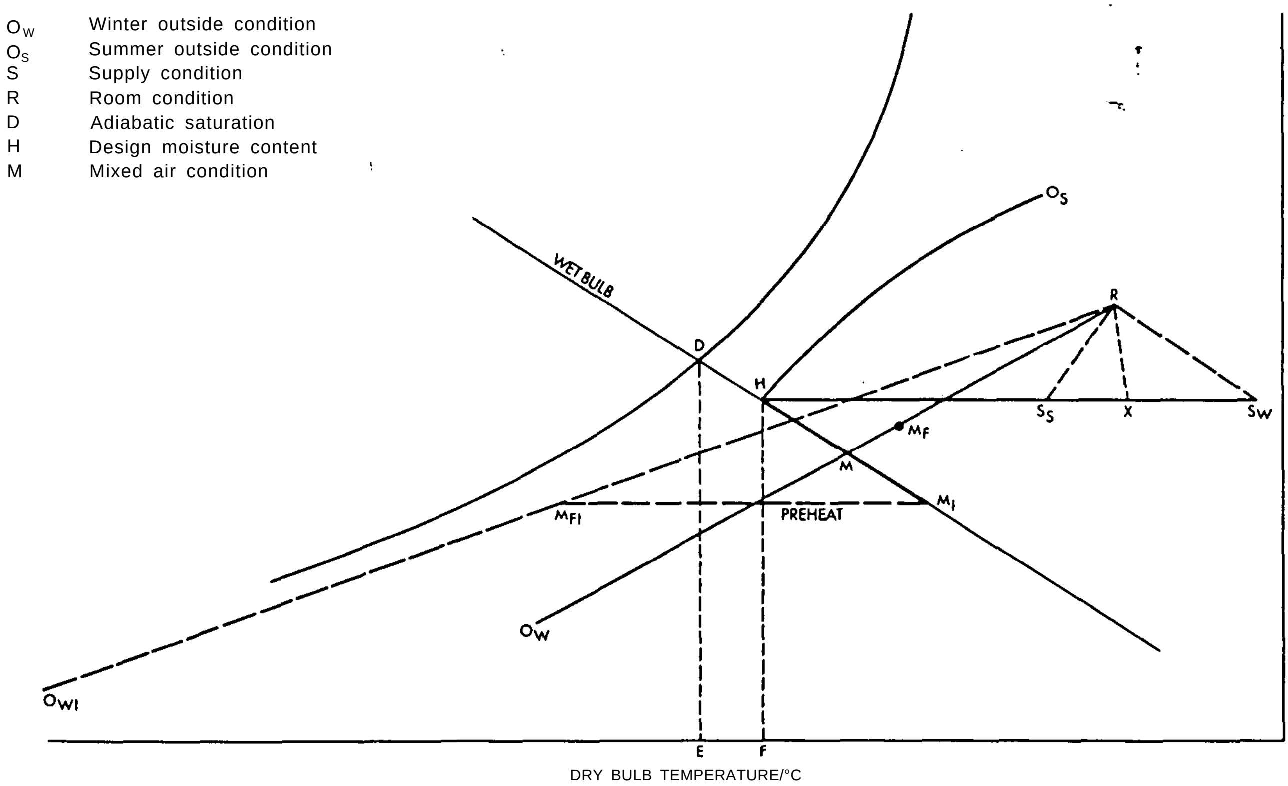



**Fig. B11.53.** Psychrometric process for a dewpoint plant.

as it passes through the spray coil, moving the condition point along the wet bulb line to H. An ideal spray coil would saturate to point D where the wet- and dry-bulb settings are identical, but in practice the spray is not perfect and the dry-bulb detector (T1) is set at F to ensure that H corresponds to the required moisture content. When selecting the dewpoint, allowances should be made for the inefficiency of the spray coil and the latent gains to the space (R-X).

In a very cold situation, such as shown at $O_{W1}$, the room condition R, at start up, may be below the line D-M and the mixing point $M_1$ cannot be achieved. Alternatively the mixing condition may not be possible because the minimum fresh air position $M_{F1}$ is below the line D-M. In either case, the dampers are modulated to $M_{F1}$ (points on line $M_{F1}$-R not being permitted) and the preheater control valve is then modulated from closed towards the open position to reach the wet-bulb line at $M_1$, where adiabatic saturation again takes place to move the condition point to H. In any situation where point M (or $M_1$, etc.) can be achieved by the use of mixed fresh and recirculating air, with or without the preheat, and with no output from the cooling coil, the term 'free cooling' is used, as the dewpoint can then be maintained without the assistance of mechanical cooling. Thus in very cold weather the dampers are in the minimum fresh air position to minimise preheating and, as the external temperature rises, the preheater valve closes and the dampers modulate to the full fresh air condition.

In summer conditions such as $O_S$, the preheater is off, the dampers are in the full fresh air position and the cooler valve $V_2$ is modulated to achieve point H. Although each of the previous steps have been described as discrete operations the three stage controller permits continuous sequential operation of each stage according to load conditions. When point H is achieved it is necessary to reheat the air to the desired discharge condition, point $S_W$ or $S_S$ for winter and summer conditions respectively.

A reduction in mechanical cooling load can be provided by overriding the damper control in certain conditions. For all practical purposes either wet-bulb or specific enthalphy can be used to measure the total heat content of the air. Where the enthalpy of the outside air exceeds that of the return air and it is more economic to cool return air than outside air. A detection device is therefore required to measure the total heat (enthalpy) or wet-bulb temperature. The enthalpy of the outside air can be measured directly (T4a), see Fig. B11.53, as being above the design room value or by dual detectors (T4a/T4b) which compare the room and outside air total heat conditions. In either case, when the room total heat is exceeded by the outside air, a signal from the device drives the dampers to the minimum fresh air position.

Free cooling may also be utilised to satisfy the secondary cooling requirements of terminals which may operate with water chilled to only about 12°C. In such circumstances it is common to feed the terminal circuit from the return from the main cooling coil (points A and B in Fig. B11.53). It is then necessary to maximise the use of fresh air in winter to provide water at a suitable temperature by running the primary chilled water pumps and spray pumps without energising the chillers or the condenser pumps.

The sequence for this (utilising V4, T6, T7, T8 and T9) and the functions of T5, T10, H and P are described in the Applications Manual. The detailed interlocking for dewpoint plants can be complex.

The use of the basic dewpoint arrangements, or variations with other forms of humidity control, can be applied to a wide range of generically named systems, including:

Single duct constant volume systems,
Dual duct and multi-zone units,
Induction units,
Fan-coil units,
Variable air volume systems.

Each of these is described in detail in the CIBSE Applications Manual, including specific problems associated with variable air volume systems.

**Solar Panel Systems**
The advent of solar collector systems has indicated one specific control need which is particularly relevant to these systems. This is the provision of suitable differential temperature controllers and detectors to ensure that fluid is only pumped through the collectors when useful heat can be gained. Accurate and matched temperature sensors are required in the form of thermocouples or resistance elements, such as thermistors or resistance thermometers. In all cases the sensors should be installed so that they are sensitive to changes in the measured fluid and their accuracy should be constant with changing ambient temperature. More sophisticated controllers can incorporate additional circuits to prevent short cycling of the pump, to prevent overheating of the storage vessel, to activate heat dispersal systems, or to operate drain-off valves for frost protection.

**Heat Reclaim Systems**
The increasing need for energy conservation has encouraged the use of various types of reclaim device. The control procedures which can be adopted for some of the more commonly used heat reclaim systems are identified below. Double bundle condensers and heat exchangers in exhaust flues are already in general use and their control is adequately covered elsewhere.

Heat reclaim or recovery systems are frequently part of a sophisticated installation and care should be taken to ensure that the proposals described below are an inherent part of the design philosophy. Always check that the energy apparently saved by the heat transfer capability of the device takes account of the additional ‘parasitic’ energy required to drive the device.

*Thermal Wheels*
These are rotating air to air heat transfer devices between two separate air streams, in parallel and adjoining ductwork. The speed of rotation will not normally exceed 0.3 rev/s and the heat recovered decreases with increasing speed. The control of energy transfer may therefore be carried out by varying either the speed of rotation of the wheel or the quantity of exhaust air passing through.

*Liquid Coupled Indirect Heat Exchangers (Run-round Coils)*
In its simplest form the pump may be controlled by an externally mounted thermostat which operates the pump whenever the external temperature is below the design exhaust air temperature. Alternatively, a differential thermostat may be used, which operates the pump only when the temperature of the exhaust air is higher than the supply air. A further alternative is to use a detector downstream of the preheater coil which modulates a three-port control valve to maintain the desired temperature. In this case the pump is switched off by means of a micro-switch in the valve actuator when the valve reaches the full bypass position. All three schemes would operate only during the normal plant running periods.

*Cross Flow Stationary Recuperator (Air to Air Heat Exchanger)*
This device is an alternative to the thermal wheel but provides only sensible heat transfer. The control system being similar to the bypass damper arrangement described for the thermal wheel.

*Heat Pumps*
The range of heat pump applications and the availability of suitable models is increasing rapidly and the control arrangements are normally included as part of the manufacturer’s package. The most common form of heat pump incorporates a refrigerant compressor unit with evaporator and condenser where the functions of the latter two elements may be reversed, dependent on whether a heating or cooling cycle is required. The basic control requirement for the changeover system is a four port refrigerant reversing valve which is used to select either the cooling or heating mode of operation according to load. The other controls are effectively those which would normally be provided with chillers and condensers. The changeover system is commonly used in air conditioned buildings where a series of heat pumps having individual air circulating fans are coupled to a water circulating system, details of which are given in the Applications Manual.

**Frost Protection and Associated Interlocking**
The use of frost protection systems may in general be split into those which are used for heating systems and those which are relevant to ventilation and air conditioning plants. The two elements which normally require protection are the plant, with its associated pipework, and the treated spaces.

*Heating Systems*
Protection of heating systems to ensure minimum energy consumption is best accomplished by a two or three stage scheme which starts only the relevant items of plant. The basic functions are as follows. If the external temperature falls to 2°C when the plant is off the circulating pumps are started to ensure that freezing will not occur at any exposed points in the pipework. When the pumps are operating in this way the immersion thermostat, in the common return to the boilers, monitors the water temperature and if it falls to, say, 20°C the boiler system is started to ensure that sufficient heat is supplied to the system to protect it. The thermostat should be of a type having an adjustable differential which will prevent short cycling of the boilers by holding the boilers on until the return water temperature rises to, say, 30°C. An additional level of protection may be provided by a room thermostat in an exposed corner of an upper floor, set at approximately 10°C. This will start both the heating and circulating systems independently of the first two stages of protection and the plant will run until the space temperature reaches minimum acceptable level. Variations of this system may be employed.

*Plenum and Ventilation Systems*

With ventilation or plenum heating, the major area of protection is related to the prevention of freezing in heating coils. In plenum plants using only fresh air, a hand reset frost protection thermostat with a serpentined detecting element across the face of the battery is essential to prevent freezing, either due to incorrectly sequenced starting times of the fan and the boilers/pumps, or a failure in the latter. The thermostat is set at approximately 3°C and if the temperature falls to this level the thermostat stops the fan and initiates an alarm. The hand reset feature ensures that the reason for the failure is investigated before the plant is restarted. With such systems the interlocking should be arranged so that the control valve on the heating coil is motored to the open position during periods of plant shut down. This ensures that the frost protection described for heating systems can circulate water through the coil in cold conditions to prevent it being frozen by very cold air being naturally drawn or blown over the coil.

*Air Conditioning Systems*

In full air conditioning plants there are several considerations related to frost protection. The system often incorporates a preheater but in temperate climates this may be unnecessary as sufficient preheat capacity is normally available from the recirculated air. Where a preheater is used, a frost protection thermostat with serpentined detecting element would be installed upstream of the preheater. The element should be capable of reacting to the lowest temperature anywhere in the duct cross section. If there is no preheater, the thermostat would be positioned adjacent to the dewpoint detector. In both cases the dampers should be interlocked with the system. In addition, the spray pump and chiller should be held off until the boost heating cycle has been completed. Where the preheater or reheater are adjacent to the cooler coil consider whether opening the heater valves at night might sufficiently heat the water in the cooling coil to damage the chiller on start up, due to a warm slug of water through the evaporator. The use of ‘free cooling’ arrangements in air conditioning plants may also affect the arrangements. When independent spray units are used upstream of the cooling coil, instead of spray coils, additional care must be taken to prevent freezing of the sprays under free cooling conditions.

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# SECTION B12 SOUND CONTROL




*This edition of Section B12 first published: 1986*

# SECTION BI2 SOUND CONTROL

**Introduction**

Sound is oscillatory variation of atmospheric pressure within the limits of amplitude and frequency to which the human ear responds. It is created by radiation of energy from a vibrating source and is propagated through solids and fluids in the form of pressure waves. Sound which is undesirable to the listener is termed Noise.

This Section of the *Guide* considers acoustics in the design of building services. Section Al, *Environmental Criteria for Design*, relates to sound and vibration in terms of human response and the ‘comfortable’ acoustic environment.

## ACOUSTIC TERMINOLOGY

### The Decibel

The logarithmic ratio between two quantities proportional to power is:

$$R = \log_{10} \frac{x}{y} \quad \ldots \quad \text{B12.1}$$

where :

| | | | |
|---|---|---|---|
| $R$ | = logarithmic ratio | .. .. .. | B |
| $x$ | = a quantity | | |
| $y$ | = another quantity with the same units | | |

This may also be expressed as:

$$R_1 = 10\,R = 10 \log_{10} \frac{x}{y} \quad \ldots \quad \text{B12.2}$$

where :

| | | | |
|---|---|---|---|
| $R_1$ | = decibel ratio | .. .. .. .. | dB |

The ear responds logarithmically to a very wide range of energy intensities between $10^2$ W/m$^2$ (painful) and $10^{-12}$ W/m$^2$ (just detectable). It is therefore convenient to use a logarithmic index to measure sound levels. Using this convention a level is expressed in relation to a reference quantity, usually the lowest detectable by the ear. The reference levels usually chosen are $10^{-12}$ W for sound power, $10^{-12}$ W/m$^2$ for sound intensity and 20μ Pa for sound pressure. The reference levels for sound intensity and sound pressure both correspond to the threshold of hearing. These reference levels are used in the *Guide*.

For a given uniform sound, the power of the source is constant, but its effects, e.g. sound intensity and pressure, depend on the point of measurement.

*Calculations with Decibels*

Absolute quantities such as power, power intensity or pressure can be added or subtracted, but *levels* of power, intensity or pressure expressed in decibels must follow the logarithmic convention, e.g. the total power level of two equal sources is:

$$L_{WT} = 10 \log_{10}\left(\frac{2W}{W_o}\right) \quad \ldots \quad \text{B12.3}$$

$$= 10 \log_{10}\left(\frac{W}{W_o}\right) + 10 \log_{10}(2) \quad \ldots \quad \text{B12.4}$$

$$\approx L_w + 3 \quad \ldots \quad \text{B12.5}$$

where :

| | | | |
|---|---|---|---|
| $L_{WT}$ | = total power level | .. .. .. | dB |
| $W$ | = power of each source | .. .. .. | W |
| $W_o$ | = reference power | .. .. .. .. | W |
| $L_W$ | = power level of one source | .. .. | dB |

Alternatively, use may be made of Fig. B12.1.

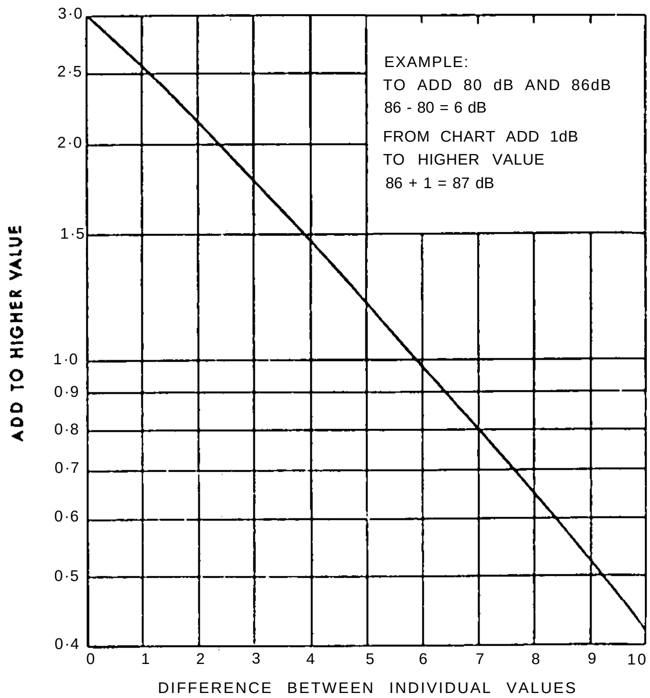



**Fig. B12.1.** Addition of decibels.

**Sound Power**

The sound power of a source is the rate at which acoustic energy is transferred from a vibrating source to a medium.

*Sound Power Level*

The sound power level of a source is given by:

$$L_W = 10 \log_{10}\left(\frac{W_s}{W_o}\right) \quad \text{B12.6}$$

where:

$L_W$ = sound power level of source .. .. dB

$W_s$ = sound power of source .. .. W

$W_o$ = reference sound power .. .. W

**Sound Intensity**

The sound intensity of a source is the mean rate of acoustic energy flow through unit area normal to the direction of propagation.

*Sound Intensity Level*

The sound intensity level of a source is given by:

$$L_I = 10 \log_{10}\left(\frac{I_s}{I_o}\right) \quad \text{B12.7}$$

where

$L_I$ = sound intensity level of source .. dB

$I_s$ = sound intensity of source .. .. $W/m^2$

$I_o$ = reference sound intensity .. .. $W/m^2$

**Sound Pressure**

The sound pressure of a source is the amplitude of the pressure variations in the wave.

*Sound Pressure Level*

The sound pressure level of a source is given by:

$$L_p = 10 \log_{10}\left(\frac{P_s}{P_o}\right)^2 \quad \text{B12.8}$$

$$= 20 \log_{10}\left(\frac{P_s}{P_o}\right) \quad \text{B12.9}$$

where

$L_p$ = sound pressure level of source .. .. dB

$P_s$ = sound pressure of source .. .. Pa

$P_o$ = reference sound pressure .. .. Pa

*Equivalent Continuous Sound Pressure Level*

The equivalent continuous sound pressure level is the sound pressure level of a continuous sound which has the same mean square sound pressure as the sound under consideration, the level of which varies with time. It is given by the following equation:

$$L_{eq} = 10 \log_{10} \frac{1}{T} \int_o^T \frac{P^2 (t)}{P_o^2} \, dt \quad \text{B12.10}$$

where:

$L_{eq}$ = equivalent continuous sound pressure level over time interval $T$ .. .. dB

$p(t)$ = instantaneous sound pressure.. Pa

$p_o$ = reference sound pressure $(20 \times 10^{-6})$.. Pa

If the measurement is performed using an A-weighted network (see *dBA Values,* below) the symbol $L_{Aeq}$ is used.

**Inverse Square Law**

In uniform free space, sound intensity and sound pressure follow an inverse square law as the distance from the source changes. For each doubling of the distance from the source, the *level* decreases by approximately 6 dB. It should be noted that this is not the case when sound is confined in a duct or in a room.

**Octave Band Spectra**

The ear responds to frequencies in the range 15 Hz to 20 kHz. For convenience in calculation and comparison, acoustic data are specified in frequency bands normally one octave wide. The preferred octave bands are: 31·5 Hz, 63 Hz, 125 Hz, 250 Hz, 500 Hz, 1 kHz, 2 kHz, 4 kHz, 8 kHz and 16 kHz. The end bands (31·5 Hz and 16 kHz) are rarely employed.

**dBA Values**

Measuring instruments can give single-figure values covering the whole frequency range. These may be un-weighted (referred to as linear) or weighted. The most commonly used weighted scale is dBA, the measuring of which corresponds to the 40-phon loudness level of Fig A1.12, see Section Al.

**NR and NC Curves**

Noise criterion curves are a method of using octave bands for rating the background noise level for annoyance and speech intelligibility in a given environment, and are attempts to express equal human tolerance in each frequency band. Beranek[1] and Kosten[2] have carried out experiments with office workers and home dwellers using broad-band noise sources. They produced the well-known NC and NR curves, the latter being commonly used in Europe (see Section Al).

NR curves, which are accepted throughout this *Guide*, give a slightly more stringent requirement than NC curves in the higher frequencies. For all practical purposes, NR and NC curves may be regarded as mutually interchangeable. The recommended noise ratings given in Section Al apply to broad-band continuous noise.

**Specification of Acoustic Data**

*Octave Band Spectra and Pure Tones*

Equipment should be rated in terms of the sound power level relative to $10^{-12}$ W and stated for each octave band. Levels may be stated for the 63 Hz band but the data can be expected to be less accurate. The tests' tolerances should be stated particularly for the lower and upper bands where measurements may be less accurate.

Special attention should be given to any pure tones present in the sound spectrum and also the working or design conditions which aggravate their generation (e.g. excessive air velocities, damaged fins on heat exchangers). It can be useful to quote the weighted sound pressure level but this should not be used in design calculations; it merely gives 'a feel' for the noisiness and is useful for comparing broad-band sound levels.

*Precautions*
Manufacturers are advised to state precautions necessary in installation to avoid poor acoustic performance. Equipment can both attenuate and generate noise and both characteristics should be taken into account.

*Sound Reduction Index*
A sound reduction index is a statement of the ability of a construction to reduce the transmission of sound through it. However, the noise level difference between the two sides also depends on the acoustic conditions on the listening sides (see equation B12.36). A sound reduction index should be stated as a function of frequency.

## NOISE FROM BUILDING SERVICES

The characteristic travel of continuous sound waves in solids, liquids and gases, also the influence of boundaries are discussed elsewhere.[3] Typical air-borne and structure-borne sound transmission paths in buildings are illustrated in Fig. B12.2. Noise sources may be regarded as falling into the following classifications:

In order to assess the noise level reaching a noise sensitive room, it is necessary to determine the sound power level of the individual sources of noise, the respective modifying characteristics of their transmission paths and the acoustic properties of the room and then sum the resulting levels.

## NOISE SOURCES

### Central Plant Noise

*Fans*
Fans are a major source of noise in ventilating systems. Noise is generated in a fan by:

(*a*) rotation of blades in air,

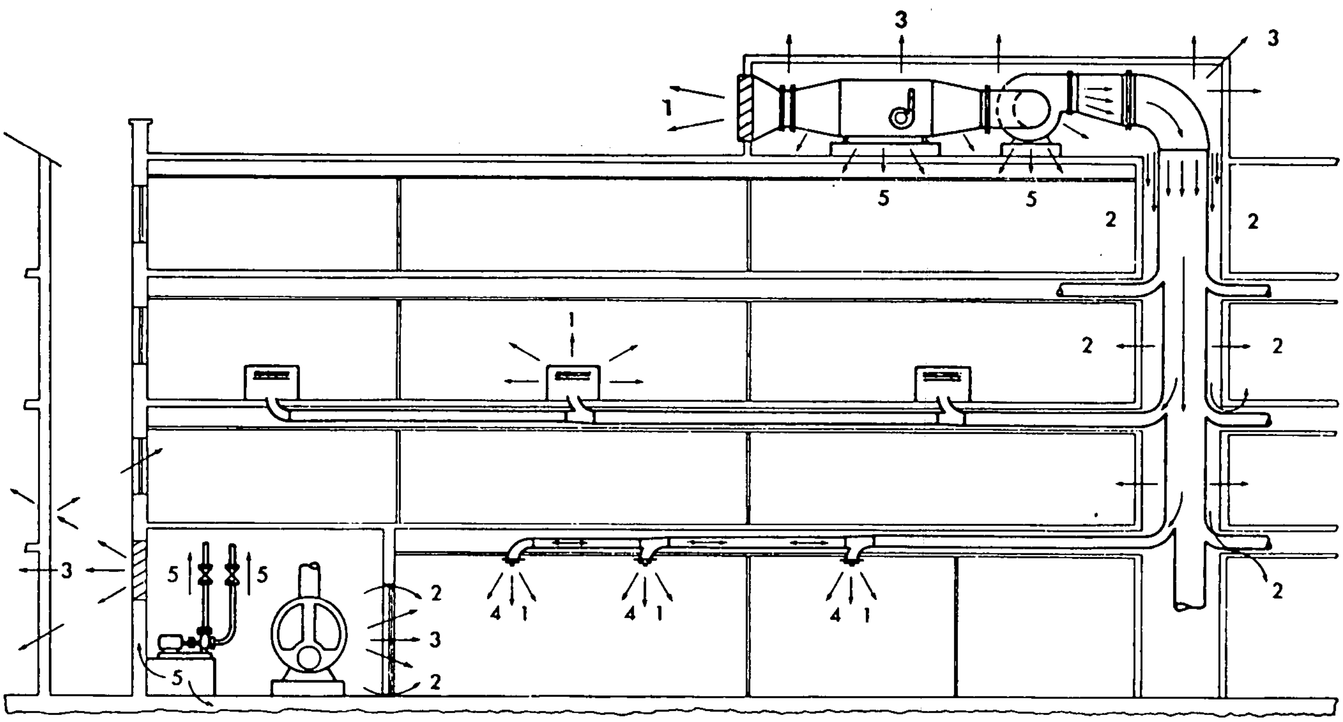
(*b*) fluctuating air pressures in the airstream and their interaction with the surfaces of the fan,

(*c*) resonant vibration of the fan casing,

(*d*) vibration excited by the fan drive and motor.

The level of the noise generated increases with the fan speed and the spectrum shape can change as the operating point moves along the constant speed characteristic of a fan. As a general rule for a constant fan speed the minimum noise level occurs when the fan is operating at about maximum efficiency.

The pressure fluctuations of turbulent flow are random in character and give rise to broad-band noise covering the audible frequency range. Steady blade rotation generates pure tones at the blade passage frequency and its higher harmonics. Induced vibrations also radiate pure tones. Fan noise then consists of a broad-band spectrum plus some pure tones. If the pure tones are below the broad-band level then normally only the broad-band noise need be considered. If they are above the broad-




(1) Airborne plant noise distributed through ducts.
(2) Airborne plant noise transmitted through gaps to occupied spaces directly or indirectly.
(3) Airborne plant noise transmitted through structure to occupied spaces and neighbourhood.
(4) Airborne room noise transfer to adjacent occupied spaces via interconnecting ducts.
(5) Plant vibration transmitted to building structure, pipes and ducts and thence to occupied areas.

**Fig. B12.2.** Noise transmission paths.

band level (by say 3 dB more than mean of the adjacent one third octave band levels) then they present a special problem and they need to be separately considered.

The fan noise data required for assessing the performance of a ventilating system are the octave band sound power levels:

(*a*) radiated through the fan inlet,

(*b*) radiated through the fan outlet,

(*c*) radiated through the fan casing and from an external drive.

BS 848: Part 2, Fan noise testing[4] sets out procedures for measuring (*a*) and (*b*) and gives recommendations for assessing the levels of (*c*) and any audible pure tones present. Tests carried out in accordance with this standard give octave band sound power levels in frequency bands with centre frequencies of 125, 250, 500, 1000, 2000 and 4000 Hz. The tests use airways that provide uniform flow conditions. Good connections like these are of importance and where space restrictions lead to poor connections, higher noise levels can be expected. There is little data available on the effect of connections but where the duct disturbs the flow at the inlet, up to 5 dB should be added to the noise spectrum determined by the standard test. Preferably the fan noise spectrum used in any noise assessment should be that obtained from the British Standard tests. If this is not available then an estimate of the sound power generated can be made from a computed value of the overall sound power level and a typical shape of the noise spectrum for that type of fan.

Typical fan noise spectra for centrifugal and axial flow fans are illustrated in Fig. B12.3. These give the octave band level relative to the overall fan sound power level with an accuracy of about ± 4 dB.

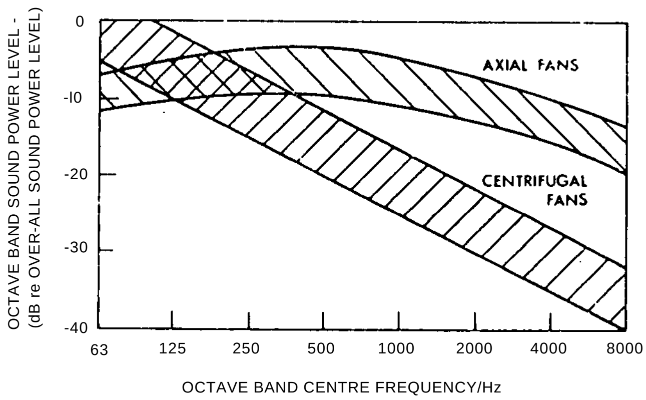



**Fig. B12.3.** Typical fan noise spectra.

The sound power level for each of the inlet or outlet connections can be estimated from the aerodynamic performance of the fan by one of the following empirical equations:

$$L_w = 67 + 10 \log_{10} M + 10 \log_{10} h \quad \text{B12.11}$$
$$= 40 + 10 \log_{10} V + 20 \log_{10} h \quad \text{B12.12}$$
$$= 95 + 20 \log_{10} M - 10 \log_{10} V \quad \text{B12.13}$$

where:

$V$ = delivered volume .. .. .. $m^3/s$

$h$ = fan static pressure .. .. .. $N/m^2$

$M$ = rated fan motor power .. .. kW

The total sound power level of a fan is the sum of the inlet and outlet sound power levels. In the above method of estimation the total sound power level becomes 3 dB greater than the inlet or outlet sound power levels since these have been assumed to be equal.

This method of estimating the sound power levels of a fan assumes a broad-band spectra. There may be pure tones arising from the blade passage frequency and therefore it is recommended that 5 dB should be added to the computed level of the octave band containing the blade passage frequency and 3 dB to its first harmonic. These frequencies are:

$$f_b = b \times n \quad \text{B12.14}$$
$$h_1 = 2 f_b \quad \text{B12.15}$$

where:

$f_b$ = blade passage frequency .. .. Hz

$b$ = number of blades

$n$ = fan speed .. .. .. .. rev/s

$h_1$ = first harmonic .. .. .. .. Hz

*Cooling Towers*

The components of cooling tower noise arise from:

(*a*) the fan

(*b*) the fan drive assembly

(*c*) water splashing into the sumps.

The noise can be transmitted through the water circulation pipes, the air intake or air discharge louvres and the cooling tower walls. The motor and fan may also give rise to structure borne sound. Investigation into the nature of cooling tower noise by Dyer and Miller[5] suggests that in general the spectrum is that as shown in Fig. B12.4. The low frequency energy arises from the fan and the high frequency energy is due to splashing of the water. The overall sound power level can be estimated from:

$$L_w = 11{\cdot}5 + 10 \log_{10} P \quad \text{B12.16}$$

where :

P = total rated fan power output .. W

The sound pressure level at a distance from the cooling tower for each octave band can be calculated from Fig. B12.4 and the expression:

$$L_p = L_w - 10 \log_{10} 4\pi r^2 + D \quad \text{B12.17}$$

where:

$r$ = distance from cooling tower .. m

$D$ = directivity index of cooling tower .. dB

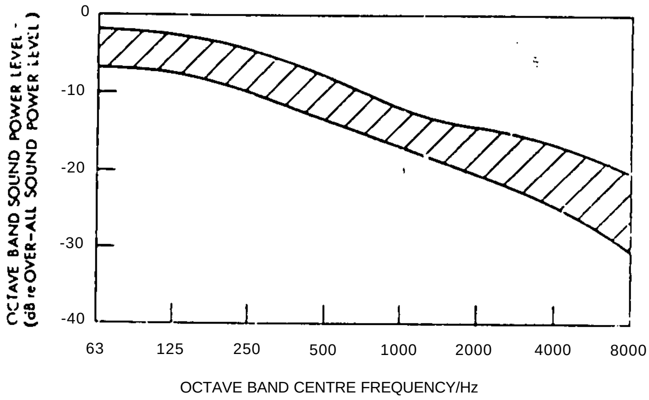



**Fig. B12.4.** Typical cooling tower noise spectrum.

*Refrigeration Equipment*

Some typical spectra are shown in Fig. B12.5. The noise radiated by a centrifugal compressor is characterized by the compressor blade passage frequency (see equation B12.14). For reciprocating compressors, one-third octave band analysis reveals that pure tones can be expected to occur at the piston stroke frequency and its harmonics, where:

$$f_p = n \times C \quad \ldots \quad \text{B12.18}$$

where:

$f_p$ = piston stroke frequency .. .. Hz
$C$ = number of cylinders
$n$ = compressor speed .. .. .. rev/s

Great care must be taken to minimize the structure-borne sound which may be transmitted through the floor slab on which the equipment is mounted or through suspended piping:

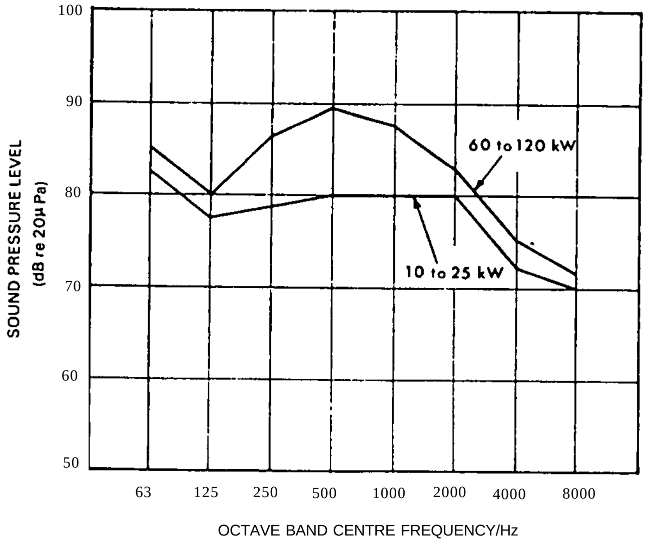



**Fig. B12.5.** Typical refrigerating equipment noise spectra.

*Boilers*

Sound power levels tend to depend on the type of burner, very little variation occurring for different types of fuel.

The sound pressure level (dBA) at 1 metre from the fuel burner may be taken as:

$$L_{pA} = 12.5 \log_{10} Q + 20 \quad \ldots \quad \text{B12.19}$$

where:

$Q$ = thermal output of burner .. .. W
$L_{pA}$ = sound level .. .. .. .. dB

The accuracy of this level is about ± 5 dBA.

The octave band sound pressure levels may be assessed from the dBA value using the corrections of Table B12.1. Some reduction in boiler noise may be achieved by fitting acoustic baffles to draught stabilizers and acoustic enclosures to the burner unit and ancillary equipment.

**Table B12.1.** Spectrum corrections for burners.

| Boiler range (kw) | Correction (dB) Octave band centre frequency (Hz) 125 | 250 | 500 | 1000 | 2000 | 4000 |
|---|---|---|---|---|---|---|
| Less than 150 | −4 | 0 | −1 | −3 | −8 | −13 |
| 150 to 350 | −7 | −3 | −1 | −4 | −7 | −11 |
| 350 to 3500 | −5 | +2 | −4 | −6 | −6 | −7 |

The sound pressure level (dBA) in chimneys may be estimated from:

$$L_{pA} = 9 \log_{10} Q + 45 \quad \ldots \quad \text{B12.20}$$

The octave band sound pressure levels may be assessed from the dBA value using the corrections of Table B12.2.

**Table B12.2.** Spectrum corrections for chimneys.

| Boiler range (kW) | Correction (dB) Octave band centre frequency (Hz) 125 | 250 | 500 | 1000 | 2000 | 4000 |
|---|---|---|---|---|---|---|
| Less than 150 | + 5 | +2 | +1 | −2 | − 6 | − 9 |
| 150 to 350 | + 6 | +3 | −2 | −3 | − 6 | − 9 |
| 350 to 3500 | +10 | +6 | 0 | −6 | −10 | −11 |

These sound pressure levels may be attenuated by the flue which can be considered as a duct. The sound power level in the chimney may be estimated from:

$$L_w = L_p + 10 \log_{10} A \quad \ldots \quad \text{B12.21}$$

where:

$A$ = cross sectional area of chimney .. $m^2$

The sound pressure level at a distance from the discharge may be estimated from:

$$L_p = L_w - 10 \log_{10} 4\pi r^2 + D \quad \ldots \quad \ldots \quad \text{B12.22}$$

where:

$r$ = distance from discharge .. .. m

$D$ = directivity index of chimney .. .. dB

*Motors*

Equipment driven by an electric motor will make more noise than the motor, but every motor, independent of pole number and speed, emits a fundamental tone equal to twice the supply frequency.

*Plant Rooms*

Some work by BSRIA is shown in Fig. B12.6 to 8 which compare the noise levels to be expected in boiler, fan and compressor plant rooms.

*Other Central Station Plant Noise*

Noise can be generated by air flow through central station equipment and by water sprays in the humidifier section but this is not normally significant.

**Noise in Water Systems**

Noise is generated in water systems from many causes, some of which are listed below, the most usual being either (*a*) or (*b*):

(*a*) Boilers, burners, induced or forced draught fans, pumps and motors. The noise may be transmitted through the water or the pipe walls over long distances or may be radiated directly to the surroundings (i.e. airborne noise) or may be transmitted through supports or contact building structure (i.e. structure borne noise) where the energy can be re-radiated from the surface as airborne noise.

(*b*) Valves and fittings in the water distribution network. The noise generated depends on the flow conditions and the type of fittings. Water hammer is a particular noise that occurs when very sudden pressure fluctuations take place.

(*c*) Waterflow.

*Pumps*

For an impeller as for a fan, the blade passage frequency is given by equation B12.14. The harmonics of this fundamental frequency can also be transmitted through the water system. This might be a problem with high-pressure pumps.

Flanking transmission may be minimized by using flexible connections on the pump inlet and discharge. The pump may also need vibration isolation at support points. The flexible connections should be at least five times the pipe diameter in length. Purpose made flexible connectors are available and the advice of the manufacturer should be sought. The pump vibration becomes more noticeable when the impeller becomes unbalanced due to dirt, sediment or wear.

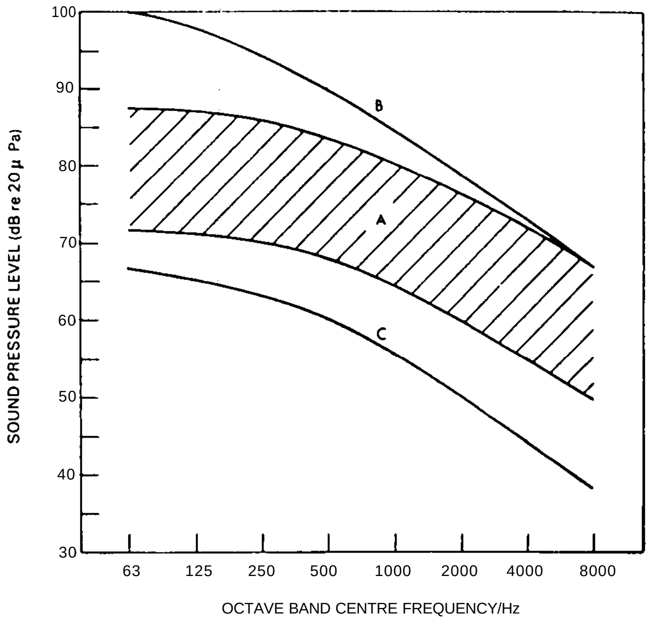



**Fig. B12.6.** Typical boiler room noise spectrum.

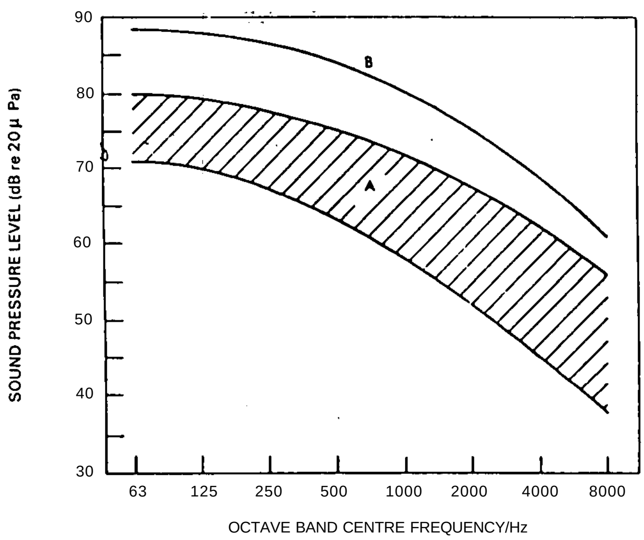



**Fig. B12.7.** Typical fan room noise spectrum.

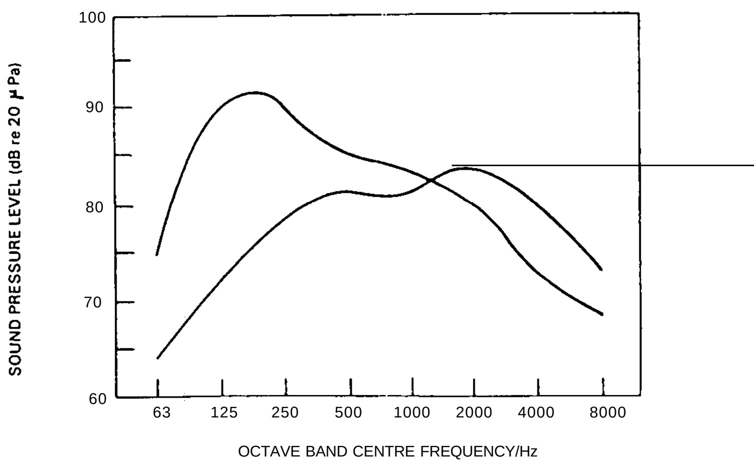



**Fig. B12.8.** Typical compressor room noise spectrum.

In Fig. B12.6 band A represents normal cases, curve B represents severely cramped boiler rooms and curve C represents unusually spacious boiler rooms. In Fig. B12.7. derived from 11 plant rooms, 8 fell within band A and curve B indicates the highest levels.

*Valves and Fittings*

The noise generated at these resistances is due to the disturbed water flow conditions created by them. Eddy flows and vortex formation give rise to noise but the highest noise levels are due to cavitation and water-hammer. Cavitation noise occurs when vapour bubbles are formed. Waterhammer is a function of the rate of pressure change; in this respect quick acting valves are to be avoided. It has also been found that corroded pipes give higher noise emissions than clean pipes.[6] It may be concluded that:

(*a*) Sound pressure level is inversely related to pipe diameter for given water velocity.

(*b*) The shape of the fittings is more important than the number used.

The sound pressure level is related to the energy loss (i.e. flow rate times pressure drop).

*Water Generated Noise*

In practice sound energy is about $10^{-1}$ to $10^{-6}$ of the total energy lost. The sound originates from eddies and vortices formed in the water stream and is at frequencies about 1000 Hz. The probability of forming such sound sources depends on the flow conditions and the pipe surface roughness.

Kristensen's[7] work suggests velocities of 2 to 4 m/s would be acceptable from an acoustic point of view in small diameter pipes in many situations.

*Isolation of Pipelines*

The blade passage frequency of the pump may excite the mains into their bending frequency modes of vibration. This may cause high stresses in the mains and/or in the supports and may also cause the transmission of structure borne noise and vibration into the building. To avoid these problems it is necessary to space the supports of the mains so that the fundamental bending frequency mode is not the same or within ±20% (±50% with resilient supports) of the pump speed and the blade passage frequency. Fig. B12.9 shows the fundamental bending frequency as a function of the distance between supports.

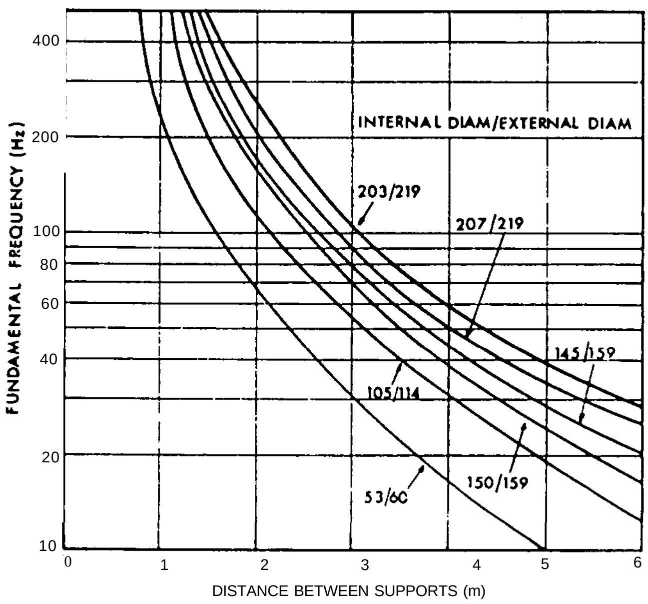



**Fig. B12.9.** Fundamental bending frequency (steel pipe).

Even with flexible connections in the mains and with the appropriate support spacing, noise may be transmitted to the structure from the mains. It is therefore desirable to isolate the mains wherever they are adjacent to areas requiring a noise level below about NR 35.

When pipes are passing through walls they are usually sleeved and should be isolated from the wall by suitable materials. It is worthy of note that the annulus between pipe and sleeve is a potential path for airborne noise and should be packed when necessary.

**External Noise**

*Road Traffic*

Road traffic is the most prevalent external noise source but other major sources are aircraft, railways, road works, factories, refuse disposal services. Each case must be treated on its merits and advice sought about the external noise climate near a building. Research carried out at the Building Research Station enables the minimum distance desirable between houses and major roadways to be estimated using the method given in the DOE publication, Calculation of Road Traffic Noise.[8]

*External Noise from Equipment*

Plant either mounted externally or connected to outside through openings in the building fabric add to the external noise climate. If these constitute a significant part of the external noise climate at any time, day or night, then the sound level generated should be checked to see that it does not infringe social and legal obligations (e.g. BS 4142). Where necessary the plant or its arrangement should be modified to meet such obligations.

For noise radiating freely into space, the sound pressure level a distance from the source is given by:

$$L_p = L_w - 10 \log_{10} 4\pi r^2 + D \quad \ldots \quad \text{B12.23}$$

where:

| | | | |
|---|---|---|---|
| $D$ | = | directivity index of source | dB |
| $r$ | = | distance from source | m |

If the source is omni-directional, i.e. it radiates uniformly in all directions, then the directivity index depends on the mounting situation, typically see Table B12.10. If the source has directional radiation properties, then values in the direction of concern may be different. Information on directivity of equipment noise should be obtained from the manufacturer.

Equation B.12.23 gives a sound pressure level reduction of 6 dB for each doubling of the distance from the source. Over large distances this reduction can be influenced by physical obstacles, ground absorption and weather conditions. These invariably lead to an increase in the attenuation rate but without detailed information such factors should be ignored in a design analysis.

## NOISE IN AIR FLOW SYSTEMS

Noise is generated by the plant and by air flow through the ducts, principally at fittings and the terminals. The procedure for assessing the sound power levels radiated from the terminals of a ventilating system is:

(*a*) Quantify the sound power level radiated into the duct by the plant (essentially fan noise).

(*b*) Subtract natural attenuation of system between the plant and the first significant noise source in the system (e.g. a partly closed branch damper) to arrive at sound power level before this fitting. Add (logarithmically) the sound power level generated at this position to obtain the sound power level immediately after the fitting. The noise level of a fitting is significant if it is within 6 dB of the system noise.

(*c*) Repeat (*b*) for each section of system between noise sources up to the terminal.

(*d*) At terminal subtract end reflection (see later) and add (logarithmically) the sound power level of the terminal. This then gives the sound power level radiated into the rooms.

Octave band sound power data should be used in this procedure.

### Sound Attenuation in Duct Systems

The attenuation of sound through the components of a ducted system is as follows:

*Straight Sheet Metal Ducts*

Data on the attenuation of conventional plain rectangular and circular sheet metal ducts are given in Table B12.3. There is some evidence that, for rectangular ducts, the damping effect of thermal insulation firmly attached to the sides will approximately double the attenuation given for the 125 and 250 Hz octave bands.

*Straight Builders-work Ducts*

No attenuation should be allowed in builders-work ducts unless lined with an absorbing material.

**Table B12.3.** Attenuation in straight sheet metal ducts.

| Section | Mean duct dimension or diameter (mm) | Attenuation (dB/m) Octave band centre frequency (Hz) 125 | 250 | 500 and above |
|---|---|---|---|---|
| Rectangular | Up to 300 | 0·6 | 0·5 | 0·3 |
| | 300–450 | 0·6 | 0·3 | 0·3 |
| | 450–900 | 0·3 | 0·3 | 0·2 |
| | Over 900 | 0·3 | 0·2 | 0·1 |
| Circular | Up to 900 | 0·1 | 0·1 | 0·1 |
| | Over 900 | 0·03 | 0·03 | 0·06 |

*Bends*

Radiused bends, with or without splitters, provide negligible attenuation. Square-back bends reflect some of the sound energy back towards the noise source and this can significantly reduce the noise being propagated away from the source. This attenuation is shown in Table B12.4.

**Table B12.4.** Attenuation provided by 90° mitred bends without turning vanes.

| Minimum duct side (mm) | Attenuation (dB) Octave band centre frequency (Hz) 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
|---|---|---|---|---|---|---|---|
| 150 | 0 | 0 | 1 | 6 | 7 | 4 | 4 |
| 300 | 0 | 1 | 6 | 7 | 4 | 4 | 4 |
| 450 | 0 | 3 | 7 | 5 | 4 | 4 | 4 |
| 600 | 1 | 6 | 7 | 4 | 4 | 4 | 4 |
| 750 | 3 | 7 | 5 | 4 | 4 | 4 | 4 |

*Branches*

For most typical duct branches it can be assumed that the acoustic energy divides in direct proportion to the areas of the duct branches, see Fig. B12.10. That is the total acoustic energy remains constant but it is divided amongst the branches so that, in any one branch, the acoustic energy is less than that in the approach duct. The effective attenuation for a branch duct is given by:

$$A = 10 \log_{10} \left( \frac{a_1 + a_2}{a_1} \right) \quad .. \quad .. \quad .. \quad \text{B12.24}$$

where:

$a_1$ = area of branch .. .. .. $m^2$

$a_2$ = area of main duct after branch .. $m^2$

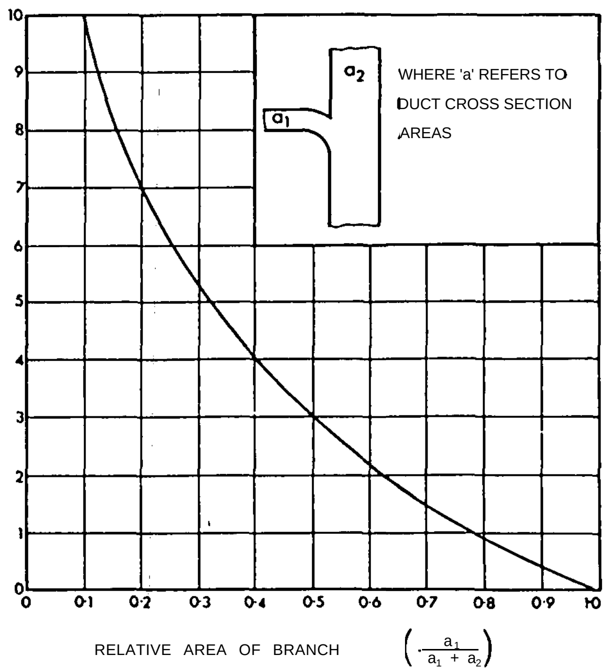



**Fig. B12.10.** Effective attenuation of a duct branch.

When the volumes divide and velocities are approximately equal, the calculation may be simplified by using air volume flow ratios instead of area ratios.

Where two or more such branches ultimately serve the same room the acoustic energy which divided amongst the branches becomes additive again within the room.

*End Reflection*

Sounds of wavelengths that are long with respect to the dimensions of the duct outlet tend to be reflected back within the duct rather than pass into the room. The theoretical value of the attenuation provided by end reflection at an opening is shown graphically in Fig. B12.11. In Fig. B12.11 the duct dimension is the diameter, for circular ducts, and $1{\cdot}13 \times \sqrt{\text{area}}$, for rectangular ducts.

It is recommended that this information be applied as follows:

*Grilles*

Use Fig. B12.11 with duct dimension determined from core area of grille. For high aspect ratio grilles (linear grilles) this is not strictly accurate but may be used in absence of specific information.

*Ceiling Diffusers*

Use Fig. B12.11 with duct dimension taken as 1·25 × neck diameter.

*Induction Units*

Fig. B12.11 should not be applied to induction units. These have substantial insertion loss and this includes some end reflection. This insertion loss can be at least 10 dB in all octave bands, accurate data should be obtained from the manufacturer.

If a bend immediately precedes a grille or diffuser the end reflection may be reduced. Where such an arrangement occurs the attenuation provided by end reflection should be taken as 50% of that recommended above.

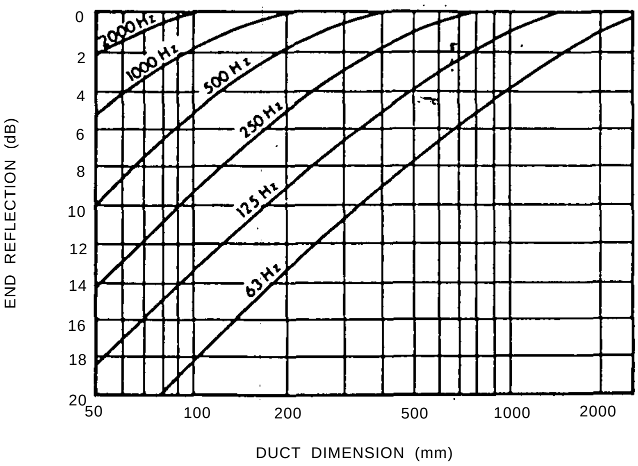



**Fig. B12.11.** Theoretical value of end reflection attenuation.

**Air Flow Generated Noise**

The movement of air through a ducted system generates noise. This noise can emanate from:

(*a*) Surface set in vibration by turbulent air flow. This can be significant at lower frequencies when surfaces are vibrating in phase at their natural frequencies producing a rumble noise. This is not normally a problem with adequately stiffened circular ducting and can be limited in high-velocity rectangular ducting by sensible design.

(*b*) Interaction between the fluctuating forces of turbulent flow and the internal surfaces of the ducts. This is generally the most significant source of airflow generated noise. It is most pronounced where turbulent flow impinges on or separates from such fittings as dampers, turning vanes and terminal equipment.

The magnitude of air-flow generated noise of type (*b*) depends strongly on the air velocity and the intensity of turbulence generated in the system. High levels of turbulence intensity can be prevented if flow separations in the

**Table B12.5.** Corrections for low turbulence duct fittings.

| Duct fitting | Value of C (dB) | Remarks | Octave band power level corrections (dB) — Octave band centre frequencies (Hz): 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
|---|---|---|---|---|---|---|---|---|---|
| Straight duct | –10 | No internal projections | –2 | – 7 | – 8 | –10 | –12 | –15 | –19 |
| 90° radiused bend | 0 | Aspect ratio <2:1<br>Throat radius ≥ $\frac{w}{2}$ | –2 | – 7 | – 8 | –10 | –12 | –15 | –19 |
| 90° square bend with turning vanes | +10 | Close spaced short radius single skin turning vanes | –2 | – 7 | – 8 | –10 | –12 | –15 | –19 |
| Gradual contraction | +1 | Area ratio 3:1 *A* and *v* taken for smaller duct | 0 | –10 | –16 | –20 | –22 | –25 | –30 |
| Sudden contraction | +4 | Area ratio 3:1 *A* and *v* taken for smaller duct | 0 | –10 | –16 | –20 | –22 | –25 | –30 |
| Butterfly damper | –5 | *A* and *v* apply to minimum free area at damper | –3 | – 9 | – 9 | –10 | –17 | –20 | –24 |

system are avoided, that is if duct fittings are selected that provide smooth transitions. The overall sound power level generated by a duct fitting can be approximately expressed by the equation:

$$L_w = C + 10 \log_{10} A + 60 \log_{10} v \quad \text{.. .. B12.25}$$

where:

$C$ = constant that depends on fitting and the flow turbulence

$A$ = minimum flow area of fitting .. m$^2$

$v$ = maximum velocity in fitting .. .. m/s

Typical values of $C$ for a range of duct fittings in low turbulence flow conditions and the corrections to obtain the octave band power levels are given in Table B12.5. These data provide some guidance on air-flow generated noise for velocities between 10 m/s and 30 m/s.

The octave band sound power levels for air-flow generated noise at grilles and diffusers should be obtained direct from the manufacturer. This information will refer to uniform flow conditions at entry to the terminal, for a bad entry higher generated noise levels can be expected.

In the absence of manufacturer's test data some idea of the generated noise level can be obtained from the following empirical relationships. However, these should not be used for an accurate assessment.

*Grilles*

$$L_w = 17 + 10 \log_{10} A + 60 \log_{10} v \quad \text{.. B12.26}$$

*Diffusers*

$$L_w = 32 + 13 \log_{10} A + 60 \log_{10} v \quad \text{.. B12.27}$$

where:

$A$ = minimum open area of fitting .. m$^2$

$v$ = air velocity at opening .. .. m/s

If the vanes of the grille are set for a wide spread, the constant in equation B12.26 should be increased to 21. To determine the spectrum from equations B12.26 and 27, use Table B12.6.

**Table B12.6.** Corrections to overall sound power level for diffusers and grilles.

| $\frac{\sqrt{A}}{v}$ (a) | Corrections (dB) | | | | | | |
|---|---|---|---|---|---|---|---|
| | Octave band centre frequency (Hz) | | | | | | |
| | 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
| 0·01 | −10 | −8 | −7 | − 6 | − 6 | − 7 | −12 |
| 0·02 | − 8 | −7 | −6 | − 6 | − 7 | −12 | −21 |
| 0·04 | − 7 | −6 | −6 | − 7 | −12 | −21 | −30 |
| 0·06 | − 6 | −5 | −6 | −10 | −17 | −26 | −35 |
| 0·08 | − 6 | −6 | −7 | −12 | −21 | −30 | −40 |
| 0·10 | − 5 | −6 | −8 | −15 | −24 | −33 | −43 |

## ASSESSMENT OF ROOM SOUND LEVEL

The propagation of sound power through a ventilating system will set up a sound pressure field within the terminal room. The sound pressure at any position in the room can be considered as two components; one due to sound directly radiated to that position (direct sound) and the other due to sound reaching the position by reflections from the walls, ceiling and floor (reverberant sound). The magnitude of the sound pressure level at a position in a room is the decibel sum of these two components and can be estimated from:

(*a*) the sound power emitted from the air terminal device(s),

(*b*) the position and radiation characteristics of the air terminal device(s),

(*c*) the acoustic properties of the room.

### Room Sound Power Level

The sound power entering the room via an air terminal device can be estimated from the fan noise, system attenuation and system generated noise as illustrated below. Where there are several air terminal devices in the room (including both supply and return) the total sound power is the sum of the sound powers propagated from all the terminals.

*Calculation Procedure for Sound Power Levels Entering Rooms*

1. Estimate fan sound power level.
2. Deduct, arithmetically, the attenuation of ducts, bends, branches, etc.
3. Add, logarithmically (decibel sum) the sound power level generated by fittings, etc. Each item must be added at the correct stage of the path from fan to room so that it will receive the attenuation of all remaining 'room side' components.
4. Add, logarithmically, the sound power level entering the room from all outlets served by the fan.

Note that the combination of 1, 2 and 3 above equals the sound power level radiated from one terminal. This data will be required for the calculation of sound pressure level at a particular receiving position.

### Room Sound Pressure Level

The sound pressure level at a receiver in a room due to sound being radiated from a terminal device (or unitary equipment) is given by:

$$L_p = L_w + 10 \log_{10}\left[\frac{4}{R} + \frac{Q}{4\pi r^2}\right] \quad \text{.. .. B12.28}$$

where:

$R$ = room constant .. .. .. .. m$^2$

$Q$ = directivity factor for terminal

$r$ = distance of receiver from terminal .. m

In this equation, $\frac{4}{R}$ represents the sound pressure in the room due to reflections from the walls, ceiling and floor (diffuse-field) and $\frac{Q}{4\pi r^2}$ represents the sound pressure at distance $r$ due to direct (free-field) radiation from the terminal device.

*Directivity Factor Q*

This is the ratio of the intensity of sound radiated in a particular direction to the corresponding intensity if the unit radiated freely in all directions. Terminal devices have directivity factors greater than unity because adjacent surfaces prevent free radiation and also since high frequency noise tends to beam in the direction of propagation.

*Room Constant R*

This is a measure of the absorption of sound within the room. It is a function of the room size and the absorption coefficient for the room. The room constant will vary with frequency and must be calculated for each of the relative octave bands. For a given band it is:

$$R = \frac{S\,\bar{a}}{1 - \bar{a}} \quad \text{.. .. .. .. ..} \quad \text{B12.29}$$

where:

$S$ = total area of room surfaces .. .. m$^2$
 = $S_1 + S_2 + S_3 + \ldots\ldots S_n$

$\bar{a}$ = average random incidence sound absorption coefficient for the room

$$\bar{a} = \frac{S_1a_1 + S_2a_2 + S_3a_3 + \ldots . S_na_n}{S} \quad \text{..} \quad \text{B12.30}$$

where:

$a_1$, $a_2$, $a_3$, . . . . $a_n$ are octave band random incidence absorption coefficients for room surfaces $S_1$, $S_2$, $S_3$, . . . . $S_n$ respectively.

In an existing room, the room constant in each octave band can be determined by either measuring the reverberation time of the room which gives:

$$S\bar{a} = \frac{0{\cdot}161\ V}{T} \quad \text{.. .. .. .. ..} \quad \text{B12.31}$$

where:

$V$ = room volume .. .. .. .. m$^3$

$T$ = reverberation time .. .. .. s

whence $R$ can be determined from equation B12.29, or by introducing into the room a known source of sound power and directivity and measuring the sound pressure level at a given distance. The room constant can then be determined from equation B12.28.

**Determination of Room Sound Pressure Level**

The following procedure for determining the sound pressure level in a room arising from sound radiation from a terminal may be found simpler to use than the basic equation B12.28.

*Reverberant Room Sound Pressure Level*

Reverberant room sound pressure level can be computed as:

$$L_{pr} = L_w + C_7 + C_8 \quad \text{.. .. ..} \quad \text{B12.32}$$

where:

$L_{pr}$ = reverberant room sound pressure level dB

$C_7$ = correction for room acoustic characteristics (see Table B12.7) .. .. dB

$C_8$ = correction for total room surface area (see Table B12.8) .. .. .. dB

**Table B12.7.** Corrections for room acoustic characteristics.

| Description of room acoustic characteristics | Corrections (dB) | | | | | | |
|---|---|---|---|---|---|---|---|
| | Octave band centre frequency (Hz) | | | | | | |
| | 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
| Live | +16 | +15 | +14 | +12 | +13 | +15 | +16 |
| Medium live | +11 | +11 | + 9 | + 7 | + 6 | + 6 | + 6 |
| Average | +11 | + 9 | + 7 | + 5 | + 4 | + 3 | + 3 |
| Medium dead | + 9 | + 6 | + 5 | + 3 | + 2 | + 1 | + 1 |
| Dead | + 6 | + 4 | + 2 | 0 | − 1 | − 1 | − 1 |

*Live*

Rooms with hard and heavy surfaces, with no soft furnishings and without any acoustical treatment or fittings of absorbing material.

*Medium Live*

Rooms with hard surfaces e.g. panel construction with no special acoustical treatment but with some absorbent content e.g. people, covered chairs, or limited soft furnishings.

*Average*

Rooms which have acoustical ceilings or appreciable soft furnishings e.g. carpeted or upholstered and furniture and soft drapes.

*Medium Dead*

Rooms which have both acoustical ceilings and appreciable soft furnishings.

*Dead*

Rooms which have been specially treated to absorb sound.

**Table B12.8.** Corrections for total room surface area.

| Total room surface area (m$^3$) | Correction (dB) |
|---|---|
| 25 | −8 |
| 50 | −11 |
| 100 | −14 |
| 250 | −18 |
| 500 | −21 |
| 1 000 | −24 |
| 2 500 | −28 |
| 5 000 | −31 |
| 10 000 | −34 |

*Direct Room Sound Pressure Level*

This is a function of the distance between the terminal and the receiver and the directivity factor for the terminal. It can be computed as:

$$L_{pd} = L_w = C_9 + C_{10} \quad \text{.. .. ..} \quad \text{B12.33}$$

where:

$L_{pd}$ = direct room sound pressure level .. dB

$C_9$ = correction for distance between terminal and receiver (see Table B12.9) dB

$C_{10}$ = correction for directivity (see Table B12.10) .. .. .. .. .. dB

**Table B12.9.** Corrections for the distance between the terminal and receiver.

| Distance (m) | Correction (dB) |
|---|---|
| 1 | −11 |
| 1·5 | −14 |
| 2 | −17 |
| 2·5 | −19 |
| 3 | −21 |
| 4 | −23 |
| 5 | −25 |
| 7 | −28 |

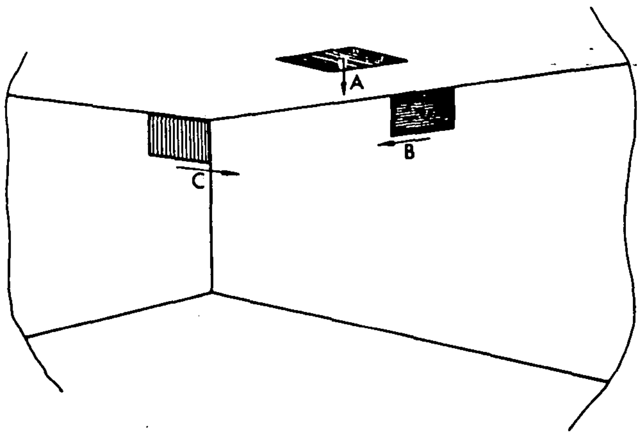



**Fig. B12.12.** Outlet locations.

The correction for directivity depends on both the mounting position and the area of the outlet. For surface mounted terminals three possible arrangements are specified as shown in Fig. B12.12. These are:

Position A—more than 1 metre from any other major room surface.

Position B—closer than 1 metre to one other major room surface.

Position C—closer than 1 metre to two other major room surfaces.

*Total Room Sound Pressure Level*

The sound pressure level at a position in a room is computed as the decibel sum of the reverberant and direct sound pressure levels. In practice for a particular position the nearest single outlet only is normally considered. However if another outlet is no more than a further 1 m distance this should be included in the determination of the direct sound pressure level for that position.

*Room Sound Pressure Levels due to Outside Noise*

The sound pressure level in a room due to outside noise is given by:

$$L_{p2} = L_{P1} - SRI + 10 \log_{10}\left(\frac{S_e}{A}\right) \quad .. \quad \text{B12.34}$$

where:

$L_{p1}$ = sound pressure level outside .. dB

$L_{p2}$ = sound pressure level inside .. dB

$SRI$ = sound reduction index of, say, wall

$A$ = total absorption in room (= $S\bar{a}$)

$S_e$ = area of, say, wall exposed to external noise .. .. .. .. $m^2$

Sound reduction indices will be found in Table B12.18.

**Table B12.10.** Corrections for directivity.

| Mounting position | Outlet area ($m^3$) | Corrections (dB) Octave band centre frequency (Hz) 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
|---|---|---|---|---|---|---|---|---|
| A | 0·01 | +3 | +4 | +5 | +6 | +7 | +8 | +8 |
| | 0·05 | +4 | +5 | +6 | +7 | +8 | +8 | +9 |
| | 0·1 | +4 | +6 | +7 | +8 | +9 | +9 | +9 |
| | 0·25 | +5 | +7 | +7 | +8 | +9 | +9 | +9 |
| | 1·0 | +7 | +8 | +8 | +9 | +9 | +9 | +9 |
| | 10·0 | +9 | +9 | +9 | +9 | +9 | +9 | +9 |
| B | 0·01 | +6 | +6 | +7 | +7 | +8 | +8 | +9 |
| | 0·05 | +7 | +7 | +8 | +8 | +9 | +9 | +9 |
| | 0·1 | +7 | +8 | +8 | +9 | +9 | +9 | +9 |
| | 0·25 | +8 | +8 | +8 | +9 | +9 | +9 | +9 |
| | 1·0 | +8 | +9 | +9 | +9 | +9 | +9 | +9 |
| | 10·0 | +9 | +9 | +9 | +9 | +9 | +9 | +9 |
| C | All areas | +9 | +9 | +9 | +9 | +9 | +9 | +9 |

**Determination of Room Sound Power Level**

The need for attenuation is normally assessed on whether the sound power level at a duct terminal will meet a specified noise rating in the room. The procedure for doing this is outlined below.

(1) List permissible room octave band sound pressure levels.

(2) Determine the difference between sound power level and sound pressure level for the room as described above in each octave band. For a multiple outlet arrangement this value should be based on the minimum distance between any terminal and a room occupant.

(3) Octave band power levels at the terminal are then equal to (1) + (2).

If there are a number of similar terminals in the room this gives the total sound power level from all the terminals. The maximum permissible sound power output for a single terminal is equal to:

$$L_{wm} = L_{wt} - 10 \log_{10} N_t \quad \text{.. .. ..} \quad \text{B12.35}$$

where:

$L_{wm}$ = maximum sound power level per terminal .. .. .. .. .. dB

$N_t$ = number of terminals in room

$L_{wt}$ = total sound power level .. .. dB

**Example Calculations**

As there may be several noise generating components in any system, each having a different amount of system attenuation between itself and the room, it is convenient to break down the calculation of room sound power level into sections. Each section should be related to a specific noise generating component and the result of these calculations can be compared to a permissible sound power level for a terminal and the sample calculations are shown on this basis.

Fig. B12.13 shows the system on which the sample calculations are based. The fan is assumed to be centrifugal and to generate 88 dB re $10^{-12}$ W sound power level, which will have been calculated from equations B12.11, 12 or 13. It is also assumed to have a blade passage frequency falling in the 250 Hz octave band. The conference room is assumed to have average acoustic conditions and the diffuser F is assumed to be so situated that the distance to the nearest listener is 2 m.

*Fan Sound Power Level Entering System*
Corrections for a typical spectrum and the blade passage frequency are made, see Table B12.11.

*Fan Sound Power Level Leaving Diffuser F*
The attenuation of each system is shown separately, and in the correct sequence, from the fan to diffuser F, see Table B12.12.

*Sound Power Level generated by 90° Mitre Bend with Turning Vanes*
Using the data given earlier, the sound power level per octave is calculated and the system attenuation on the room side of the bend is subtracted giving the bend-generated sound power level leaving diffuser F, see Table B12.13.

*Sound Power Level generated by Diffuser*
Using the data given earlier, the sound power level is calculated. All the sound power is discharged to the room, see Table B12.12.

*Permissible Sound Power Level per Terminal*
To obtain values per octave for the permissible sound power level per terminal, the sound pressure level at a selected point in the room is calculated for a specific sound power level input. The difference between the input level and the room sound pressure level is the room effect, and this added to the required NR rating for the

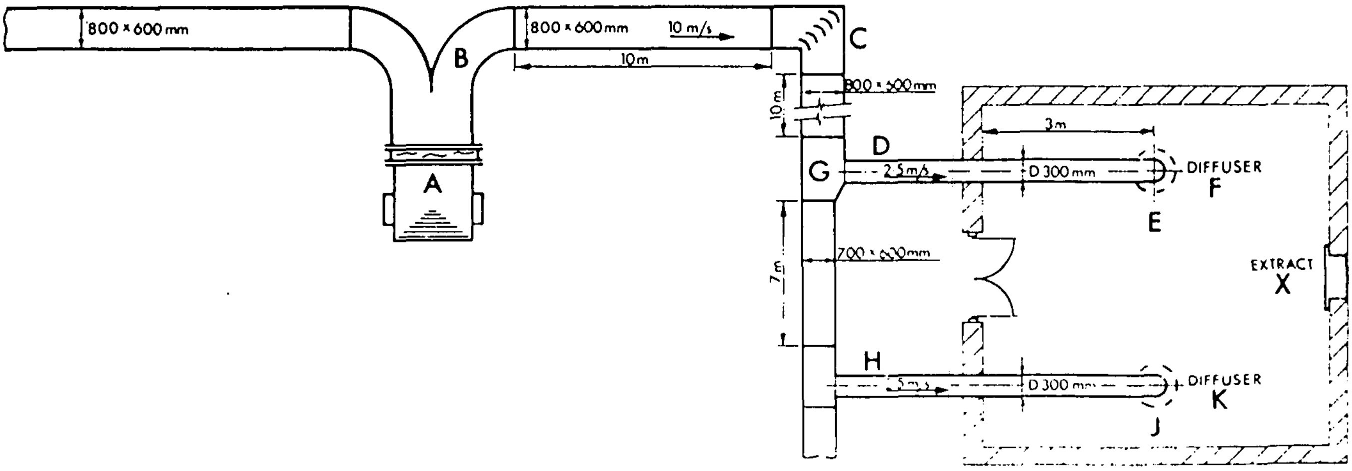



**Fig. B12.13.** Example air-conditioning system.

room gives the maximum permissible sound power level per terminal. In the sample calculation the specific sound power level input used is that already calculated for the fan sound power level leaving diffuser F. However, any arbitrarily chosen input values could be used as it is only the room effect which is required and not absolute values, see Table B12.15.

Note that a correction appears in the calculation of reverberant sound pressure level for the extract system noise. This correction does not apply to the direct sound pressure level in the example but it would do so if an extract system terminal was equidistant, with diffuser F, from the selected point for calculating the room sound pressure level.

**Table B12.11.** Fan sound power level entering system.

| Sound component | Text reference | Sound level (dB) Octave band centre frequency (Hz) 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
|---|---|---|---|---|---|---|---|---|
| Fan total sound power level .. .. .. | | 88 | 88 | 88 | 88 | 88 | 88 | 88 |
| Corrections for typical fan noise spectrum .. | Fig. B12.3 | –2 | –7 | –12 | –17 | –22 | –27 | –32 |
| Corrections for blade passage frequency .. .. | Equation B12.14 | 0 | +5 | + 3 | 0 | 0 | 0 | 0 |
| Total corrections to fan sound power level .. | | –2 | –2 | – 9 | –17 | –22 | –27 | –32 |
| Fan sound power level delivered to system .. | | 86 | 86 | 79 | 71 | 66 | 61 | 56 |

**Table B12.12.** Fan sound power level leaving diffuser F.

| Sound component | Text reference | Sound level (dB) Octave band centre frequency (Hz) 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
|---|---|---|---|---|---|---|---|---|
| Fan sound power level entering system .. .. | Table B12.11 | 86 | 86 | 79 | 71 | 66 | 61 | 56 |
| Branch attenuation at B .. .. .. .. | Equation B12.24 | –3 | –3 | –3 | –3 | –3 | –3 | –3 |
| Straight duct, 20m × 800mm, × 600mm, attenuation | Table B12.3 | –6 | –6 | –4 | –4 | –4 | –4 | –4 |
| Total attenuation prior to branch D .. .. | | –9 | –9 | –7 | –7 | –7 | –7 | –7 |
| Sound power level prior to branch D .. .. | | 77 | 77 | 72 | 64 | 59 | 54 | 49 |
| Branch attenuation at D to 300mm diameter duct | Equation B12.24 | –8 | –8 | –8 | –8 | –8 | –8 | –8 |
| Bend attenuation at D to 300mm diameter duct | Table B12.4 | 0 | –1 | –6 | –7 | –4 | –4 | –4 |
| Straight duct, 3m × 300mm diameter attenuation | Table B12.3 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| End reflection attenuation .. .. .. .. | Fig. B12.11 | –9 | –5 | –2 | 0 | 0 | 0 | 0 |
| Total attenuation from D to diffuser F .. .. | | –17 | –14 | –16 | –15 | –12 | –12 | –12 |
| Fan sound power level leaving diffuser F .. | | 60 | 63 | 56 | 49 | 47 | 42 | 37 |

**Table B12.13.** Sound power level generated by mitre bend.

| Sound component | Text reference | Sound level (dB) Octave band centre frequency (Hz) 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
|---|---|---|---|---|---|---|---|---|
| Sound power level generated by 90° mitred bend with vanes .. .. .. .. .. | | 67 | 67 | 67 | 67 | 67 | 67 | 67 |
| Octave band corrections .. .. .. .. | Table B12.5 | – 2 | – 7 | – 8 | –10 | –12 | –15 | –19 |
| Bend generated sound power level delivered to system .. .. .. .. .. .. | | 65 | 60 | 59 | 57 | 55 | 52 | 48 |
| Straight duct, 10 m × 800 mm × 600 mm, attenuation .. .. .. .. .. .. | Table B12.3 | – 3 | – 3 | – 2 | – 2 | – 2 | – 2 | – 2 |
| Attenuation from D to diffuser F .. .. | Table B12.12 | –17 | –14 | –16 | –15 | –12 | –12 | –12 |
| Total attenuation from bend to diffuser F .. | | –20 | –17 | –18 | –17 | –14 | –14 | –14 |
| Bend generated sound power level leaving diffuser F .. .. .. .. .. .. | | 45 | 43 | 41 | 40 | 41 | 38 | 34 |

*Summary and Attenuation Decisions*

Table B12.16 below summarizes the sound power level delivered to the room by diffuser F from the three noise generating system components considered in this example. Also shown is the permissible sound power level per terminal.

It will be seen that the diffuser is 1 dB over the permissible values in the octave bands centered on 2 kHz and 4 kHz. 1 dB excess is not considered significant and this diffuser is therefore acceptable.

As the room sound pressure level is a consequence of the decibel sum of the three sets of sound power levels delivered to the room, suitable attenuation must be selected to achieve this result. Table B12.16 suggests how this could be done but other solutions are possible, for instance in the 250 Hz octave band the fan sound power level could be reduced to only 46 dB. The total sound power level delivered from this terminal is then 46 + 33 + 33 = 46 dB (decibel sum) and the bend sound power level reduced to 33 dB. The decisions as to attenuation methods depend on economy, space restrictions, control of induct noise levels to limit the radiation of noise to other areas, etc.

**Table B12.14.** Sound power level generated by diffuser.

| Sound component | Text reference | Sound level (dB) Octave band centre frequency (Hz) 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
|---|---|---|---|---|---|---|---|---|
| Sound power level generated by diffuser .. .. | | 41 | 41 | 41 | 41 | 41 | 41 | 41 |
| Octave band corrections .. .. .. .. | Table B12.6 | –10 | –8 | –7 | –6 | –6 | –7 | –12 |
| Diffuser generated sound power level delivered to system .. .. .. .. .. .. | | 31 | 33 | 34 | 35 | 35 | 34 | 29 |

**Table B12.15.** Permissible sound power level per terminal.

| Sound component | Text reference | Sound level (dB) Octave band centre frequency (Hz) 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
|---|---|---|---|---|---|---|---|---|
| Fan sound power level leaving diffuser F .. | Table B12.14 | 60 | 63 | 56 | 49 | 47 | 42 | 37 |
| REVERBERANT ROOM EFFECT | | | | | | | | |
| Correction for extract system .. .. .. | See above | +3 | +3 | +3 | +3 | +3 | +3 | +3 |
| Correction for room surface area .. .. | Table B12.8 | –19 | –19 | –19 | –19 | –19 | –19 | –19 |
| Correction for room acoustic conditions .. | Table B12.7 | +11 | +9 | +7 | +5 | +4 | +3 | +3 |
| Total correction .. .. .. .. .. | | –5 | –7 | –9 | –11 | –12 | –13 | –13 |
| Reverberant sound pressure level corrected for extract system .. .. .. .. .. | | 55 | 56 | 47 | 38 | 35 | 29 | 24 |
| DIRECT ROOM EFFECT | | | | | | | | |
| Correction for adjacent terminals of any systems .. .. .. .. .. .. | See above | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| Correction for distance to listener .. .. | Table B12.9 | –17 | –17 | –17 | –17 | –17 | –17 | –17 |
| Correction for directivity .. .. .. | Table B12.10 | +4 | +5 | +6 | +7 | +8 | +8 | +9 |
| Total correction .. .. .. .. .. | | –13 | –12 | –11 | –10 | –9 | –9 | –8 |
| Direct sound pressure level at listener's position | | 47 | 51 | 45 | 39 | 38 | 33 | 29 |
| Total room sound pressure level due to fan (equal to the decibel sum of the two corrected values given above) .. .. .. .. .. | | 56 | 57 | 49 | 42 | 40 | 34 | 30 |
| Room effect (equal to the fan sound power level leaving diffuser F minus the total room sound pressure level) .. .. .. .. .. | | 4 | 6 | 7 | 7 | 7 | 8 | 7 |
| Octave band sound pressure levels of NR 30 .. | Fig. A1.11 | 48 | 40 | 34 | 30 | 27 | 25 | 23 |
| Permissible sound power level per terminal (equal to NR 30 + room effect) .. .. .. | | 52 | 46 | 41 | 37 | 34 | 33 | 30 |

The attenuation decisions summarized in Table B12.17 require the addition of attenuation by means of duct lining, silencers, etc. to the amounts shown in Table B12.16.

The required attenuation for the bend could consist of duct lining after the branch at D. As attenuation is additive this treatment at D will reduce the need for fan attenuation to levels also shown in Table B12.16. The required fan attenuation could not easily be provided by duct lining and a packaged duct silencer would seem appropriate. The noise generation of the packaged duct silencer would have to be checked for the particular air velocity and size selected.

*When to Calculate*

Common sense will indicate that the detailed calculations set out above will not be necessary in every case.

In low pressure, low velocity systems the fan may normally be considered as the only significant noise source unless, as in the example, components such as 90° mitred bends with turning vanes, dampers, etc. are situated fairly close to the terminals or unavoidably awkward pieces of ducting may cause turbulence.

High pressure, high velocity systems will generally require more detailed calculation. Duct noise radiation may be serious over false ceilings and casing radiation from terminal reheat boxes and pressure reducing valves may also need attenuation.

**Cross-talk**

Cross-talk is said to occur when any sound is transmitted, at a noticeable level, directly through ducts or unsealed openings between one room and another. As sound can travel through ducts regardless of the direction of air flow consideration must be given to preventing cross-talk in both directions through a duct.

The assessment of additional duct attenuation, if any, necessary to control cross-talk needs a careful examination of the relevant factors. The following particular cases are considered:

(*a*) Cross-talk between immediately adjacent rooms where the duct opening in the noisy room is remote from the noise source in that room. The rooms may be either horizontally or vertically adjacent and occasionally either room may be noisy, but at different times. Such a situation may arise in a tall office with a ceiling or high level wall grille. The required total attenuation in the interlinking duct to prevent the sound insulation of the separating wall being impaired is given by the following formula:

$$K = SRI - 10 \log_{10} \frac{S_p}{S_o} + C \quad .. \quad .. \quad \text{B12.36}$$

where:

$K$ = attenuation .. .. .. .. dB

**Table B12.16.** Attenuation decisions.

| Sound component | Text reference | Sound level (dB) Octave band centre frequency (Hz) 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
|---|---|---|---|---|---|---|---|---|
| Fan sound power level leaving diffuser F .. | Table B12.12 | 60 | 63 | 56 | 49 | 47 | 42 | 37 |
| Bend generated sound power level leaving diffuser F .. .. .. .. .. .. | Table B12.13 | 45 | 43 | 41 | 40 | 41 | 38 | 34 |
| Sound power level generated by diffuser F .. | Table B12.14 | 31 | 33 | 34 | 35 | 35 | 34 | 29 |
| Total sound power level leaving diffuser F .. | Fig. B12.1 | 60 | 63 | 56 | 50 | 48 | 43 | 39 |
| Permissible sound power level per terminal .. | Table B12.15 | 52 | 46 | 41 | 37 | 34 | 33 | 30 |
| ACTION REQUIRED | | | | | | | | |
| Reduce fan sound power level to new level .. and | | 51 | 43 | 37 | 30 | 21 | 20 | 22 |
| Reduce bend sound power level to new level .. | | – | – | 37 | 30 | 21 | 20 | 22 |

**Table B12.17.** Summary of attenuation requirements.

| Sound component | Text reference | Sound level (dB) Octave band centre frequency (Hz) 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
|---|---|---|---|---|---|---|---|---|
| Total fan sound power level attenuation required | Table B12.16 | 9 | 20 | 19 | 19 | 26 | 22 | 15 |
| Bend sound power level attenuation required .. | Table B12.16 | 0 | 0 | 4 | 10 | 20 | 18 | 12 |
| Fan net attenuation required, taking into account the bend attenuation downstream .. .. .. | | 9 | 20 | 15 | 9 | 6 | 4 | 3 |

$SRI$ = sound reduction index of party wall or floor .. .. .. .. dB

$S_p$ = area of party wall or floor .. .. $m^2$

$S_o$ = area of duct opening on receiving side $m^2$

$C$ = constant. A value of 6 is normal but if full integrity of the sound insulation is required this should be increased to 10.

(*b*) Similar to (*a*) except that the listeners are close to the duct opening:

$$K = SRI + C \quad \text{B12.37}$$

(*c*) cross-talk between rooms not adjacent to each other:

$$K = L_{p1} + 10 \log_{10} S_o + D - L_{p2} + (L_w - L_p) \quad \text{B12.38}$$

where:

$L_{p1}$ = band sound pressure level in noisy room .. .. .. .. dB

$L_{p2}$ = band sound pressure level at the required NR number for the receiving room .. .. .. .. dB

$S_o$ = cross sectional area of duct opening in noisy room .. .. .. $m^2$

$D$ = 6 for frequencies $> \dfrac{330}{d}$

$D$ = 0 for frequencies $< \dfrac{300}{d}$

$d$ = maximum dimension of duct opening in noisy room .. .. m

$(L_w - L_p)$ from equation B12.28.

*Cross-talk Attenuation*

Cross-talk attenuators may be required to augment the natural attenuations in the ducts to provide the required level of sound insulation.

It is sometimes difficult to establish a reasonable value for the sound pressure level in a noisy room. As most cross-talk problems are concerned with speech transmission the following average values of a male speech spectrum may be of assistance, see Table B12.19.

**Table B12.19.** Average values of male speech spectrum.

| Octave band centre frequency (Hz) | Sound pressure level (dB) |
|---|---|
| 125 | 55 |
| 250 | 59 |
| 500 | 66 |
| 1000 | 65 |
| 2000 | 60 |
| 4000 | 52 |
| 8000 | 40 |

**Table B12.18.** Sound reduction index of partitions.

| Nature of partition | Weight ($kg/m^3$) | Sound reduction index at frequency (Hz) (dB) 125 | 250 | 500 | 1000 | 2000 | 4000 |
|---|---|---|---|---|---|---|---|
| 115 mm brick, plastered on both faces to thickness of 12 mm .. .. | 260 | 31 | 36 | 41 | 48 | 54 | 61 |
| 230 mm brick, plastered on both faces to thickness of 12 mm .. .. | 465 | 38 | 44 | 51 | 54 | 62 | 68 |
| 280 mm cavity brick wall with butterfly ties and both outer faces plastered | — | 29 | 40 | 45 | 62 | 72 | 84 |
| 75 mm hollow clay blocks plastered on one face to thickness of 12 mm | 68 | 28 | 32 | 33 | 35 | 38 | 43 |
| 200 mm cavity wall with two leaves of 75 mm hollow blocks and 50 mm airspace, outer faces plastered .. .. .. .. .. .. | 136 | 38 | 48 | 52 | 57 | 68 | 78 |
| 75 mm clinker concrete slabs unplastered .. .. .. .. | 83 | 17 | 18 | 20 | 24 | 30 | 42 |
| 75 mm clinker concrete slabs with both faces plastered to thickness of 12 mm .. .. .. .. .. .. .. .. | 146 | 26 | 33 | 40 | 50 | 57 | 58 |
| 200 mm cavity wall with two leaves of 75 mm clinker concrete slabs and 50 mm airspace, outer faces plastered .. .. .. .. .. | 195 | 41 | 44 | 46 | 52 | 61 | 69 |
| Single sheet (24 oz) glass in wood or sealed metal frame .. .. | 6·8 | 15 | 20 | 22 | 27 | 29 | 25 |
| Single 6 mm plate glass in wood or sealed metal frame .. .. .. | 16·6 | 20 | 21 | 26 | 29 | 29 | 38 |
| Double glazing–two sheets of (24 oz) glass in common frame with 7 mm airspace .. .. .. .. .. .. .. .. .. | 15·1 | 19 | 16 | 21 | 27 | 30 | 29 |
| Double window–two sheets of (24 oz) glass in separate frames 185 mm apart with absorbent in reveals .. .. .. .. | 13·1 | 23 | 37 | 42 | 46 | 53 | 50 |
| T. and G. floorboards on wooden joists with plasterboard ceiling .. | 39 | 21 | 28 | 33 | 36 | 44 | 46 |
| 100 mm thick reinforced concrete floor .. .. .. .. .. | 235 | 39 | 37 | 41 | 49 | 57 | 66 |
| 6 mm asbestos cement board in steel frame with cover strips .. .. | 8·3 | 12 | 18 | 24 | 29 | 33 | 34 |
| 10 mm plasterboard on wooden frame .. .. .. .. .. | 9·3 | 16 | 20 | 24 | 30 | 33 | — |
| 10 mm plasterboards on both sides of 100 × 50 mm studs at 400 mm centres, both faces plastered to thickness of 12 mm. Overall thickness 150 mm .. .. .. .. .. .. .. .. .. | 61 | 24 | 28 | 34 | 47 | 39 | 56 |
| 12 mm fibreboard on wooden frame .. .. .. .. .. | 3·8 | 12 | 15 | 20 | 24 | 30 | 29 |
| 12 mm fibreboard on 25 × 50 mm battens at 400 mm centres on one side of 115 mm brick plastered on both faces and plastered on fibreboard to 12 mm thickness .. .. .. .. .. .. | 288 | 29 | 35 | 46 | 51 | 61 | 66 |
| Two sheets of 6 mm plywood bonded together by plywood lattice. Overall thickness 45 mm .. .. .. .. .. | 9·8 | 14 | 17 | 19 | 18 | 24 | 29 |
| Paper pulp between two sheets of 2 mm plywood. Overall thickness 50 mm .. .. .. .. .. .. .. .. .. | 14 | 13 | 19 | 26 | 30 | 25 | 31 |

## NOISE CONTROL IN AIR-FLOW SYSTEMS

The possible transmission paths for noise to reach rooms served by ventilating plants are described earlier (see Fig. B12.2). Should the transmission of noise along one or more of these paths give too high a noise level in terminal areas additional noise control features will be necessary.

### Control of Airborne Noise in Ducts

The reduction of airborne noise in a duct is achieved by absorbing acoustic energy or reflecting some of it back towards the noise source. Such reduction is effected by:

(*a*) packaged attenuators

(*b*) lined plenum chambers

(*c*) lined ductwork.

The performance of an attenuator placed in a ducted system should always be specified by its 'insertion loss'. The insertion loss is the difference in sound power downstream of a proposed attenuator location with and without the attenuator in position.

The insertion loss is normally expressed to octave band sound power levels. In the case of attenuators tuned to suppress a pure tone the insertion loss at the frequency of the pure tone should also be given.

Attenuators like other components of the duct system also generate noise.

### Sound Absorption Properties of Materials

The acoustic characteristics of absorbing materials are normally specified by the absorption coefficient $\alpha$ for sound waves impinging with random incidence. The absorption coefficient varies with frequency and is influenced by the type of material, its thickness and density and by the method of mounting. A list of octave band absorption coefficients is given in Table B12.20. Materials selected for sound absorption should also meet the relevant requirements for:

(*a*) fire protection

(*b*) erosion by air flow

(*c*) vermin resistance.

**Table B12.20.** Absorption properties of materials.

| Material | Nominal thickness (mm) | Absorption coefficient at frequency (Hz) | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | | 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
| FIBROUS MATERIALS | | | | | | | | |
| Fibreglass super-fine mat .. .. .. .. .. | 50 | 0·15 | 0·40 | 0·75 | 0·85 | 0·80 | 0·85 | — |
| Fibreglass scrim-covered sewn sheet .. .. .. | 40 | 0·40 | 0·80 | 0·95 | 0·95 | 0·80 | 0·85 | — |
| Fibreglass bitumen-bonded mat .. .. .. .. | 25 | 0·10 | 0·35 | 0·50 | 0·55 | 0·70 | 0·70 | 0·75 |
| Fibreglass bitumen-bonded mat .. .. .. .. | 50 | 0·30 | 0·55 | 0·80 | 0·85 | 0·75 | 0·80 | — |
| Fibreglass resin-bonded mat .. .. .. .. .. | 25 | 0·10 | 0·35 | 0·55 | 0·65 | 0·75 | 0·80 | 0·75 |
| Fibreglass resin-bonded mat .. .. .. .. .. | 50 | 0·20 | 0·50 | 0·70 | 0·80 | 0·75 | 0·80 | 0·80 |
| Fibreglass resin-bonded board .. .. .. .. | 25 | 0·10 | 0·25 | 0·55 | 0·70 | 0·80 | 0·85 | 0·95 |
| Stillite semi-rigid slabs .. .. .. .. .. | 25 | 0·05 | 0·20 | 0·45 | 0·75 | 0·80 | 0·75 | — |
| Stillite slabs, muslin-faced .. .. .. .. .. | 50 | 0·15 | 0·55 | 0·70 | 0·80 | 0·85 | 0·85 | 0·95 |
| Stillite scrim-covered quilts on 50 mm thick battens .. | 50 | 0·50 | 0·70 | 0·90 | 0·90 | 0·85 | 0·80 | — |
| Jones & Broadbent resin-bonded mineral wool felt .. | 25 | 0·10 | 0·20 | 0·50 | 0·65 | 0·90 | 0·85 | — |
| Lintafelt muslin-covered cotton felt .. .. .. | 25 | 0·15 | 0·45 | 0·70 | 0·85 | 0·95 | 0·85 | — |
| Paxfelt on 50 mm thick battens .. .. .. .. | 50 | — | 0·55 | 0·65 | 0·75 | 0·80 | 0·80 | — |
| Paxfelt semi-rigid asbestos slab on 50 mm thick battens .. | 25 | — | 0·50 | 0·55 | 0·65 | 0·70 | 0·75 | — |
| Isoverbel Hagerwood mattress on 50 mm battens .. | 50 | 0·25 | 0·60 | 0·80 | 0·85 | 0·85 | 0·90 | 0·95 |
| Isoverbel Hagerwood mattress on 50 mm battens with polythene cover .. .. .. .. .. .. .. | 50 | 0·25 | 0·60 | 0·85 | 0·80 | 0·65 | 0·50 | 0·30 |
| EXPANDED PLASTICS AND RUBBERS | | | | | | | | |
| I.C.I. flexible polyurethane foam .. .. .. .. | 50 | 0·25 | 0·50 | 0·85 | 0·95 | 0·90 | 0·90 | — |
| I.C.I. rigid polyurethane foam .. .. .. .. | 50 | 0·20 | 0·40 | 0·65 | 0·55 | 0·70 | 0·70 | — |
| Jablite expanded polystyrene on 45 mm thick battens .. | 12 | 0·05 | 0·15 | 0·40 | 0·35 | 0·20 | 0·20 | — |
| Polyzote expanded polystyrene on 50 mm battens .. | 25 | 0·10 | 0·25 | 0·55 | 0·20 | 0·10 | 0·15 | — |
| Rubazote foam rubber with smooth skin on 25 mm battens | 12 | 0·10 | 0·40 | 0·20 | 0·10 | 0·10 | 0·15 | — |
| Rubazote foam rubber with skin removed on 25 mm battens .. .. .. .. .. .. .. | 10 | 0·05 | 0·20 | 0·30 | 0·15 | 0·10 | 0·15 | — |

**Table B12.20.** Absorption properties of materials—*continued*

| Material | Nominal thickness (mm) | Absorption coefficient at frequency (Hz) 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
|---|---|---|---|---|---|---|---|---|
| BOARDS AND PANELS | | | | | | | | |
| Pinex insulating fibreboard stuck on 6 mm Uralite .. | 12 | 0·10 | 0·15 | 0·25 | 0·30 | 0·30 | 0·40 | — |
| Pinex insulating fibreboard on 100 mm thick battens .. | 12 | 0·45 | 0·25 | 0·25 | 0·25 | 0·30 | 0·35 | — |
| Asbestolux asbestos board on 150 mm wood blocks .. | 6 | 0·23 | 0·11 | 0·05 | 0·03 | 0·02 | — | — |
| Plasterboard .. .. .. .. .. .. .. | — | 0·20 | — | 0·10 | — | 0·04 | — | — |
| Stramit (strawboard) with paper lining removed on 100 mm thick battens .. .. .. .. .. | 50 | 0·25 | 0·20 | 0·35 | 0·40 | 0·45 | 0·50 | — |
| Pimberite chipboard on 16 mm battens .. .. .. | 20 | 0·20 | 0·25 | 0·20 | 0·20 | 0·15 | 0·20 | — |
| Plywood in framework with 30 mm air space behind .. | 12 | 0·35 | 0·20 | 0·15 | 0·10 | 0·05 | 0·05 | — |
| PERFORATED TILES | | | | | | | | |
| Acoustic-Celotex fibreboard tiles .. .. .. .. | 12 | 0·10 | 0·20 | 0·40 | 0·50 | 0·45 | 0·50 | — |
| Paxtile asbestos tiles .. .. .. .. .. .. | 25 | — | 0·55 | 0·75 | 0·85 | 0·80 | — | — |
| Perfonit fibreboard tiles .. .. .. .. .. | 20 | 0·20 | 0·50 | 0·70 | 0·85 | 0·75 | 0·65 | — |
| Unitex Acoustile fibreboard tiles .. .. .. .. | 12 | 0·20 | 0·55 | 0·60 | 0·60 | 0·65 | 0·80 | — |
| COMPOSITE TREATMENTS | | | | | | | | |
| (Panels and perforated covers over absorbent) | | | | | | | | |
| Plasterboard 10 mm thick backed with 25 mm thick bitumen-bonded fibreglass on 50 mm battens .. .. .. | — | 0·30 | 0·20 | 0·15 | 0·05 | 0·05 | 0·05 | — |
| Plasterboard 10 mm thick, perforated 8 mm diameter holes 2755/m$^2$. 14% open area backed with 25 mm thick bitumen-bonded fibreglass on 90 mm battens .. .. .. | — | 0·25 | 0·70 | 0·85 | 0·55 | 0·40 | 0·30 | — |
| Plywood 12 mm thick with 30 mm thick fibreglass backing between 30 mm battens .. .. .. .. .. | — | 0·40 | 0·20 | 0·15 | 0·10 | 0·10 | 0·05 | — |
| Plywood 12 mm thick perforated 5 mm diameter holes 6200/m$^2$. 11% open area with 60 mm deep airspace behind .. | — | 0·20 | 0·35 | 0·55 | 0·30 | 0·25 | 0·30 | — |
| Plywood 12 mm thick perforated 5 mm diameter holes 6200/m$^2$. 11% open area backed with 60 mm thick fibreglass between mounting battens .. .. .. .. .. | — | 0·40 | 0·90 | 0·80 | 0·50 | 0·40 | 0·30 | — |
| Frenger ceiling 0·8 mm unperforated metal panels backed with 25 mm thick resin-bonded fibreglass, mounted on 22 mm diameter pipes 135 mm from wall .. .. .. .. | — | 0·50 | 0·35 | 0·15 | 0·05 | 0·05 | — | — |
| Frenger ceiling 0·8 mm perforated metal panels 3·5 mm diameter holes 3445/m$^2$. 3% open area backed with 20 mm thick bitumen-bonded fibreglass mounted on 25 mm square pipes 150 mm from wall .. .. .. .. .. .. | — | 0·20 | 0·45 | 0·65 | 0·45 | 0·35 | 0·25 | — |
| Burgess tiles 0·8 mm perforated metal panels 2 mm diameter holes 29440/m $^2$. 13% open area backed with 25 mm thick resin-bonded fibreglass slab. No air space .. .. .. | — | 0·10 | 0·30 | 0·60 | 0·75 | 0·80 | 0·80 | 0·80 |
| Burgess tiles 0·8 mm perforated metal panels 2 mm diameter holes 29440/m $^2$. 13% open area backed with 25 mm thick Stillite semi-rigid mineral wool slabs. No air space .. | — | 0·10 | 0·30 | 0·60 | 0·75 | 0·80 | 0·90 | 0·70 |
| MISCELLANEOUS BUILDING SURFACES | | | | | | | | |
| Brickwork .. .. .. .. .. .. .. | — | 0·05 | 0·04 | 0·02 | 0·04 | 0·05 | 0·05 | — |
| Concrete (constructional) .. .. .. .. .. | — | 0·02 | 0·02 | 0·02 | 0·04 | 0·05 | 0·05 | — |
| Unplastered breeze block .. .. .. .. .. | 75 | 0·20 | 0·45 | 0·50 | 0·40 | 0·45 | 0·40 | — |
| Gypsum plaster on solid wall .. .. .. .. | — | 0·03 | 0·03 | 0·02 | 0·03 | 0·04 | 0·05 | — |
| Granwood flooring blocks .. .. .. .. .. | 12 | — | 0·05 | 0·05 | 0·05 | 0·05 | — | — |
| Eldorado cork tiles .. .. .. .. .. .. | 14 | — | 0·05 | 0·15 | 0·25 | 0·25 | — | — |
| Sheet rubber (hard) .. .. .. .. .. .. | 6 | — | 0·05 | 0·05 | 0·10 | 0·05 | — | — |

## VIBRATION AND STRUCTURAL NOISE CONTROL

Section A1 suggests the criteria given in Fig. A1.14 and 15 for acceptable vibration levels in buildings. This Section considers how to achieve these criteria in practical situations. The theory of vibration isolation is discussed briefly, though not rigorously, so that the difficulties will be clearly seen, and a method of approach is presented which will minimize the risks of failure. Warnings will be given where possible of the factors for which exact design is not a practical possibility at the present time.

### The Theory of Vibration Isolation

The usual method of isolating the vibration of a machine is to support the machine on a resilient system such as steel or rubber springs. The incorrect use of such supports can however cause the transmission of vibration to be worse than with a rigid support. The specification and selection of resilient supports is therefore of critical importance. The most important characteristic of a resilient support is the natural frequency of vibration which it provides for the equipment mounted on it. A natural frequency of vibration is the frequency that will be found if a resiliently supported body is displaced from its position of rest and then suddenly released so that it oscillates before coming to rest. For a given body on a given resilient support this frequency will always be the same.

In fact, most resiliently supported machines have several natural frequencies each being related to vertical, horizontal and rotational motions. Typically a machine will have six natural frequencies, however for most problems it is the vertical mode which is of greatest importance as floors have greatest flexibility in this direction. The frequency of vertical natural vibration is controlled solely by the weight of the machine and the stiffness of the resilient supports. For example, in the case of conventional simple helical spring or other support, whose deflection is proportional to the applied load (and whose dynamic stiffness equals its static stiffness), the frequency of vertical natural vibration is more simply stated as a function of the deflection of the support caused by the applied load.

Some materials, such as rubber and cork for instance, do not exhibit a linear relationship between applied load and deflection and, furthermore. their stiffness under an oscillating load is greater than a static load test would indicate. For such materials the dynamic stiffness must be used rather than the static deflection.

Fig. B12.14 and 15 show theoretical resonance curves for differing degrees of damping. It will be seen that damping reduces the response at a frequency ratio of 1, but increases the response at ratios above $\sqrt{2}$.

(*a*) *Isolation efficiency* (*sometimes called transmissibility*)
Depending on the forced/natural frequency ratio (using the natural frequency provided to the system by the resilient supports) the transmission of vibration from the machine is either amplified or reduced. Fig. B12.14 shows that for a reasonably large reduction of the vibration, a ratio of about 3 or greater is necessary. Ratios of $\sqrt{2}$ or below are at best ineffective and can be positively detrimental by substantially increasing the vibration.

(*b*) *Floor motion*
The same graph (Fig. B12.14) illustrates the effect of a vibratory force on the movement of the floor. The frequency ratio now is the forced vibration frequency over the natural frequency of the floor itself. Again it is seen that floor motion depends on this ratio. It is always difficult and frequently impossible to be sure of the natural frequency of a floor, especially at the design stage. A method of estimating it is given below.

(*c*) *Machine motion*
If a machine which generates unbalanced forces is supported resiliently it will vibrate. Fig. B12.15 shows the effect of the frequency ratio on the motion of the isolated machine. Once more it is seen that a large ratio (>3) is favourable.

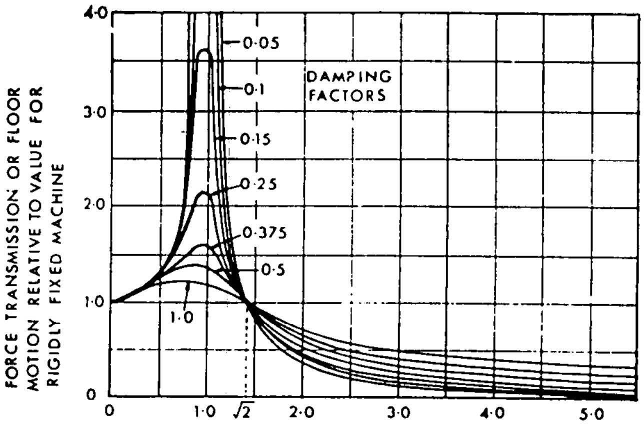



**Fig. B12.14.** Isolation efficiency and floor motion.

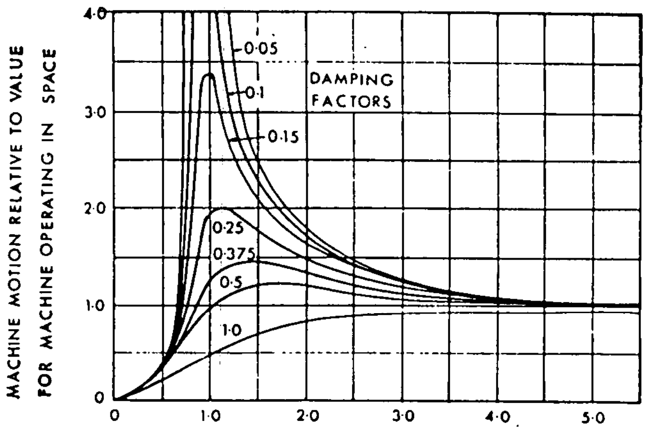



**Fig. B12.15.** Machine motion.

It must be emphasized that the graphs in Fig. B12.14 and 15 are only approximate as they are based on a simplified theory of vibration isolation. They are however good indicators of the trends due to changes in the frequency ratio.

Summarizing, it can be said that good isolation, low floor motion and small machine motion all depend on the realization of suitably large frequency ratios and for springs with a linear stiffness characteristic the vertical natural frequency is a function only of the deflection of the springs under the static weight applied to them. For materials with a non-linear stiffness characteristic and/or with a higher dynamic than static stiffness, dynamic data must be obtained and used.

**Determining the Six Natural Vibration Frequencies**

The vertical natural frequency is, as stated above, a function of the applied load and vertical stiffness of the mountings.

For linear supports the vertical natural frequency is:

$$f_{nat} = \frac{16}{\sqrt{s}} \quad \text{B12.39}$$

where:

$f_{nat}$ = vertical natural frequency .. Hz
$s$ = static deflection .. .. .. mm

For non-linear supports the formula is:

$$f_{nat} = 5{\cdot}0 \sqrt{\frac{k}{w}} \quad \text{B12.40}$$

where:

$k$ = isolator vertical dynamic stiffness kg/m
$w$ = sprung mass .. .. .. kg

The other natural modes of vibration, for a rectangular body mounted resiliently at its lowest corners, are as shown in Fig. B12.16. The calculation of the exact values for these natural frequencies is a fairly complex matter[9, 10] and involves such factors as mass moment of inertia, vertical and transverse spring stiffnesses.

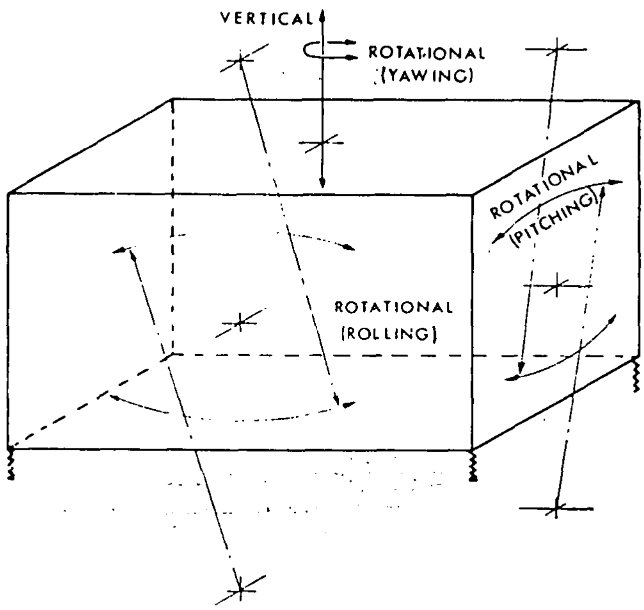



**Fig. B12.16.** Modes of vibration.

There are technical advantages in arranging the resilient supports so that they are in the same horizontal plane as the centre of gravity of the supported body. The greater expense of this arrangement will normally exclude its consideration but it may be justified in special cases.

**A Practical Approach to Vibration Isolation**

The problems of vibration isolation are eased if the types of installation met with in practice are categorized and guide lines given for suggested approaches. The following categories are commonly encountered:

(1) Balanced machines on raft floors (i.e. slabs resting on ground).
(2) Unbalanced machines on raft floors.
(3) Balanced machines on suspended floors.
(4) Unbalanced machines on suspended floors.
(5) Isolation of connections to machines, e.g. pipes, ducts, conduits.

Balanced machines include rotary machines, such as fans and motors, and those reciprocating machines whose primary forces and couples have been eliminated by counter-balancing. Such machines will be defined as statically and dynamically balanced. Unbalanced machines will be defined as those that cannot be statically and dynamically balanced since their primary forces and couples cannot be eliminated, e.g. single cylinder reciprocating engines.

Primary forces and couples are those occurring at the running speed of the machine; the manufacturer will provide the necessary data.

*Balanced Machines on a Raft Floor*

Such machines will be defined as statically and dynamically balanced, and the floor will consist of a concrete slab resting on the ground.

When a slab resting on the ground is set into vibration some part of the ground vibrates with the slab. This adds substantially to the effective weight which opposes the vibration and for this category of machine no problems of excessive transmission of vibration and structure borne noise need be feared provided:

(*a*) isolation systems, having a vertical natural frequency of 10 Hz to 35 Hz, are used; these may be layers of resilient material cast into concrete plinths. It is not necessary to ensure that the vertical natural frequency is $\frac{1}{3}$ of the forcing frequency to avoid resonance nor even that it is below the forcing frequency. There is no serious risk provided that the vertical natural frequency is at least 25% above or below the forcing frequency and is below, say, 35 Hz to avoid the audio frequencies,

(*b*) there are no rooms adjacent to, or above, the plant room requiring very low noise levels, say NR 35 or below.

It is not recommended that a full calculation of all six natural vibration frequencies is performed. In unexceptional circumstances a design based on the vertical frequency only will have every probability of success. If condition (*b*) above is not satisfied and there are critical areas nearby, it is safer to use the recommended procedure for unbalanced machines on suspended floors.

*Unbalanced Machines on Raft Floors*

The safest method of isolation will be achieved by using the procedure for unbalanced machines on suspended floors. Nevertheless, it must be acknowledged that there are many successful installations on the isolation system proposed as above for balanced machines and it is reasonable to adopt that method if:

(*a*) there are no rooms adjacent to or above the plant room requiring noise levels of NR 35 or below,

(*b*) the machines in question are small (say less than 250 kg total mass),

(*c*) the machine is fixed to a mounting block, above the resilient material, of sufficient mass effectively to oppose the unbalanced forces and/or couples. A mass equal to the total weight of the machine and drive is usually sufficient.

If these conditions are not satisfied, use the procedures for unbalanced machines on suspended floors.

*Balanced Machines on Suspended Floors*
For this category it is safest to subdivide into (*a*) machines running at 12 rev/s or below and (*b*) machines running at above 12 rev/s.

(*a*) Machines running at 12 rev/s or below. The force produced by a machine is proportional to the square of the speed. Thus, for nominally balanced machines, the amount of unbalanced force produced at low speeds will be very small. Nevertheless if a resonant condition was experienced, the transmission could be excessive. It is recommended that the vertical natural frequency is $\frac{1}{3}$, or less, of the rotational frequency; unit isolators are more likely to be suitable than layer isolation. Additionally, it is recommended that the vertical deflection of the isolator should not be less than 30 mm in order to avoid accidental resonance with the natural frequency of the floor.

(*b*) Machines running at above 12 rev/s should be fully calculated as for unbalanced machines on suspended floors.

*Unbalanced Machines on Suspended Floors*
Full calculation of all the six natural frequencies is necessary. Ideally all natural frequencies will be $\frac{1}{3}$, or less, of the forcing frequency. If, however, it is certain that one or more of the six natural modes of vibration will not be excited by the unbalanced forces and/or couples, then the natural frequency of these unexcited modes may be regarded as satisfactory if they are at least 25% above or below the forcing frequency, to avoid resonance.

It is also necessary in this case to establish whether the motion of the machine, in answer to its own unbalanced forces and/or couples, will be acceptable in terms of visual impact and performance.

Calculation of the motion of a machine can be a fairly complex and time consuming matter. It is probably best left to the specialist but the following simplified procedures will indicate whether a problem is likely to exist. These procedures assume that all relevant natural frequencies are $\frac{1}{3}$ of the forcing frequency.

(*a*) Vertical or horizontal force acting along a line passing through the centre of gravity.

$$e = \frac{28{\cdot}5 \times F}{M \times f_f^2} \quad \ldots \quad \text{B12.41}$$

where:

| | | | |
|---|---|---|---|
| $e$ | = | displacement amplitude | mm |
| $F$ | = | unbalanced force | N |
| $M$ | = | sprung mass | kg |
| $f_f$ | = | forcing frequency | Hz |

(*b*) Vertical or horizontal force acting along a line net passing through the centre of gravity.

$$e = \frac{28{\cdot}5\ F\ (d^2 + 12ab)}{Md^2\ f_f^2} \quad \ldots \quad \text{B12.42}$$

where:

| | | | |
|---|---|---|---|
| $a$ | = | perpendicular distance between line of force and centre of gravity | m |
| $b$ | = | distance between centre of gravity and part of machine whose motion is to be calculated | m |
| $d$ | = | diagonal of equivalent rectangle enclosing bulk of machine | m |

(*c*) Vertical or horizontal couple.

$$e = \frac{341 \times c \times b}{M \times d^2 \times f_f^2} \quad \ldots \quad \text{B12.43}$$

where:

| | | | |
|---|---|---|---|
| $c$ | = | couple | N m |

If the simplified procedures above indicate the possibility of an unacceptable displacement amplitude, as defined below, specialist advice should be taken. The calculated amplitude may be regarded as acceptable if:

$$A \leq \frac{1{\cdot}0}{f_f} \quad \ldots \quad \text{B12.44}$$

where:

| | | | |
|---|---|---|---|
| $A$ | = | calculated amplitude | mm |

As with balanced machines on suspended floors, mountings with a deflection of less than 30 mm should not be used even on good, stiff floors. If flexible floors such as hollow pot, thin concrete or timber are to be used, specialist advice should be taken.

The motion of the floor under anti-vibration mountings is difficult to predict but some guide is given by using the equation B12.41. The value for $F$ should be taken as 40% of the unbalanced force of the machine (to make a conservative allowance for the effect of the isolators) and the value for $W$ as the weight of the floor under the vertically projected area of the machine. The answer will only be of the correct order if the natural frequency of the floor is about $\frac{1}{3}$ of the forcing frequency. The floor frequency can be estimated from:

$$f \simeq \frac{0{\cdot}5}{\sqrt{d}} \quad \ldots \quad \text{B12.45}$$

where:

| | | | |
|---|---|---|---|
| $d$ | = | expected deflection of the floor under full loading | m |

(If no value of $d$ can be obtained, assume a value of 1/600 of the shorter span between vertical supports)

*Vibration Isolation of Pipes and Duct Supports*
Some flexibility in all pipe, duct and other connections to vibration isolated machines is essential. Failure to provide this flexibility may have two results:

(*a*) the vibration forces generated by the machine may be transmitted into the structure through stiff connections thus making the vibration isolation valueless,

(*b*) the motion of the machine on its isolators may cause unacceptable stresses in stiff pipes and other connections.

The required flexibility may be provided by:

(*a*) fitting flexible connections in pipe and duct runs. If flexible hoses are used in high-pressure pipe lines connected to vibration isolated machines, allowances must be made for the force tending to stretch or compress the flexible hose. The force is given by the internal pipe area ($m^2$) × pressure ($N/m^2$). This force will be transferred to the vibration isolators which will consequently deflect, (*b*) for pipe runs only, arranging the run adjacent to the vibration isolated machine so that it is not rigidly supported and thus capable of flexing easily in any direction. This will normally involve the resilient supports of the pipe for a length which includes two right angled (or nearly so) bends separated from each other by a straight length some 10 times the pipe diameter. The pipe must be capable of accepting the stresses set up by this motion. If resilient supports are used the above principles of vibration isolation apply.

The use of resilient supports for pipe and duct connections to machines is also desirable if the supports have to be taken from a plant room wall, floor or ceiling which is the partition to a quiet area (say < NR 35).

**Insulation of Noise within Buildings**

The interest of the building services engineer in the transmission of noise within buildings is primarily the containment of plant noise within plant rooms, builders work services shafts, etc. It must also be ensured that the transmission of noise via the mechanical services system itself is compatible with the standards of transmission loss necessary for the efficient and enjoyable use of the building. This is mostly related to cross-talk problems which have been discussed above.

*Plant Room Noise*

The acoustic design of the structure of plant rooms may be the responsibility of the building services engineer, an architect or acoustic consultant. In every case, the first step in arriving at a satisfactory design must be to establish the likely sound pressure levels which will exist in the plant room and this must be the responsibility of the building services engineer.

Typical noise spectra for compressors, boiler and fan plant rooms have been illustrated in Fig. B12.6 to 8. More specific predictions may be made by considering the noise of each item of plant in a plant room and summing to obtain a total value. Manufacturers should be asked for sound power level output data relevant to radiation into a plant room. For a ducted fan this will differ substantially from the sound power level radiated into the duct system because of the insulation of the duct wall and fan casing. An approximation of casing radiation may be made by deducting the sound reduction index of a duct (see Table B12.21) from the fan's total generated sound power level.

Given sound power level data for all equipment in the plant room, the resulting sound pressure level may be determined by the procedure set out above. Note, however, that the direct sound pressure level component must be related to a specific item(s) of plant close to the wall, floor or ceiling of the plant room through which the transmission of noise would be critical.

The Room Acoustic Characteristics may be taken to lie between Live and Medium Live using the value for Medium Live only if substantial areas of open louvres are present, as these provide a degree of equivalent sound absorption by releasing the noise to the exterior. Allowance must also be made for external noises entering the plant room via louvres.

**Table B12.21.** Sound reduction indices for duct walls.

| Mass of wall per unit area ($kg/m^2$) | Sound reduction indices (dB) | | | | | | |
|---|---|---|---|---|---|---|---|
| | Octave band centre frequency (Hz) | | | | | | |
| | 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
| 5 | 6 | 10 | 13 | 17 | 22 | 33 | 38 |
| 10 | 10 | 17 | 22 | 28 | 33 | 38 | 40 |
| 15 | 13 | 20 | 25 | 32 | 36 | 40 | 40 |
| 20 | 17 | 22 | 28 | 33 | 38 | 40 | 40 |

In the absence of sound power level data, sound pressure level at a specified distance from the item of plant may be obtainable from the manufacturers or by measurement of similar items. The summation of such sound pressure levels can only be done by a commonsense approach.

Consideration is also necessary of the measurement distances, plant room sizes (of both the measurement site and the plant room which is the subject of the design) and the location of the various plant items relative to each other. Some plant rooms will contain pockets of especially high sound pressure levels. Chambers for fans with open inlets or discharges will receive the whole sound power level from one side of the fan and little attenuation will be found through heating or cooling coils, washers etc. into adjacent chambers. High sound pressure levels in such chambers may excite structural vibration[11] and vibration isolation of the chamber may be justified if sensitive occupied areas are close.

*Plant Room Noise Control*

Table B12.22 indicates the likely noise reduction through floors, walls and ceiling slabs of various superficial densities. Subtraction of the reduction values from the plant room sound pressure levels and comparison with the required NR on the opposite side will indicate the necessary superficial density. If the proposed structure cannot meet this requirement then some or all of the following may have to be adopted:

(*a*) Provide a floating floor over the whole or part of the plant room area.

(*b*) Suspend the sub-ceilings of rooms below the plant-room or a sub-ceiling in the plant room with resilient hangers, and incorporate an acoustic barrier weighing at least 50 $kg/m^2$ between the slab and the suspended ceiling with mastic seals between the barrier and the walls. The void should be at least 500 mm with all fittings, ducting and air diffusers below the barrier.

(*c*) To avoid leakage apply an acoustic seal at the head and base of plant room partition walls.

(*d*) Provide special acoustic treatment inside the structural walls.

(*e*) Consider the provision of an unpierced concrete-slabbed void between the plant chamber and occupied rooms if a service route is required.

(*f*) Line all or part of the inside surfaces of the plant room with sound-absorbent material.

(*g*) Enclose particularly noisy machines in specially designed acoustic enclosures.

Care must be taken to provide an acoustic seal at all points where ducts and pipes pass through the plant room structure. Such seals may have to be flexible to allow thermal movement and to avoid the transmission of vibration to the structure.

**Table B12.22.** Reduction in sound pressure level due to transmission through various structures.

| Description of structures | Noise reduction (dB) Octave band centre frequency (Hz) 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
|---|---|---|---|---|---|---|---|
| Solid wall or floor, no windows or openings, superficial density of: | | | | | | | |
| 100 kg/m² .. .. .. .. .. .. .. .. | 27 | 31 | 35 | 39 | 43 | 45 | 47 |
| 250 kg/m² .. .. .. .. .. .. .. .. | 33 | 37 | 41 | 45 | 49 | 51 | 53 |
| 500 kg/m² .. .. .. .. .. .. .. .. | 39 | 43 | 48 | 51 | 54 | 56 | 58 |
| 700 kg/m² .. .. .. .. .. .. .. .. | 41 | 43 | 50 | 54 | 57 | 59 | 61 |
| 1000 kg/m² .. .. .. .. .. .. .. .. | 43 | 47 | 51 | 56 | 59 | 61 | 63 |

Plant room doors will be an inevitable weak point in the insulation to the remainder of the building. Where possible a 'sound lock' of 2 doors (or pairs of double doors) in series should be provided. The lobby between the doors should be as large as possible but not less than 1 m long and its walls and ceiling must be lined with sound absorbing material. All doors must be made to close tightly on good seals including the threshold; some proprietary draught excluders are beneficial. Table B12.23[12] shows the effect on overall insulation value for partitions having a 7% door area. The construction of the door becomes more important as the insulation value of the partition is increased. If the acoustic conditions are severe then special acoustically insulating doors may be necessary. When a plant room is generally quiet but contains a very noisy fan chamber, the door to that chamber may be of a proprietary air-tight, sound insulating type.

**Table B12.23.** Insulation values of partitions with doors

| Construction | Insulation value of partition (dB) 25 | 30 | 35 | 40 | 45 | 50 |
|---|---|---|---|---|---|---|
| Any door with large gaps round edges (15 dB) | 23 | 25 | 27 | 27 | 27 | 27 |
| Light door with edge-sealing treatment (20 dB) | 24 | 28 | 30 | 32 | 32 | 32 |
| Heavy door with edge-sealing treatment (25 dB) | 25 | 29 | 33 | 35 | 37 | 37 |
| Double doors with sounding lock (air-space or lobby) (40dB) | 25 | 30 | 35 | 40 | 44 | 48 |

*Services Shafts*

The likely sound pressure level in a services shaft may be roughly estimated by calculating the sound pressure level in the ducts which it contains, subtracting the Sound Reduction Index of the duct wall and adding 10 dB. If the shaft is open-ended to a plant room, the sound pressure level entering may be assumed to be that existing in the plant room less a reasonable allowance of attenuation per metre of shaft if the ducts and pipes are externally lagged with fibrous materials.

Having obtained a spectrum of likely sound pressure level in the shaft the construction required for the walls can be assessed from Table B12.18 and the required NR rating for the adjacent occupied spaces.

Pipes and ducts must not be fixed to lightweight panels lest they be set into vibration; isolating supports are desirable close to sensitive areas.

## INSULATION AND ISOLATION OF BUILDINGS FROM EXTERNAL NOISE AND VIBRATION

### Noise

The planning of a building with respect to its external noise environment is not normally a function of the building services engineer. The architect, or other planning authority, will consider such factors as:

(i) reducing the noise at source
(ii) positioning the building far from the source
(iii) screening the building from the source
(iv) using high acoustic transmission loss materials or structures for the external cladding of the building.

The building services engineer must ensure, however, that the noise climate within the building due to an external source(s) is not worsened in consequence of the presence of the building services. Generally, this will involve consideration of the entry of external noise via any and all perforations in the external structure such as ducted air inlets and outlets, louvres, etc.

This problem is seldom serious but consideration should be given to it and may follow the steps listed below.

(1) Establish the likely octave band sound pressure level spectrum outside the building. This may be obtainable from the architect or other planning authority, but if it is not available then measurements may be taken on site or some degree of prediction is possible.

*Measurements on Site*

(*a*) If the level of noise is substantially constant (e.g. machine noise), measure the mean level per octave band.

(*b*) If the level of the noise fluctuates by say 5 dB in any octave band which would determine the NR rating of the noise use the value of the equivalent continous sound pressure level, $L_{eq}$.

(*c*) If the external noise is due to road traffic, estimate and use a level per octave band which is not exceeded for more than 10% of the time (known as the $L_{10}$ value) relevant to the occupation of the building. (In critical cases, special measuring equipment may be used to obtain accurate values for $L_{10}$ levels and more complex rating methods are available but require specialist advice.

(*d*) If the external noise contains occasional loud noises above a normal background level (e.g. rail traffic or aircraft) complex rating methods may be desired but an assessment in terms of $L_{eq}$ will generally be satisfactory.

*Predictions*

(*a*) Predictions of the relevant levels per octave band may be obtained from measurements at another similar site.

(*b*) Measurements may be taken at a different distance from the source and the relevant levels obtained by extrapolation or interpolation (–6 dB per distance doubling for peak noises and/or localized sources, or –3 dB per distance doubling for continuous traffic flow roads or sources which are large relative to the distance to the building).

(*c*) Many books, articles, etc. on acoustics contain typical spectra of noise which may be useful.

(*d*) Consideration must be given to possible increases in the external noise climate in the future due to road widening, new roads, increased traffic density, new industrial development etc.

(2) Calculate the sound power levels entering, say, the plant room of the building via, say, the louvres.

$$L_w = L_p + 10 \log_{10} S \quad \text{B12.46}$$

where:

$S$ = free area of the louvre or opening m$^2$

From these sound power levels calculate the room sound pressure (see earlier part of this section) and add these, logarithmically, to any sound pressure levels due to plant.

(3) Deduct the insulation between the, say, plant room and the occupied spaces.

(4) Compare the resulting sound pressure levels in the occupied spaces with the NR criterion. If the noise is too high, and is largely a consequence of the sound power levels due to external sources, then noise control measures will be required. Noise control within buildings is discussed below.

As an alternative to steps 1 to 4 above, the following procedure can be used:

(i) obtain from the architect or planning authority (or deduce from data given in Table B12.18) the likely reduction in noise level outside to inside via the main structure.

(ii) Ensure that the path via louvres, etc. has at least an equal attenuation.

**Vibration**

The technique of isolating complete buildings from surface or underground rail or road traffic vibration by the use of resilient building supports is finding increasing application in congested city centres. This technique has implications for the building services, plumbing, etc. as sufficient flexibility must be provided in the services, where they pass the resilient support, to avoid by-passing the isolation. In some cases, the flexibility must be sufficient to allow the future jacking-up of the complete building for inspection and replacement of the resilient building supports.

Individual consideration must be given to each case.

## MEASUREMENT

Measurement of noise may be necessary to establish noise level conformity, or non-conformity, with a specification and/or to provide engineering data for the correction of a noisy installation. Measurements may be necessary within a building or, if neighbourhood complaints have been received, in the open.

### Sound Measurement

*Precautions Off Site*

(i) All measuring equipment should be calculated before and after each usage and, by the manufacturers or a laboratory, every six months or if damage is suspected. It should go without saying that the strength of the batteries should also be checked.

(ii) Proper record sheets are essential. The record must state the position of measurement, the condition of the plant at the time and the settings of the noise measuring equipment, e.g. frequency of weighting and time weighting.

(iii) The correct position for the microphone will depend on the purpose of the noise measurement. For instance, in checking the acceptability of noise in an office, the correct position would be where an occupant of the office could reasonable be seated, usually not less than about 1·5 m from an air outlet. In a plant room, however, the interest is usually in the highest noise levels adjacent to a barrier (wall, floor, ceiling) between the plant room and a critically quiet area. Careful judgement must dictate the correct measurement position. Some further points are made below.

*Precautions On Site*

(i) Follow the instructions given with the instrument.

(ii) Before switching the instruments on, turn the amplifier to the highest setting so that it will not be overloaded, then adjust the amplifier downwards until a reading is obtained.

(iii) Microphones are often directional; make sure that they face the source of noise and are not shielded by objects or by oneself or others who wish to see the dial reading.

(iv) Take readings on the A and C or linear scales, when available, even when performing an octave band frequency analysis. This can provide a useful check on the analysis, see Interpretation below.

(v) Take measurements in several positions as the noise may vary significantly from place to place due to standing waves.

(vi) Make sure that the cause of the noise being measured is known. External noises may compromise accuracy, see *Extraneous Noise* below. Running individual components of a complex plant singly can be very informative.

(vii) Be sure that the microphone is not situated in a moving airstream as air turbulence will generate completely misleading readings. Windshields are available for most microphones and the manufacturers will indicate the maximum air velocity for which they are suitable.

(viii) Meter readings are likely to fluctuate especially at low frequencies. If the swing is more than 6 dB a fair reading is -3 dB from frequent peaks or consider measuring in termms of $L_{eq}$. Be careful to listen to extraneous sounds which may influence the reading. For swings of less than 6 dB take the average reading.

(ix) Vibration of the microphone and/or meter can generate spurious signals. Whenever possible hold the instrument in the hand, otherwise place it on a soft resilient pad. High sound pressures (>110 dB) can also generate signals within the instrument independently of the microphone signal. To check the effect of vibration and high sound pressures, disconnect the microphone from the instrument and recheck the reading. If it is within 10 dB of the reading with the microphone connected then the instrument is being affected and accuracy will suffer.

(x) Beware of unusual heat or humidity as they may damage the instruments; so may dirt or corrosive atmospheres.

*Extraneous Noise*

Noise from sources other than those which it is required to measure may compromise accuracy. Take a full spectrum without the machines running and compare this to a spectrum taken with the machines running. If the difference between machines on and off is less than 10 dB down apply the corrections listed in Table B12.24.

**Table B12.24.** Corrections for extraneous noise.

| Differential (dB) | Correction to be applied to level taken with machine running (dB) |
|---|---|
| 3 | – 3 |
| 4–5·5 | – 2 |
| 6–9 | – 1 |

If the difference between readings taken with the machine on and with the machine off is less than 3 dB, the measurements cannot be regarded as having any value because the machine noise is so small a part of the total noise.

*Interpretation*

(i) Plot the octave readings on a graph of the NR curves or compare the readings per octave to a table of NR values. The highest NR curve reached in any octave will give the NR values of the noise.

(ii) It is useful to add logarithmically by the method given earlier, the values of readings in each of the eight octave bands. If the summation is not within ± 2 dB of the linear or dBC reading, it is likely that an error or instrument malfunction has occurred and the readings should be repeated.

(iii) Readings are unavoidably taken from time to time in rooms not yet furnished. The effect of furnishing can be appreciable especially at the higher frequencies. The calculation procedure for Effect of Receiving Room will give some guidance as to the probable effect

(iv) In comparing noise readings to NR curves an excess of 2 dB in some octave bands should not be regarded as a serious breach of the criterion.

**Vibration Measurement**

Vibration levels are usually stated in terms of displacement, velocity or acceleration at a given frequency. Measurement of any one of these factors over a range of frequency may be used to establish vibration level. It should be noted that vibration levels can vary substantially within short distances and over short periods of time, so measurement results should be treated with circumspection. Techniques for measurement can broadly be subdivided into mechanical, electrical, improvised and subjective.

*Measurement Procedure*

The following points should be noted:

(*a*) All instruments should be regularly calibrated at about 6 to 8 month intervals by the manufacturers or a recognized laboratory to ensure that the readings are reliable. Calibration should also be required if the instruments are accidentally subjected to impact or extremes of temperature or humidity. Always consult the manufacturer’s recommendations concerning the conditions in which the instruments may be used.

(*b*) Do not attempt to measure the vibration of any panel or component whose weight is not very much greater than the pick-up or probe. The weight of the pick-up or probe can significantly affect the vibration; as an example of this, window pane vibration cannot normally be measured without special lightweight pick-ups. Similarly, the weight of the test engineer on a lightweight steel platform or floor can change its vibration characteristics and give misleading results.

(*c*) Some pick-ups and instruments, if improperly screened, can receive and record extraneous signals. Check for this by observing the instrument with the pick-up disconnected—if there is still a reading then an extraneous signal is being received. Also check for an extraneous signal by observing whether there is an equal reading when the pick-up is removed from the machine under test.

(*d*) When using a probe, there is a temptation to measure in potentially dangerous places. It is important that all safety precautions are taken.

(*e*) Ensure that the measurement taken is of a reasonably solid part of the structure or machine whose motion is to be determined. Lightweight components, such as sheet metal covers, may vibrate much more than the structure to which they are attached.

(*f*) Ensure that a complete statement of the conditions at the time of each test is written down immediately. This statement must give date, time, site of measurement, instruments used, direction of measurement (i.e. vertical, horizontal, etc.), numbers and types of machines running, etc. There is no substitute for an orderly and neat record sheet.

(*g*) In most cases, the reading from a dial will not be constant but will vary continuously over a range. Take the average reading if the variations are quite small; take the average and the extreme readings if the variations are large or in critical cases.

(*h*) If possible, arrange for small changes in machine speed and measure changes in vibration level. If a resonant condition exists a small change in machine speed can cure a vibration problem without significantly affecting performance.

(*i*) As with all types of noise and vibration measurements it is desirable when tracing the source of an excessive disturbance to arrange for machines to be run individually to check whether a single, easily cured source is present rather than a summation of several sources each of which by itself would be acceptable.

*Interpretation of Measurements*

Site measurements must be compared to the specified maximum levels and, if necessary, action taken in accordance with the procedure given above to correct excessive vibration. In the absence of specified levels assess the site measurements against the recommendations in Section A1.

Comparison of site measurements against the data for subjective reactions to vibration given in Section A1 serves as a useful check against errors due to mishearing, or in writing down errors called out, on site over the noise of machines.

## EQUIPMENT

### Attenuators

*Packaged Attenuators*

These are factory made attenuators designed to achieve a large insertion loss over a relatively short length of unit. Designs differ according to the type of insertion loss spectrum they give, i.e. whether they predominantly absorb low, mid and/or high frequency noise. Manufacturers should provide measured insertion loss data for the performance of their packaged attenuators (see BS 4718[13]). The insertion loss spectrum of a packaged attenuator is a function of the absorbing material, its arrangement and mounting.

Attenuators are normally for broad band performance but can also be made to selectively absorb noise over a narrow frequency range. Where the requirement is for the attenuation of significant pure tones the manufacturer should be consulted to ensure that the attenuator is properly matched to the system.

*Lined Plenum Chambers*

A plenum chamber can provide substantial insertion loss if the walls of the chamber are lined with a sound absorbing material. The insertion loss for the arrangement shown in Fig. B12.17 can be calculated as follows:

$$L = 10 \log_{10} \left[ \frac{1}{a \left( \frac{S}{2\pi S^3} + \frac{1-a}{A a} \right)} \right] \quad .. \quad \text{B12.47}$$

where:

| | | | |
|---|---|---|---|
| $L$ | = | insertion loss .. .. .. .. | dB |
| $a$ | = | area of duct at entry to plenum .. | m$^2$ |
| $s$ | = | perpendicular distance between entry and exit planes .. .. .. | m |
| $S$ | = | slant distance, entry to exit .. .. | m |
| $A$ | = | total area of absorbing surface in plenum | m$^2$ |
| $a$ | = | random incidence absorption coefficient of lining | |

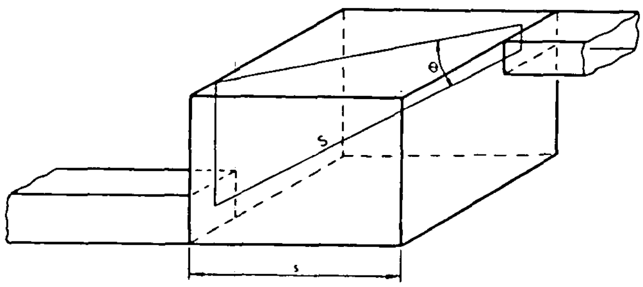


**Fig. B12.17.** Plenum chamber.

*Lined Ductwork*

The simplest application of sound absorption within a ventilating system is the lining of an air duct with a sound absorbing material. In addition to lining the walls of the duct absorbing splitters can be added to improve the insertion loss. In the absence of specific acoustic performance data for a lined straight rectangular or circular duct, the following formula has been found to give a reasonable prediction of the insertion loss when the total length of lined duct does not exceed about 2 m.

$$IL = \frac{P}{A} \, l \, a^{1.4} \quad .. \quad .. \quad .. \quad .. \quad \text{B12.48}$$

where:

| | | | |
|---|---|---|---|
| $IL$ | = | insertion loss .. .. .. | dB |
| $P$ | = | perimeter of lined duct .. .. | m |
| $A$ | = | free cross-sectional area of duct .. | m$^2$ |
| $l$ | = | length of lined duct .. .. .. | m |
| $a$ | = | random incidence absorption coefficient of lining material when fixed to rigid backing | |

For lined straight lengths greater than 3 m this equation is likely to overestimate seriously the insertion loss at high frequencies, i.e. at frequencies above 700/d Hz where *d* is the maximum length of side of the duct (in metres).

In a duct incorporating lined splitters, each air passage represents a separate lined duct to which the above formula can be applied. However for a splitter separating two air passages, the equivalent lining thickness is half the splitter thickness.

The effectiveness of a lining material is increased if applied to a radiused bend. Typical values are given in Table B12.25.

**Table B12.25.** Insertion loss of lined, radiused bends.

| Bend width (mm) | Insertion loss (dB) | | | | | | |
|---|---|---|---|---|---|---|---|
| | Octave band centre frequency (Hz) | | | | | | |
| | 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
| 150 | 1 | 2 | 5 | 8 | 11 | 11 | 8 |
| 300 | 1 | 3 | 5 | 12 | 15 | 15 | 12 |
| 450 | 1 | 3 | 9 | 13 | 18 | 18 | 15 |
| 600 | 5 | 8 | 10 | 17 | 25 | 25 | 18 |

*Notes:*
These figures ‘are for a lining of low density, 25 mm thick glass fibre.
The inside bend radius is equal to half the bend width.

*Influence of Air Flow on Attenuators*
Air flow has two effects on the performance of an attenuator:

(*a*) It generates noise within the attenuator.

(*b*) It modifies the propagation of sound waves, hence changes the rate of absorption.

The self-noise of an attenuator is a function of the design and the air velocity. For packaged attenuators the self-noise characteristics should be obtained from the manufacturers.

The effect of air flow on the propagation of sound in attenuators arises because it changes the velocity of propagation and distorts the wave motion through the presence of boundary layers and turbulence. Since the velocity of sound (340 m/s) is much greater than conventional air velocities this effect can be neglected if the air velocity does not exceed 20 m/s.

*Flanking of Attenuators*
The insertion loss of an attenuator can be reduced if noise short circuits the attenuator or the noise insulation of the downstream duct is not adequate to prevent the intrusion of external noise.

Flanking noise through or around the casing of attenuators may well limit the maximum insertion loss that can be obtained from a unit to about 45 dB. If a higher insertion loss is required then such features as vibration breaks and damped casings will be necessary.

External noise can intrude into a duct downstream of an attenuator if the noise insulation of the duct walls is inadequate or if the duct passes through a noisy area, e.g. a plant room. In particular ordinary flexible joints provide very little noise insulation. This can be improved by using heavier flexible materials.

For ducts passing through a noisy area the following method may be used to assess the sound power level transmitted into the duct:

$$L_w = L_{po} - SRI + 10\log_{10}\left(\frac{A}{4} + \frac{AS_w}{R}\right) \qquad \text{B12.49}$$

where:

$L_w$ = sound power level transmitted into duct dB
$L_{po}$ = sound pressure level in space outside duct .. .. .. .. .. dB
$SRI$ = sound reduction index of duct wall (see Table B12.21) .. .. dB
$A$ = duct cross-sectional area .. .. $m^2$
$S_w$ = area of duct wall exposed to noise $m^2$
$R = \dfrac{(2A + aS_w) \times (2A + S_w)}{(2A + aS_w) + (2A + S_w)}$
$a$ = absorption coefficient for internal surface of duct

Table B12.25 shows that insulating the ducts with a heavy material substantially increases the transmission loss of the walls, thereby restricting the infiltration of noise.

Noise may also break out through the walls of the duct and create excessive noise within a room even though the duct does not serve the room.

This possibility may be checked as follows:

(*a*) assess the sound power level per octave band in the duct as it enters the room,

(*b*) convert to sound pressure level per octave band within the duct using:

$$L_p = L_w - 10\log_{10} A \quad .. \quad .. \quad .. \qquad \text{B12.50}$$

where:

$A$ = cross sectional area of duct .. $m^2$

(*c*) assess room sound pressure level per octave band using the total surface area of the duct within the room as the area of the ‘partition’ and the sound reduction index of the duct wall given in Table B12.21,

(*d*) check calculated octave band sound pressure level against the permissible room sound pressure level.

## Noise Measuring Equipment

Noise measuring equipment normally consists of a microphone, amplifier, frequency analyser and an analogue or digital display or recorder. A brief discussion is given below of each significant component in a measuring system.

*Microphones*
The task of a microphone is to generate an electrical signal proportional to the incident sound pressure and to pass this signal to the remainder of the measuring system but, for example, temperature and humidity can adversely affect microphone performance. The manufacturers of the equipment will specify the safe limits of use. If these limits are accidentally exceeded the microphone should be returned to the manufacturer for checking. The possibility of mechanical damage must also be watched. Extension leads for microphones may be used when convenient but the power loss this involves normally necessitates a correction to the reading.

*Amplifiers*
The amplifier will normally be part of a proprietary noise measuring instrument; no discussion of the types available will be attempted here. As with all other electronic equipment care must be taken to avoid damage.

*Frequency Analysers*
Analysers may form part of a single noise measuring instrument or may be a separate component fed from, say, a sound level meter. Many types of frequency analyser are available. Some measure the strength of a single frequency of noise, others of a constant percentage around a nominal frequency. However, for the purposes described earlier in the section, the most useful analyser is that which measures the strength in each of the octave bands which are used in determining the spectra of the Noise Rating curves (see Section A1). Sometimes a more detailed analysis is desirable and for this a $\frac{1}{3}$ octave band, or narrow band analyser can be used.

*Sound Level Meters*
The most commonly used instrument is the sound level meter (SLM) which incorporates a microphone, amplifier, frequency weighting network(s) and read-out device (either analogue or digital) within a single unit. Many SLMs also incorporate 1/3 octave and octave filters or permit the connection of a filter set.

Integrating sound level meters enable $L_{eq}$ to be determined directly and there are also hand held instruments which provide percentile levels such as $L_{10}$ or $L_{90}$.

Specifications for sound level meters are given in BS 5969[14].

*Tape Recorders*
Tape recorders may be used to record noise on site for later and more convenient analysis. Care has to be taken to avoid overloading and to provide a reference signal on the tape of known sound pressure against which the analysed noise may be assessed. Alternatively a sound level meter may be used on site and a note made of the actual sound pressure measured. Caution is also dictated by the fact that the frequency response of the tape recorder itself may be a compromising factor in the final statement of noise spectra. Due allowance for this frequency response must be made.

**Vibration Measuring Equipment**

*Mechanical Measuring Systems*
These usually comprise a probe coupled by levers to a stylus which records the wave form on a strip of paper which is moved past the stylus's position at a controlled speed. From this record, the frequency and displacement of simple vibration can be obtained by counting the number of cycles in a given length/time of the paper record and measuring the displacement from the paper record, making any necessary corrections due to the effect of the lever system.

Complex vibration consisting perhaps of disturbances from several sources, some or all of which may be changing in frequency and transmitted force, is not easily analysed by mechanical equipment though maximum displacements can be obtained. When, as in most heating or ventilating installations, individual items of plant can be run singly and thus produce a fairly simple vibration trace, mechanical measuring equipment will usually be found adequate and will be favoured for its simplicity. One limitation must be emphasized; mechanical measuring instruments have their own internal natural frequencies of vibration and will not measure reliably at these frequencies, typically the internal natural frequency will be about 5 Hz (but check with the manufacturers) and this is in the speed range of, for instance, slow speed centrifugal fans.

Reed vibrometers are simple frequency measuring devices. They consist of a calibrated variable cantilever, whose natural frequency will vary with the length of the cantilever. By holding the vibrometer against the vibrating object and adjusting the length of the cantilever until a resonant motion of the reed is observed, a close estimate of the frequency of the vibration can be obtained. Continue to check over the whole frequency range of the instrument as many machines will generate vibration at more than one frequency.

*Electrical Measuring Systems*
Many types of electrical measuring systems are available. They vary between simplicity, which may be misleading, and complexity, which may provide an excess of data. There are some reasonable compromises. Most manufacturers of noise measuring equipment provide vibration pick-ups which may be interchanged with the microphone. The pick-ups are normally accelerometers, i.e. they generate an electrical signal proportional to the acceleration to which they are subjected. Often these attachments are switchable to a reading of displacement or velocity as an alternative to acceleration. The noise level meter will be calibrated in decibels and it will be necessary to convert these to the appropriate units for acceleration, displacement or velocity. If an octave, $\frac{1}{3}$ octave or other frequency analyser is available with the noise measuring equipment it may be used in conjunction with the vibration pick-up and an indication obtained of vibration levels in the various frequency bands. With this type of equipment, however, the low frequency limit is about 20 Hz (i.e. the lower audible frequency limit) and many vibration problems are concerned with lower frequencies.

Purpose made vibration meters and analysers can be obtained and can be used to measure much lower frequency vibration. A minimum requirement for purpose made instruments is a frequency response down to 5 Hz.

*Improvised Measuring Systems*
For large displacement, low frequency vibration it is usually possible to obtain a reasonable measurement of displacement provided that a stationary reference point can be found to close the required measuring position.

This reference point may, for instance, be a building column close to a floor which is vibrating or the building floor underneath vibration isolators supporting a machine which is thought to vibrate to excess. As an example of an improvised measuring system, consider a possible procedure for the latter case:

(i) Hold a sheet of paper against a convenient, flat, vertical surface of the machine.

(ii) Arrange a sharp pencil, at a height to mark the paper, by holding it tightly against a stick propped off the floor. The pencil will thus be static for it is supported from the floor but the paper will vibrate with the machine; a mark will be made on the paper which represents the full displacement of the machine.

(iii) Draw an arc on the paper with the stick-supported pencil and measure roughly the time taken to draw the arc. Counting the number of cycles of vibration in the arc will give a rough estimate of the frequency of vibration.

*Subjective Measurement*

Estimates of displacement from visual observation are frequently excessive and should be disregarded. As the frequency of vibration is generally known from machine speeds, etc., an estimate of displacement can often be made by assessing the vibration as 'just perceptible', 'unpleasant' etc. and referring to Section A1. However, because some machines produce significant levels of vibration at multiples of their rotational speeds, this assessment may be in error.

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# SECTION B13 COMBUSTION SYSTEMS




*This edition of Section B13 first published: 1986*

# SECTION B13 COMBUSTION SYSTEMS

## INTRODUCTION

This Section deals with various aspects of fuel utilisation including supply, storage, distribution, chimney and flue sizing and construction, boiler house ventilation and general combustion data, together with commonly used equipment. The characteristics of particular fuels and fundamental data on combustion appear in Section C5.

## SOLID FUELS

The term 'solid fuel' is applied to a wide range of combustible materials including naturally occurring fuels, e.g. coal, lignite and peat, manufactured fuels, e.g. coke and brickettes and other materials such as wood waste and town refuse.

Although formerly widely used, especially in small heating plants, coke is no longer popular because of its high cost.

In the UK, the most widely used solid fuel for the industrial and commercial sectors is coal. Its properties, both physical and chemical, vary considerably and these are dealt with in detail in Section C5. However, the main characteristics are described in the following.

### Coal Classification

Coal is classified in terms of 'rank', a high rank coal having a high carbon content. The other major combustible element, hydrogen, occurs in the form of hydrocarbons which are released as combustible gasses when the coal is heated. Carbon which does not pass-off as volatile matter is referred to as 'fixed carbon'. The higher the volatile content the lower the rank. The properties of various coals, along with their NCB rank numbers, are given in Section C5, Table C5.1.

As coal is heated the particles swell and bind together, known as 'caking', to form coke. The extent to which caking occurs varies, as does the quality of the coke produced. Caking and swelling of different coals can be measured on a comparative basis by means of the Gray King assay; this test is used to define additional classes.

All coals contain both ash and moisture, neither of which produce heat. Washed coals have lower ash content and higher volatile content than the untreated coals from which they are prepared. In classifying coals such variations are eliminated by calculating the properties for a fuel containing neither ash nor moisture. Thus the data in the classification table refer to 'dry mineral matter free' (DMMF) coal.

The total moisture content is made up of surface and inherent moisture. Only surface is affected by climatic changes.

*Calorific Value*

The calorific value of coal does not affect its rank number. For coals having the same rank number the variation in calorific value for the DMMF fuel is very small. The variations in calorific value for 'as fired' coals are due to differing proportions of ash and moisttire. As fired, the calorific value of coal ranges from 21 to 31 MJ/kg.

With mechanical stokers the rate at which coal can be burnt varies widely and no tabulated data are available. The design output or rating of coal fired equipment is generally based on a calorific value of 26 MJ/kg and ratings should be adjusted when coals of higher or lower calorific value are used.

*Ash Fusion Temperature*

The melting point of coal ash depends on its composition and is highest when the main constituents are alumina and silica. Other constituents may act as fluxes to reduce the melting point. Coal with a low ash fusion temperature is more likely to produce troublesome clinker.

Ash fusion temperature is given in terms of the deformation temperature, hemisphere temperature and flow temperature of a specially prepared sample when heated. The test method is prescribed in BS 1016[1].

### Presentation of Data

*Description of Coal*

Coal is described for sales purposes by a combination of the name of the preparation plant and the grade name, the latter denoting a size category, e.g. 'Daw Mill Singles'.

*Pricing Specification of Coal*

The pricing specification of industrial coals provides data on the main characteristics of the fuel from which its suitability for a particular purpose may be assessed, as well as constituting the basis for pricing. It includes the nominal size, rank and calorific value, together with the volatile, moisture, sulphur and ash contents. There may also be data on ash fusion temperatures.

*Proximate and Ultimate Analysis*

Proximate analysis gives moisture, volatile matter, ash and fixed carbon contents. Ultimate analysis gives the carbon, hydrogen, nitrogen and sulphur contents.

*Full Specification*

This entails considerable laboratory work and is provided only where warranted by technical considerations relating to industrial plant. Usually it includes a full chemical analysis, calorific value, rank number, ash fusion data and a detailed size grading analysis.

### Effects of Trace Elements

*Sulphur*

The average sulphur content of British coals is 1.6% but a proportion of the sulphur generally remains as sulphate in the ash. Sulphur is also associated with low-temperature corrosion since oxides of sulphur may combine with water to produce acids which condense on surfaces having a temperature below the acid dew-point. The sulphur contents of coal also affects the design of chimneys, see *Chimney and Flue Sizing.*

*Sodium, Potassium and Phosphorus*

These elements are associated with bonded deposits. The chemical reactions which initiate deposit formation are complex. However, in general, a bonded deposit on the metal surfaces of a boiler takes the form of a layer, in contact with the metal, containing a high proportion of sodium and potassium salts. At high gas temperatures these salts remain molten on their outer surfaces whilst solidified and adhere to the cooler metal. As combustion progresses, fly ash in the gas tends to stick to the initial deposit which builds up into a larger mass. The most common example occurs in some types of boiler where deposits form at the inlet to the first bank of smoke tubes, thus restricting heat transfer.

The presence of sodium or potassium in a coal does not in itself predispose it to bonded deposit formation. Furthermore, methods exist by which deposit formation may be reduced where deposition is likely due to the coal composition and combustion conditions.

*Chlorine*

Chlorine usually occurs as sodium or potassium chloride. The presence of chlorine is determined more easily than sodium and potassium. Therefore, chlorine content is an indication of the presence of sodium and potassium.

### Coal Delivery

*Delivery Vehicle Types*

Except for very large installations, delivery to industrial and commercial consumers is generally by road vehicle. Unloading may be by tipping, mechanical or pneumatic conveyor, or some combination of these methods.

*Tipper Vehicles*

Tipper vehicles unload at the rear, with payloads from 8 to about 20 tonnes. Smaller loads can be delivered but this may increase the delivered fuel costs.

*Mechanical Conveyor Vehicles*

Conveyor delivery vehicles have sloped sides so that the coal falls directly onto a belt conveyor. At the rear of the vehicle a second conveyor receives the fuel and delivers it to the bunkerage. The second conveyor can be rotated horizontally, so that the vehicle may unload to the rear or to either side, and inclined to discharge up to 2.8 m above the ground. Payloads range from 8 to 15 tonnes.

*Pneumatic Delivery Vehicles*

Pneumatic delivery vehicles discharge graded coals up to 50 mm in size into a stream of compressed air which carries the fuel through pipes to the bunker. The air is supplied by a blower at pressures up to 90 kPa and the coal is introduced into the air stream through a rotary valve. Hose diameters are between 100 and 130 mm. The rate of delivery varies with the hose length and the size of fuel but rates up to 40 tonne/h can be achieved.

*Access Requirements*

Access roads between the public highway and the point of discharge should be designed to suit the dimensions and turning circle of the type of delivery vehicle likely to be used. When internal roadways are accessed via a gateway to which a direct approach is possible, the minimum opening should be 3.7 m. Where the vehicle is required to enter at an angle the width of the opening should be increased to 4.3 m.

Where tipped delivery is used there should be adequate space for reversing and the vehicle must be able to reach its unloading position without excessive manoeuvring. Adequate headroom for tipping must be available.

For conveyor delivery, less headroom is required at the point of discharge and it may not be necessary to provide turning space for the vehicle since discharge to either side is possible. Pneumatic delivery vehicles do not require direct access to the delivery point. The maximum distance from the vehicle to the point of discharge should not be greater than 27.5 m, measured along the pipe. For pneumatic/tipper vehicles, the height of the vehicle fully tipped must be allowed. The maximum length of flexible tube carried by these vehicles is 9 m.

### Coal Stocks

Coal may be stored at the consumer's premises in various ways, e.g. bunkers, silos, open stock piles. Estimation of coal stocks by volume should be based on the data given in Table B13.1.

**Table B13.1.** Bulk density and specific volume of various coal types.

| Coal type | Size (mm) | Bulk density ($t/m^3$) | Specific volume ($m^3/t$) |
|---|---|---|---|
| Graded | >12.5 | 0.8 | 1.25 |
| Dry smalls | 12.5 – 0 | 0.83 | 1.20 |
| Washed smalls | 12.5 – 0 | 0.88 | 1.14 |

### Bunkers

A total storage capacity equivalent to at least 100 hours operation at maximum plant output is recommended, with a minimum bunker size of 20 tonnes. The usable capacity of a bunker without resort to manual trimming depends on three factors:

(*a*) The materials of construction which affect the angle of repose.

(*b*) The method of delivery which determines the height to which the bunker may be charged.

(*c*) The method by which coal is withdrawn from the bunker.

*Bunker Linings*

The main structure is usually steel, brick or concrete. The application of a suitable lining reduces abrasion and friction between the coal and the bunker walls to improve bunker utilisation.

Corrosion occurs in steel bunkers storing bituminous coal owing to the moisture contained, but this can be avoided by spraying on a waterproof concrete lining immediately after erection. Alternatively, glass, ceramic or basalt tiles may be bonded to bunker walls. Another lining material in common use is ultra-high molecular weight polyethylene, either in sheet or reconstituted form. The fixing of bunker linings requires special techniques.

*Angle of Repose*

Rectangular flat-based bunkers cannot be emptied completely and the ‘dead’ coal acts as a reservoir. The calculation of usable and reserve volume is based on the ‘angle of repose’ taken up by the coal. Experience shows that this angle is variable and unpredictable. However, some typical values are given below. Bunkers with hopper bottoms can be emptied completely.

*Tipper Delivery*

Delivery costs are least when simple tipper vehicles are used. However, the bunker must be below road level and the costs of excavation, lining and tanking can be high. On sloping sites it may be possible to achieve a large height difference between bunker base and finished ground levels. Provided that a direct vehicle approach is possible, the clear opening for tipping need not exceed 3.7 m. A substantial stop 230 mm high should be provided and the distance from the stop to the rear bunker wall should be at least 2.5 m. The opening height should be such that the vehicle body in its fully tipped position does not foul the upper structure. Vertical roll shutter doors prevent contamination of the coal and improve the appearance.

The bunker should be covered with a grid screen to prevent the ingress of large objects which may damage the coal extraction equipment. A square mesh grid (100 mm) is usual and operators must be able to stand on the grid in safety to free bridging across the bunker outlet should this occur. If vehicles are likely to stand on the grid, the structure must be strong enough to take the expected axle weights. For smalls or singles, the grid could consist of flat bars spaced 65 to 75 mm apart, with 16 mm diameter round bars and ferrules.

*Conveyor Delivery*

Conveyor delivery vehicles permit greater flexibility in the design of bunkerage and the bunker can be filled from a side opening near its top. The conveyor can be backed in to the opening and advantage taken of the throw and radial movement. The optimum throw of these conveyors is obtained with a boom lift of 1.5 m.

The most efficient storage is achieved with the lower edge of the bunker hatch located at a height of 1.2 m. The hatch should be wide enough to allow sufficient radial movement on the conveyor to give an even spread. The hatch should be arranged to close progressively upwards from its lower edge as the conveyor is withdrawn.

Multiple boiler installations with wide bunkerage should be provided with continuous or self-continuous openings. This allows better use of the angular movements of the secondary conveyor together with the movement of the vehicle.

*Pneumatic Delivery*

Pneumatic delivery, although possible only with graded coals, is the most flexible method, the major advantage being that direct access to the point of delivery is not required and large stocks can be held above ground level without on-site handling equipment. It should be noted that pneumatic delivery is extremely noisy.

The essential bunker furnishings are:

(*a*) A fixed 127 mm diameter delivery pipe with long radius bends terminating in a suitable coupling as near as possible to the standing point of the delivery vehicle. The maximum length of this pipe in either a vertical or horizontal direction is limited by the blower pressure available at the vehicle.

(*b*) A coal discharge device consisting of a plain open end on the delivery pipe, particularly suitable where it is desired to make maximum use of bunker storage volume.

(*c*) A glass fibre filter through which air is exhausted from the bunker. This is particularly important if the exhaust is into the boiler house. The most effective exhaust points are directly adjacent to

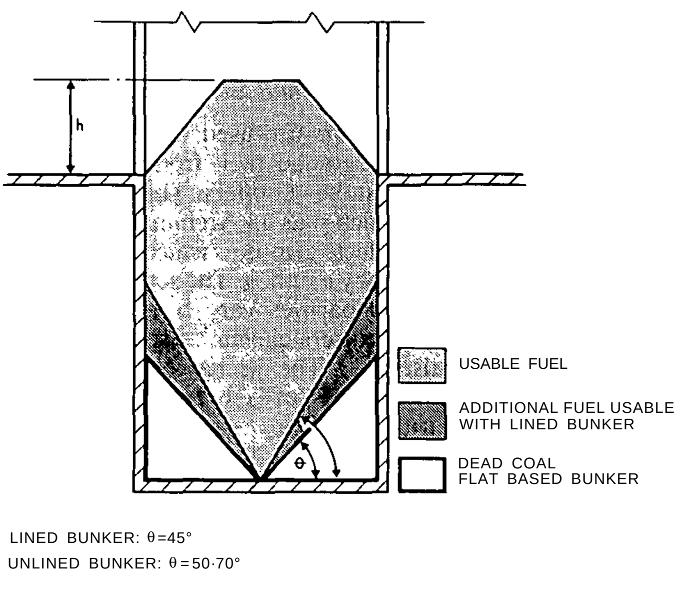



**Fig. B13.1.** Typical angles of repose.

the bunker filler pipe and on the bunker roof. Filters may be placed in any convenient position provided they have a minimum filter area of 0.37 $m^2$ and will not be covered with coal on the inside of the bunker.

(*d*) An access door.

*Bunker Bases and Fuel Extraction*

Whatever the method of delivery, the bunker base should be designed to suit the method of coal extraction. Overhead bunkers normally have tapered bases; other bunkers may have a flat base in which case the coal takes up its own angle of repose within the bunker leaving 'dead' coal in the corners. These pockets should be avoided because of the risk of spontaneous combustion. However, bunkers of this type normally store singles, for which the risk is low, and the 'dead' coal is regarded as standby fuel.

Bunker discharge may be downwards by gravity or a conveyor may be used to extract coal horizontally or at an angle to suit the requirements of the combustion appliance. Coal may discharge from hopper or tapered bases, or to bucket elevators.

*Bunker Outlet Chutes*

Chutes may be lined with either glass or plastic to reduce their angle of repose. The cross-sectional dimensions should be such as to allow a free flow of coal without bridging.

*Bunker Drainage*

Where washed grades of fuel are used, adequate drainage should be provided to deal with free moisture which may percolate to the bunker base. The shape of the bunker base will dictate the drainage method employed. For bunkers with sloping sides, moisture may be drained easily from the lowest point. In flat based bunkers, land drain pipes can be used, set below the coal extraction screw.

*Bunker Level Controls*

Various systems are available to indicate the fuel level remotely and operate handling equipment automatically to maintain maximum and minimum levels.

*Safety*

A notice should be displayed at all bunker entrances to the effect that no person should enter the bunker for any purpose without duly noting the requirements of the Health and Safety at Work (etc) Act 1974.

**Silos**

Concrete stave silos have been used successfully for the storage of coal for many years in the USA and, to a lesser extent, in the UK. However, vitreous enamel coated steel silos have been in use for much longer in the UK and silo storage is now accepted as a means of overcoming limitations on storage space.

The detail of silo design is not covered here and specialist advice should be sought. The following points should be considered.

*Spontaneous Combustion*

Spontaneous combustion is unlikely in normal use. Experience of coal storage in open yards indicates that 2 to 3 months is the danger period for build-up of heat to spontaneous combustion conditions and after this period the coal can be regarded as safe provided that the conditions remain unchanged. The most reliable way to avoid the problem during summer shutdown is to empty the silo of all but a few tonnes retained for contingencies. Regular monitoring for carbon monoxide over the first four months of a period of shutdown warns in advance of a heat build-up.

*Dust Explosions*

A cloud of dust of any flammable material will explode where:

(*a*) The concentration of dust in the air falls within the explosive limits.

(*b*) There is an ignition source of sufficiently high energy.

*Gas Explosions*

Methane is emitted from freshly mined coal, most actively after crushing in deep mines. If a gas pocket accumulates, there is a high risk of ignition by impurities in the coal causing a spark. Pneumatically delivered coal containing iron pyrites is a potential source of ignition energy. Methane rises, being less dense than air, and the installation of a cyclone is recommended to avoid a critical accumulation of gas. A cyclone must be located at the highest point in the silo roof to avoid gas pocket formation.

*Safety Recommendations*

Access ladders and catwalks are essential to inspect furnishings in accordance with local regulations. Maintenance must be carried out according to recommended procedures.

**Solid Fuel Handling Equipment (Small Plant)**

For single shell boiler and small central heating plants, the following systems provide simple and cheap automatic handling facilities.

*High-speed Screw Conveyors and Elevators*

High-speed screw conveyors are used to elevate fuel, either washed smalls or graded fuel, from firing floor level up to mechanical stoker hoppers. They consist of a drawn steel tube in which runs an Archimedian screw. The head carries an electric motor which drives the screw either by gears or by a pulley. The screw is driven at 6 to 12 rev/s and capacities vary from 0.5 to 10 tonne/h. Some manufacturers elevate at speeds as low as 2 rev/s to minimise wear. The discharge is by a single chute for single-flue boilers or a breeches piece for delivery to twin-furnace boilers.

If the base of the elevator is to be placed in a ground hopper, agitation should be considered to avoid 'rat-holing' which can take place with wet washed smalls. This can be done by installing a gyrator or by vibrating the hopper.

If a graded coal is to be handled by screw conveyor, the tube should be 150 mm in diameter; some makers have standardised on a diameter of 130 mm for all fuels. These machines are of simple construction, the working parts are easy and cheap to replace and they may be supported by a simple wire rope from a convenient overhead beam. However, screw conveyors can be very noisy and may cause some fuel degradation.

The BCURA* open-screw feeder was developed as a means of providing a positive feed from a flat storage area to the base of a screw elevator. It is especially suitable for fuel reclamation from rectangular fuel bunkers for small boiler installations. The feeder consists of two screws mounted side by side which traverse through the coal heap. The screws are driven in opposite senses, so that when the machine reaches the end of one traverse, the other screw continues to provide a positive feed of fuel to the base of the elevator.

*Overhead Monorail*

A versatile handling system is the electric hoist and grab on an overhead monorail. This can be used to reclaim fuel from stock, convey it to the boilerhouse and then discharge it into stoker hoppers or bunkers. One such system can be programmed to charge the hoppers in sequence.

*Skip Hoist*

This has the advantage of few moving parts. Although its lack of fine control renders it unsuitable for some boiler installations, it is particularly appropriate for re-charging magazine boilers and is unaffected by the abrasiveness of coke. These systems may be automatic or semi-automatic.

*Pneumatic Handling*

Static pneumatic handling systems may be used for transferring coal from a delivery bunker to an overhead bunker. Combined pneumatic coal and ash handling systems are sometimes employed, powered by a single blower or exhauster.

**Solid Fuel Handling Equipment (Large Plant)**

*Chain and Bucket Elevator*

This is commonly employed to deliver fuel from a ground hopper to the stoker hoppers of one or two shell boilers; where more boilers are employed, the elevator discharges to a cross-conveyor feeding the individual stoker hoppers.

The elevator consists of a continuous chain of malleable iron links, either belt-driven or directly coupled to a geared motor through a slipping coupling. The buckets attached to the chain may be from 130 to 610 mm wide but for most industrial boiler plant a width of 180 to 250 mm is adequate. An elevator incorporating a 180 mm bucket running with a chain speed of 0.25 m/s has a capacity of 1.5 tonne/h and a 300 mm bucket with a chain speed of 0.5 m/s has a capacity of 10 tonne/h.

*British Coal Utilisation Research Association

This type of elevator is usually automatically controlled by a diaphragm switch fitted in the side of the delivery chute or a pair of probes fitted at high and low levels. The elevator is automatically started when the bottom probe is uncovered and switched off when the high level probe in the delivery chute is again covered by fuel. When installed, the elevator angle can vary between 60° and 90° to the horizontal.

*Belt and Bucket Elevator*

This is similar in application to the chain and bucket elevator, but is normally installed vertically. This is a limitation but there are the advantages in that higher speeds can be used and it is quieter in operation.

*Belt Conveyor*

This is commonly used for handling materials in bulk. It is silent, economical in power consumption and simple to operate and maintain. When used for handling coal, it is normally limited to transferring the fuel horizontally as the maximum elevation is about 30°.

Conveyor capacity depends on the belt speed and belt width and the rate of loading. A 300 mm wide belt running at a speed of 0.5 m/s has a carrying capacity of 10 tonne/h.

*Drag-link Conveyor*

Drag-link conveyors are the most common system for moving coal from a single point over the top of mechanical stoker hoppers. They are strongly constructed and consist of an endless chain of mild steel or malleable iron U-shaped links. The bottom strand of the chain runs in a cast iron channel and returns over cast iron idlers. The chain carries the fuel from the supply point to a series of openings into the hopper chutes controlled by independent rack and pinion operated gates. As each chute is filled the supply is diverted to the next opening until all the hoppers have been filled. Normally the drag-link conveyor is manually controlled but automatic operation can be achieved by similar methods as those for chain and bucket elevators.

*Screw Conveyor*

Screw conveyors are sometimes used as an alternative to drag link conveyors. They operate at 1 to 2.5 rev/s and have diameters in excess of 230 mm. They are cheaper to install but tend to cause greater degradation of the fuel. They can have multiple discharge points and have the advantage of being able to discharge in two directions simultaneously by employing opposite-handed flights on either side of the elevator discharge point. These conveyors are ideal for washed small coal.

*Combined Elevator and Conveyor*

The combined elevator and conveyor comprises an endless chain of interlocking malleable iron links totally enclosed in steel trunking. Noise generation maintenance requirements and power consumption can all be greater than for other conveying systems.

These conveyors can both convey and elevate without the necessity of discharging from one system to another. Therefore, they can be used for withdrawing from flat bottom bulk fuel storage areas, elevating to the required level and then discharging to a series of overhead bunkers.

**Coal Firing Methods**

A variety of firing appliances are available for solid fuel. It is important that the size and rated output of the equipment matches the capacity of the boiler or furnace, subject to agreement with the boiler or furnace manufacturer. The choice of equipment for a particular application depends largely on the type and size of boiler or furnace and its load characteristics, and on the types of coal locally available. The final choice of firing equipment can influence the choice of complementary coal and ash handling plant.

*Hand Firing*

Although hand firing with bituminous coal is no longer practiced, it may still be employed with smokeless fuel. For example, the fire box of a sectional boiler can be charged two thirds full with anthracite, but mechanical draught will usually be needed. Hand firing is seldom used, the greater efficiency and convenience of automatic combustion appliances outweighing the higher capital costs.

*Gravity Feed Combustion*

The simplest combustion equipment delivers coal by gravity directly into the fire bed. For successful firing, the fuel should not take or swell on reaching the combustion zone; it must be hard, flow easily, and have a low volatile content so that under correct draught conditions there is no flame to move back up the feed tube.

Anthracite, a natural smokeless fuel, meets these requirements. It differs from bituminous coal in that the volatile content is much lower and the calorific value is higher. This means that boilers and burners designed for burning anthracite can have heat transfer surfaces close to the fire bed. Such appliances are not suitable for burning bituminous coals because the higher volatile content is chilled by the heating surfaces before it has had time to burn completely.

*Gravity Feed Burners and Pre-burners*

These consist of an external fuel hopper feeding a delivery tube which passes down through the boiler front plate at an angle of 45°. Inside the boiler the flue forms a cone over a water cooled hemispherical hearth. A single fan supplies both primary and secondary air; primary through the hearth from an annulus and secondary through an orifice in a horseshoe-shaped tube mounted over the fire. This system of combustion makes full use of the radiant heating surface within the boiler and the water cooled hearth augments the boiler heat transfer surface.

Depending on the configuration, boilers with capacities as high as 1.1 MW can be equipped with this system. Frequently the height of the hopper is such that a screw elevator is necessary, in which case the size of the hopper can be reduced by incorporating a level switch.

Pre-burner units are available in which anthracite is burnt in a water cooled furnace with an integral fuel hopper fitted in front of the boiler. Hot gases from this furnace pass into the boiler and the cooling water rejoins the boiler water system.

Gravity fed anthracite appliances are simple in principle and operation. They have a high turndown ratio and can produce high thermal efficiencies. Often anthracite fuel beds, dormant for up to sixty hours, become active immediately the air supply fan is turned on. When the fan is switched off, air leaking through the stationary impeller under the natural chimney draught enables the fire to remain alight with very low fuel consumption.

*Underfeed Stoker*

This type of stoker is available for sectional, shell and small water-tube boilers up to 1.76 MW rating. Underfeed stokers consists of a retort or pot surrounded by a plenum chamber from which air passes through tuyeres into the retort, see Fig. B13.2.

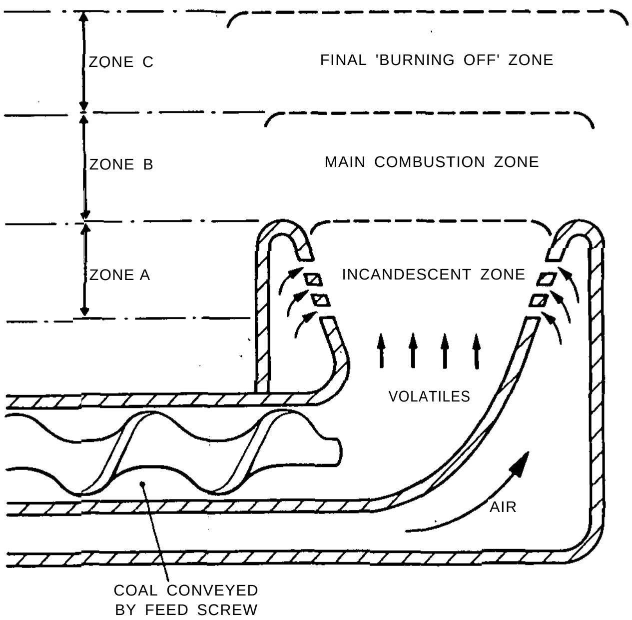



**Fig. B13.2.** Combustion of coal by underfeed stoker.

Coal is delivered into the bottom of the retort by an Archimedian screw. After ignition, the coal is pushed upwards into zone A, meeting the primary air supplied by the forced draught fan, and becomes incandescent. Below this zone there is some heating of the fuel and the volatiles burn rapidly and completely in passing through the incandescent zone. When it reaches zone B, the fuel is almost free from volatiles. The bed is completely incandescent and this forms the main combustion zone from which there is intense radiation. Zone C is the area in which combustion is completed, leaving only the residue of inert material in the form of clinker and ash.

The recommended fuel is bituminous singles of low ash content with slight to medium caking properties. The minimum ash fusion temperature should be 1200°C for Rank 700 to 900 coals, and 1315°C for Ranks 201 and 202.

Installation of underfeed stokers should follow the recommendations of BS CP 3000[2]. In addition, boiler house layout must provide:

(*a*) Access to the boiler front and stoker side for removal of ash, cleaning of flueways and convection passes.

(*b*) Adequate maintenance access to the stoker for feed screw and retort screw withdrawal.

(*c*) A simple arrangement for ash extraction. For smaller plant this will involve mounting the boilers on a plinth 610 mm high which permits ash to be raked from the combustion chamber along a small portable chute into a dustbin.

(*d*) Access to underfeed stoker installations. This is best obtained when boilers are fired from the rear. This involves the provision of a larger boiler house to provide space at the rear of the boiler for withdrawal of the stoker screws. For boilers fired from the front, the stoker screw should be fitted in a trench covered by a checker plate at floor level.

(*e*) Access to front and rear of boiler for cleaning. When the fuel bunker is adjacent to the side of the boiler, it may be more convenient to introduce the stoker through the side of the boiler.

Conventional underfeed stokers are manually de-ashed. The ash takes the form of fused clinker which can be removed easily. Several manufacturers offer optional automatic de-ashing in a stoker which combines the retort with a reciprocating bar grate to the side. Volatiles are distilled and burnt off in the retort but residual coke falls to the grate where combustion is completed, the ash being discharged by a screw conveyor below. Manual attention is not wholly eliminated, but is reduced to the control of fire bed conditions about every 3 days.

*Coking Stoker*

These are available in ratings from 1 to 10 MW. They are usually applied to shell and small water-tube boilers and are readily adapted for fully automatic combustion control. A slight to medium caking coal is desirable; highly caking coals of Ranks 300 to 500 are not ideal. A minimum of 8% ash is required with normal cast-iron fire bars, but chrome alloy bars can be supplied to improve heat resistance. Over-fire draught requirements usually involve the installation of induced draught fan equipment.

The coking stoker consists of a feed hopper at the bottom of which is a ram feeding mechanism which pushes the coal onto a coking plate above the front of the grate where partial distillation of the volatiles takes place. From the top coking plate the coal moves down onto a bottom set of coking plates resting on a moving bar grate.

This design produces a wedge of un-ignited coal. Ignition takes place from both the top and bottom of the furnace. A small refractory arch at the top of the mechanical stoker assists top ignition.

The release of volatiles at the front of the furnace turns the coal into coke before it is carried forward by the moving grate. The term ‘coking stoker’ is used because the coking method of hand-firing is reproduced mechanically. A coking stoker does not necessarily require a coking, i.e. ‘swelling’, type of coal and can deal with a wide variety of grated coals and washed smalls.

Burning and heat release rates vary with the type of coal used. With some low rank coals, it is desirable to design for burning rates above 0.0678 $kg/m^2s$. This compensates for the lower calorific values and ensures enhanced thermal efficiencies by reducing the smoke emission associated with low burning rates.

*Chain Grate and Travelling Grate Stokers*

Chain grate stokers are available for use on all types of shell boiler and for water-tube boilers up to about 80 MW output rating. Travelling grate stokers, in which the tension on the grate links is relieved by support bars, have output ratings up to 75 MW.

Fully automatic combustion control is usually achieved by varying the speed of the grate and damper settings on the forced and induced draught fans. Automatic ash extractors can be fitted on some types. The units burn a wide range of coals but the minimum ash content required to maintain fire bar cooling is about 8%. For values below this, special heat resistance links should be considered for the grate. The ideal fuel is 13 mm washed smalls, with an upper limit of 25 mm.

These stokers consist of an endless chain or ‘moving mat’ which feeds coal continuously to the boiler furnace. The rate of feed is regulated by the speed of the grate and the thickness of the fire is controlled by an adjustable guillotine door. In the chain grate stoker the surface of the grate is under tension. In a travelling grate, driving links are under tension with a mat of grate links simply resting on the driving links. For larger stokers the travelling grate construction is more widely used.

Both chain grate and travelling grate stokers incorporate a feed hopper, a refractory arch over the front of the grate and a forced draught fan which delivers primary air into a wind box below the grate and secondary air above the fire. As the coal passes under the guillotine door and enters the furnace, its temperature rises due to heat radiated from the refractory arch. First moisture and then volatile matter leave the coal. Ignition is established and as the higher temperature zone travels down through the bed, the rising temperature drives off the remaining volatile matter to burn above the grate, leaving coke. The fixed carbon in the coke burns off towards the back of the grate leaving clinker and ash which are finally passed into the ash pit. By installing a conveyor below the grid, ashes can be brought to the front of the boiler automatically.

*Spreader Stoker*

The spreader stoker is well proven on large water-tube boilers but has only recently been produced in sizes suitable for shell boilers. Metered fuel is thrown into the furnace by high speed rotors. Shaped vanes on the rotor produce a suitable fuel distribution pattern along a line stretching from front to rear of the furnace.

The unit responds rapidly to fluctuating loads, and can accept a wide range of coals, including those with low (3 to 4%) ash content, high sulphur and chlorine content, high caking properties and low ash fusion characteristics. The most suitable grade is washed singles. Unwashed fuels or those with significant fines content are not suitable. These stokers can be adapted to fully automatic control.

*Pulverised Fuel*

Most pulverised fuel applications are for large water-tube boilers with a limited application to shell boilers. The system components are a raw coal feeder which feeds a pulverising mill at a rate regulated to match heat demand. The mill feeds special burners, usually of the venturi grid nozzle type for shell burners. The pulverised fuel is normally ignited by gas torches which are removed when stable ignition is achieved.

The power requirements for the fan and pulveriser vary between 70 and 100 J/kg. Regular soot blowing is required and high efficiency multi-cyclone grid arresters are necessary to give an acceptable dust burden of less than 0.687 g/m$^3$ in the flue gas outlet. The free moisture content of the fuel should not exceed 6 to 7% for a volatile content greater than 20% and the fuel should be ground to a particle size of 76 µm for 85% of the material, with no particles larger than 295 µm.

Pulverised fuel forms an explosive mixture with air in certain concentrations. Therefore the system must be designed carefully to ensure safe operation. Almost any type of fuel can be used provided that the specification of the fuel is known at the design stage.

Important characteristics of the fuel are:

*(a)* 'Grindability', which determines the maximum mill output at the required fineness and power consumption.

*(b)* Moisture content, which dictates transport air temperature and volume rate.

*(c)* Ash content, which affects boiler fouling, precipitator design and mill output.

*(d)* Chlorine and sulphur content, which affect boiler fouling.

*Fluidised Bed Combustion*

Fluidised combustion of coal takes place in a fire bed comprising inert materials, such as ash or sand, the individual particles of which are maintained in suspension by a strong upward flow of air, see Fig. B13.3. F1uidised bed boilers can operate at atmospheric or raised pressures and have the following features:

*(a)* a distribution plate through which fluidising air is blown

*(b)* immersed steam raising or water heating tubes which extract heat directly from the bed

*(c)* tubes above the bed which extract heat from the hot gases before they enter the flue.

The fire bed as a whole behaves as a fluid, with vigorous circulation of the particles within it. The bed temperature can be controlled closely and is normally maintained between about 750 and 950°C. At these temperatures, the ash neither fuses nor clinkers so that surfaces in contact with the bed remain clean. There is rapid transfer of heat from the fire bed to surfaces in contact with it. Some designs take advantage of this by immersing the boiler tube surface in the bed.

Fluidised beds can operate at lower excess air levels than are possible with mechanical stokers and a small increase in boiler thermal efficiency is possible. However, a fluidised bed needs greater fan power than a conventional stoker. In countries where sulphur dioxide emissions are closely controlled, fluidised beds offer a means of control by adding limestone to the bed. The resulting chemical reaction reduces the amount of sulphur dioxide discharged to the atmosphere. Fluidised bed combustors are not selective of fuel and, if purpose designed, can accommodate a wide range of solid fuel characteristics.

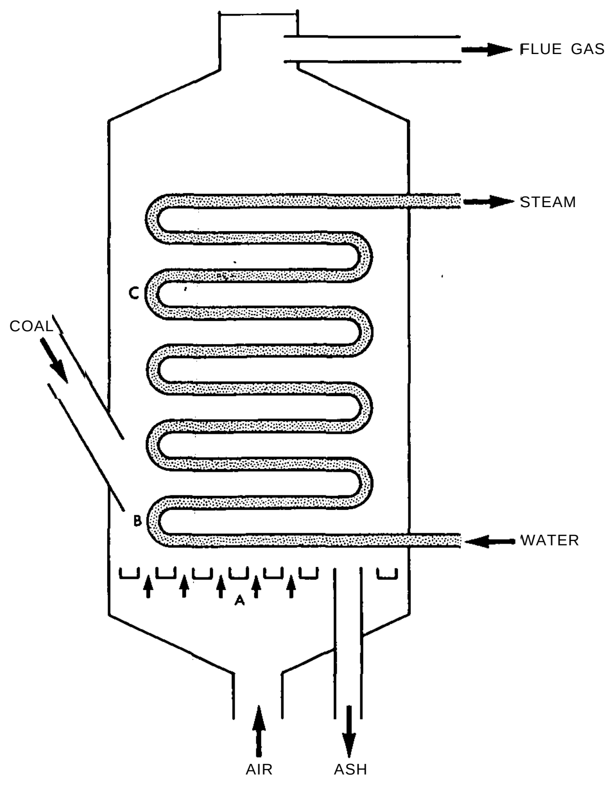



**Fig. B13.3.** Fluidised bed boiler.

*Special Combustion Systems*

Several proprietary systems for burning solid fuels do not fall into the above stoker classifications. One such system feeds the coal into the combustion chamber from a top feed point where primary air is admitted as a vortex. Thus combustion takes place both in suspension and on the grate. Other systems are variations of the cyclone configuration or the coking and spreader systems described above.

## Fuel Ash Handling Equipment

Smaller solid fuel steam raising boilers of recent design usually include provision for automatic ash removal and many new hot water boilers also have sufficient internal storage to obviate daily ash removal. Various methods have been devised to remove ash from larger boilers.

*Ash Extraction from Boilers*

Depending on the firing equipment, ash may have to be extracted from the boiler by hand. Sectional boilers are commonly fired by underfeed stokers and ash and clinker must usually be raked out by hand. However, fully automatic ash removal is provided on some boilers using miniature chain grate stokers. The boiler should be mounted on a plinth of a suitable height so that ash can be discharged directly into replaceable containers at the rear.

Horizontal shell boilers fitted with chain grate or coking stokers may have completely mechanised ash withdrawal. Both of these moving grate stokers discharge ash at the rear. The ash is then brought forward out of the furnace tubes. Three types of ash extractor are commonly used:

*(a)* *Drag-Link*
A series of scraper bars or drag-links are attached to and driven by continuous chains. The scraper bars move across the surface of a plate located beneath the grate to collect the ash. The drive is geared to the stoker so that the extractor speed matches the rate of combustion.

*(b)* *Plough*
A plate beneath the grate collects the ash. Periodically a plough blade is driven forward into the ash by chains. When the plough reaches the rear of the plate, the motor reverses and on the return stroke the ash is scraped out of the furnace tube.

*(c)* *Vibratory*
The ash is collected in a fuel trough beneath the grate which is vibrated, either magnetically or mechanically. This causes the ash to be brought forward and, once clear of the combustion equipment, the ash can be disposed of in various ways.

*Bins and Trolleys*

When ash is deposited directly into bins or trolleys some means of moving these to the disposal point should be provided. In a basement boilerhouse a small hoist, lift or pulley blocks may be used. Large wheeled bins may be used if the boilerhouse layout permits and these can be lifted and emptied mechanically by local authority refuse collection vehicles.

Ash may also be emptied into skip hoist trolleys which are raised and discharged into an overhead silo which is emptied periodically.

*Screw Conveyors*

Screw conveyors are often used to move ash from a pit beside the boiler to a disposal container. Wear is reduced by running the screw conveyors at low speeds, 1 rev/s or less, and by keeping angular inclinations below 40°. Special materials may be used for the flights to further reduce wear. It may be necessary to install a crusher before a screw conveyor if large pieces of clinker are present. A 150 mm conveyor can handle pieces up to 40 mm in size but clinker size tolerance increases with the diameter of the conveyor.

Combined elevators and conveyors, as described earlier, may also be used for moving ash.

*Vibratory Conveyors*

Sealed tubular vibratory conveyors are often used to convey ash. They are simple in construction and require little maintenance.

*Vacuum and Pneumatic Ash Disposal*

Vacuum ash disposal systems are simple, clean and effective. One type uses a steam air ejector to create the vacuum by means of a venturi mounted at the delivery end of the pipe. Contact between the ash and the steam is momentary and there is no risk of packing or freezing the stored ash. Steam at a gauge pressure of about 700 kPa is supplied to the venturi. The mass of ash handled is approximately ten times the mass of steam supplied. Alternatively, a turbo-exhauster may be used to create the vacuum. The exhauster draws air, via a filter, from the storage silo and discharges it to atmosphere.

A pneumatic system, used on large installations, blows the ash towards the dispersal point with air at positive pressure. Another system uses a smaller positive pressure to fluidise the ash as it leaves the boiler. enabling it to slide down sloping ducts to a hopper.

*Submerged Conveyors*

Submerged drag-link and belt conveyors are used on large industrial boiler plant. The chain or belt is a continuous loop and runs in a water-filled trough. The ash falls from the boilers down steel chutes ending below the water surface. On contact with the water, the ash is immediately quenched and dust cannot escape. The conveyor may be inclined at one end to raise the ash to a disposal silo or a separate skip hoist or elevator.

## Selection of Combustion Equipment

All types of combustion equipment can be ignited and controlled automatically. Selection is influenced largely by the class of boiler to which the equipment is to be fitted.

*Sectional and Other Small Heating Boilers*

Underfeed stokers are commonly chosen for boilers up to 1.76 MW. They require little attention and are simple to operate and maintain. Fluidised bed firing is also suitable for these boilers, as is gravity fed equipment where anthracite is to be burned.

*Shell Boilers*

The choice of boiler will often dictate the selection of the combustion equipment. The chain grate stoker is relatively expensive but can use a wide range of smalls for smokeless operation. Extra space is required for removal and repair of the grate but automatic control and mechanical ash removal can be readily accommodated.

Coking stokers can burn a wider variety of fuels than most other firing appliances and can be controlled automatically. However, ash can be removed automatically only when the stoker operates in a drop-tube type boiler. Adequate overfire draught is necessary for smokeless operation.

Fluidised bed firing is available for some 'economic' type boilers as well as those of novel configuration (often including water-tube surfaces) which have been specifically designed to accommodate this method. The latter have high output ratings for their size.

Pulverised fuel firing can be applied to shell boilers. This enables higher output from a given boiler size than that possible with mechanical stokers and it is of particular interest where conversion from oil or gas is undertaken. However, the comparatively high cost of pulverised fuel must be taken into account.

Composite boilers combine a water-tube furnace with a shell-type section and are available in the range 4.5-30 MW. They are fired by mechanical stokers or fluidised bed combusters.

*Water-tube Boilers*

The choice of firing equipment is determined by the boiler size. For ratings up to 9 MW, coking stokers are sometimes installed but the most widely used methods for ratings up to 80 MW is the chain or travelling grate stoker or, alternatively, the spreader stoker.

Pulverised fuel firing may be considered as an alternative to mechanical stokers, especially in conversions from liquid or gaseous fuelled boilers. For boiler ratings over 80 MW, pulverised fuel firing predominates because the furnace dimensions are too large for mechanical stokers.

Fluidised bed firing can be applied to purpose designed water-tube boilers of any size, see *Gaseous Fuels.*

## GASEOUS FUELS

### Town and Natural Gas*

Under the provisions of The Gas Act 1948, the British Gas Corporation is obliged to supply and continue to supply gas to any premises within 23 m of a gas distribution main. The cost of making a connection, laying the service pipe and maintaining it up to a maximum length of 9 m on public land are met by British Gas and that of any remainder on public land, and all on private land, by the applicant.

Where a new or increased supply of gas is requested for uses other than lighting and domestic requirements. and this involves alteration or enlargement of the existing supply main, a written undertaking to receive and pay for a minimum quantity of gas over a minimum period is required. The quantity and period are related to the cost of the works and these are stipulated by British Gas. A capital payment and securities for sums due under the contract may also be required.

The Gas Act 1980 relieved British Gas of the obligation to supply gas to new applicants whose proposed consumption would exceed 25 000 therms per year. These applicants now require a special agreement with the Corporation for a service connection and are likely to be charged according to special tariffs.

*The Gas Safety Regulations*

The Gas Safety Regulations 1972 apply throughout the UK with the exception of the inner London Boroughs, which fall under certain provisions of the London Gas Undertakings Regulations 1954 made under the London Gas Undertakings (Regulations) Act 1939. The Regulations lay down requirements for the installation of gas pipes, meters, appliances and other fittings on consumers' premises.

The Gas Safety (Installation and Use) Regulations 1984 impose requirements as to the nature, installation and use of gas fittings to ensure that the installation and fittings are safe. The Regulations also impose requirements on persons installing or working on gas fittings and prohibit the new supply of gas to buildings unless adequate emergency controls are provided. These Regulations substantially amend the Gas Safety Regulations 1972. Designers, installers and users of gas should take care that installations and appliances meet the requirements of the Gas Safety (Installation and Use) Regulations.

*Town Gas Pressure*

Domestic appliances on town gas are designed to operate at inlet pressures between 0.75 and 2.5 kPa. The area Boards of British Gas are obliged to maintain a stated minimum pressure in the district main.

In certain circumstances gas is supplied to domestic and other premises at pressures above 2.5 kPa, sometimes as high as 200 kPa. Regulators are fitted to reduce the supply pressure to the working pressure within premises

*Natural Gas Pressures*

Domestic appliances designed for natural gas operate at pressures between 1.75 and 2.5 kPa.

Natural gas is normally supplied to premises at pressures up to 5 kPa and governed at the meter to give a nominal pressure between 2 and 2.5 kPa within the premises. The pressure losses in the system from the meter outlet to the furthest appliance connection should not exceed 100 Pa.

In larger commercial and industrial plants, where burner equipment is fan assisted or pressurised, higher operating pressures are needed and a pressure booster in the gas supply may be necessary, see *Gas Pressure Boosters.*

* The legislation referred to in this section may change as a result of the re-establishment of the British Gas Corporation as a public limited company.

*Service Pipes*

The Gas Safety Regulations require that installation or any work on a service pipe must be carried out by British Gas or its approved contractors.

Each building must have a separate service pipe. In multi-storey flats or maisonettes a separate service pipe should be provided for each lift of dwellings within the block. For a single property, the service pipe should enter the premises as close as possible to the meter position. In multi-story buildings, each service pipe should enter as close as possible to a position where it can readily be extended vertically through the building, branches or tees being fitted on each floor to serve individual flats. Branches to separate meter positions must be kept to a minimum length.

*Meters*

Primary meters should be as close as possible to the point of entry of the service pipe and allow easy access without undue disturbance to the consumer. For the meter to function accurately it should be protected from:

- *(a)* the risk of physical damage
- *(b)* continuous dampness or wet conditions
- *(c)* excessive temperature changes
- *(d)* contact with flames or electrical sparks, etc.

Meters must not be installed on or under stairways or in any other part of a building with two or more floors above the ground floor, where the stairway or the other part of the building provides the only means of escape in cases of fire. For buildings with one floor above the ground floor, meters must not be installed on or under a stairway or other part of the building which constitute the only means of escape in case of fire unless:

- *(a)* the meter is of fire resistant construction
- *(b)* the meter is housed in a compartment with automatic self-closing doors and which is of fire resisting construction
- *(c)* the pipe immediately upstream of the meter incorporates a device which cuts off the flow of gas automatically if the temperature of the device exceeds 95°C.

Electricity and gas meters may be housed together provided the relevant regulations are observed and they are separated by a fire resistant partition if they are sited within 150 mm of each other. Gas meters or installation pipes should not normally be nearer than 50 mm to an electrical conduit or other conductor unless insulated.

The requirements for secondary meters are the same as those for primary meters.

Meters housed in a meter box attached to or built into the external face of the outside wall of a building must be so constructed and installed that any gas escaping within the box cannot enter the building or any cavity in the wall, but disperses to the outside air.

Ideally, domestic meters, both credit and prepayment, should be mounted 1.35 m from floor level. Larger meters are usually at floor level.

For large plants having very high consumption it may be difficult to accommodate a large direct measurement meter. Shunt (inferential) meters, which require less space, are available for these applications.

It is a mandatory requirement that a permanent emergency notice be displayed prominently near the meter. Where secondary meters are installed, the emergency notice must indicate their number and location. Reference should be made to the relevant British Standards, particularly BS 4161[3] and BS 6400[4].

*Emergency Controls*

A new gas supply will not be provided for use within any building unless an emergency control is provided, with adequate access, to comply with the Gas Safety Regulations.

*Internal Installation Pipes*

Internal installation pipes are those between the point consumer control and the points of connection to the appliances. The Gas Safety Regulations* require that only competent persons shall undertake the installation of gas pipes and appliances. In particular, the following points should be noted:

- *(a)* Pipes in walls, floors or standings of solid construction must be installed so as to be protected against failure caused by movement.
- *(b)* Pipes must not be installed inside cavity walls.
- *(c)* Pipes must not be installed under the foundations of a building nor in the ground under the base of a wall or footings.
- *(d)* Pipes must not be installed in an unventilated shaft, duct or void.
- *(e)* Pipes must take the shortest practicable route through a solid structure and be enclosed in a gas-tight sleeve.

All gas pipes must be electrically bonded. During work on a gas pipe or fitting, a suitable bond must be provided to maintain electrical continuity. Refer to the Gas Safety Regulations throughout.

*Ventilation*

The Gas Safety Regulations require that all gas appliances have adequate means of ventilation to ensure proper combustion within the appliance and the safety of the persons using it, see *Combustion and Ventilation Air*.

*Gas Pressure Boosters*

In large commercial and industrial plants where burner equipment is of the fan assisted or pressure type, higher pressures are required. In such cases, it may be necessary to fit a booster to the gas supply to increase

* See footnote on page B13-12

the pressure. Where this is done, suitable controls must be fitted to ensure that the gas pressure in the surrounding mains cannot be reduced below a specified minimum.

Schedule 4 of the Gas Act 1972 enables British Gas to require customers using a gas booster or compressor to fit and maintain a device to prevent pressure fluctuations in the supply mains. British Gas must be consulted when considering gas pressure boosting and the Act requires that the Corporation are notified if a booster or compressor is fitted.

*Sizing of Boosters*

If the purpose of the booster is to enable smaller pipes to be used, the booster is sized in the usual way to provide the required volume of gas against the total pipe resistance. The permissible pressure drop per equivalent metre of pipe run governs the pressure supplied by the booster. The booster must be large enough to supply the required quantity of gas when the supply pressure is at its lowest. Therefore a pressure governor will normally be required.

If the purpose of the booster is to increase the pressure available at the burner nozzle, the booster should be sized to provide the required quantity of gas at a pressure at the governor greater than that required by the burner. For example, if the excess gas pressure is 250 Pa above that required by the burner, and the governer requires a pressure of 3.75 kPa, the booster would increase the pressure in the supply to 4.0 kPa plus that required to overcome the resistance from the booster to the governor inlet.

*Pressure Boosting Equipment*

Fig. B13.4 shows a basic booster system. If a positive displacement compressor is used, protection is required against the following:

*(a)* damage to the compressor if all the outlets are closed

*(b)* damage to the meter from suction

*(c)* disturbance to the gas supply by suction, leading to unsafe conditions

*(d)* damage to a low pressure meter by excessive pressure.

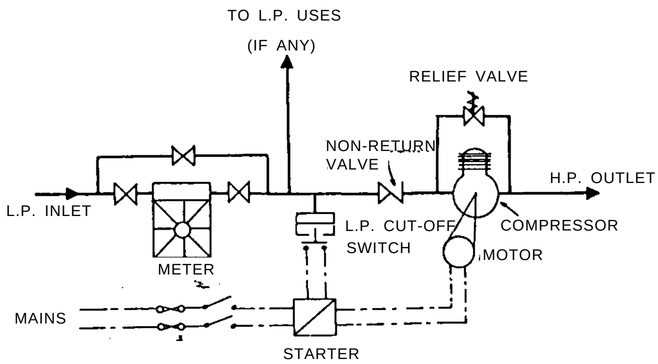



**Fig. B13.4.** Basic gas booster system.

In the more elaborate arrangement shown in Fig. B13.5. protection from the following is required in addition to the above conditions:

*(a)* damage to a low pressure receiver by excess pressure

*(b)* damage to a high pressure receiver by excess pressure.

The system is controlled by:

*(a)* A low pressure cut-off switch on the gas supply to the booster, to ensure that the booster is shut off if the mains pressure falls below a stipulated value. The low pressure switch should have manual reset and be wired to switch off the booster.

*(b)* A pressure switch on the pressure side of the booster to ensure adequate pressure is available before the automatic gas burner control opens. This is desirable but not mandatory.

*(c)* A non-return valve in the gas line to ensure that pressure surges cannot be transmitted into the gas supply pipework. This is a mandatory. requirement and the valve must be of a type acceptable to British Gas with a pressure lift greater than 7 kPA.

*(d)* A by-pass around the compressor which includes a pressure relief valve.

As an alternative to the low pressure cut-off switch *(a)*, an integral relief valve and a lock-shut cut-off valve may be used.

On starting the compressor, temporary pressure starvation on low pressure burner installations may cause the low pressure cut-off switch to operate, thus interrupting the supply. Therefore, where a compressor takes most or all of the metered gas supply, one of the following additional precautions may be required during starting:

*(a)* Temporarily override the normal outlet pressure control relief valve to ensure that the relief is wide open before starting.

*(b)* Install a second by-pass around the compressor incorporating a manual valve which is open before starting and closes slowly when running.

*(c)* Install a second by-pass incorporating a quick acting governor sensing the compressor inlet pressure and set to open if this falls below its normal working value.

*(d)* Install an anti-pulsation valve.

*(e)* Install a low pressure container, fitted with a drain cock.

*Note:* by-passes in *(b)* and *(c)* must have the inlet connection between the compressor inlet and the non-return valve.

The use of an anti-pulsation valve is the simplest solution provided that the resulting pressure loss is tolerable. If not, the low pressure container should be considered. Where either of these is used it must be inserted immediately downstream from the low pressure cut-off switch.

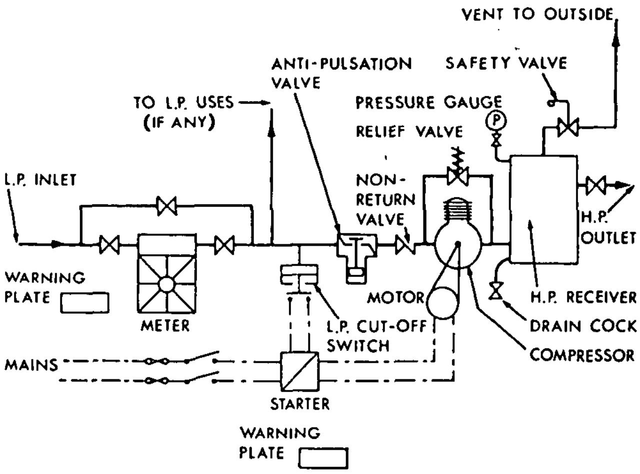



**Fig. B13.5.** Complete booster system.

It is sometimes desirable to smooth out fluctuations in the outlet pressure and/or provide for sudden changes in demand. These problems are most common where the compressor is close to the point of use and, in consequence, has a small pipeline capacity. Both may be overcome by installing a high pressure container, fitted with a pressure gauge, safety valve and drain cock. The system shown in Fig. B13.5 includes both an anti-pulsation valve and a high pressure receiver.

In addition to the protective devices discussed above, warning plates should be placed at the meter inlet valve and compressor starter. Where there is a varying demand, or an oversized compressor or fan, the possibility of overheating should be considered and provision made for compressor idling, or a recycle cooler should be included.

## Liquefied Petroleum Gas (LPG)

The two principal types of LPG are butane and propane. Their properties are described in Section C5. The specifications for commercial propane and butane are given in BS 4250[5]. LPG has a higher calorific value and requires higher operating pressures than natural gas and it is chemically reactive with some materials.

Since LPG is more dense than air, any leakage will accumulate at low level and may percolate into basements, drains, ducts etc, where it will remain until deliberately dispersed. Such an accumulation represents a serious fire and explosion hazard because of the dilute flammability range in air (2 to 11% gas). Internal installations must be totally gas tight and there is *no* acceptable level of leakage for LPG.

### *Delivery of LPG*

LPG is delivered by road or rail tanker. The flexible delivery hoses have special couplings which connect directly to the storage vessel or off-loading facility.

### *Storage of LPG*

The installation of static bulk storage systems is covered by a code of practice[6] issued by the LPG Industry Technical Association.

An industrial LPG installation usually consists of one or two tanks of at least one tonne capacity mounted horizontally on concrete foundations. Where warranted by the size of installation; two small tanks are preferred to a single large tank so that inspection and maintenance may be carried out on one while the other remains in service.

Bulk tanks must not be sited too close to adjoining buildings and there must be adequate space between tanks in multi-tank installations, see Table B13.2.

Tanks must be sited to be accessible to the delivery tanker for topping-up and it is a requirement that the tanker driver can stand at the tank whilst keeping the vehicle in view during filling. The normal effective length of filling hoses is 30 m, but the distance between the tanker and tank should be kept to a minimum.

Tanks are sited clear of buildings to minimise the risk of overheating should the building catch fire. Heating the contents raises the pressure inside the vessel and although the relief valves will vent excess pressure, it is highly undesirable that they should be called upon to operate.

**Table B13.2.** Installation distances for LPG storage tanks.

| Tank capacity ($m^3$) | Minimum distance/(m) | |
|---|---|---|
| | From buildings | Between vessels |
| < 0.45 | 0* | 0.6 |
| 0.45 – 2.25 | 3 | 0.9 |
| 2.25 – 9.0 | 7.5 | 0.9 |
| > 9.0 | 15.0 | 1.5 |

* A tank of capacity 0.45 $m^3$ can be sited close to a building, allowing space for maintenance, with a minimum distance of 2.5 m from the tank filling valve to any opening in the building.

Conventional bund walls, common with oil fuel storage, should not be constructed around LPG storage tanks. In the event of a leak such an enclosure could cause a serious hazard by trapping vapour which would then have no means of escape. Equally, tanks should not be installed under trees or under any enclosure.

Bulk storage capacity should be sufficient for a minimum of six weeks' supply at maximum take-off rates. This will provide adequate reserves to meet bad weather conditions.

The pressure within storage vessels is about 690 kPa at 15°C. The pressure range under normal conditions in the UK is from 200 to about 900 kPa.

There are two methods of supply to the user, the choice being dictated by the application and the take-off required.

(*a*) *Portable Containers*
These are available in various sizes with capacities ranging from 45 to 550 kg to suit

applications with differing consumption rates; they may readily be transported to site as required. Alternatively they can be manifolded together in multi-container assemblies as a fixed outdoor installation to be used in conjunction with an automatic changeover device so that continuous supply is assured, see Fig, B13.6.

(*b*) *Bulk Storage*
Bulk installations consist of one or more static storage vessels, usually cylindrical in form. Common capacities range between 1 and 60 tonnes. However, for very large take-offs cylindrical tanks up to 100 tonne capacity are installed.

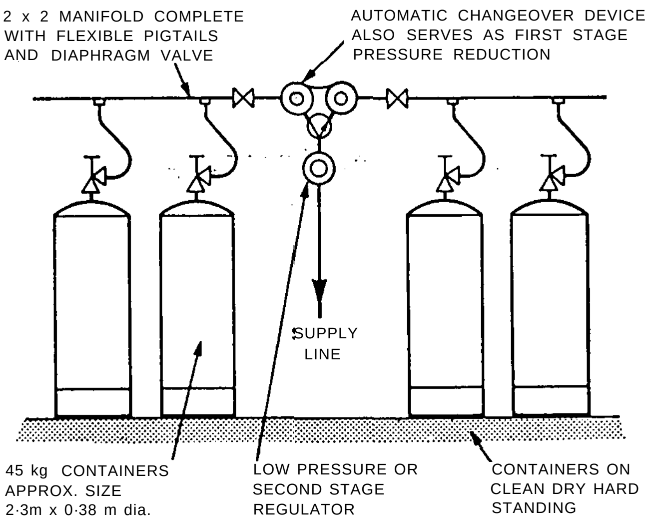



**Fig. B13.6.** Arrangement of LPG manifold.

Portable containers are made in accordance with specifications agreed by the Explosives Department of the Home Office. They are normally filled by the producing refineries and/or distributors. It must be stressed that LPG containers and bulk storage tanks are never completely filled; the quantity of gas is closely controlled according to a filling ratio laid down in BS 5355[7]. This is defined as the ratio of the maximum mass of LPG permitted in the vessel to the mass of water required to fill it at 15°C.

The filling ratio varies slightly according to the specific gravity of the particular LPG and is adjusted according to the size of the vessel. For example, in a bulk tank the ratio is 44% for propane and 52% for butane. For portable containers the ratios are 42% and 50% respectively. On this basis a tank of 9 $m^3$ water capacity holds about 4 tonnes of LPG.

Storage tanks for LPG are manufactured in accordance with BS 5500[8]. Propane vessels of over 0.9 $m^3$ constructed to these specifications are invariably to service duty requirements, having a maximum permitted working pressure of 1.45 MPa, but for butane medium duty vessels are made for a maximum working gauge pressure of 500 kPa as this is considered adequate. For storage vessels of 0.9 $m^3$ and under, these gauge pressures are increased to 1.75 MPa for propane and 600 kPa for butane.

*Handling*

LPG can be handled safely provided that good engineering standards are adopted and the same degree of caution is exercised as with other gaseous fuels. However, it is important that users are familiar with the properties of LPG so that the correct safety procedures are followed in the event of leaks. Particular care must be taken to avoid skin contact since freeze burns can result. All installations must comply with the Gas Safety Regulations as detailed above.

Propane tanks are equipped with a vapour outlet to high pressure reducing valves which are duplicated for reliability. These are generally set to discharge into the supply lines at gauge pressures of 70 to 170 kPa. Any further pressure reductions required are made at or near the point of consumption according to the characteristics of the burner. The high vapour pressure and calorific value of these gases permit the use of small diameter distribution pipework.

Unless very large quantities of propane are required, ambient temperatures are normally sufficiently high in the UK to cause vaporisation from the storage vessel. Under no circumstances should external heat be applied to assist the output. It may be necessary to design an installation for liquid take-off in conjunction with a vaporiser, the units being direct fired or heated by hot water or steam, as is common practice with butane.

*Installation Pipes*

The service pipe from the tank to the building is generally installed underground at a minimum depth of 0.5 m.

LPG installations inside buildings should comply with BS 5482[9], The Gas Safety Regulations, The Building Regulations 1985 and the requirements of the supplier.

It is essential to ensure an adequate supply of fresh air to LPG burning appliances and permanent ventilators of the correct size must be installed. All appliances must be connected to a flue to ensure removal of the products of combustion.

**Synthetic Natural Gas (SNG)**

Synthetic or manufactured natural gas varies in chemical composition. It is generally used for peak load levelling only. The properties of SNG must be such that when it is substituted for natural gas the effect on the performance of gas appliances is minimal. Therefore, SNG should give similar heat input, flame stability and combustion without readjusting the appliance.

**Gaseous Fuel Burners**

All equipment approved by British Gas is fitted with a burner unit suitable for the appliance and only where an appliance is converted to burn another fuel is there a need for consideration to be given to the type of burner necessary. The advice of either the local Gas Board or the boiler/burner manufacturer should be obtained before any conversion is carried out. There are several types of gas-fired burners suitable for air heaters, boilers, etc. and some of the more frequently used are discussed below. In the case of liquefied petroleum gas, the appropriate fuel supplier should be contacted for advice.

*Non-aerated Jet*

Non-aerated jets are used predominantly to burn town gas. Gas is supplied to the burner at low pressure and all the air for combustion is obtained from the air surrounding the flame. The flame is luminous. Non-aerated burners are used on all types of water heaters (e.g. storage heaters, sink and multipoint instantaneous heaters), central heating boilers, radiant and convector fires, air heaters, etc.

*Aerated Jet*

In low pressure aerated, or bunsen type, burners, approximately half the air required for combustion is entrained by gas issuing from an injector and it mixes with the gas before reaching the burner head. The remaining air for combustion is obtained from the air surrounding the flame. Aerated burners are in general use on town gas cooker hotplates, grills, boiler burners and refrigerators. At present, with natural gas and liquefied petroleum gas, aerated burners are used on all types of gas appliance.

*Atmospheric Burners*

This type of burner uses the momentum of the gas at normal meter outlet pressure to mix primary air and gas. The mixture is burnt at the burner port, with the addition of secondary air. These burners are only suitable for installations where ample and stable flue draughts are available, where there is no down flow of combustion products from the flue or the possibility of trapping flue gases within the boiler. A draught diverter is essential.

*Fan-assisted Burners*

This type of burner is designed so that all the air required for combustion is supplied by a fan. The fan pressure is intended to overcome any resistance through the burner due to air flow. It can also be sized to overcome boiler resistance where the burners are used on pressurised combustion chambers. These burners have the advantage of close combustion air control, i.e. quantity and the point of supply. It is also possible to exercise control over flame shape and dimensions. These factors can often permit the use of smaller, more compact combustion chambers.

*Pre-mix Burners*

Pre-mix burners, which are usually equipped with a suitable control system and air blower, are designed to produce a combustible mixture of gas and air that can be burnt at the burner port without the addition of any secondary air. This type of burner can reduce the combustion space still further. Pre-mix burners are widely used in industrial furnace applications.

*Multi-gas Burners*

Burners of each of the foregoing types are available for operation on town gas, natural or liquified petroleum gases and can be provided in a form which is easily convertible from one gas to another. The basic changes are injector orifice size and operating pressure.

*Gas/Oil Burners*

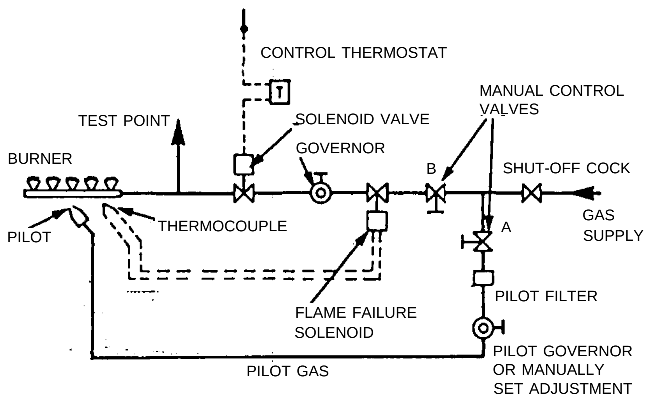
Normally gas/oil burners are supplied as part of a packaged, forced, or induced draught, boiler unit, but they can also be applied to existing boilers. Dual fuel burners of the gas/oil type can be easy to change over with minimum adjustment. Dual fuel burners can be designed for fully automatic change over if required.

*Controls*

All boilers, air heaters and other similar appliances should be fitted with a flame failure device. This shuts off the gas supply to the burner in the event of a flame failure. The control sequences for gas fired appliances vary due to different systems employed by various manufacturers, but they can be divided into two main types, those used with a natural draught type burner and the controls required for automatic gas burners with forced or induced draught.

Controls for natural draught applicances are manufactured in multi-functional or integrated units and are normally fitted on appliances up to 45 kW. B13.7 shows the valving arrangement.

On natural draught burners over 45 kW, output safety control layout varies with each boiler manufacturer but generally is similar to that shown in Fig. B13.8. The manufacturers should be consulted for details of their controls and setting information.




VALVES A & B MAY BE COMBINED IN ONE UNIT.

**Fig. B13.7** Gas burner control (<45 kW).

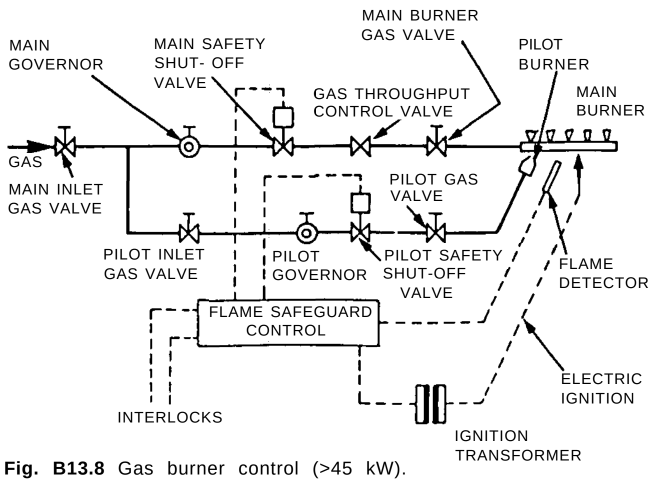



**Fig. B13.8** Gas burner control (>45 kW).

To provide for a minimum basic control scheme to enable a boiler to work efficiently and safely, the following controls, or components performing a similar duty, are essential:

(a) *Electrically Energised Components*

(1) A dual purpose boiler thermostat, embodying two separate thermostats which fulfil the functions of a separate control and a limitstat with manual reset.

(2) A valve with slow opening and rapid closing characteristics for the main burner supply.

(3) A safety shut-off valve in main and pilot supplies.

(4) A switch operated by a fusible or electric thermal link.

(5) A flame failure control which preferably should be of the flame rectification type.

(b) *Gas Controls*

(1) A manually operated main shut-off valve.

(2) A manually operated valve for the main burner supply.

(3) A constant pressure governor for the main burner supply.

(4) A manually operated shut-off valve for the pilot burner supply.

(5) A constant pressure governor for the pilot burner supply.

Conditions arise where it is necessary to install a control system to conform with the special requirements of a public or local authority which may require all boilers, where installed in public places of entertainment, to be fitted with components which will ensure a rapid and complete cessation of the gas supply in the event of flame failure. The time required varies from 15 to 2 seconds according to the output of the boiler. These conditions can be achieved by installing an electronic scheme which may comprise one of the following:

(a) Press button start.

(b) Electric (spark) ignition.

(c) Flame rectification, photo-cell or other quick response flame failure system.

(d) Ignition by two aerated pilots which must be located so as to ignite readily the main gas burner and to project across the flame sensing probe of the flame failure unit.

*Automatic Gas Burners with Forced and Induced Draught*

For details of control requirements for these burners see ‘Standards for Automatic Gas Burners, Forced and Induced Draught’.[10] The Standards relate primarily to the safety aspect of automatic control. They do not provide a complete specification for automatic burners and their ancillary equipment.

The Standard applies to:

(a) Automatic gas burners employing forced or induced draught i.e. forced or induced draught burners in which, when starting, gas to the main burner is turned on automatically.

(b) Burners as in *(a)* of both packaged and non-packaged types.

(c) Burners as in *(a)* of all thermal and input ratings.

(d) Burners as in *(a)* for all fuel gases.

*Conversion of Equipment*

Where equipment is converted from solid, liquid or gaseous fuel to an alternative gaseous fuel, technical advice is available from British Gas, LPG suppliers, burner and/or equipment manufacturers, and competent authorities should be consulted to ensure that the correct type of burner and control equipment is selected.

**Condensing Boilers**

Conventional boiler design aims to avoid gas-side corrosion by condensation. This limits boiler efficiency, based on gross calorific value, to about 85%. The main combustion products from clean gases such as natural gas are carbon dioxide and water vapour. Provided the dew-point temperature is not reached, corrosion will not occur.

The principle of the condensing boiler is that the flue gases pass through a secondary heat exchanger (or recuperator) which ensures that the flue gas temperature is below the dew-point. Latent heat is also removed and this increases the operation efficiency to 93% which compares with an average seasonal efficiency of 75% for conventional boilers.

The condensate has a pH of about 4 but with appropriate heat exchanger materials corrosion is avoided. However, the choice of materials for the drainage system must be considered as it is essential to ensure that they are not affected by the acidity of the condensate, see Sections B7 and B8. By extracting heat from the flue gases, their temperature is too low to rely on natural draughts and mechanical ventilation is necessary to ensure that the products of combustion are removed.

The efficiency of condensing boilers depends on the return temperature of the heating water. If this is as low as 25°C, the leaving flue gas temperature will be about 4°C higher giving an efficiency of about 97%, see Fig. B13.9. However, higher return water temperatures are more likely which reduces the efficiency in operation. The lower return temperatures mean that larger heat emitters are required but full use can be made of floor warming systems.

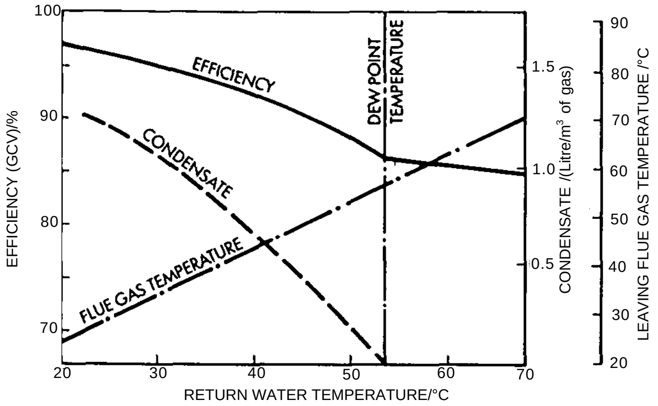



**Fig. B13.9.** Relationship between return water temperature and condensing boiler efficiency.

## LIQUID FUELS

The major petroleum derived fuels used for domestic, commercial and industrial applications are distillate fuels (classes C and D), residual fuels (classes E, F and G) and a special grade (class H) for very large applications, e.g. power plants, steelworks etc. These grades are all supplied according to BS 2869[11].

Coal tar fuels (CTF), derived from coal distillation, comprise six grades; CTF 50, 100, 200, 250, 300 and 400 according to BS 1469[12]. These fuels are used mainly in industrial applications.

Systems for the storage and handling of liquid fuels should be installed to comply with the recommendations given in BS 799[13] and BS 5410[14] and also the recommendations of the burner manufacturer and fuel supplier.

### Liquid Fuel Storage Equipment

Storage tanks for liquid fuels are mainly of two types, vertical cylindrical and horizontal cylindrical, made from welded mild steel. Tanks for domestic purposes are generally of a horizontal rectangular pattern of limited capacity and must be installed to conform with BS 5410[14], local bye-laws and The Building Regulations.

Larger commercial/industrial storage tanks must be installed on a concrete base and within a retaining (bund) wall capable of holding the capacity of the tank in the event of leakage. The catchpit walls must be oil-proof and capable of withstanding the considerable pressure of liquid in the event of an overflow. A permanent drain is not permitted in the catchpit but facilities should be provided to remove accumulated rainwater.

Often more than one storage tank will be required. Part 5 of BS 799[13] recommends that the minimum net storage capacity should be calculated by taking three weeks supply at the maximum rate of consumption, or the equivalent of a normal delivery plus two weeks supply at the maximum rate of consumption, whichever is the greater.

*Tanks of capcity not exceeding 3.5 $m^3$*

For installations up to 12.5 kW, a rectangular mild steel tank of 1.1 $m^3$ capacity with 'tented top' to shed water is satisfactory. The tank should be capable of accepting deliveries of a minimum of 2.2 $m^3$ to take advantage of the cost benefits of bulk delivery.

These tanks should meet the particular requirements of BS 799[13] according to type, as follows.

Type III: Rectangular tanks.

Type II: Horizontal cylindrical tanks.

*Tanks of Capacity 3.5 to 54.5 $m^3$*

These tanks should conform to BS 799[13] according to type, as follows.

Type J: Rectangular tanks for a design pressure equivalent to a head of water not exceeding 0.5 m (i.e. 4.9 kPa).

Type K: As above but for a design pressure equivalent to a head of water not exceeding 0.75 m (i.e. 73.6 kPa).

Type A: Horizontal cylindrical tanks for a design pressure equivalant to a head of water not exceeding 3.5 m (i.e. 34.3 kPa).

Type G: Vertical cylindrical tanks with flat bottom plates for a design pressure equivalent to a head of water not exceeding 0.5 m (i.e. 4.9 kPa).

*Note:* In the above definitions the head of water is measured from the top of the tank.

Tanks can be welded on site where required. The BS 799[13] type number/letter must be stated. The pressure condition on the tank must be considered before the type is selected. A design pressure of 4.9 kPa (as Types J, G above) will suffice for normal outdoor tanks above ground but basement installations may require higher design pressures (Types K and A) unless special pressure release measures are taken. It is important to consider the pressure conditions which can apply to the tank in the case of over-filling where oil can rise up the vent pipe and over-pressurise the tank.

The siting of storage tanks must comply with the requirements of BS 799[13] and BS 5410[14], together with the Building Regulations and any local authority requirements. Large vertical cylindrical tanks of capacities over 54.5 $m^3$ are site erected and should be designed to BS 2654[15]. Underground reinforced concrete storage tanks with oil resistant or tiled linings can be used.

Buried steel tanks are usually protected by bitumenised paint and the excavation backfilled with sharp sand. Storage tanks should not be buried directly in contact with the soil since this may result in corrosion. They should be fixed to a concrete base of adequate strength to prevent the tank being forced from its mounting by ground water pressure. Tanks can also be encased in concrete in which cast the outer tank surface is wire brushed and painted with a corrosion inhibiting primer.

**Table B13.3** Fill pipe dimensions.

| Class of oil | Total length of fill pipe (m) | Minimum bore of fill pipe (mm) | Hose coupling BSPT (mm) |
|---|---|---|---|
| C and D<br>CTF 50 | Up to 12 | 32 for tanks up to and including 1·35 $m^2$<br>50 for tanks over 1·35 $m^3$ | 50<br>50 |
| | Over 12 | 50 for tanks up to and including 1·35 $m^3$<br>65 for tanks over 1·35 $m^3$ | 50<br>65 |
| E | Up to 12 | 65 | 80 |
| | Over 12 | 80 | 80 |
| F and G<br>CTF 100<br>CTF 200<br>CTF 250 | | 80 | 80 |

*Fill Pipe*

Table B13.3 gives fill pipe dimensions for different classes of oil. For fill pipes over 10m long, using class E, F or G oils, the pipe should be lagged and provided with trace heating to cater for high viscosity oils. Where the fill point is below the level of the fill line, a screw down fill-way oil valve should be installed and the inlet protected with a non-ferrous screw-on cap secured by a chain. A drain cock and catchpit should be provided under the inlet. The oil fill point should be within 30 m of the delivery vehicle position.

*Vent Pipe*

A separate vent pipe, of internal diameter at least equal to that of the fill pipe, must be fitted to each tank. It should rise continuously to a height above the tank not exceeding the design pressure condition of the tank and terminate in open air where odours during filling will not cause inconvenience. The open end of the vent should be fitted with a downward facing bend and protected by open wire mesh. Where the vent pipe rises to a considerable height, excessive internal pressure on the tank may result due to the pressure head of oil should overfilling occur. An unloading device must be installed to limit the head pressure on the tank and discharge within the catchpit area.

*Isolating Valves*

Isolating valves should be fitted at the tank outlet, the outlet tapping being positioned a minimum of 76 mm above the tank base to prevent water and sediment entering the delivery pipe.

*Drain Valve*

A screw-down gate valve or glanded cock must be fitted at the lowest part of the underside of the tank to remove water and sediment.

*Contents Indicator*

An oil indicator of one of the following types should be installed depending upon the size and function of the tank:

(*a*) Glass/plastic sight tube with suitable shut-off arrangement for domestic installations.

(*b*) Dipstick.

(*c*) Float gauge.

(*d*) Hydrostatic gauge.

An overfill alarm is advisable to warn the operator when the tank is full and thus prevent oil spillage and/or damage to the tank.

*Ullage*

There should always be a small ullage, or air space, above the oil level when the contents gauge registers full. This prevents discharge of oil from the vent pipe due to any frothing and surging of the liquid during delivery. The ullage should provide not less than 100 mm between the oil surface and the top of the tank or the equivalent of 5% of the total contents, whichever is the greater.

*Tank Heating*

With oils of classes E, F, G and H, the tank must be fitted with heaters to provide the storage temperatures indicated in Table B13.4. These consist of steam coils, hot water coils or electric immersion heaters. It is usual to maintain tanks at the storage temperature recommended for the particular grade of oil and then raise the temperature of the oil to be used by means of a separate outflow heater. These are sized for maximum oil usage, the temperature being raised from storage temperature to handling temperature. A combination unit incorporating electric elements must be fitted in one tank, for start-up.

Care must be taken to ensure that the heater elements and the control thermostat are positioned below the oil draw-off point. This ensures that they are always covered with oil.

**Table B13.4.** Storage temperatures for fuel oils.

| Class | Minimum temperature/°C | |
|---|---|---|
| | Storage | Outflow |
| E | 10 | 10 |
| F | 25 | 30 |
| G | 40 | 50 |
| H | 45 | 55 |

**Central Oil Storage and Distribution**

Central oil storage is a method of storing domestic fuel oil in bulk and distributing supplies to individual dwellings by way of an underground pipework distribution network to a small meter at each dwelling. The site is fed from one or more oil storage tanks installed at some convenient point and of such a capacity as to afford adequate supply for all of the dwellings on the site for several weeks, together with a margin for possible weather factors that could delay the otherwise scheduled delivery.

Where the storage is above ground, distribution can usually be by gravity, provided the relative levels of the tank outlet and the properties of the oil to be supplied are favourable. This represents the simplest method and is that most suitable where the individual heating systems are equipped with appliances employing non-automatic vaporising burners and where it is desirable that neither the burners nor the fuel supply are dependent upon electrical supply. Where it is necessary to place the main storage tanks below ground, oil lifting pipes are required to serve a small capacity header tank which would then provide the necessary static head for gravity distribution through the site.

Other distribution systems are of the pump pressure one pipe type, or pumped ring main, from above ground or below ground main storage tank(s). All underground distribution pipes must be adequately protected against corrosion and, where pipes cross roadways, be sleeved. Attention must be paid to the levels of the pipe, providing adequate air-venting and draining facilities where necessary.

The oil supply to each dwelling is metered, a meter box usually built into the outside of the wall of the dwelling in order that meters can be read at any time by the operating fuel suppliers, and a shut-off valve incorporated in the meter box is accessible to the fire authorities in the event of fire. The meter box front is provided with a glass panel to permit emergency use of the shut-off valve.

Since most domestic burner oil level control units are limited to a maximum gauge pressure of 20 to 30 kPa, pressure reducing valves are fitted in the meter boxes and a safety cut-off valve is provided to protect the appliance in the event of failure of the pressure reducing valve. These safety cut-off valves are usually bellows operated from a detecting tube connected on the outlet pipe from the pressure reducing valve, and will not permit oil to flow again until reset by the service engineer.

The entire layout and construction of a central oil storage installation must be carefully planned and conform with the relevant local authority requirements and bye-laws. the Pipe Lines Act 1962. and meet with the approval of the relevant fire prevention officers.

**Oil Supply Systems**

Pipework for both petroleum and coal tar fuels should be of mild steel (non-galvanised) with malleable cast iron or wrought iron fittings. Materials such as brass and alloys of copper and zinc should be avoided. Natural and most synthetic rubbers are unsuitable for seals and the advice of specialist manufacturers should be sought before using these materials.

Skin contact or inhalation of vapours should be avoided at all times. Persistent contact with fuel oils may give rise to dermatitis.

For all grades of fuel, the pump capacity should be such as to maintain oil circulation at three times the rate of consumption at the burner. This avoids pressure fluctuations at the burner due to changes in the consumption rate.

*Class C and D Oils*

For domestic installations using class C or D fuels, a single pipe delivery system is suitable. Annealed copper tube may be used with class C fuel and, to some extent, with class D fuel. However in some cases, copper pick-up can contaminate class D fuels leading to fuel degradation.

Vaporising burners require a single pipe system with a pressure head maintained in the tank at all times to provide a positive pressure at the metering valve. Pressure jet atomising burners may use a single pipe system provided that a positive oil pressure of 2.5 kPa is available at the burner pump suction at all times. Under extreme conditions, some class D heating oils may require heating to ensure flow of oil. Both tank heating and trace heating (with pipes suitably lagged) may be required to maintain flow temperature between 0 and 5°C.

For larger installations a ring main system may be appropriate, see Fig. B13.10, especially for strategic supply applications. The recommendations of BS 6380[16] on the cold weather use of distillate oils should be followed.

*Class E Oil*

This light residual fuel oil should be supplied from a simple heated storage tank and circulating ring main system as shown in Fig. B13.11. The oil handling temperature should be maintained above 10°C. For this system it is assumed that pre-heating of the fuel to atomising temperature takes place by means of a heater at the burner.

*Class F and G Oils*

These classes require a hot oil ring main system as shown in Fig. B13.12. This involves heating the oil in the tank and using an outflow heater to raise the oil to pumping temperature. Circulating pumps, filters valves and pipework must all be trace heated and lagged. The oil may be raised to atomising temperature by means of a heater at the burner but is generally heated close to the atomising temperature by a line heater or pre-heater tank as illustrated. The final atomising temperature is achieved by a trim heater on the burner which would be a pressure jet or rotary cup type for this class of fuel. On the largest installations. where high pressure steam is available, twin Fluid atomisers are more common.

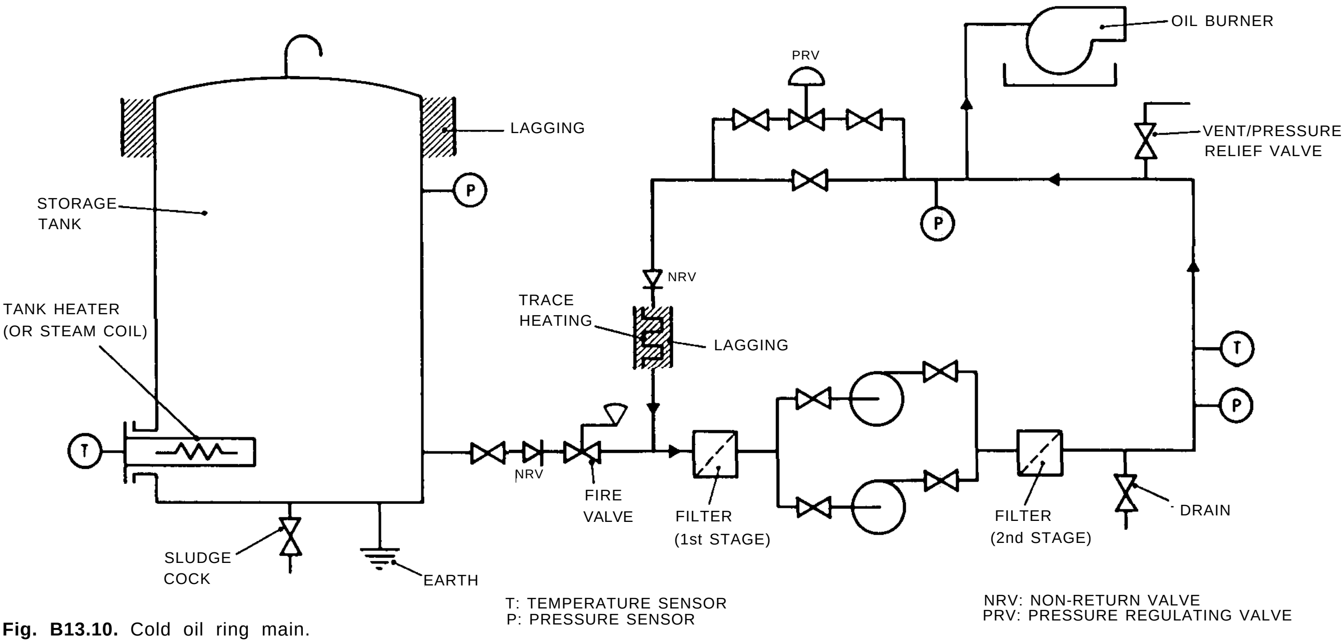



**Fig. B13.10.** Cold oil ring main.

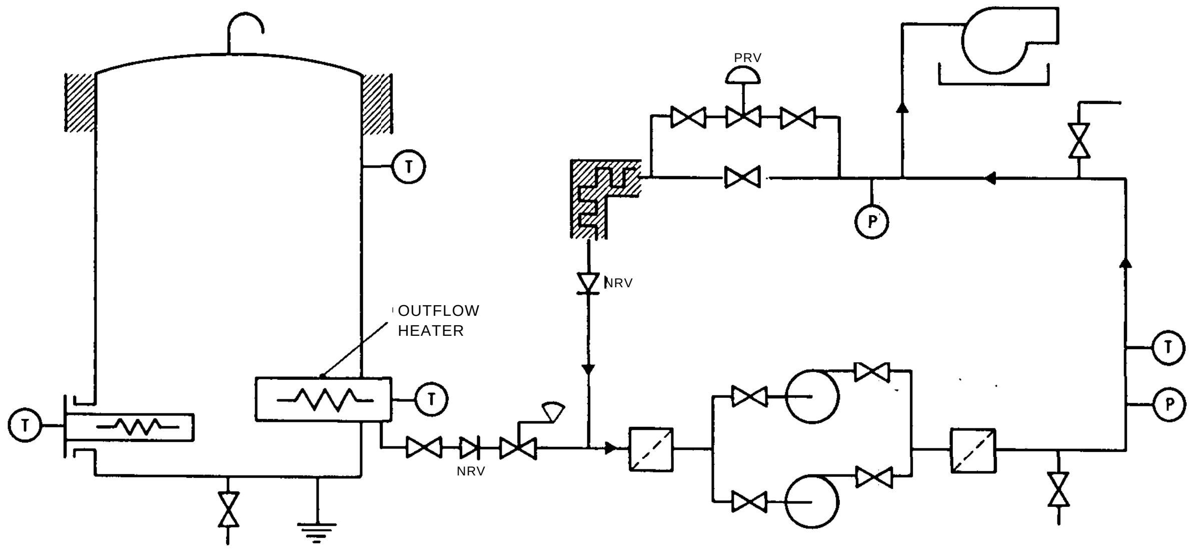



**Fig. B13.11.** Warm oil ring main.

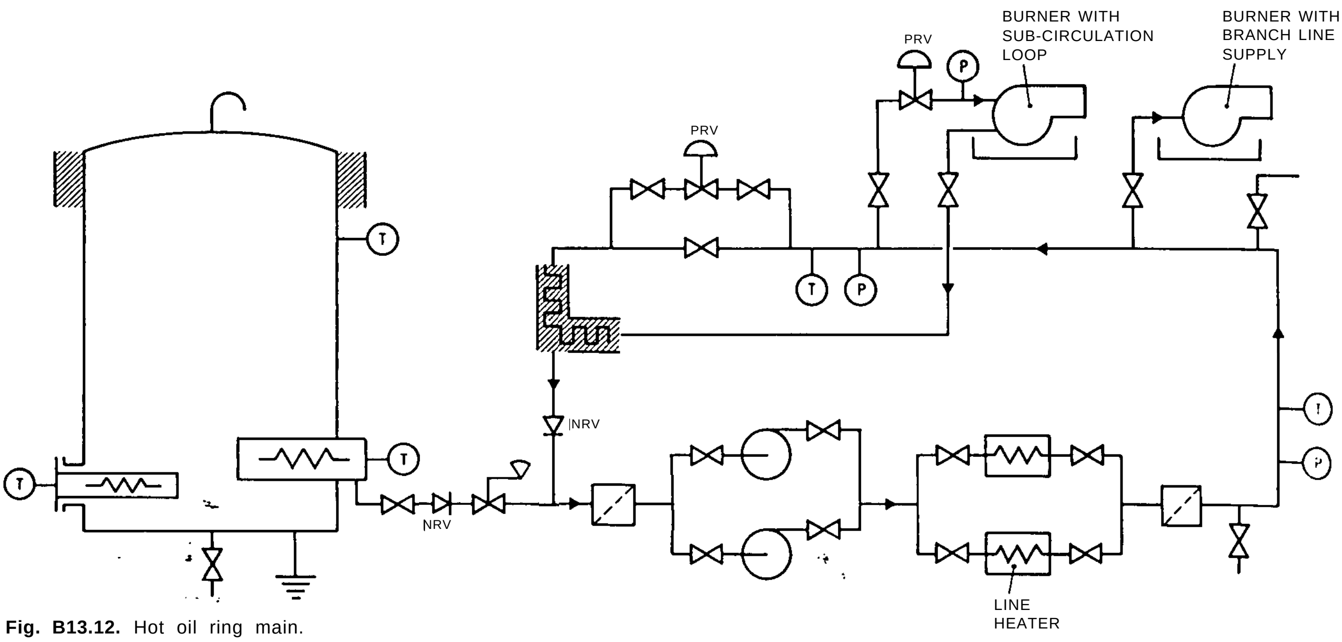



**Fig. B13.12.** Hot oil ring main.

Oil fuel which has been raised to higher atomising temperatures should return to the suction side of the pump, but must *not* be returned to the storage tank. Pre-heating of oils should be in accordance with Part 4 of BS 799[13].

Once raised to handling/atomising temperature, hot oil should be kept circulating. Static hot oil may give rise to accumulation of deposits in the heaters. When not in use, either the pipework system should be drained or oil circulated at the lowest handling temperature.

Where oil lines are trace heated or oil heaters are used, a pressure relief valve must be fitted to prevent build-up of pressure in the heated sections if isolation valves are closed. The system for any particular application should be approved by both the burner manufacturer and the fuel supplier.

*Coal Tar Fuels (CTF)*

The lightest grade, CTF 50, may be supplied using a system similar to that for class D petroleum, i.e., single pipe. However, all pipework and components in contact with CTF must be of ferrous metal. Appropriate grades of neoprene are satisfactory for seals but specialist advice should be sought from the manufacturers of pumps, valves etc.

CTF 100 requires a higher handling temperature than CTF 200, see Table B13.4, due to the presence of naphthalene which crystallises out below 35°C. This can be achieved using trace heating and lagging by means of electricity or steam (not superheated steam), as for oils of grades E, F and G. Pre-heating of CTF 100 should be limited to a heat release rate of 1.6 $W/cm^2$ of element sheath surface, in accordance with BS 799[13], Part 4. Pre-heating of heavier grades, CTF 200 and above, should be limited to 1.0 $W/cm^2$. Experience indicates that coal tar fuels should only be used with mechanical and twin-fluid atomisers.

Filtration of both petroleum and coal tar fuels should also be in accordance with BS 799[13], Part 4. Both basket and self cleaning types may be used, usually in duplex form. A final fine mesh filter is usually incorporated just prior to the burner or on pressure jets and internal mixing twin-fluid atomisers using residual fuels. For the heavier grades of CTF, which contain finely divided solids in suspension, such filtration is not possible and external mixing twin-fluid atomisers must be used, in which the fuel leaves the atomising fluid clear of the nozzle tip.

**Liquid Fuel Burners (Small Installations)**

The following types of burner are available for the domestic range up to approximately 40 kW.

*Pot Burner*

In its simplest form this comprises an oil reservoir and a vaporising chamber in the base of which oil is heated and vaporised. The vapour is mixed with primary combustion air and burnt in the body of the pot. Primary air for combustion is induced progressively via a series of holes in the wall of the burner and in some types of burner, also through a similarly perforated hollow pillar set coaxially in the pot. The rate of burning in this type of burner depends on the amount of oil vaporised, which in turn depends on the temperature and size of the pot, the distance of the flame from the base of the pot and the amount and velocity of the air drawn into the burner.

*Short Drum Burner*

The short drum burner consists of one or more pairs of concentric perforated sheet metal drums placed above interconnecting grooves in a cast-iron burner case. These grooves are fitted with kindler wicks to initiate combustion. As with a pot burner, the oil is vaporised in the base of the burner and the vapour rises to a level in the annular space where sufficient air for complete combustion is induced. The annular space between pairs of drums is covered by a metal or refractory lid in order to direct all the air induced by chimney effect of the hot burner to enter the gas annuli and mix with the vaporised oil. These covers also provide some extra radiating surface.

*Multiple Stage Burner*

Multiple stage burners differ from pot burners in shape but not greatly in operating principles. The burner consists of an elongated metal trough the sides of which are perforated in a similar manner to the pot burner. The trough is divided into lateral superimposed chambers by means of suitably shaped baffles. Depending on the fuel flow rate to the burner, combustion takes place at any one of the various levels in the burner. This burner is probably slightly more efficient, cleaner in operation and more flexible than the conventional pot burner.

This burner, like the pot and short drum burners, is usually operated on a non-automatic principle. Ignition is manual and the burning rate is controlled by a manually operated metering valve. However, in some instances the metering valve is controlled remotely on a high/low basis by a thermostat operated by the boiler water temperature. Pilot flame or electrical ignition are not normally used.

*Fan Assisted Burner*

This burner is, in principle, similar to the natural draught pot burner except that air required for combustion is supplied from an electrically driven fan. Thus a typical fan assisted burner consists of a vaporising burner pot encased in a metal housing into which the fan blows air for combustion. Air from this box diffuses into the burner pot. The majority of these burners operate on the fully automatic principle i.e. on high flame/off. The oil is metered automatically, ignited electronically by hot wire or spark ignition and the fan operated only during the period in which the burner is operated.

Burner air supply is a critical feature of the natural draught burner. A suitable, adequate chimney to induce this air supply must therefore be provided for a boiler using this type of burner and automatic draught regulation is essential. Even in the fan assisted vaporiser burner the air supply is not entirely independent of chimney draught.

*Rotary Vaporising Burner*

Rotary vaporising burners, a further form of the fan assisted burner, are known by other names including 'wallflame'. An electrically driven centrifugal spinner distributes fuel uniformly on to a short stainless steel wall mounted on a circular refractory hearth. The small amount of fuel distributed per unit area of wall makes it possible for ignition to be achieved initially by high tension spark or hot wire glow plug. Combustion quickly becomes self-supporting as the wall is heated by radiation from the flame and the coarsely atomised spray of fuel from the spinner vaporises at the wall. Air, induced by a fan located on the spinner, is directed towards the wall, deflected upwards mixing with the vapour, and combustion takes place on or below a series of sheet metal stabilising grilles placed above the wall. In one form of this burner some of the combustion gases are recirculated in a toroidal manner by using a specially shaped fan. A further measure of flame stabilisation is achieved by this means.

This type of burner is invariably fully automatic in operation since initial ignition and both air and fuel supply can be controlled electrically by means of a boiler thermostat or other forms of electrical control.

*Constant Level and Fuel Metering Controls for Vaporising Burners*

A simple form of a constant level and fuel metering control for vaporising burners operates as follows:

Fuel from the main tank is supplied via a filter to the chamber of the control. The fuel inlet valve to the control, a needle valve, is balanced by a float and thus maintains a constant fuel level. This inlet valve needle can also be raised or lowered by a lever or trigger and thus serves to regulate fuel supply to the control. The fuel supply to the burner is regulated by a metering stem or valve, the outflow of which can either be controlled manually or automatically by thermostat, or preset to a known amount. A spring loaded safety trip device in the control operates to close the inlet valve should the fuel rise to an excess level in the chamber for any reason.

*Pressure Jet Burner*

Pressure jet burners are comparitively simple in construction and operation. Basically a pressure jet gun-type burner consists of a short tubular air director at one end of which, coaxially mounted, is a special nozzle. This nozzle is so designed that a high rotational velocity is imparted to the fuel during its passage through the nozzle and this energy of motion causes the fuel to break up into a cloud of fine droplets when it leaves the nozzle. At the other end of the tube a centrifugal air fan and a high-pressure fuel pump are mounted, both driven by the same electrical motor.

Oil fuel is pumped at relatively high gauge pressure, usually greater than 450 kPa, through the nozzle thus producing the spray. Low pressure air blown along the tube by the fan mixes with the oil droplets at the nozzle end of the tube and the mixture is ignited by a high tension spark from a pair of electrodes mounted near the nozzle. Combustion very quickly becomes self-supporting and the electrical ignition may be automatically switched off.

Variations of this basic type of burner contain so called 'combustion heads'. These consist of specially shaped air baffles and directors designed to promote better air/fuel mixing thus giving more efficient combustion and a more stable flame pattern. In the smaller ranges, this burner is used only on a fully automatic on/off basis. The operation of the burner can be controlled electrically by means of a boiler thermostat or external control and is also regulated by a flame failure device usually of the photoelectric type.

**Liquid Fuel Burners (Large Installations)**

For larger installations (generally above 30 kW) the oil burners employed are of the atomising type, which deliver fuel into the combustion chamber in a finely divided state.

*Mechanical Atomisers*

These are burners of the pressure jet type. The fuel is delivered at a gauge pressure between 450 and 1500 kPa through the nozzle of the burner which ejects the oil in a cloud of finely divided particles. A general description of pressure jet burners has been given previously.

These burners are the simplest in construction since no high pressure fan or compressor is used for atomisation. Pressure jet burners can be supplied to burn quantities of fuel ranging from 0.5 g/s to 1.5 kg/s. Since output depends on the pressure of oil at the burner nozzle, which can normally be varied between 450 and 2000 kPa, the oil output range on turndown in a simple pressure jet is limited to approximately 2:l. Modified pressure jet burners incorporating such features as spill jet nozzles permit an improvement in range of turndown up to 7:l.

The nozzle only converts the oil into a cloud of small particles, so that the design of the air director is all important to ensure complete mixing of combustion air with atomised fuel oil, and provide the flame shape required. Various types of head design are available to achieve the optimum conditions for a particular combustion chamber and heat exchange surface configuration.

The above class of burners probably constitutes the major usage in the normal heating applications, taking individual boiler units up to, say, 6 MW.

*Rotary Burners*

Fuel oil is distributed into an air stream from the lip of a rapidly rotating cup. The centrifugal force of the rotating cup arrangement throws off the oil as a fine mist, where it meets an airstream at right angles to its plane of motion, thus causing a shearing action which completes atomisation, and give a good degree of turndown. They are popular for shell boiler installations, particularly those of an industrial nature.

*Twin Fluid Atomisers*

This class of burners can be subdivided as follows depending upon the oil atomising medium employed.

(*a*) *Low Pressure Air Burners (LPA)*
Utilise air pressure at 1.5 to 7.5 kPa as the atomising medium. The air is directed to atomise the oil and at the same time provide combustion air requirements usually from the same fan. A variety of special nozzles and arrangements have been produced for this type of burner.

(*b*) *Medium Pressure Air Burners (MPA)*
Work on a similar principle to the LPA burner except that air for atomisation is provided from a compressor at between 50 and 200 kPa pressure with oil at about the same pressure. The balance of combustion air is provided from a low-pressure air fan in the normal way. Approximately 5 to 10% of total air requirement is provided from the compressor at high pressure. These units give a large turndown with good atomisation and stack solids performance.

(*c*) *High Pressure Burners*
These normally use steam as the atomising medium at pressures above 200 kPa. They are normally used on large water-tube type installations.

*Emulsion Systems*

Some burner manufacturers also produce emulsion systems to enable low percentages of water (typically 5%) to be added to the fuel for combustion of an oil-water mixture. This improves atomisation and, hence, the combustion of heavy residual fuels. Careful control of viscosity is required.

## CHIMNEYS AND FLUES

The chimneys and flues for any boiler plant must be considered, together with the boilers and combustion equipment, as an integrated design.

### Functions of the Chimney

The chimney must provide sufficient suction or draught to enable the particular type of plant installed to operate under optimum firing conditions.

The chimney produces a suction at its base due to the difference in the density between the column of hot gas within the chimney and the outside air. This can be expressed by the formula:

$$D = (\varrho_a - \varrho_g)\, g_n \quad .. \quad .. \quad .. \quad .. \quad \text{B13.1}$$

where:

$D$ = draught produced per metre of chimney height .. .. .. .. .. Pa
$\varrho_a$ = density of ambient air .. .. kg/m$^3$
$\varrho_g$ = mean density of flue gases .. .. kg/m$^3$
$g_n$ = gravitational acceleration .. .. m/s$^2$

Thus the draught produced is proportional to the height of the chimney and the temperature of gas within it. Fig. B13.13 gives data for winter and summer ambient conditions at various chimney temperatures. This gross draught is available to provide the energy required to move the flue gases through the particular boiler, flue and chimney system. Clearly, this available energy must be greater than the overall system resistance to gas flow in order that the plant will operate satisfactorily under natural draught conditions.

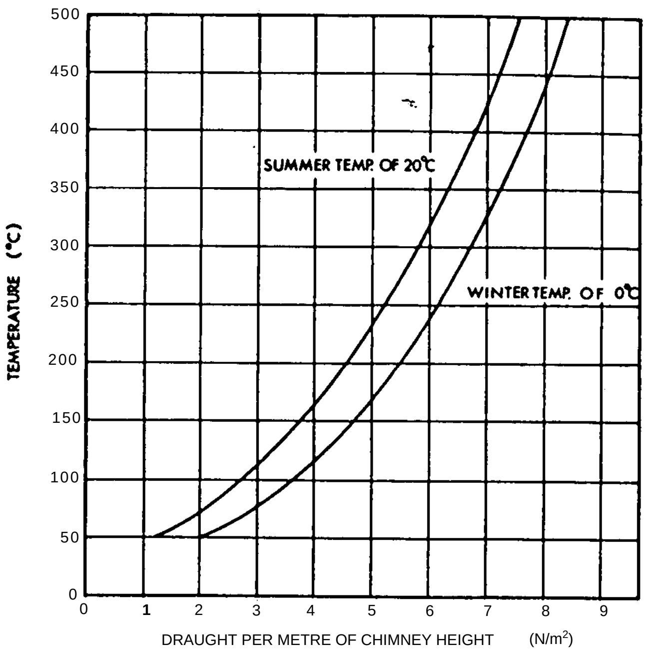



**Fig. B13.13.** Chimney draught (0°C and 20°C ambient).

The chimney must also provide effective flue gas dispersion so as to limit the ground level concentration of sulphur dioxide in the vicinity, having regard to the characteristics of the surrounding area. This is a public health requirement registered under the Clean Air Acts 1956 and 1968. Under these Acts the local authority must agree the proposed chimney height for any new furnace or extension to existing furnaces used for any purpose or any type of building.

### Design Factors

The following interrelated factors must be considered when carrying out chimney and flue design.

*System Resistance*

The chimney/flue areas must be selected so as to give flue gas velocities such that the overall maximum system resistance to gas flow is not greater than the available draught, whether this be natural or fan assisted, whilst at the same time achieving the maximum possible efflux velocity from the chimney terminal. Therefore it is important that the flue layout is carefully considered and designed so as to limit shock losses at bends etc. In general the following principles should be observed in flue design:

(*a*) Position the boilers as close as possible to the chimney to limit friction and heat losses in the connecting flue system.

(*b*) Avoid all short radius 90° bends in flue systems.

(*c*) Avoid abrupt section changes and use transformation sections with 15° included angles.

(*d*) Arrange the flue/chimney entry section to slope at 45° or more to the horizontal.

(*e*) Avoid protrusion of the flues beyond the inner face of the chimney or main flue connection.

(*f*) Make flues circular or square and avoid aspect ratios greater than 1.5 to 1, width to depth.

(*g*) Where possible flues should be sloped upwards towards the chimney.

(*h*) Provide clean-out doors at each bend in the flues at the chimney base, and adjacent to fans and dampers to aid maintenance.

(*i*) Avoid long 'dead' chimney pockets under the flue entry points, which are corrosion zones, and can cause harmonic pulsation problems.

*Chimney Efflux Velocity*

Unless chimney gas efflux velocities are maintained above a minimum value, depending upon the wind condition across the chimney outlet, downwashing of flue gases will occur on the leeward side. The Clean Air Act Memorandum[17] on chimney heights stipulates minimum full load efflux velocities of 6 m/s for natural draught and 7.5 m/s for fan forced or induced draught installations.

A further problem with low efflux velocities is the probability of inversion, where cold air enters the top of the chinmey, and flows downward, being heated by the rising hot gases and chimney walls. This can reduce chimney internal skin temperatures below the acid dew-point, resulting in acid smut emission.

The maintenance of an adequate efflux velocity at all loads is difficult where more than one boiler is served by one chimney in view of the turndown involved, particularly if each has high/low or modulating firing equipment.

For practical purposes the minimum efflux velocity to limit downwashing and inversion is approximately 6 m/s.

*Chimney Outlet Nozzles*

It may not always be possible to achieve efflux velocities of 6 m/s on natural draught plant, particularly if the whole flue and chimney system is designed on this velocity basis, due to the excessive system resistance involved. In such cases, the system can be designed for a lower velocity and a nozzle fitted at the chimney outlet to increase efflux velocity to the extent that the excess available draught allows. Nozzles should be designed with a maximum included angle of 30° as indicated on Fig. Bl3.14. They can be used for raising chimney gas efflux velocities where fans are employed for draught production.

**Flue Corrosion and Acid Smut Formation**

When sulphur free fuels are burned there is a temperature at which moisture begins to condense out in flue gases. This temperature is known as the water dew-point and its value varies according to the type of fuel, amount of excess air and combustion intensity.

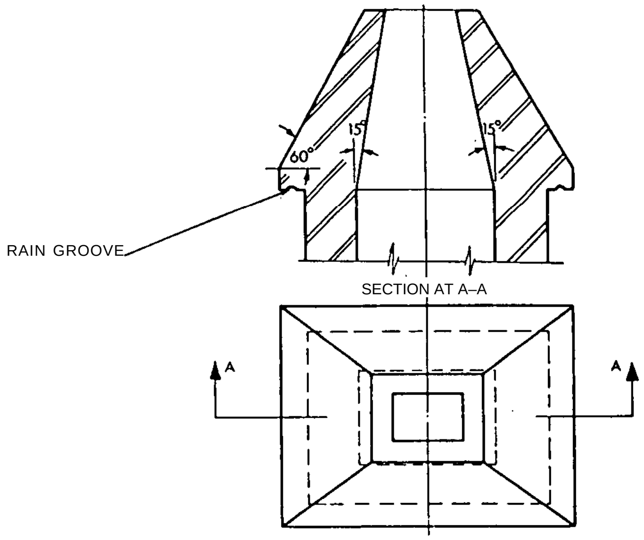



**Fig. B13.14.** Nozzle design.

With sulphur bearing fuels a second dew-point occurs at a higher temperature and this is called the acid dew-point, the value again depends on the type of fuel. amount of excess air, sulphur content and combustion intensity. The sulphur in the fuel is oxidized to $SO_2$ during the combustion process and a proportion of this is oxidised further to $SO_3$, with subsequent formation of sulphuric acid. Fig. B13.15 illustrates these points.

A peak rate of corrosion tends to occur some 30 to 40°C below the acid dew-point and a dramatic increase in corrosion rate occurs below the water dew-point. Acid dew-points generally lie in the range 115 to 140°C for the type of boiler plant used in heating work but depend upon excess air used, flame temperature, sulphur content etc. A significant depression in acid dew-point temperature occurs where fuels have less than 0.5% sulphur content. It can also be reduced or eliminated by stoichiometric combustion conditions which can only be approached on very large plants.

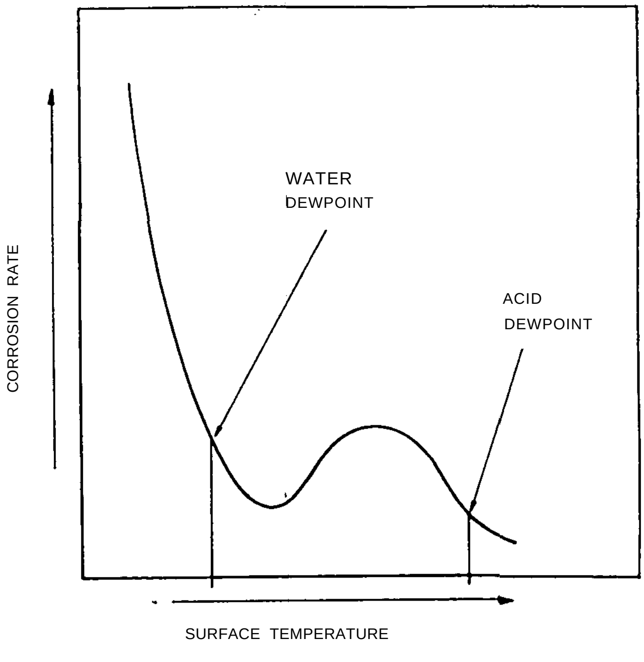



**Fig. B13.15.** Water and acid dew-points.

*Smuts*

A smut is an acid agglomerate of carbon particles resulting from a combination of stack solids and low temperature corrosion products. If the inner surface of any flue/chimney falls below the acid dew-point temperature of the waste gases an acidic film forms on the surface to which stack solids adhere and build up into loose layers which are dislodged and ejected from the chimney as the firing rates change.

**Flue/Chimney Area and Siting**

Where chimneys are oversized, or where more than one boiler is used with one flue/chimney, the cooling effect is accentuated at low loads and, apart from the undesirable effects mentioned above, the inner chimney surface temperatures can fall below acid dew-point conditions, even with insulation applied.

*It is essential to install one flue/chimney per boiler, correctly sized for maximum practicable full load flue gas velocity if these problems are to be avoided.*

Chimney outlets should not be positioned such that air inlets into the building are on the leeward side of the chimney for the prevailing wind direction.

Generally internal chimneys have less heat dissipation than free standing units but where external chimneys are used they should, where possible, be positioned on the leeward side of the building or site, considering the prevailing wind direction.

When the flue/chimney area is reduced to give high flue gas velocities and a pressurised flue system, the construction of flues and chimneys must be carefully considered. With a mild steel flue/chimney system all joints should be welded or otherwise permanently sealed. Expansion should be accommodated by means of bellows type expansion joints and all explosion relief doors, clean out doors etc. fitted with the requisite joints to withstand pressurised flue conditions where concrete, or brick chimneys with lining bricks are used, they should generally be sized to be under suction conditions unless the construction is specifically designed for operation under pressurised flue conditions. By combining the chimneys into one insulated envelope the cooling losses are reduced and the effective chimney plume height increased. The chimney outlet should be at a minimum height of 3 m above the highest point of the adjacent building roof level in order to limit wind pressure variations on the flue outlet and present the minimum face area to the prevailing wind.

**Cold Air Admission**

The admission of cold air into the flue/chimney system reduces the flue gas temperatures and hence the available natural draught. Draught stabilisers deliberately introduce cold air to regulate the draught by this means. However, reduction of the flue gas temperature with high sulphur fuels can also produce the corrosion and acid smut emissions described, and for normal boiler plants burning these types of fuels the use of draught stabilisers is not recommended.

Modulating dampers should be used for draught regulations on these plants with safety interlocks fitted to prevent firing against a closed damper. With high chimneys the damper should be arranged to close when the firing equipment is off-load, so as to isolate the boiler and limit cold air ingress to the system. This procedure limits the cooling effect on the internal flue and chimney system, and the corrosion mechanism within the boiler gas-side heating surfaces. In the latter case, cold moist air entering the boiler can set up very rapid corrosion within the water dew-point range (below 60°C) by combination with the gas-side flue deposits which remain in the boiler. Arrangements must be made on the water circuits by means of shunt pumps etc. to ensure that the return water temperature to the boilers is always above a minimum 63°C. A permanent bleed of this return water should be arranged through idle boilers by means of a small by-pass connection fitted with a pre-set regulating valve.

**Heat Loss**

To enable the correct chimney construction to be selected it is necessary to predict the minimum internal surface temperature likely to be obtained at the chimney terminal under all loads. The following is a method of obtaining an approximate value. It should be remembered that average values are used for some parameters and that radiation from the gases to the chimney is ignored in order to simplify calculations. The heat loss from the chimney or duct is given by

$$q = U\,A\,(t_g - t_{ao}) \quad \text{B13.2}$$

where:

| | | |
|---|---|---|
| $q$ | = heat loss | W |
| $U$ | = overall thermal transmittance | W/m$^2$ K |
| $A$ | = surface area | m$^2$ |
| $t_g$ | = mean waste gas temperature | °C |
| $t_{ao}$ | = outside air temperature | °C |

The lowest outside air temperature is usually assumed to be −10°C.

The overall thermal conductance is derived from:

$$\frac{1}{U} = \frac{1}{h_o} + \frac{l_1}{l_1} + \frac{l_2}{l_2} = \quad \cdots \quad + \frac{1}{h_i} \quad \text{B13.3}$$

where:

| | | |
|---|---|---|
| $h_o$ | = external film coefficient | W/m$^2$ K |
| $h_i$ | = internal film coefficient | W/m$^2$ K |
| $l$ | = thickness of chimney layer | m |
| $\lambda$ | = thermal conductivity of chimney layer | W/m K |

Figs. B13.16 and 17 give values of $h_o$ and $h_i$ for various conditions. The heat loss can also be deduced from:

$$q = W\,C_p\,(t_{g1} - t_{g2}) \quad \text{B13.4}$$

where:

| | | |
|---|---|---|
| $W$ | = mass flow rate of gases | kg/s |
| $C_p$ | = specific heat of waste gases at constant pressure | J/kg K |

$t_{g1}$ = temperature of gases entering base of chimney .. .. °C

$t_{g2}$ = temperature of gases leaving top of chimney .. .. .. °C

$t_g$ = $\frac{1}{2}(t_{g2} + t_{g1})$ .. .. .. K

Alternatively, the volume of flow of waste gases ($m^3/s$) is sometimes used in conjunction with the specific heat (in $J/m^3$ K). The specific heat is usually taken a 1.22 $kJ/m^3$ K at 20°C.

For conditions of thermal equilibrium, equations B13.2 and 4 must give the same heat loss, so they may be combined:

$$U\ A\ (t_g - t_{ao}) = W\ C_p\ (t_{g1} - t_{g2}) \quad \text{B13.5}$$

If the temperature of the waste gases entering the chimney or duct is known or estimated, the temperature of the gases leaving the chimney may be determined from equation B13.5. The minimum surface temperature may then be established from:

$$h_i\,(t_{g2} - t_{si}) = U\ (t_{g2} - t_{ao}) \quad \text{B13.6}$$

where:

$t_{si}$ = temperature of inside surface of chimney °C

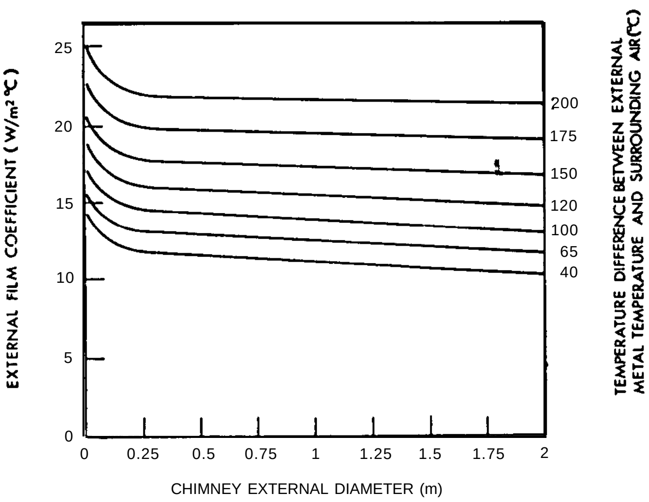



**Fig. B13.16.** Values of external film coefficient.

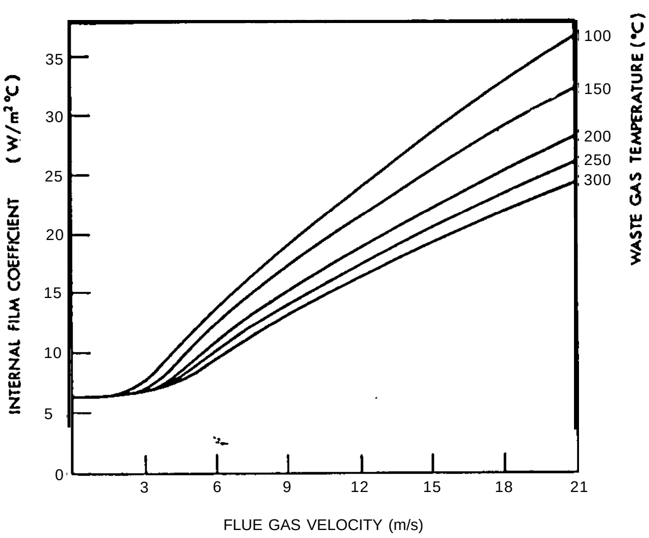



**Fig. B13.17.** Values of internal film coefficient.

Chimney and Flue Sizing

In order to size chimneys and flues correctly, the following information is required.

(*a*) The type of fuel used, its calorific value and the percentage sulphur content.

(*b*) The type and rated output of the boiler.

(*c*) The overall thermal efficiency of the boiler based on gross calorific value.

(*d*) The boiler flue gas outlet conditions at high and low fire, i.e. gas outlet temperatures and percentage carbon dioxide.

(*e*) The draught requirements at the boiler outlet at high and low fire.

(*f*) The height of the installation above sea level. (Gas volumes are increased by approximately 4% for every 300 m above sea level and allowance must be made in specifying volumes of forced and induced fans, etc. for installations at more than 600 m above sea level).

(*g*) Location of plant and the character of surroundings, viz. topography, height of buildings surrounding plant, prevailing wind direction and velocities and the position of the boiler (i.e. basement or roof top).

(*h*) Winter and summer extremes of ambient temperature.

(*i*) Proposed general chimney construction to assess the cooling effect on gases.

In the case of boiler plants burning sulphur bearing fuels where the total sulphur dioxide emission at full load exceeds 0.38 g/s, the chimney height is determined by the requirements of the Clean Air Acts 1956 and 1968, as interpreted by the method put forward by the Department of the Environment publication ‘Chimney Heights’, third edition[17]. For smaller boiler plants burning sulphur bearing fuels with a sulphur dioxide emission of less than 0.38 g/s, the chimney height is determined by combustion draught requirements with the proviso that such chimneys should terminate at least 3 m above the surrounding roof level or at a higher level if the public health authority so require.

For gaseous fuels having negligible sulphur content, the chimney heights must be assessed from other criteria. A method using other combustion products as the guiding criteria (e.g. oxides of nitrogen, aldehydes, etc.) has been proposed[18], sizing the chimney to ensure an acceptable low limit of ground level concentration and this method is described below. The height so obtained must also be checked as providing the necessary draught for the boiler plant operation.

The aim of the legislation and proposed methods is to limit the ground level concentration of products (e.g. nitrogen and sulphur oxides, aldehydes) and every chimney must be discussed with the Public Health Department of the local authority concerned at the design stage, and their agreement to the proposal obtained.

**Chimney Heights for Sulphur Bearing Fuels**

The maximum fuel burning rate at full plant loading is given by:

$$W_m = \frac{100 \times R}{CV \times h} \quad \text{.. .. .. ..} \quad \text{B13.7}$$

where:

| | | |
|---|---|---|
| $W_m$ = maximum fuel burning rate | .. | kg/s |
| $R$ = rated boiler output | .. .. | MW |
| $CV$ = calorific value of fuel | .. .. | MJ/kg |
| $h$ = thermal efficiency of boiler | .. | % |

The maximum sulphur dioxide emission for fired equipment is:

$$E_m = K_1 \times W_m \times S \quad \text{.. .. ..} \quad \text{B13.8}$$

where:

| | | |
|---|---|---|
| $E_m$ = maximum sulphur dioxide emission | | g/s |
| $S$ = sulphur content of fuel | .. .. | % |
| $K_1$ = 20 for oil firing | | |
| = 18 for coal firing | | |

Equations B13.7 and 8 may be combined for easy reference as:

$$E_m = K_2 \frac{R}{h} \quad \text{.. .. .. ..} \quad \text{B13.9}$$

where:

$K_2$ = a constant representing the type of fuel, its calorific value and sulphur content as read from Table B13.5.

**Table B13.5.** Properties of fuels and values of $K_2$ for equation B13.9

| Type of fuel | Properties | | $K_3$ in equation B13.9 |
|---|---|---|---|
| | Calorific value (MJ/kg) | Sulphur content (%) | |
| **Liquid fuel** | | | |
| Class D, Gas oil | 45·5 | 1·0 | 43·9 |
| Class E, Light fuel oil | 43·4 | 3·2 | 147·5 |
| Class F, Medium fuel oil | 42·9 | 3·5 | 163·2 |
| Class G, Heavy fuel oil | 42·5 | 3·8 | 178·8 |
| **Solid fuel** | | | |
| Anthracite 101 and 102 | 30·0 | 1·1 | 66·0 |
| Dry steam coal 201 | 30·5 | 1·1 | 64·9 |
| Coking steam coal 202 and 204 | 30·7 | 1·1 | 64·5 |
| Medium volatile coking coal 301a and 301b | 30·5 | 1·3 | 76·7 |
| Low volatile coal 200H | 29·9 | 1·3 | 78·3 |
| Medium volatile coal 202H and 303H | 30·0 | 1·3 | 78·0 |
| Very strongly caking coal 401 | 29·5 | 1·9 | 115·9 |
| Strongly caking coal 501 and 502 | 29·4 | 1·9 | 116·3 |
| Medium caking coal 601 and 602 | 27·6 | 1·9 | 123·9 |
| Weakly caking coal 701 and 702 | 26·7 | 1·8 | 121·4 |
| Very weakly caking coal 802 | 25·2 | 1·9 | 135·7 |
| Non-caking coal 902 | 23·8 | 1·8 | 136·1 |

Where the sulphur dioxide emission is equal to or greater than 0.38 g/s, decide the area and category from the following alternatives:

A: An undeveloped area where development is unlikely, where background pollution is low, and where there is no development within 800 m of the new chimney.

B: A partially developed area with scattered houses, low background pollution and no other comparable industrial emissions within 400 m of the new chimney.

C: A built-up residential area with only moderate background pollution and without other comparable industrial emissions.

D: An urban area of mixed industrial and residential development, with considerable background pollution and with other comparable industrial emissions within 400 m of the new chimney.

E: A large city, or an urban area of mixed heavy industrial and dense residential development, with severe background pollution.

Refer to Fig. B13.18 to obtain the uncorrected chimney height and, for fuels with more than 2% sulphur content, add 10% to this height. If the height obtained is more than 2½ times the height of the building or any building in the immediate vicinity, no further correction is required.

Where this is not so, the final chimney height is obtained by substitution in the following formula:

$$H = [0.56h_a + 0.375h_b] + 0.625h_c \quad \text{..} \quad \text{B13.10}$$

where:

| | | |
|---|---|---|
| $H$ = final chimney height | .. .. | m |
| $h_a$ = building height or greatest length whichever is the lesser | .. .. | m |
| $h_b$ = building height | .. .. .. | m |
| $h_c$ = uncorrected chimney height | .. | m |

Table B13.6 provides solutions to the term in brackets against known values of $h_a$ and $h_b$. The final chimney height may then be obtained by adding this result to the appropriate value read from the scale on the right hand side of Fig. B13.18. Note that 10% must be added to this latter value for fuels with more than 2% sulphur content.

Where the sulphur dioxide emission is less than 0.38 g/s proceed as follows:

(*a*) Assess the height of buildings through which the chimney passes or to which it is attached.

(*b*) Add 3 m to this height to obtain the preliminary chimney height.

(*c*) Where the particular building is surrounded by higher buildings, the height of the latter must bc taken into consideration as above.

(*d*) Select a trial flue gas velocity (from Table B13.7) and calculate the flue and chimney resistance.

(*e*) Compare this with the available chimney draught (Fig. B13.13) and adjust the chimney height to suit. recalculating where necessary to give the highest possible efflux velocity from the chimney.

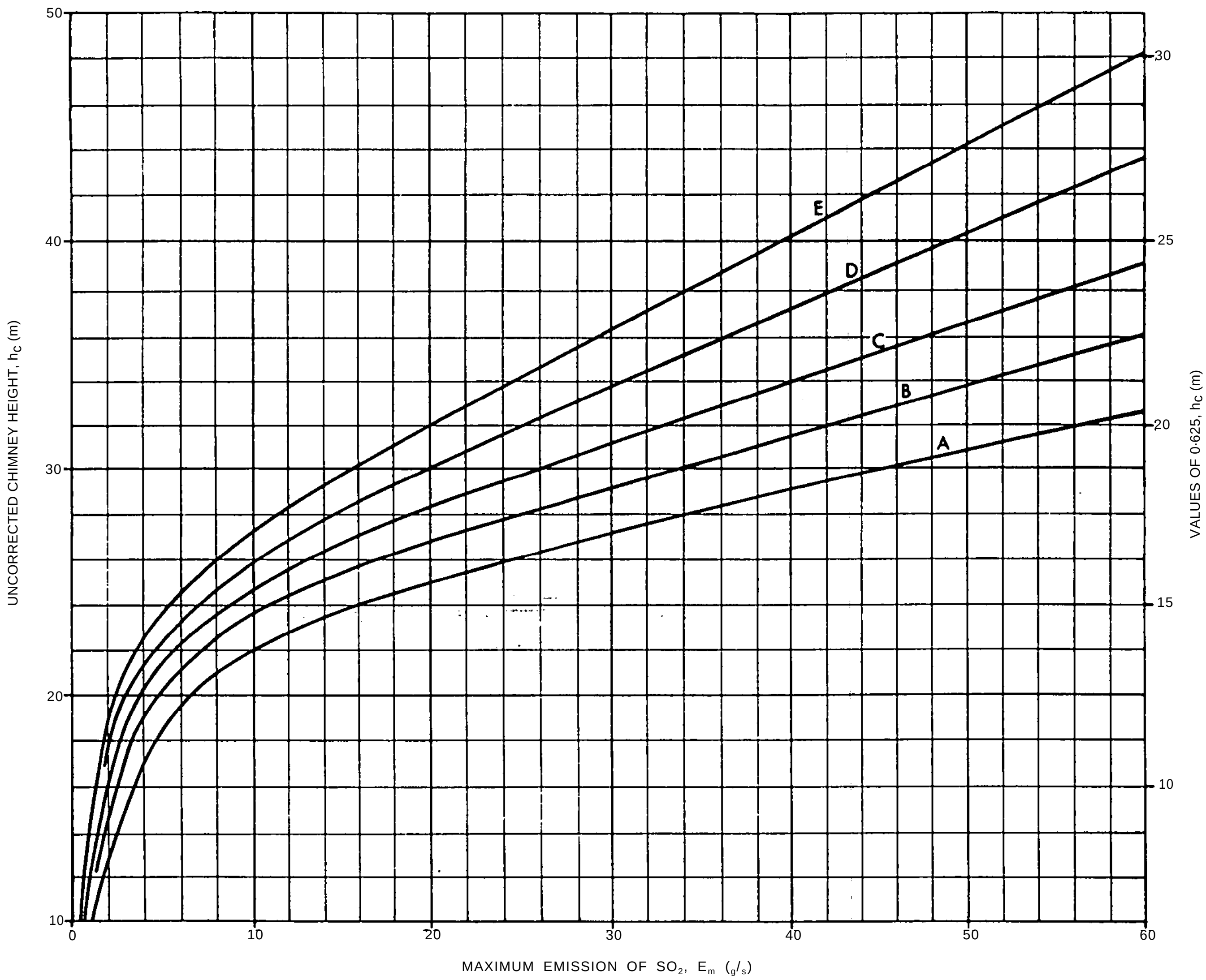



**Fig. B13.18.** Uncorrected chimney heights.

**Table B13.6.** Solutions to part of equation B13.10.

| Building height $h_b$ (m) | Values of $[0{\cdot}56h_a + 0{\cdot}375h_b]$ in equation B13.10 for listed values of $h_a$ (m)—building height or length (m) | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 9 | 12 | 15 | 18 | 21 | 24 | 27 | 30 | 33 | 36 | 39 | 42 | 45 | 48 | 51 | 54 | 57 | 60 |
| 9 | 8·4 | 10·1 | 11·8 | 13·5 | 15·1 | 16·8 | 18·5 | 20·2 | 21·9 | 23·5 | 25·2 | 26·9 | 28·6 | 30·3 | 31·9 | 33·6 | 35·3 | 37·0 |
| 12 | 9·5 | 11·2 | 12·9 | 14·6 | 16·3 | 17·9 | 19·6 | 21·3 | 23·0 | 24·7 | 26·3 | 28·0 | 29·7 | 31·4 | 33·1 | 34·7 | 36·4 | 38·1 |
| 15 | 10·7 | 12·3 | 14·0 | 15·7 | 17·4 | 19·1 | 20·7 | 22·4 | 24·1 | 25·8 | 27·5 | 29·1 | 30·8 | 32·5 | 34·2 | 35·9 | 37·5 | 39·2 |
| 18 | 11·8 | 13·5 | 15·2 | 16·8 | 18·5 | 20·2 | 21·9 | 23·6 | 25·2 | 26·9 | 28·6 | 30·3 | 32·0 | 33·6 | 35·3 | 37·0 | 38·7 | 40·4 |
| 21 | 12·9 | 14·6 | 16·3 | 18·0 | 19·6 | 21·3 | 23·0 | 24·7 | 26·4 | 28·0 | 29·7 | 31·4 | 33·1 | 34·8 | 36·4 | 38·1 | 39·8 | 41·5 |
| 24 | 14·0 | 15·7 | 17·4 | 19·1 | 20·8 | 22·4 | 24·1 | 25·8 | 27·5 | 29·2 | 30·8 | 32·5 | 34·2 | 35·9 | 37·6 | 39·2 | 40·9 | 42·6 |
| 27 | 15·2 | 16·8 | 18·5 | 20·2 | 21·9 | 23·6 | 25·2 | 26·9 | 28·6 | 30·3 | 32·0 | 33·6 | 35·3 | 37·0 | 38·7 | 40·4 | 42·0 | 43·7 |
| 30 | 16·3 | 18·0 | 19·7 | 21·3 | 23·0 | 24·7 | 26·4 | 28·1 | 29·7 | 31·4 | 33·1 | 34·8 | 36·5 | 38·1 | 39·8 | 41·5 | 43·2 | 44·9 |
| 33 | 17·4 | 19·1 | 20·8 | 22·5 | 24·1 | 25·8 | 27·5 | 29·2 | 30·9 | 32·5 | 34·2 | 35·9 | 37·6 | 39·3 | 40·9 | 42·6 | 44·3 | 46·0 |
| 36 | 18·5 | 20·2 | 21·9 | 23·6 | 25·3 | 26·9 | 28·6 | 30·3 | 32·0 | 33·7 | 35·3 | 37·0 | 38·7 | 40·4 | 42·1 | 43·7 | 45·4 | 47·1 |
| 39 | 19·7 | 21·3 | 23·0 | 24·7 | 26·4 | 28·1 | 29·7 | 31·4 | 33·1 | 34·8 | 36·5 | 38·1 | 39·8 | 41·5 | 43·2 | 44·9 | 46·5 | 48·2 |
| 42 | 20·8 | 22·5 | 24·2 | 25·8 | 27·5 | 29·2 | 30·9 | 32·6 | 34·2 | 35·9 | 37·6 | 39·3 | 41·0 | 42·6 | 44·3 | 46·0 | 47·7 | 49·4 |
| 45 | 21·9 | 23·6 | 25·3 | 27·0 | 28·6 | 30·3 | 32·0 | 33·7 | 35·4 | 37·0 | 38·7 | 40·4 | 42·1 | 43·8 | 45·4 | 47·1 | 48·8 | 50·5 |
| 48 | 23·0 | 24·7 | 26·4 | 28·1 | 29·8 | 31·4 | 33·1 | 34·8 | 36·5 | 38·2 | 39·8 | 41·5 | 43·2 | 44·9 | 46·6 | 48·2 | 49·9 | 51·6 |
| 51 | 24·2 | 25·8 | 27·5 | 29·2 | 30·9 | 32·6 | 34·2 | 35·9 | 37·6 | 39·3 | 41·0 | 42·6 | 44·3 | 46·0 | 47·7 | 49·4 | 51·0 | 52·7 |
| 54 | 25·3 | 27·0 | 28·7 | 30·3 | 32·0 | 33·7 | 35·4 | 37·1 | 38·7 | 40·4 | 42·1 | 43·8 | 45·5 | 47·1 | 48·8 | 50·5 | 52·2 | 53·9 |
| 57 | 26·4 | 28·1 | 29·8 | 31·5 | 33·1 | 34·8 | 36·5 | 38·2 | 39·9 | 41·5 | 43·2 | 44·9 | 46·6 | 48·3 | 49·9 | 51·6 | 53·3 | 55·0 |
| 60 | 27·5 | 29·2 | 30·9 | 32·6 | 34·3 | 35·9 | 37·6 | 39·3 | 41·0 | 42·7 | 44·3 | 46·0 | 47·7 | 49·4 | 51·1 | 52·7 | 54·4 | 56·1 |

**Table B13.7.** Trial flue gas velocities.

| Chimney height (m) | Trial flue gas velocity (m/s) | |
|---|---|---|
| | Natural draught boilers* | Boilers with pressurized combustion chambers |
| Up to 12 | 3·6 | 6·0 |
| 12 to 20 | 4·5 | — |
| 12 to 24 | — | 7·5 |
| 20 to 30 | 6·0 | — |
| 24 to 30 | — | 9·0 |
| over 30 | 7·5 | 12·0 |

*Note:*
For brick flues/chimneys less than 0·3 m square, the velocity should be limited to 4·5 m/s for natural draught boilers and 6 m/s for pressurized boilers.
*Gas resistance less than 37 N/m$^2$.

## Chimney Heights for Non-sulphur Bearing Fuels

The following procedure should be followed:

*(a)* Assess the boiler plant heat input rate.

*(b)* For single, free standing, chimneys, read the corresponding chimney height from Fig. B13.19.

*(c)* For single chimneys passing through, or adjacent to buildings (the more usual case), the additional height may be read off from Fig. B13.19 and added to then building height to give the final chimney height.

Where two or more chimneys are in close proximity, the height of each should be increased slightly as follows:

*(a)* Find the individual final chimney heights as before.

*(b)* Express the separation between a pair of chimneys as a multiple of the free standing height of the smaller chimney.

*(c)* From Fig. B13.20, read off the height correction factor. The required increase in height is then given by:

$$DH_2 = h \times H_f \quad .. \quad .. \quad .. \quad \text{B13.11}$$

where:

$DH_2$ = increase in height .. .. m

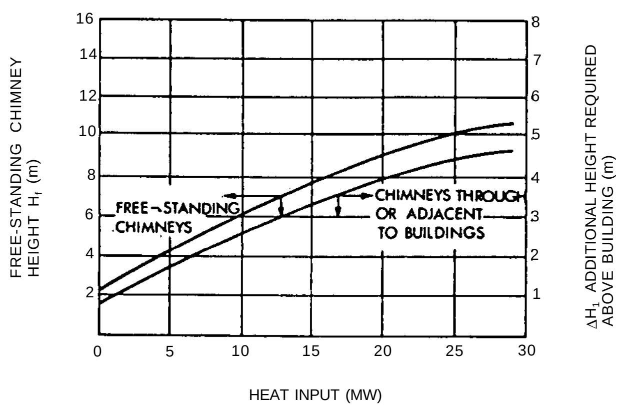
$h$ = height correction factor (see Fig. B13.20)

$H_f$ = free standing height of the taller chimney .. .. m

*(d)* Repeat these steps for each pair of chimneys.

*(e)* Add the largest increase in height found to the final height of each chimney found as before.

Check that the height of each chimney provides the required combustion draught.



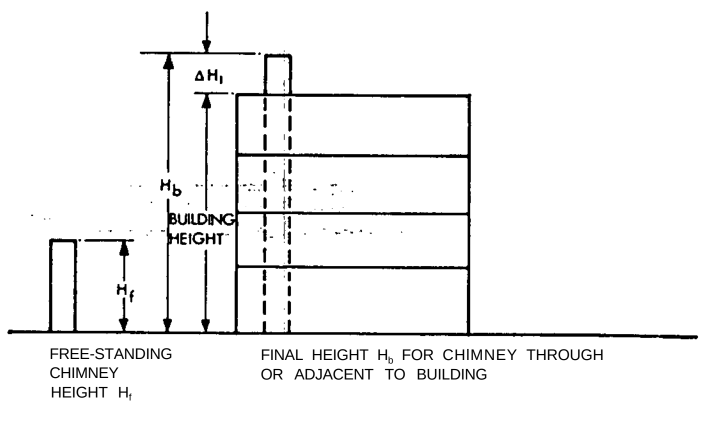
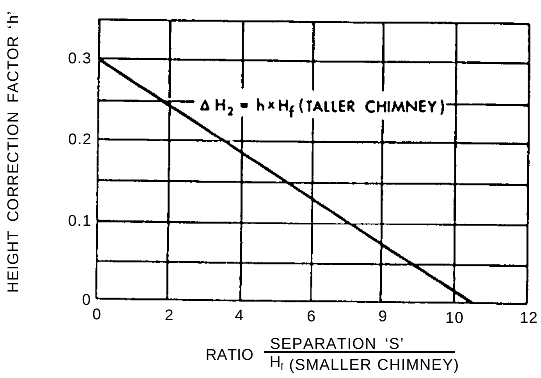




**Fig. B13.19.** Heights for single chimneys.



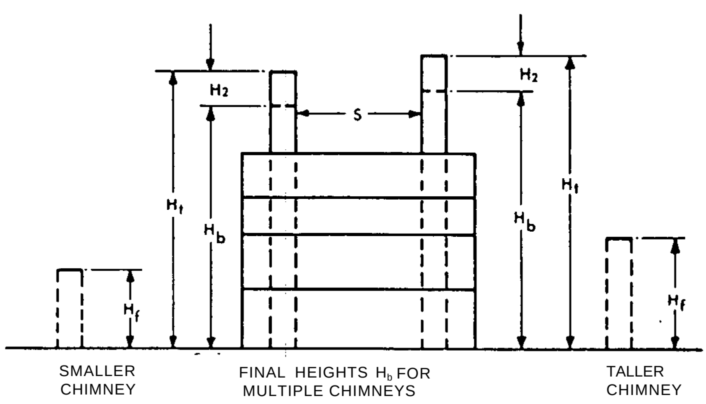




**Fig. B.13.20.** Heights for adjacent chimneys.

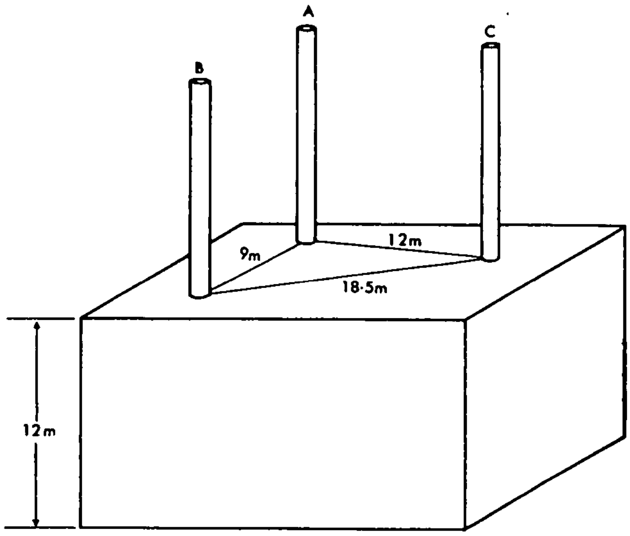



**Fig. B13.21.** Diagram for example.

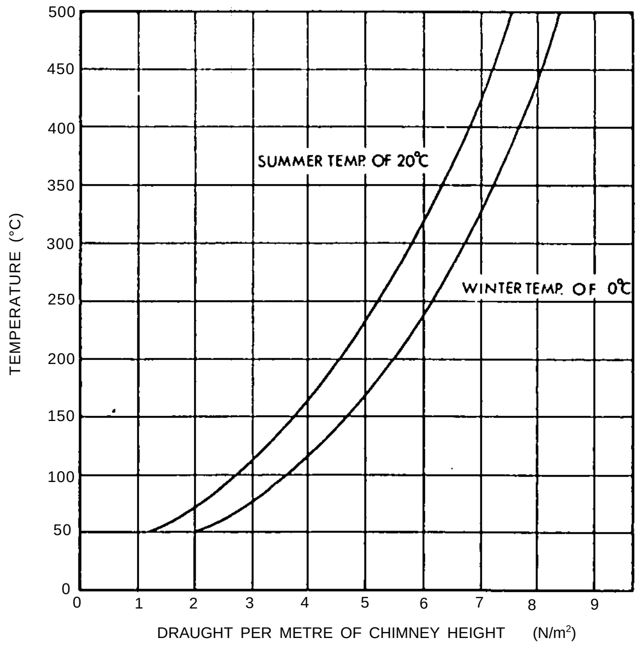



**Fig. B13.22.** Chimney draught (0°C and 20°C ambient).

*Example*

Fig.B13.21 shows a building 3 m high with three chimneys passing through it. If the heat inputs are 6 MW to A, 15 MW to B and 3 MW to C, determine the chimney heights.

*Solution*

The free standing heights (from Fig. B13.19) are A = 4.6 m, B = 7.8 m and C = 3.4 m. The heights to be added to that of the building are (from Fig. B13.21) A = 1.9 m, B = 3.3 m and C = 1.4 m.

The separations of pairs of chimneys expressed as a multiple of the smallest free standing height are AB = 2.6.

BC = 5.5 and CA = 2.7. From equation B13.11, the required increases in height, considering each pair in turn, are AB = 1.8 m, BC = 1.1 m and CA = 1.0 m Since the largest value is 1.8 m, the required chimney heights are:

A = 13.9 + 1.8 = 15.7 m
B = 15.3 + 1.8 = 17.1 m
C = 13.4 + 1.8 = 15.2 m

*Available Chimney Draught*

From the chimney height assessed from Fig. B13.19 it is possible to obtain the equivalent available chimney draught at various flue gas temperatures by reference to Fig. B13.22. These curves are based upon ambient temperatures of 0 and 20°C.

**Determination of Flue/Chimney Area**

The area must be selected to provide the highest possible flue gas velocity and the smallest cooling area, bearing in mind the available draught and frictional resistance of the flue and chimney considered. In order to avoid downwash, a chimney efflux velocity of approximately 7.5 to 9 m/s is required, but this cannot always be achieved on natural draught plant. The procedure is as follows:

*(a)* Calculate the flue gas volume flow rates to be handled at full and low fire conditions at the temperatures involved at the particular boiler outlet.

*(b)* Select a flue gas velocity which appears reasonable for the plant considered (see Table B13.7) and obtain area equivalant from:

$$A = \frac{Q_f}{v} \qquad \text{.. .. .. .. B13.12}$$

where:

$A$ = area equivalent (Table B13.8) $m^2$
$Q_f$ = flue gas volume flow rate at full fire .. .. .. $m^3/s$
$v$ = flue gas velocity .. .. m/s

*(c)* Calculate the flue/chimney resistance to gas flow at this velocity for the particular flue layout proposed. This resistance comprises energy losses at bends, restrictions, section changes etc., efflux into the atmosphere and skin frictional resistances. The friction loss per metre run of the flue/chimney is given by:

$$DP_f = \frac{C \times p_v \times F}{A} \qquad \text{.. .. .. B13.13}$$

where:

$DP_f$ = frictional loss per metre run Pa/m
$C$ = circumference or perimeter of flue .. .. .. m
$P_v$ = velocity pressure factor (see Table B13.9) .. .. .. Pa
$F$ = friction factor (Table B13.10)

The energy loss at a bend etc. is given by:

$$DP_e = P_v \times \zeta \qquad \text{.. .. .. B13.14}$$

where:

$DP_e$ = energy loss .. .. Pa
$\zeta$ = pressure loss coefficient for the particular item (see Table B13.11)

**Table B13.8.** Effective area of chimney sections.

| Circular and square sections | | | Rectangular and elliptical sections | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| a (m) | Area ($m^3$) | | a × b (m) | Area ($m^3$) | | a × b (m) | Area ($m^3$) | | a × b (m) | Area ($m^3$) | |
| | Circle | Square | | Ellipse | Rectangle | | Ellipse | Rectangle | | Ellipse | Rectangle |
| 0·3 | 0·07 | 0·09 | 0·3 × 0·4 | 0·09 | 0·12 | 0·9 × 1·0 | 0·71 | 0·90 | 1·5 × 1·6 | 1·88 | 2·40 |
| 0·4 | 0·13 | 0·16 | 0·3 × 0·5 | 0·12 | 0·15 | 0·9 × 1·0 | 0·78 | 0·99 | 1·5 × 1·7 | 2·00 | 2·55 |
| 0·5 | 0·20 | 0·25 | 0·3 × 0·6 | 0·14 | 0·18 | 0·9 × 1·2 | 0·85 | 1·08 | 1·5 × 1·8 | 2·12 | 2·70 |
| 0·6 | 0·28 | 0·36 | 0·4 × 0·5 | 0·16 | 0·20 | 1·0 × 1·1 | 0·86 | 1·10 | 1·6 × 1·7 | 2·13 | 2·72 |
| 0·7 | 0·39 | 0·49 | 0·4 × 0·6 | 0·19 | 0·24 | 1·0 × 1·2 | 0·94 | 1·20 | 1·6 × 1·8 | 2·26 | 2·88 |
| 0·8 | 0·50 | 0·64 | 0·4 × 0·7 | 0·22 | 0·28 | 1·0 × 1·3 | 1·02 | 1·30 | 1·6 × 1·9 | 2·39 | 3·04 |
| 0·9 | 0·64 | 0·81 | 0·5 × 0·6 | 0·24 | 0·30 | 1·1 × 1·2 | 1·04 | 1·32 | 1·7 × 1·8 | 2·40 | 3·06 |
| 1·0 | 0·79 | 1·01 | 0·5 × 0·7 | 0·27 | 0·35 | 1·1 × 1·3 | 1·12 | 1·43 | 1·7 × 1·9 | 2·53 | 3·23 |
| 1·1 | 0·95 | 1·21 | 0·5 × 0·8 | 0·31 | 0·40 | 1·1 × 1·4 | 1·21 | 1·54 | 1·7 × 2·0 | 2·67 | 3·40 |
| 1·2 | 1·12 | 1·44 | 0·6 × 0·7 | 0·33 | 0·42 | 1·2 × 1·3 | 1·22 | 1·56 | 1·8 × 1·9 | 2·68 | 3·42 |
| 1·3 | 1·33 | 1·69 | 0·6 × 0·8 | 0·38 | 0·48 | 1·2 × 1·4 | 1·32 | 1·68 | 1·8 × 2·0 | 2·83 | 3·60 |
| 1·4 | 1·54 | 1·96 | 0·6 × 0·9 | 0·42 | 0·54 | 1·2 × 1·5 | 1·41 | 1·80 | 1·8 × 2·1 | 2·97 | 3·78 |
| 1·5 | 1·77 | 2·25 | 0·7 × 0·8 | 0·44 | 0·56 | 1·3 × 1·4 | 1·43 | 1·82 | 1·9 × 2·0 | 2·98 | 3·80 |
| 1·6 | 2·01 | 2·56 | 0·7 × 0·9 | 0·49 | 0·63 | 1·3 × 1·5 | 1·53 | 1·95 | 1·9 × 2·1 | 3·13 | 3·99 |
| 1·7 | 2·27 | 2·89 | 0·7 × 1·0 | 0·55 | 0·70 | 1·3 × 1·6 | 1·63 | 2·08 | 1·9 × 2·2 | 3·28 | 4·18 |
| 1·8 | 2·54 | 3·24 | 0·8 × 0·9 | 0·57 | 0·72 | 1·4 × 1·5 | 1·65 | 2·10 | 2·0 × 2·1 | 3·30 | 4·20 |
| 1·9 | 2·83 | 3·61 | 0·8 × 1·0 | 0·63 | 0·80 | 1·4 × 1·6 | 1·76 | 2·24 | 2·0 × 2·2 | 3·45 | 4·40 |
| 2·0 | 3·14 | 4·00 | 0·8 × 1·1 | 0·69 | 0·88 | 1·4 × 1·7 | 1·87 | 2·38 | 2·0 × 2·3 | 3·61 | 4·60 |

**Table B13.9.** Velocity pressure.

| Velocity (m/s) | Velocity pressure at stated temperatures, °C Pa | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | 50 | 100 | 150 | 200 | 250 | 300 | 350 | 400 | 450 | 500 |
| 3 | 4·9 | 4·2 | 3·7 | 3·4 | 3·0 | 2·8 | 2·5 | 2·4 | 2·2 | 2·0 |
| 4 | 8·7 | 7·5 | 6·7 | 6·0 | 5·4 | 4·9 | 4·5 | 4·2 | 3·9 | 3·6 |
| 5 | 13·7 | 11·8 | 10·4 | 9·3 | 8·4 | 7·7 | 7·1 | 6·5 | 6·1 | 5·7 |
| 6 | 19·7 | 17·0 | 15·0 | 13·4 | 12·1 | 11·1 | 10·2 | 9·4 | 8·7 | 8·3 |
| 7 | 26·8 | 23·1 | 20·4 | 18·2 | 16·5 | 15·1 | 13·8 | 12·8 | 11·9 | 11·1 |
| 8 | 35·0 | 30·2 | 26·6 | 23·8 | 21·5 | 19·6 | 18·1 | 16·7 | 15·6 | 14·6 |
| 9 | 44·3 | 38·2 | 33·8 | 30·2 | 27·3 | 24·9 | 22·9 | 21·2 | 19·7 | 18·4 |
| 10 | 54·5 | 47·1 | 41·7 | 37·2 | 33·6 | 30·8 | 28·2 | 26·1 | 24·3 | 22·8 |
| 11 | 66·0 | 56·9 | 50·4 | 45·0 | 40·6 | 37·2 | 34·1 | 31·6 | 29·4 | 27·5 |
| 12 | 78·5 | 67·8 | 60·0 | 53·5 | 48·4 | 44·1 | 40·5 | 37·6 | 35·0 | 32·8 |
| 13 | 92·2 | 79·6 | 70·4 | 62·9 | 56·9 | 51·9 | 47·6 | 44·1 | 41·1 | 38·4 |
| 14 | 107 | 92·4 | 81·7 | 73·0 | 65·9 | 60·3 | 55·4 | 51·2 | 47·6 | 44·6 |
| 15 | 123 | 106 | 93·5 | 83·8 | 75·6 | 69·2 | 63·5 | 58·8 | 54·6 | 51·2 |
| 16 | 139 | 121 | 107 | 95·2 | 86·0 | 78·8 | 72·3 | 66·9 | 62·1 | 58·1 |
| 17 | 158 | 136 | 121 | 107 | 97·1 | 88·7 | 81·4 | 75·4 | 70·3 | 65·6 |
| 18 | 177 | 153 | 135 | 121 | 109 | 99·5 | 91·5 | 84·4 | 78·8 | 73·8 |

The total resistance of the flue/chimney is given by:

$$DP_t = (DP_f \times L) + S\,DP_e + D \quad .. \quad \text{B13.15}$$

where:

$DP_t$ = total resistance .. .. Pa

$L$ = length of flue/chimney .. m

$S\,DP_e$ = sum of individual energy losses at bends etc. .. .. Pa

$D$ = draught required at boiler outlet .. .. .. Pa

The total resistance obtained from equation B13.15 must been compared with the available chimney draught. If the residual chimney draught is excessive, the flue areas can be recalculated using a higher flue gas velocity or a nozzle can be fitted to the chimney to take up the excessive draught by providing for increased efflux velocity. Downwash of gases and inversion can occur at low velocities and a minimum efflux velocity of 7.5 m/s will obviate these problems in general. Such a velocity may not be possible on natural draught plants of small size with non-pressurised combustion chambers as the above calculations will demonstrate. In such cases the maximum practicable efflux velocity should be sought.

Where high velocities are required, or where the flue run gives high resistance, the use of increased forced draught fan power and/or induced draught fans must be considered. The calculations are performed in a similar manner to ascertain the fan duties required to overcome the flue system resistances involved at the selected velocity.

The resistance to gas flow on low fire should be assessed and compared with the available chimney draught. There may be excessive suction at the boiler outlet on low fire resulting in the need for control dampers/draught controllers where the values fall outside the manufacturers' stated limits.

Under high velocity flue conditions, the flues and chimneys will probably be under pressurised conditions. Extra care must be taken in flue/chimney construction where such running conditions are required and it is not good practice to pressurise flues/chimneys of brick construction.

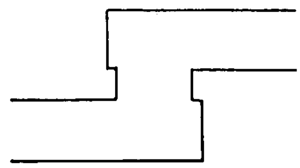
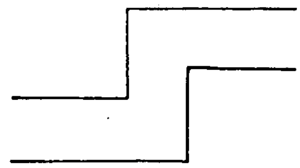
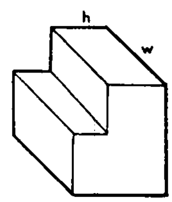
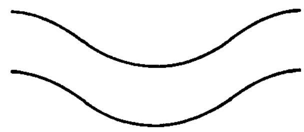
**Table B13.10.** Friction factors for mean gas temperatures of 180 to 340°C.

| Flue diameter (mm) | Mean gas velocity (m/s) | Smooth concrete or welded steel | Riveted steel or smooth cement pargeting | Brick or rough cement pargeting |
|---|---|---|---|---|
| 150 | 1·5 | 0·0054 | 0·0069 | 0·0190 |
| | 3·0 | 0·0048 | 0·0066 | 0·0187 |
| | 4·5 | 0·0045 | 0·0064 | 0·0185 |
| | 6·0 and over | 0·0044 | 0·0063 | 0·0180 |
| 230 | 1·5 | 0·0047 | 0·0058 | 0·0180 |
| | 3·0 | 0·0043 | 0·0057 | 0·0176 |
| | 4·5 | 0·0040 | 0·0056 | 0·0171 |
| | 6·0 and over | 0·0039 | 0·0055 | 0·0168 |
| 305 | 1·5 | 0·0042 | 0·0054 | 0·0146 |
| | 3·0 | 0·0039 | 0·0051 | 0·0145 |
| | 4·5 | 0·0037 | 0·0050 | 0·0144 |
| | 6·0 and over | 0·0036 | 0·0049 | 0·0143 |
| 355 | 1·5 | 0·0040 | 0·0050 | 0·0138 |
| | 3·0 | 0·0037 | 0·0049 | 0·0135 |
| | 4·5 | 0·0035 | 0·0048 | 0·0130 |
| | 6·0 and over | 0·0034 | 0·0046 | 0·0130 |
| 460 | 1·5 | 0·0037 | 0·0047 | 0·0130 |
| | 3·0 | 0·0035 | 0·0045 | 0·0128 |
| | 4·5 | 0·0033 | 0·0044 | 0·0127 |
| | 6·0 and over | 0·0032 | 0·0043 | 0·0127 |
| 610 | 1·5 | 0·0036 | 0·0043 | 0·0125 |
| | 3·0 | 0·0033 | 0·0040 | 0·0123 |
| | 4·5 | 0·0031 | 0·0039 | 0·0120 |
| | 6·0 and over | 0·0030 | 0·0038 | 0·0119 |
| 1220 | 1·5 | 0·0030 | 0·0035 | 0·0111 |
| | 3·0 | 0·0027 | 0·0034 | 0·0110 |
| | 4·5 | 0·0026 | 0·0033 | 0·0109 |
| | 6·0 and over | 0·0025 | 0·0032 | 0·0108 |
| 1830 | 1·5 | 0·0027 | 0·0031 | 0·0101 |
| | 3·0 | 0·0025 | 0·0029 | 0·0098 |
| | 4·5 | 0·0024 | 0·0028 | 0·0097 |
| | 6·0 and over | 0·0023 | 0·0027 | 0·0095 |

*Notes:*
For other gas velocities and flue diameters interpolate. For constructions other than those shown, select the figure on the basis of similarity of the interior finish of the flue.

Flue areas are based on circular sections. Where other than cicular sections are used, the 'effective' area is taken as that of the ellipse or circle that can be enclosed within the actual rectangle or square. Table B13.8 gives these values.

**Table B13.11.** Values for pressure loss factors for flue.

| Flue fitting | ζ |
|---|---|
|  | 4 to 5 |
|  | 3 |
|  | 2 to 2·5 |
|   | $\frac{w}{h}$ / ζ: 0·5 / 2·0; 1·0 / 1·5; 2·0 / 1·0; 3·0 / 0·8; 5·0 / 0·6 |
|  | 0·4 (if radius of offset is over ten times diameter, ζ =0) |

*Note:*
The above details are peculiar to flues. For conventional bends etc., see Table C4.35.

## Examples

The following examples are worked out at length to demonstrate the principles and procedures detailed above.

*Example 1*

Find the economic chimney dimensions (for a mild steel insulated construction) for an oil-fired shell type package boiler plant consisting of 2 × 3 MW boilers and 1 × 1·5 MW boiler fired with Class F, 30 centistoke fuel oil (sulphur content 3%) at 80% thermal efficiency. The area involved is category 'D' and the building dimensions are 15 m high by 60 m long. Individual flues/chimneys are required as shown in Fig. B13.23. The calorific value of the fuel is 42 MJ/kg.

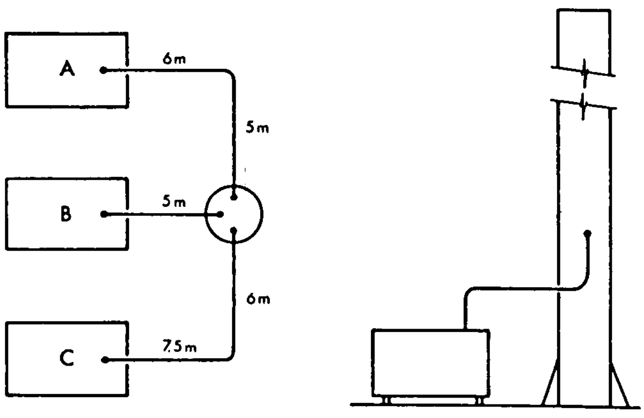



**Fig. B13.23.** Diagram for examples.

*Solution*

From equation B13.7, the maximum fuel burning rate for the plant is 0.22 kg/s.

Maximum sulphur dioxide emission, from equation B13.8, is 13.2 g/s. Using this sulphur dioxide emission the chimney height, from Fig. B13.18, is 27 m. Since the sulphur content of the fuel is greater than 2%, add 10% to this height. The result, 30 m, is less than 2½ times the building height, therefore the corrected chimney height, from equation B13.10, must be calculated (i.e. 31 m).

Assuming a flue gas temperature of 260°C, the chimney draught available (in summer), from Fig. B13.22, is 165 Pa.

Packaged boilers are used, with a chimney height of over 30 m, therefore using Table B13.7, select a trial flue gas velocity of 12 m/s. The flue gas volumes to be handled, from Table B13.12, are 30 $m^3$/kg of fuel burned. Hence, the volume of gases for each 3 MW boiler is 2.6 $m^3$/s and for the 1.5 MW boiler is 1.3 $m^3$/s.

The corresponding flue areas are, from equation B13.12, for the 3 MW boilers 0.22 $m^2$ and for the 1.5 MW boiler 0.11 $m^2$. These correspond to flue diameters of say 0.5 m and 0.4 m.

Considering boiler A and its flue/chimney system, the overall resistance to gas flow, from equations B13.13 to 15, is 225 Pa. The losses from the system comprise 2 × 90° bends ($\xi$ =2), 1 × 45° chimney entry ($\xi$ =0.5) and the chimney exit loss ($\xi$ =1).

The available chimney draught is 165 Pa, so with the conditions taken there would be a positive pressure of 60 Pa at the boiler outlet. It may be possible for the boiler manufacturer to increase the forced draught fan power to accommodate this pressure but if a balanced (zero) draught is required at the boiler outlet the above calculations must be repeated, using a lower velocity, until the desired conditions are achieved.

Considering boiler B, which has a shorter flue and only one 90° bend, the resistance for a given flue gas velocity is less (165 Pa) and this system would work with a balance at the boiler flue outlet. In the case of boiler C the resistance is 250 Pa, so the same comments apply as for boiler A.

*Example 2*

Assume the same boiler plant as for Example 1, but boilers A and B feed into a common chimney while boiler C has its own chimney.

*Solution*

The chimney heights are the same as before, i.e. 31 m. Boiler C is calculated as before.

In the case of boilers A and B feeding into one chimney, there are various considerations. To avoid downwash at low load (say ½ load on one boiler), the gas efflux velocity should not drop below 6 m/s, i.e. with both boilers at high fire this relates to an efflux velocity of 24 m/s. The frictional resistance due to this high velocity will impose a high pressure at the boiler outlets and with one boiler only working, gases may be forced into the idle boiler causing corrosion etc.

A more realistic target is an efflux velocity of 18 m/s at full load, which would give reasonable conditions for all but very light loads. This efflux velocity can be obtained by a nozzle at the chimney outlet, thus keeping the flue-chimney velocities to a reasonable figure to limit friction losses.

Proceeding as in Example 1 gives the same flue gas volumes per boiler at 2.6 $m^3$/s, the same trial velocity at 12 m/s and the same flue area per boiler at 0.22$m^2$, which gives a diameter of 0.5 m. The chimney area is calculated for the total volume of flue gases of 5.2 $m^3$/s and gives a diameter of 0.8 m.

The frictional resistance in the flues and chimney is calculated from equations B13.13 to 15 to give 215 Pa.

**Table B13.12.** Flue gas conditions at full boiler rating.

| Type of boiler | Coal fired; calorific value = 25·4 MJ/kg | | Oil fired; 'D' fuel-calorific value = 42·7 MJ/kg | | Natural gas; calorific value = 40·0 MJ/$^{3*}$ | | | |
|---|---|---|---|---|---|---|---|---|
| | | | | | Diverter† | | No diverter | |
| | Flue gas temperature (°C) | Flue gas volume per kg of fuel ($m^3$/kg) | Flue gas temperature (°C) | Flue gas volume per kg of fuel ($m^3$/kg) | Flue gas temperature (°C) | Flue gas volume per $m^3$ of fuel ($m^3/m^3$) | Flue gas temperature (°C) | Flue gas volume per $m^3$ of fuel ($m^3/m^3$) |
| Lancashire + economizer .. .. .. | 290 | 25·0 | 290 | 31·5 | — | — | 290 | 25·0 |
| Three passeconomic .. .. .. | 260 | 20·5 | 260 | 30·0 | — | — | 270 | 24·0 |
| Packaged economic .. .. .. | 260 | 20·5 | 260 | 30·0 | — | — | 230 | 22·5 |
| Flash steam generator .. .. .. | — | — | 290 | 31·5 | — | — | 260 | 23·5 |
| Vertical fire tube .. .. .. .. | 370 | 28·0 | 370 | 37·0 | — | — | 370 | 28·5 |
| Vertical cross tube .. .. .. .. | 530 | 35·0 | 530 | 45·0 | — | — | 540 | 37·0 |
| Small cast iron sectional .. .. .. | 400 | 29·5 | 370 | 37·0 | 95 | 34·0 | 350 | 27·7 |
| Large cast iron sectional .. .. .. | 370 | 28·0 | 320 | 33·0 | 95 | 34·0 | 300 | 24·5 |
| High efficiency cast iron sectional .. .. | 260 | 20·5 | 260 | 30·0 | 95 | 34·0 | 260 | 23·5 |
| Water tube + economizer .. .. .. | 260 | 20·5 | 260 | 30·0 | — | — | 260 | 23·5 |

*For town gas (calorific value = 16–20 MJ/$m^3$) the values of the flue gas volume are approximately halved.
†The flue gas temperature quoted is the average secondary flue gas temperature.

The available chimney draught is 165 Pa which results in a positive pressure of 50 Pa at the boiler outlets. This value only relates to a chimney velocity of 12 m/s, but a full load efflux velocity of 18 m/s is required.

This can be achieved by means of a nozzle on the chimney outlet, though with the addition of the equivalent velocity pressure to the resistance. The velocity pressure at 12 m/s is 47 Pa, the velocity pressure at 18 m/s is 107 Pa (from Table B13.9) hence the net additional pressure is 60 Pa. The overall resistance, therefore, is 275 Pa and the pressure at the boiler outlet is 110 Pa. On low load, the resistance to flow varies as the square of the volume. Thus, with one boiler on half fire, the resistance is given by:

$$DP = 275 \times (½)^2 \simeq 70 \text{ Pa}$$

Consequently, there is a negative pressure (suction) at the boiler outlet of 95 Pa.

This shows that from high to low load the boiler outlet is subject to a pressure variation from, say, –100 Pa to +110 Pa, so the boiler/burner unit must be checked as being capable of this duty. Modulating dampers may be necessary to produce more constant conditions.

More careful analysis enables the pressure condition to be ascertained at the flue's junction prior to chimney entry under various conditions. If excess pressure occurs with one boiler working, the sizes should be revised to prevent gases passing through the idle boiler. This example shows the merit in arranging one flue and chimney system per boiler, as in Example 1.

*Example 3*

Assume the same system and conditions as in Example 1, but with the boilers fired by natural gas (calorific value 40 MJ/m$^3$).

*Solution*

To give 7.5 MW at 80% boiler efficiency, the gas flow rate is 0.23 m$^3$/s hence, from Fig. B13.19, the chimney must be 2.5 m taller than the building, giving an overall height of 17.5 m. For a flue gas temperature of 260°C, the available chimney draught is 90 Pa, from Fig. B13.22. Table B13.7 gives a trial flue gas velocity of 7.5 m/s and from Table B13.12 the flue gas volume flow rate for the 3 MW boiler is 2.1 m$^3$/s and for the 1.5 MW boiler is 1.05 m$^3$/s. The corresponding flue areas are 0.28 m$^2$ and 0.14 m$^2$, from equation B13.12, giving flue diameters of 0.6 m and 0.4 m respectively.

For boiler A, the frictional resistance, given by equation B13.13 is 14 Pa. The loss due to 2 × 90° bends, 1 × 45° chimney entry and the chimney exit is 62 Pa, from equation B13.14. Hence, the overall resistance to gas flow is 76 Pa.

The available draught is 90 Pa, so the sizes are adequate to give 7.5 m/s flue gas velocity and provide a suction at the boiler outlet.

Similar reasoning can be used to determine the other flue sizes.

**Draught Production Equipment**

The draught necessary to move flue gases through the flue/chimney sustem and discharge them at a suitable velocity under specified firing rates, can be produced in several ways.

*Natural Draught Systems*

In these applications the chimney provides the following facilities by virtue of its height and the temperature of the gases within it:

(*a*) A specific suction condition in the boiler combustion chamber.

(*b*) Overcomes the flue and chimney resistance to gas flow, providing a minimum efflux velocity of gases at the chimney terminal.

(*c*) Overcomes boiler gas-side resistance to gas flow.

Typical plants which normally employ the natural draught system are standard open bottom cast-iron section type boilers fitted with oil or gas burners.

Due to the natural buoyancy effects employed, it is obvious that flue gas velocities must be relatively low in order to reduce system resistances to a practical level where chimneys are not of excessive height.

*Forced Draught Systems*

The firing equipment is fitted with a forced draught fan which, in the case of oil or gas burners, provides the necessary combustion air, and overcomes the burner air register resistance and the boiler resistance to gas flow. The natural chimney draught required in these cases has to overcome less overall resistance than in the natural draught case and flue gas velocities can often be increased for the same chimney height. Examples of this type are oil or gas fired packaged shell type boilers or air heaters, of a construction which permits pressurised combustion chambers to be used.

*Induced Draught Systems*

A fan is fitted at the boiler outlet to cater for the resistance requirements of the firing equipment and the boiler but also, in certain instances, the flue and chimney resistance when burning at maximum rating. Examples of this type are found in coal fired shell type boilers and certain water tube type boilers. Due to the fan power employed, draught is not dependent upon chimney buoyancy conditions and gas velocities can be increased depending upon the fan power requirement.

*Balanced Draught Systems*

A forced draught fan is fitted to provide all combustion air, and overcome the resistance of air registers, or fuel bed. An induced draught fan is fitted at the boiler outlet to take the hot gases and overcome resistance of the boiler and the flues and chimney system. It is usual to fit a draught controller which, by damper control on the fans, maintains the balanced 'zero' condition in the combustion chamber. Examples of this type are found in most coal fired shell and water tube boilers fitted with chain grate stokers, water tube boilers, oil and gas fired. Due to the fan power employed high velocities can be used in the flue system, which again is not dependent upon chimney height.

*Fan Diluted Flues*

This system has been specifically developed to overcome the problems of fluing gas appliances in ground floor shop premises in mixed developments of offices, shops and flats. Fresh air is drawn in through a duct by a fan, mixed with the products of combustion, and finally discharged to the atmosphere with a carbon dioxide content of not more than 1%. Gas, with its extremely low sulphur content, is ideal for this method of fluing.

Fan dilution systems have been used extensively for launderettes, shops, restaurants, public houses, etc. Many local authorities allow the discharge to be made at low level, above a shop doorway for instance, or into well ventilated areas with living or office accommodation above. This method of fluing offers an excellent alternative where natural draught flues are not practical.

Ideally, the air inlet and discharge louvres should be positioned on the same wall or face of the building. If the louvres are likely to be subjected to strong wind forces, some shielding is recommended. A damper or butterfly valve is fitted near the diluent air inlet to balance the installation. Protected metal sheet can be used for ducting as flue temperatures with this system are low, typically 65°C.

*Balanced Flues*

Balanced flues are applied mainly to domestic gas fired appliances, but are available for low sulphur content fuels, e.g. kerosene. The balanced flue is of compact design. The appliance is of a room sealed construction and is sited adjacent to an outside wall. The air for combustion is taken from outside of the building adjacent to the point at which the products of combustion are discharged using a common balanced flue terminal. The close proximity of air inlet and combustion products outlet makes the balanced flue terminal relatively insensitive to wind conditions and location.

At present, boilers and heaters are available only in the lower output range but special designs are possible for larger outputs. Fan assistance has been applied to balanced flue systems. This leads to a reduction of the flue assembly and permits siting of the boiler or air heater remote from an external wall. Since the balanced flue terminals are on outside walls and not generally above roof level, reference should be made to any relevant regulations or local byelaws which apply to terminal siting.

*Se-ducts and U-ducts*

These ducts increase the scope of application of room sealed gas fired appliances in multi-storey dwellings and permit the use of one flue to serve all appliances.

*Branched Flue Systems*

The branched flue system for gas appliances, sometimes called the shunt system, is designed for venting appliances of the conventional flue type. It represents considerable space-saving over venting each appliance with an individual flue. For further information on conditions and sizing see BS 5440[19].

**Chimney Linings**

The chimney construction should ideally be such that the linings provided give the following facilities:

(*a*) Provide sufficient insulation capability to maintain inner skin temperatures above the acid dew-point during normal running operations.

(*b*) Be chemically resistant to acids and flue gas deposits generally.

(*c*) Be physically capable of resisting absorbtion of moisture and its re-evaporation.

(*d*) Able to withstand fairly rapid internal gas temperature changes.

(*e*) Have small thermal capacity to limit heat up time.

(*f*) Be cheap in both installation and subsequent inspection and replacement.

*Steel Construction*

Steel chimneys are either of single or multi-flue construction, the outer windshield being designed to cater for the required wind pressures under either guyed or self-supporting design conditions. The structural requirements are covered by BS 4076[20].

With single flue construction a simple method of insulation consists of applying externally a cladding of 1.6 mm polished aluminium sheet located 6 mm from the outer mild steel chimney surface by means of asbestos spacers at 1.2 m intervals. This provides a 6 mm stagnant air space for insulation, assisted by the reflectivity of the polished aluminium.

With high sulphur fuels and chimneys having a gas volume turndown of more than 2.5:1 with modulating or two position firing equipment, this insulation is insufficient for chimney heights above 10-12 m. A mineral wool insulation at least 50 mm thick should be substituted for the 6 mm air space. With multi-flue construction the inner flues are placed within a windshield structurally calculated for wind pressures etc. as before. The internal flues are insulated with either mineral wool, or the whole space around the flues filled with a loose insulation which can be pumped into place if necessary. Thermal expansion problems must be considered in the design and provision made for replacing any one flue at a future date.

A similar mild steel multiple flue system can be installed within a concrete structural outer shell, again providing facilities for subsequent replacement.

*Brick Construction*

Brick flues/chimneys should always be lined internally For solid or liquid fuels the lining may be gunned solid insulation refractory or diatomaceous earth type insulation. The insulation standard should not be less than the equivalent of 115 mm thickness of diatomaceous earth for flue gas temperatures up to 315°C.

Where flue gas conditions dictate (e.g. low temperature, high sulphur and moisture) an acid resisting brick inner lining, backed by a lining of insulation material, can be used.

Careful attention must be paid to the lining construction and the type of jointing mortar used otherwise flue gases will leak through behind the lining and set up corrosive conditions.

The effect of pressurised operation on these linings is questioned and for general operation such chimneys should be operated under suction or balanced draught conditions. They must be carefully designed by a competent structural engineer who is aware of the combined physical/chemical effects involved.

*Concrete Construction*

Similar comments to those on brick construction apply, but the insulation thickness should generally not be less than the equivalent of 150 mm diatomaceous earth in order to limit the interface concrete temperature to a maximum of 50°C under normal boiler plant operating conditions.

*Ventilated Chimneys*

Here a ventilated air space is situated between the inner lining and outer chimney shell. The construction should not be used in general with high sulphur fuels due to the cooling effect created and the consequent danger of acid dewpoint and acid smut emission.

*Plastic Flues/Chimneys*

Plastic and resin bonded materials, being impervious to low temperature corrosion, are now being used. The materials have high thermal expansion and conductivity so that insulation is necessary to limit acid smut emission. The upper temperature limit is around 260°C so care must be taken both in the construction and the selection for flue gas temperature conditions.

In all cases the manufacturers must be consulted as to the suitability of the particular construction for the required application.

**Mechanical Dust Collectors**

The many designs of mechanical dust collector that are available may be grouped into three types:

(*a*) low efficiency collectors

(*b*) medium efficiency collectors

(*c*) high efficiency collectors.

*Low Efficiency Collectors*

Settling chambers, simple deflection collectors and simple swirl flow types fall into this category. Draught loss through these collectors is in general of the order 50 to 75 Pa.

A typical simple swirl flow collector is depicted in Fig. B13.24. Spin is imparted to the dust laden gases by fixed vanes which has the effect of centrifuging dust towards the walls of the unit. An offtake in the circumference allows the dust to escape into a dead space chamber where the dust falls to a hopper. The small amount of carrier gas which is taken off with the dust is bled back into the main stream. This type of collector is a basic form of uniflow cyclone and can be designed for vertical mounting.

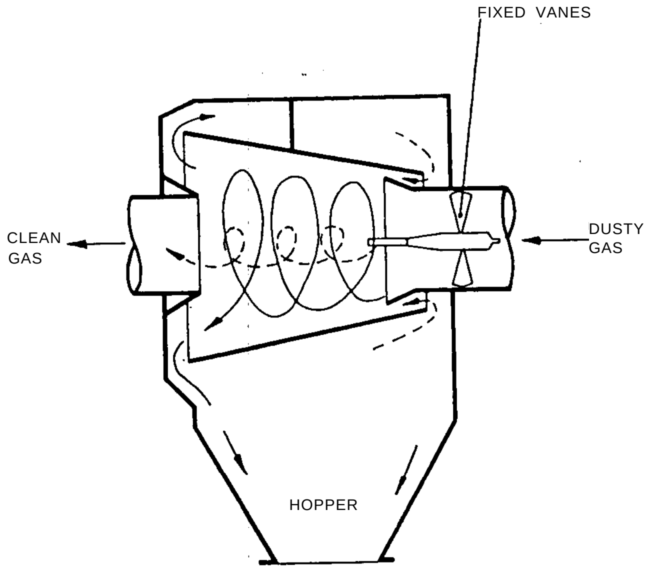



**Fig. B13.24.** Simple swirl collector.

*Medium Efficiency Collectors*

Simple cyclones, uniflow cyclones with effective secondary separating systems, scroll collectors and complicated deflection collectors such as louvred cone type fall into this group. Draught loss across these collectors will be of the order 250 to 500 Pa.

A diagrammatic view of a scroll collector is shown in Fig. B13.25. Dust laden gases are drawn into the fan. Grit/dust is thrown to the side of the casing and bled off with a small proportion of gas to the secondary separator, the cleaned gas passing to the chimney. Final extraction of the grit/dust takes place in a simple cyclone, the carrier gas being recirculated back to the fan inlet.

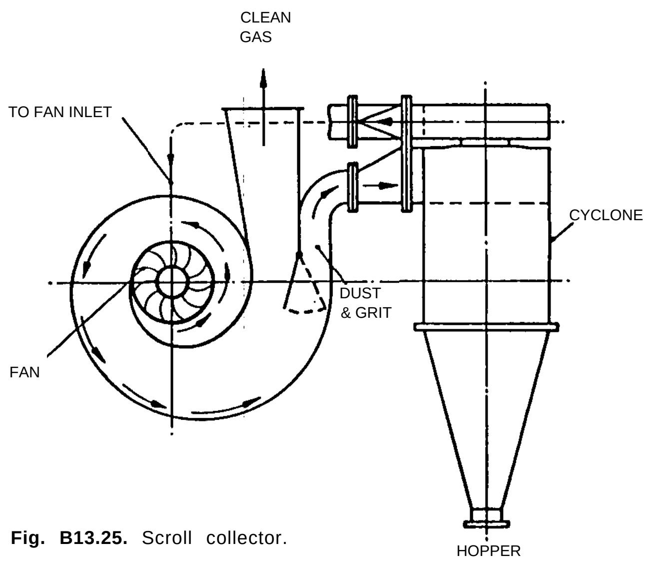



**Fig. B13.25.** Scroll collector.

The louvred cone collector is illustrated in Fig. B12.26. Dust laden gases pass at high velocity into a slotted cone where cleaned gases pass through the slots and hence to the chimney. Grit/dust remains within the cone and, together with a small amount of gases, passes to a secondary separator. As with the scroll collector this takes place in a simple cyclone.

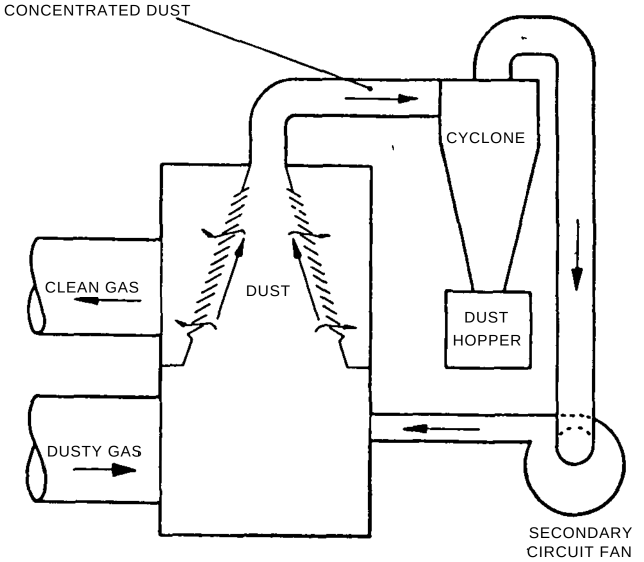



**Fig. B13.26.** Louvred cone collector.

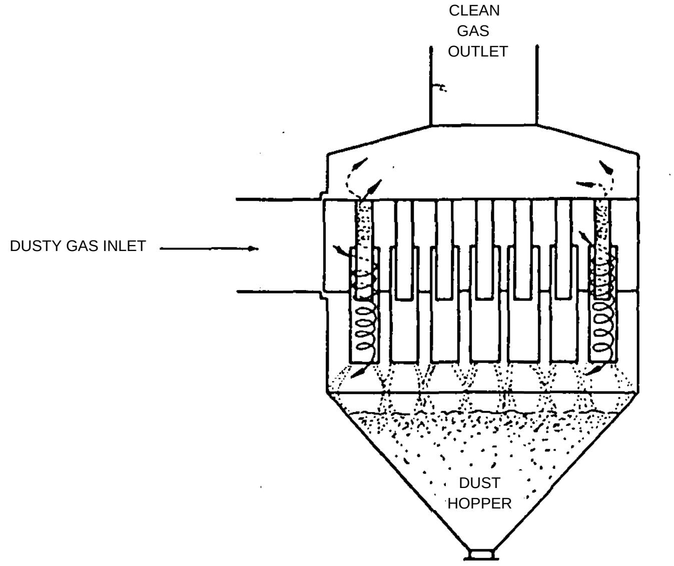



**Fig. B12.27.** Multi-cyclone collector.

*High Efficiency Collectors*

Many forms of collector fall into this category but most, if not all, employ multiple units of small return flow or uniflow cyclones set in parallel. Draught losses through these collectors can vary between 250 and 1000 Pa.

Fig. B13.27 shows a diagrammatic arrangement of a design in which vane entry, return flow cyclones are used. Dust laden gases enter the collector and impinge on the specially shaped vane entries of the individual cyclone outer walls. This centrifuges grit/dust to the outer walls and hence into the collecting hopper. Clean gas returns through the central tubes to pass into the exit chamber and thence to the chimney.

*Efficiency and Selection of Collectors*

The relative efficiency of the different groups in removing particles of various size from a gas is indicated in Fig. B13.28. A comparison of the overall efficiency that can be expected under specified conditions is given in Table B13.13.

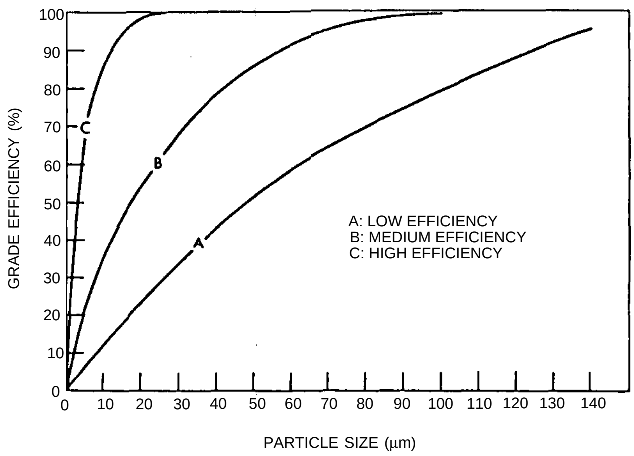



**Fig. B13.28.** Efficiency against particle size for dust collectors.

**Table B13.13.** Comparison of collector efficiencies.

| Dust size and content | | | Typical low efficiency collector | | Typical medium efficiency collector | | Typical high efficiency collector | |
|---|---|---|---|---|---|---|---|---|
| Dust size (mm) | Content by weight (%) | Concentration* before collector (g/m $^3$) | Grade efficiency (%) | Emission after collector (g/m $^3$) | Grade efficiency (%) | Emission after collector (g/m$^3$) | Grade efficiency (%) | Emission after collector (g/m$^3$) |
| Under 10 | 10 | 0·1 | 6 | 0·09 | 18 | 0·082 | 43 | 0·057 |
| 10–20 | 10 | 0·1 | 20 | 0·08 | 46 | 0·054 | 94 | 0·006 |
| Under 20 | 20 | 0·2 | 15 | 0·17 | 32 | 0·136 | 68 | 0·063 |
| Over 20 | 80 | 0·8 | 68 | 0·26 | 89 | 0·084 | 99 | 0·002 |
| 20–50 | 20 | 0·2 | 40 | 0·12 | 71 | 0·058 | 99 | 0·002 |
| Over 50 | 60 | 0·6 | 77 | 0·14 | 96 | 0·026 | 100 | — |
| 50–76 | 20 | 0·2 | 60 | 0·08 | 91 | 0·018 | 100 | — |
| Over 76 | 40 | 0·4 | 85 | 0·06 | 98 | 0·008 | 100 | — |
| Total | 100 | 1·0 | 57 | 0·43 | 78 | 0·220 | 93 | 0·065 |

*Based on an original concentration of 1.0 g/m$^3$ at 0°C and 101.3 kPa.

## CORROSION CONTROL

### Mechanisms of Corrosion

The most common cause of corrosion in boilers and chimneys is the presence of water vapour and oxides of sulphur, following combustion of fuels containing sulphur. The following considers corrosion in boilers and flues. A detailed treatment of corrosion mechanisms in general is given in Section B7.

When any fuel containing hydrogen and sulphur is burned. water vapour and sulphur dioxide ($SO_2$) are produced. A small proportion of the $SO_2$ is further oxidised to sulphur trioxide ($SO_3$) which immediately combines with water vapour to produce sulphuric acid. This will condense on any surface below the acid dew-point temperature giving rise to corrosion. The acid dew-point is the temperature at which the combustion gases become saturated with acid vapour and, when cooled without change in pressure, condense as a mist.

The acid dew-point varies with the type of acid and its concentration. Further cooling of the gases to the water dew-point may produce corrosive effects even more serious than those produced at higher (i.e. more concentrated) acid dew-points. During normal operation it is unlikely that the water dew-point (about 38°C) will be reached but this may occur for intermittently operated plant. When the method of operation is such as to reduce the water circulating temperatures to 38°C condensation is inevitable.

In addition to sulphur, other constituents such as chlorine and nitrogen react to give acidic gases which can combine with water vapour and thereby cause corrosion if allowed to condense on cooler metal surfaces.

In boilers which are shut-down, flue deposits become damp because of their hygroscopic nature and produce acid sulphates which are likely to cause corrosion. Acid corrosion is less likely to occur with coal rather than residual fuel oils, for the following reasons:

*(a)* The average sulphur content of coal is generally lower than that of residual fuel oils and about one tenth is retained in the ash.

*(b)* The hydrogen content of coal is lower than that of other fuels. Therefore the amount of water vapour produced during combustion is also lower.

*(c)* The small amounts of fly ash in the flue gases tend to absorb free $SO_2$ and thus reduce the production of corrosive acid.

The combination of less water vapour and lower levels of $SO_2$ means that lower gas temperatures may be used, resulting in a corresponding gain in plant efficiency. On large, well operated and maintained plant, the production of $SO_2$ may be minimised by controlling the excess oxygen in the combustion zone. However, precise control is necessary and this is unlikely to be achieved on small plants.

### Prevention of Boiler Corrosion

During boiler shut-down. e.g. during the summer months, all surfaces should be thoroughly cleaned and all partially burnt fuel and ashes removed. All doors and dampers should be left open to ensure that air is drawn through the boiler. Lime washing of all accessible surfaces may be beneficial and where good air circulation can be obtained, trays of a moisture absorbing material, such as quicklime, should be provided.

For plant in operation, the system should be designed so that the average boiler water temperature does not fall below about 50°C. This helps to ensure that the water dew-point is not exceeded. For details of control of boiler systems see Section B11 and CIBSE Applications Manual *Automatic Controls and their Implications for Systems Design*[21]. Under no circumstances should the boiler thermostat be used as a control thermostat to reduce the flow temperature in a heating system. Every effort should be made to ensure that good circulation is obtained in both the water and the gas sides of the boiler, avoiding cold spots.

### Prevention of Flue Corrosion

To minimise the risk of corrosion, the following points should be noted:

*(a)* Sufficient insulation should be provided to maintain inner skin temperature above the acid dew-point during normal operation.

*(b)* The flue or chimney lining should be chemically resistant to acids and flue gas deposits.

*(c)* The flue gas velocity must be sufficiently high to prevent precipitation of acids and deposits on internal flue linings.

This is dealt with in detail under *Chimneys and Flues.*

## COMBUSTION AND VENTILATION AIR

Combustion equipment must be provided with a supply of air adequate to ensure:

*(a)* Complete combustion of fuel and safe operation of the equipment at its rated output.

*(b)* Ventilation of the enclosure in which the appliance is installed to prevent overheating of the enclosure or the equipment.

The air required for these purposes should be introduced into the space through inlets sited to minimise the discomfort caused to the occupants and in a position where they are unlikely to be blocked accidently (either on the inside or outside of the building), closed deliberately or rendered inoperative by prevailing winds.

It may be necessary to pre-heat the air and introduce it into the space through diffusers, see Section B3. Alternatively, the air may be ducted from an outside source directly to the appliance. The air should not be drawn from areas where it may be contaminated by odours, corrosive or flammable fumes or the products of combustion.

The combustion and ventilation air provided for combustion equipment should not be used for other purposes such as make-up air for extract ventilation systems.

It is essential that the products of combustion are conveyed to a suitable discharge point where they will be safely dispersed. For appliances incorporating a flue, the ambient pressure inside the space enclosing the appliance must not fall below the outside pressure.

Whilst not requiring combustion air, equipment rooms which accommodate heat exchangers, electric thermal storage equipment, electrode boilers, pump rooms, service voids etc., nevertheless require sufficient ventilation air to remove excess heat in order to maintain a comfortable environment for the occupants and to protect the equipment.

**Design Recommendations**

Statutory requirements for combustion and ventilation air for heat producing appliances are contained in The Building Regulations 1985. Practical guidance on methods of compliance for solid fuel and oil burning appliances with rated *outputs* up to 45 kW and for gas burning appliances with rated *inputs* up to 60 kW are given in Approved Document J[22]. The following section provides design guidance for larger installations.

*Boiler House and Plant Room Environments*

Ventilation air requirements must take into account the heat emission from the appliance itself plus that from the associated pipes, pumps, flues, electric motors etc. For boiler houses and plant rooms having large glazed areas the effects of solar heat gains must also be considered. Uninsulated parts of the equipment will produce localised radiant heating and consideration should be given to the protection of both occupants and equipment from the effects of radiant heat.

Views on acceptable temperatures for boiler houses and plant rooms vary widely. In general, for the UK, the upper design temperature should not exceed 32°C at ceiling level and should be between 21 and 27°C in the working area. In countries where the outside temperature exceeds 32°C somewhat higher limits will be necessary, perhaps as high as 50°C at ceiling level. The upper design limits should be set bearing in mind the comfort of operating and maintenance staff and the acceptable temperature limits for installed electrical equipment. In some circumstances, ventilation alone may not be adequate and mechanical cooling may be necessary.

*Combustion Air Requirements*

Combustion air requirements are given in Fig. B13.29 for single boilers in the range of 45 to 1,500 kW and Fig. B13.30 for single boilers in the range of 1.5 to 6.0 MW. These figures are based on average data over each of the two ranges of boiler output. The main differences are due to the fuels used and the efficiencies obtained. For this reason, the two graphs give different values at the changeover between the stated ranges. For boilers of rated output 1,500 kW, Fig. B13.29 should be used.

Note that these data apply to single boilers. If an installation consists of three boilers, each rated at 1,000 kW, the combustion air requirement should be determined for each boiler separately using Fig. B13.29 and these added together to give the total requirement.

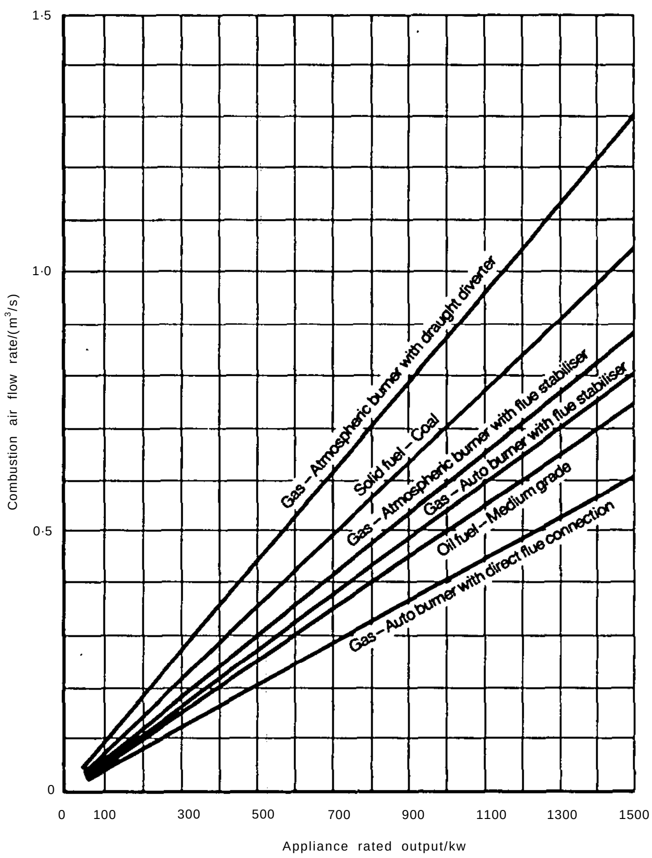



**Fig. B13.29.** Combustion air requirements for single appliances with outputs in the range of 45 to 1500 kW.

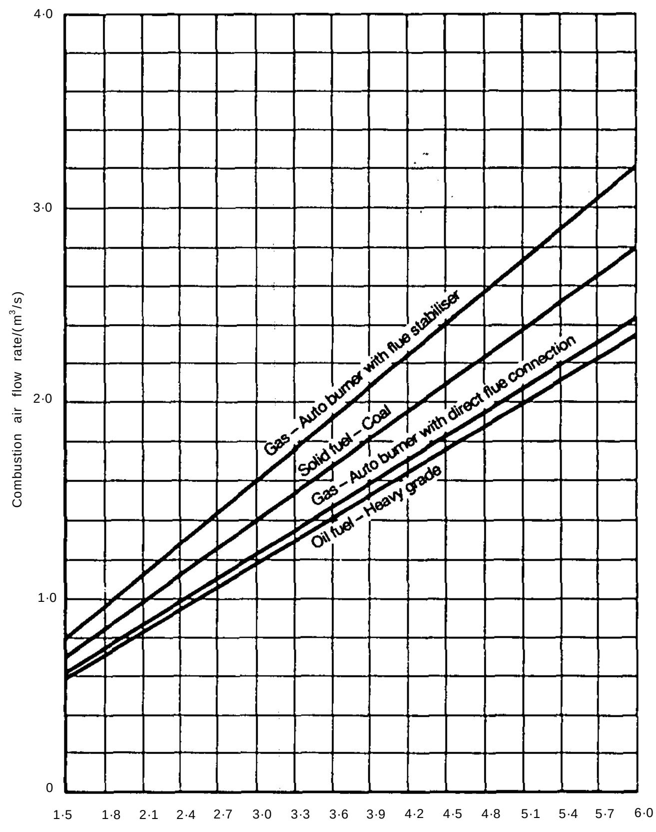



**Fig. B13.30.** Combustion air requirements for single appliances with outputs in the range of 1.5 to 6.0 MW.

**Estimation of Ventilation Air Requirements**

A guide to seasonal ventilation air requirements is given in Figs. B13.31 and 32. These are based on assumed net heat gains (due to heat loss from plant, solar gains etc.) of 5% of the rated output of the appliance during winter, spring and autumn operation, and 10% during summer operation. The seasonal inlet and outlet temperatures are based on typical data for the UK, see Section A2. These figures include a small allowance for the cooling effect of the combustion air.

For multiple appliance installations, the total requirement may be obtained by adding together the requirements estimated separately for each appliance.

For a more accurate assessment of the ventilation air requirement for a particular installation the following method should be used.

**Detailed Assessment of Ventilation Air Requirements**

The required quantity of ventilation air is obtained from:

$$Q_v = \frac{H}{\varrho\, C\, (t_{bh} - t_o)} \quad \ldots \quad \ldots \quad \ldots \qquad \text{B13.6}$$

where:

| | | | |
|---|---|---|---|
| $Q_v$ | = | ventilation air flow rate .. .. | $m^3/s$ |
| $\varrho$ | = | density of air (= 1.2) .. .. | $kg/m^3$ |
| $C$ | = | specific heat capacity of air (= 1.02) .. .. .. .. | kJ/kg K |
| $t_{bh}$ | = | boiler house ceiling design temperature .. .. .. | °C |
| $t_o$ | = | outside (supply) air temperature | °C |
| $H$ | = | heat gain to boiler house .. | kW |

The heat gain to the boiler room is obtained from:

$$H = H_p + H_s - H_f - H_c \quad \ldots \quad \ldots \qquad \text{B13.17}$$

where:

| | | | |
|---|---|---|---|
| $H_p$ | = | heat gain from hot surfaces .. | kW |
| $H_s$ | = | solar heat gain .. .. .. | kW |
| $H_f$ | = | fabric heat loss .. .. .. | kW |
| $H_c$ | = | cooling effect of combustion air | kW |

*Supply of Combustion and Ventilation Air*

Having determined the quantities of air required for combustion and ventilation the particular scheme must be examined to determine whether a mechanical ventilation system will be necessary to achieve the required total air displacement. Calculation methods for natural ventilation are given in Sections A4 and B3; mechanical ventilation is dealt with in Sections B2 and B3.

In small installations, i.e. up to 45 kW, or those where induced draught flue systems are employed, the combustion air alone may be sufficient to meet the ventilation requirements. However, in all cases, it is necessary to provide both high and low level openings for general ventilation in addition to the low level openings for the combustion air.

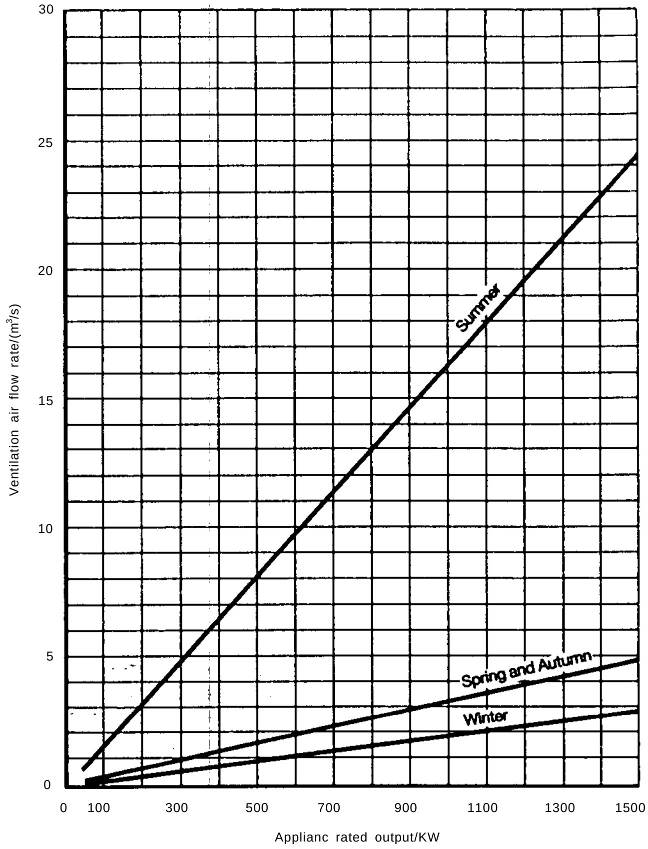



**Fig. B13.31.** Ventilation air requirements for single appliances with outputs in the range 45 to 1 500 kW.

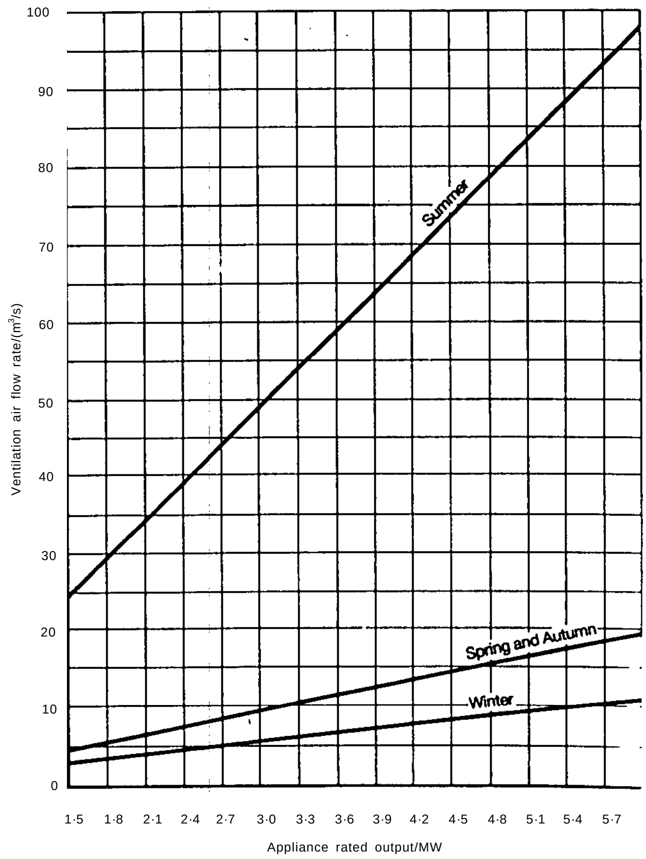



**Fig. B13.32.** Ventilation air requirements for single appliance with outputs in the range of 1.5 to 6.0 MW.

*Outside (Supply) Air Temperature*

The supply air temperature will depend on the season and the geographical location of the installation. For the UK, the following supply air temperatures are suggested:

*(a)* For plant operating during winter; 10°C.

*(b)* For plant operating during the autumn and spring; 19°C.

*(c)* For plant operating during the summer; 27°C.

The varying air volumes should be calculated for each season of operation and the inlet and outlet grilles, or fan sizes, should be chosen to cater for the largest seasonal air volume. An example calculation is given later.

In the UK, it should not be necessary to pre-heat or cool the supply air. However, if cooling is employed, the air should be introduced through diffusers to minimise discomfort, see Section B3.

*Combustion and Ventilation Air System Design*

Generally, small installations at or above ground level should have their combustion and ventilation air provided by natural means, employing both high and low level openings.

Basement, internal and large installations at or above ground level will usually require a combination of natural and mechanical ventilation. If the air flow route is difficult both supply and extract may require mechanical means.

Whether natural or mechanical, the system should be designed to provide a comfortable working environment, avoiding both horizontal and vertical temperature gradients. Inadequate ventilation at high level may result in ‘hot spots’ between beams and extract ventilators should be positioned to avoid this. Both inlet and outlet openings should be placed on opposite or adjacent sites of the building to reduce the effect of wind forces.

Air flow openings must be kept clear of all obstructions during normal operation of the plant. Shut-off dampers should not be fitted although, in some circumstances, ‘normally open’ automatic fire dampers may be required.

Where mechanical air supply is employed, electrical interlocks with the boiler plant should be provided to prevent damage in the event of failure of the supply fan(s).

*Natural Ventilation*

The sizing of openings for natural ventilation is dealt with in Section A4 and B3. Alternatively. the following equation may be used:

$$A = \frac{5.8Q}{[h\ (t_{bh} - t_o)]^{0.5}} \quad \text{.. .. .. B13.18}$$

where:

| | | | |
|---|---|---|---|
| $A$ | = | free area of opening .. .. | $m^2$ |
| $Q$ | = | air flow rate .. .. .. .. | $m^3/s$ |
| $h$ | = | height difference between centre lines of inlet and outlet grilles .. | m |

Solutions to this equations are given in Fig. B13.33. Both high and low level openings should be used; the high level (outlet) openings should be sized to cater for the total ventilating air quantity and the low level (supply) openings sized to cater for the total combined ventilating and combustion air quantity.

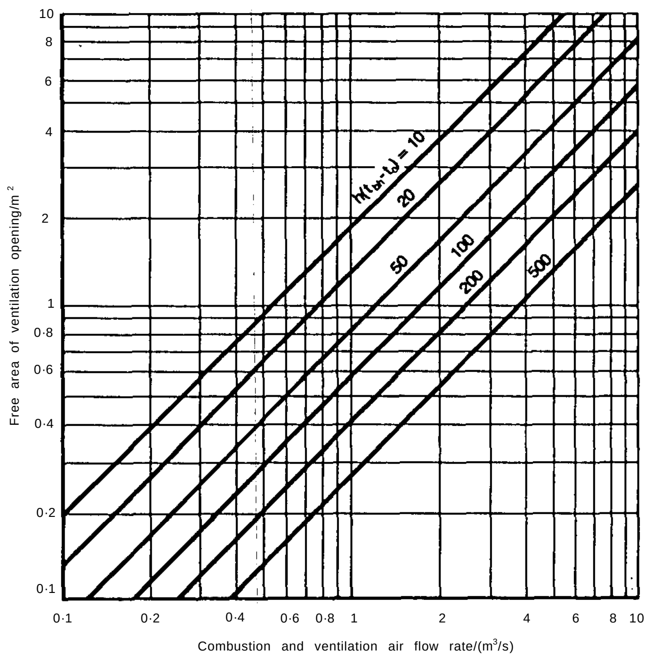



**Fig. B13.33.** Ventilation openings for natural ventilation.

*Combined Natural and Mechanical Ventilation*

A combined natural and mechanical ventilation system should allow for natural extract at high level, to take advantage of convective forces in the room, with mechanical supply at low level. The high level natural ventilators should be sized to cope with the total quantity of ventilation air, as above.

To prevent leakage of flue gases and to ensure that the flue draught is not impeded at any time, the air pressure in the boiler room must not exceed the prevailing outside pressure. Therefore, the fan duty should exceed the calculated total combined combustion and ventilation air quantity by at least 25%.

Fan powered inlets should be arranged to blow outside air into the space at a point where cross ventilation will ensure pick-up of heat without causing discomfort to the occupants.

*Full Mechanical Ventilation*

Where it is impractical to provide sufficient natural ventilation to remove the heat emitted by the plant, both mechanical supply and extract will be required.

The high level extract should be sized to cater for the total ventilating air quantity and the low level supply should exceed the total combined combustion and ventilating air quantity by at least 25%, as above.

*Mechanical Flue Draught Systems*

On large installations, i.e. above 3 MW, the heating equipment manufacturer should be consulted to ensure that adequate replacement air is available. A separate system, either natural or mechanical, may be required to ventilate the space.

*Fan Diluted Flue Systems*

On gas fired installations employing a fan diluted flue system, the diluent air must be drawn from outside the boiler room (heating chamber), preferably direct from outside, see Fig. B13.34.

On no account should the boiler room ventilation be combined with the flue dilution system due to the risk of flue gases entering the boiler room in the event of failure of the diluent air fan.

The ventilation and combustion air requirements are separate from the flue dilution air and should be calculated as above.

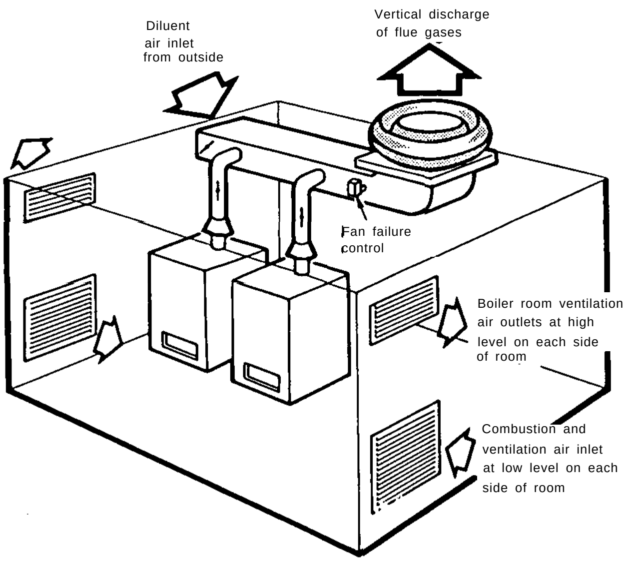



**Fig. B13.34.** Fan diluted flue system.

*Fire Protection*

Replacement air should not be drawn through pipe trenches or fuel service ducts. Where metal ducts penetrate walls and floors, effective sealing should be provided to confine the ventilation to the boiler room and to meet fire protection requirements. Penetration of fire barrier walls by ventilation ducts should be avoided if possible.

Fire dampers in ventilation ducts should be electrically interlocked with the boiler plant.

*Noise*

Care must be taken to prevent any noise generated in the boiler room emerging from natural or mechanical ventilation openings to the detriment of the surrounding environment. Particular care is necessary with mechanical flue draughts and fan diluted flue systems.

*Example*

Consider a small boiler house containing two 750 kW oil-fired heating boilers and one 100 kW oil-fired hws boiler. The total heat gain from these boilers and their associated pipes, pumps, storage cylinders etc. is determined for each season, along with the solar gains for the highest month for each season. For this example, the cooling effect of the combustion air is negligible. Hence, allowing for the heat losses through the structure the net seasonal heat gains are as follows:

| | |
|---|---|
| Winter: | 62.0 kW |
| Spring/Autumn: | 62.6 kW |
| Summer: | 10.0 kW |

The seasonal supply air and boiler room (ceiling level) temperatures are as follows:

| | *Winter* | *Spring Autumn* | *Summer* |
|---|---|---|---|
| $t_o$: | 10°C | 19°C | 27°C |
| $t_{bh}$: | 32°C | 32°C | 32°C |
| $(t_{bh} - t_o)$: | 22°C | 13°C | 5°C |

From equation B13.16, the ventilation flow rates are:

| | |
|---|---|
| Winter: | 2.3 $m^3/s$ |
| Spring/Autumn: | 3.9 $m^3/s$ |
| Summer: | 1.6 $m^3/s$ |

From Fig. B13.29, the combustion air requirements are:

Winter/Spring/Autumn:. (2×0.38) + 0.05 = 0.8 $m^3/s$
Summer (hws only): 0.05 $m^3/s$

Hence, combined combustion and ventilating air requirments are:

| | |
|---|---|
| Winter: | 3.1 $m^3/s$ |
| Spring/Autumn: | 4.7 $m^3/s$ |
| Summer: | 1.7 $m^3/s$ |

If natural ventilation is to be provided, assuming a height difference of 4 m between the centre lines of the high and low level grilles, using equation B13.18 the corresponding grille areas are:

| | *Inlet* | *Outlet* |
|---|---|---|
| Winter: | 1.9 $m^2$ | 1.4 $m^2$ |
| Spring/Autumn: | 3.8 $m^2$ | 3.1 $m^2$ |
| Summer: | 2.2 $m^2$ | 2.0 $m^2$ |

Taking the maximum sizes, the inlet should be 3.8 $m^2$ and the outlet 3.1 $m^2$.

For a combined natural and mechanical ventilation system, the outlet size would be as above. However, the supply fans would be sized to exceed this calculated supply rate by at least 25%. Therefore, in this example, a supply fan duty of 5.9 $m^3/s$ would be required.

For a fully mechanical ventilation system, the supply fan duty would be as above and the extract fan sized to displace a total 3.9 $m^3/s$.

## BOILER EFFICIENCY

### Efficiency of Combustion

For efficient combustion of the fuel, the supply of combustion air must be sufficient to ensure complete combustion of the fuel. However, any excess air will cool the flame which reduces efficiency.

Combustion air must be admitted at the correct time to ensure that the oxygen makes intimate contact with the combustible material in the fuel. The gases evolved must be maintained at a temperature at or above their ignition point until combustion is complete. Maximum combustion efficiency occurs with maximum gas temperature.

Adequate mixing of the gases is essential and the rate of expansion of the products of combustion must be sufficient to ensure complete combustion.

### Efficiency of Heat Transfer

The rate of heat transfer from the flame or flue gases must be as high as possible. If deposits of soot and carbon are permitted to build up on the boiler heating surfaces the rate of heat transfer is greatly reduced. On the water side of the boiler, heat transfer will be impeded by salt deposits and other scales, if allowed to accumulate. This may be avoided by suitable water treatment methods, see Section B7.

### Radiation Losses

Care must be taken to minimise the loss of heat from the outside surfaces of the boiler and its associated pipework by providing adequate thermal insulation.

### Stack Losses

The sensible heat loss resulting from excess air and its relation to the proportion of carbon dioxide in the flue gases must be considered. This is determined by the Siegert formula, see BS 845[23].

The formula is based on the net calorific value of the fuel which is related to gas temperatures and the proportion of carbon dioxide, thus:

$$q_f = \frac{K\ (t_2 - t_1)}{P} \quad \text{.. .. .. ..} \quad \text{B13.19}$$

where:

| | | | |
|---|---|---|---|
| $q_f$ | = | sensible heat losses in flue gas .. | % |
| $K$ | = | factor depending on fuel type | |
| $t_1$ | = | temperature or air entering boiler | °C |
| $t_2$ | = | temperature of flue gases leaving boiler .. .. .. .. | °C |
| $P$ | = | proportion of $CO_2$ in flue gases .. | % |

Values of the factor $K$ are given in Table B13.14.

The proportion of carbon dioxide varies with fuel type. Table B13.15 indicates both theoretical and target values for various fuels. Actual values of the carbon dioxide contained in the flue gases may be measured by various gas analysis techniques, see *Measurement and Analysis of Flue Gases.*

**Table B13.14.** Values of $K$ for various fuels.

| Fuel | Value of $K$ |
|---|---|
| Fuel oil | 0.56 |
| Natural gas | 0.38 |
| Anthracite | 0.68 |
| Bituminous coal | 0.63 |
| Coke | 0.70 |

**Table B13.15.** Theoretical and target values for carbon dioxide and oxygen contents of flue gases for various fuels.

| Fuel | Target $CO_2$ % | Theoretical $CO_2$ % | Target $O_2$ % |
|---|---|---|---|
| Fuel oil | 13 | 16.0 | 3.9 |
| Natural gas | 10 | 11.7 | 3.2 |
| Anthracite | 13 | 19..5 | 7.0 |
| Bituminous coal | 12 | 18.6 | 7.4 |
| Coke | 13 | 19.5 | 7.0 |
| Dry steam coal | 13 | 19.2 | 7.9 |

*Example*

Consider a boiler fuelled by bituminous coal. The temperature of the air entering the boiler is 22°C and the flue gas temperature is 230°C.

For a measured carbon dioxide proportion of 8% the Siegert formula gives:

$$q_f = \frac{0.63\ (230 - 22)}{8} = 16.4\%$$

If the proportion of carbon dioxide is increased to 13% the stack losses fall to 10.1% This shows that the stack losses may be reduced by reducing the excess air in the flue gases.

### Measurement and Analysis of Flue Gases

To maintain maximum combustion efficiency from boiler plant, the composition of the flue gases must be monitored. Normally, this analysis is limited to measuring the proportions of carbon dioxide and oxygen although, in some industrial applications, carbon monoxide is also monitored.

The flue gases from large boiler installations may be monitored continuously by means of permanent analysis and recording equipment. Automatic instruments for continuous sampling and analysis of flue gases are dealt with in BS 3048[24].

For smaller boiler plant, portable systems are used to provide spot checks. Such apparatus is an essential part of the service/commissioning engineer's equipment. The various types of apparatus and their methods of use are covered by BS 1756[25].

*Flue Gas Sampling*

The following points should be noted in order to minimise errors in flue gas measurement and analysis:

*(a)* Stratification may occur at low flow velocities so sampling should take place under turbulent conditions. However, bends should be avoided due to 'dead spots'.

*(b)* Leakage of air after combustion may cause a layer of cold dense air to stick to the flue wall.

*(c)* Variation in total gas flow during sampling may cause discrepancies in the measurements.

*(d)* The sample must be cooled to stop further reactions and to avoid damage to the measuring equipment. However, condensation due to cooling of the sample below its dew-point may cause blockages in the sampling system.

*(e)* The composition of the sample may change if delays occur between sampling and analysis. Such delays may be due to slow sampling methods or the need for large sample volumes.

*(f)* Other factors such as corrosion, blockage of sampling lines, air leakage etc.

### Boiler Efficiency by Direct Measurement

An alternative method of assessing boiler efficiency is by direct measurement of fuel use, feed water consumption and rate of steam production together with data on the combustion air temperature, steam pressure and steam dryness.

This method requires sophisticated and costly metering equipment to accurately measure and record these data. For reliable results, monitoring over a period of at least two hours is required and the various instruments must be calibrated prior to each test

## HEAT GENERATING EQUIPMENT

### Boilers

*Cast-iron Sectional Boilers*

Conventional sectional boilers are of the open bottom inverted horeshoe type. Cast-iron sections are connected together via nipples to form waterway passages: a boiler of any size, up to a maximum of 1.8 MW, can be made in this way. They require refractory combustion chambers and insulated/air or water cooled hearths designed to limit boiler house floor temperatures to a maximum 65°C. They also require adequate suction in the combustion chamber, provided by chimney or induced draught fan.

Designs are now produced with fully water cooled combustion chambers requiring no brickwork and these are independent of chimney draught to some extent. Such designs are at present taken up to ratings of approximately 4 MW and are suited to solid, liquid, or gaseous fuel firing. They are increasingly provided in matched set form where the boiler and its firing equipment have been tested and approved as a combination.

All such boilers can be installed on sites with limited access and can be extended to meet future requirements. They are suitable for LTHW systems and low pressure steam up to a maximum gauge pressure of 400 kPa. Certain units in spheroidal graphite iron can withstand a gauge pressure of 1 MPa.

*Mild Steel Rectangular Boilers*

These generally consist of a flame tube, water cooled, combustion chamber and one or two convection tube passes, built into a rectangular water walled external shell. They can be of the open bottom type, requiring suitable brickwork and hearth construction, or the water based type. They can be fired with solid, liquid or gaseous fuels and are available in ratings from 100 kW to 3 MW and gauge working pressures of 10 bar.

*Mild Steel Sectional Boilers*

Similar comments apply to these units as to cast-iron sectional boilers in general terms. Available in ratings up to 1.5 MW and a gauge working pressure of 450 kPa.

*Mild Steel Shell Boilers*

Such units are of several types, and the range extends from approximately 100 kW to 12 MW. The principal types are:

*(a)* Three- or four-pass construction, consisting of a furnace tube and two or three passes of convection tubes, They are designed with either a fully wet back, a semi-wet back or a dry back combustion chamber for use with steam or water. On the water units (heaters) certain makers fit an internal venturi type mixing device to reduce thermal stressing in the heater, by mixing cool return water with the hotter internal boiler water, and at the same time limit corrosion in the water dew-point temperature ranges.

*(b)* Two-pass units of similar construction to *(a)* above consisting of a furnace tube and one set of convection tubes.

*(c)* Units with annual cylinder construction in place of tubes, giving multi-pass arrangements, or with a combination of annular cylinder and tubes.

Shell boilers are made for water or steam gauge pressures up to 1 MPa or higher if required. They are normally provided as packaged units with firing equipment, controls etc. incorporated on the unit.

*Water Tube Boilers*

These are manufactured for steam and hot water applications in ratings above 0.5 MW and are of the factory built packaged type or site erected units. They can be forced draught or balanced draught boilers, with natural or forced circulation on the water side and can be designed to meet any output and pressure requirement likely to be encountered in practice.

*Smokeless Fuel Appliances*

Generally, smokless fuel (in sizes up to about 32 mm) is burned in gravity feed appliances. These are not strictly mechanical stokers, but on smaller boilers they offer the same degree of automatic operation and control. There are three principal types of gravity feed appliance:

(a) *Magazine Boiler*
These are self-contained units rated up to 3 MW. They usually hold 24 hours' supply of fuel, have thermostatic control, and mechanical de-ashing devices are fitted.

(b) *Gravity Feed Unit*
These are rated up to 0.75 MW and are suitable for application to sectional or similar hot water boilers. Ash is fused to a clinker and removed manually, and combustion is thermostatically controlled from the air supply fan.

(c) *Independent Gravity Feed Burner*
Suitable for application to hot water boilers rated up to 1.2 MW. a pre-boiler furnace with similar controls and ash removal to those for the gravity feed unit.

**Boiler and Burner Sizing**

The correct sizing of boilers is essential if the following objectives are to be achieved:

(a) Optimum thermal efficiency over the seasonal operation of the plant.

(b) Accurate load matching of plant output to heat demand.

(c) Incorporation of sufficient standby capacity to meet essential load requirements with one boiler off load for any reason (usually $^2/_3$ maximum design winter heat load is adequate).

An anticipated annual load curve should be assessed, bearing in mind diversity factors which can be applied to non-simultaneous loads and with particular reference to the low summer load conditions. All unnecessary boiler margins should be eliminated.

Boiler ratings should then be selceted with the object of meeting the load curve within the modulating range of the firing equipment provided so that no boilers are operating under extensive on/off conditions. This generally indicates the provision of a plant with boilers suitably graded in size, or two equally sized units with a smaller summer load unit, depending upon the particular load curve considered.*

*Multiple Boiler Installations*

Load variation during the season is clearly large; and consideration should be given to the number of boilers to be installed. Operation at low load leads to inevitable drop in efficiency and is to be avoided as far as possible. For medium-sized installations, it is usually convenient to install two similar boilers, able together to meet all demands, and separately to operate during periods of low loads and for summer domestic hot water supply.

The sizing of multiple boiler installations, and their control and sequencing are considered in Sections A9 and B11, respectively.

* Modern boilers, particularly those for domestic use, are designed to have low thermal capacity and low water content. Unless some water is constantly circulating, through the boiler the temperature may rise to a dangerous level before the limit control operates to shut off the fuel. This aspect must be borne in mind when designing a system incorporating a diverting valve which can cause all the system water to by-pass the boiler.

*Burner Sizing*

The selection depends upon the type and size of installation, the turndown facilities required, and whether modulating, high/low or on/off control is desirable. It is becoming more usual to purchase boiler and burner as a packaged unit with the manufacturer's standard firing equipment matched to his boiler.

Boiler manufacturers must be consulted before an alternative type of burner is put forward in order to ensure that it will match the boiler requirements as regards heat release, flame dimensions, turndown, and gas side resistances.

It is recommended that for general applications residual fuel oils, E, F and G,[11] are not burned below the following low fire boiler ratings:

E: 110 kW
F: 220 kW
G: 330 kW

*Load Variation on Oil*

For large installations employing boilers using residual fuel oils, it may be necessary to incorporate a single boiler operating on class D fuel to meet the summertime load. The choice of fuel may be influenced by factors other than cost, e.g. the need to minimise the emission of sulphur dioxide.

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# SECTION B14 REFRIGERATION AND HEAT REJECTION




*This edition of Section B14 first published: 1972*
*Reprinted with amendments and additions: 1986*

# SECTION B14 REFRIGERATION AND HEAT REJECTION

## Introduction

Refrigeration may be defined as a process of energy transfer from a low temperature source to a higher temperature sink, normally maintained by circulating a refrigerant through an expansion device from a high pressure side containing a condenser, to a low pressure side containing an evaporator.

The coefficient of performance (COP) of a refrigeration system is defined as the ratio of the refrigeration effect to the work done in maintaining the refrigeration effect.

## TYPES OF REFRIGERATION SYSTEM

### Mechanical Refrigeration

Mechanical refrigeration is the term generally applied to the vapour compression cycle, the components of which are shown in Fig. B14.1. The different theoretical processes involved are given below and illustrated on the pressure enthalpy diagram in Fig. B14.2. The stages in the cycle are:

1–2 Compression from dry saturated vapour at evaporating pressure to superheated vapour at condensing pressure.

2–3 Rejection of heat at constant pressure while de-superheating and condensing.

3–4 Expansion at constant enthalpy from saturated liquid at condensing pressure to evaporating pressure.

4–1 Absorption of heat at constant pressure while evaporating to become dry saturated vapour.

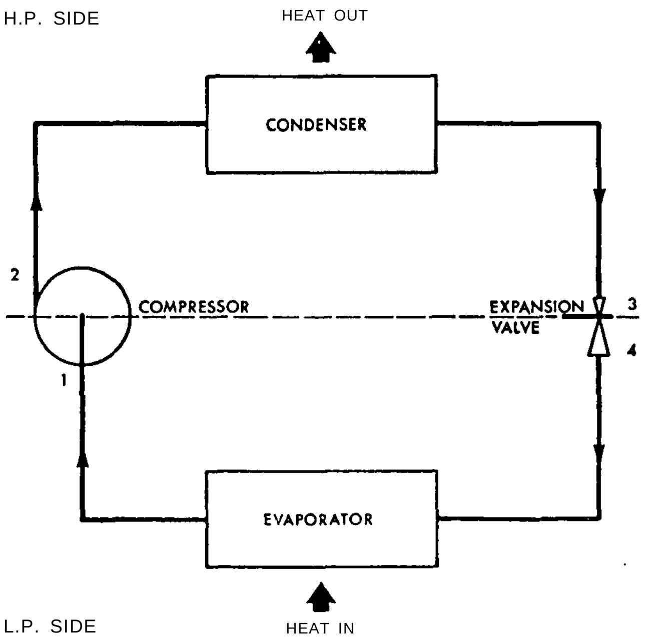



**Fig. B14.1.** Vapour compression cycle.

Where interest is centred on the heat rejected as opposed to the heat absorbed, the equipment usually is known as a heat pump.

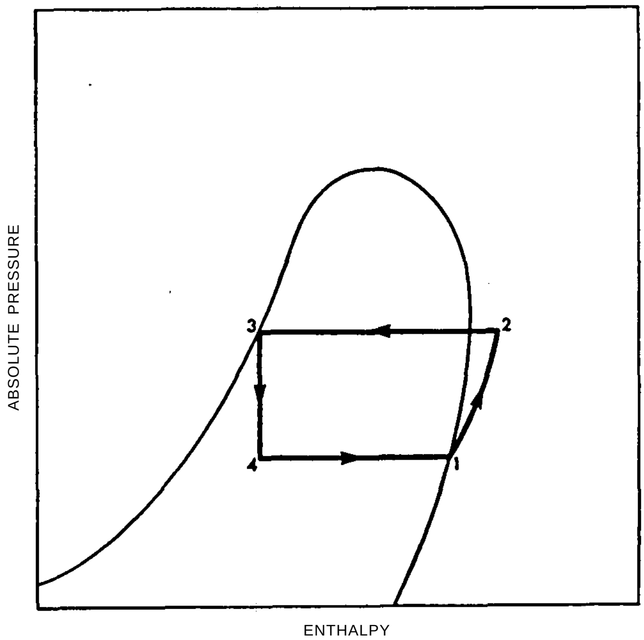



**Fig. B14.2.** Pressure enthalpy diagam of a vapour compression cycle.

When used for both heating and cooling the change over from one to the other may be performed in the air, water or refrigeration systems. Fig. B14.3 shows an air/air system using a refrigerant change-over method.

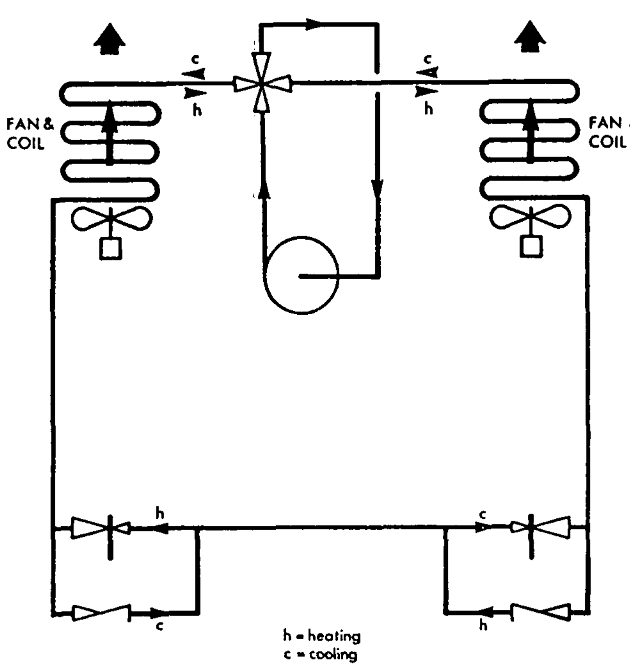



**Fig. B14.3.** Refrigerant change-over system.

### Absorption Refrigeration

The operation of the absorption cycle is similar to that of the vapour compression cycle, the only major difference being the replacement of the compressor by a heat operated absorber-generator. For large equipment, the refrigerant is usually water while the absorbent is commonly a solution of lithium bromide, the advantages of which are its high affinity for water and lasting chemical stability at the temperatures used. For the smaller sized equipment the refrigerant used is ammonia while water is used as the absorbent.

In air-conditioning systems, the absorption process is commonly employed in packaged equipment for the production of chilled water using hot water or steam as the energy source. Some smaller sized equipment (below about 50 kW duty) may be direct fired using gas or oil.

The basic components of the absorption process are shown in Fig. B14.4 and the absorbent solution cycle is shown in Fig. B14.5. The stages in the process are:

- 1–2 Refrigerant vapour is drawn from the evaporator by its absorption in the solution.
- 2–3 Some of the solution is drawn off and pumped to the generator, its pressure being raised to that of the condenser.
- 4 The addition of heat in the generator raises the temperature of the solution, evaporating some of the refrigerant and so concentrating the solution.
- 4–2 The concentrated solution flows back to the absorber through a flow metering device so maintaining the difference between the high and low pressure sides of the system.
- 4–5 Rejection of heat at constant pressure, condensing the refrigerant to saturated liquid.
- 6 Expansion of the refrigerant at constant enthalpy to evaporator pressure.
- 6–1 The addition of heat at constant pressure in evaporating to saturated vapour.

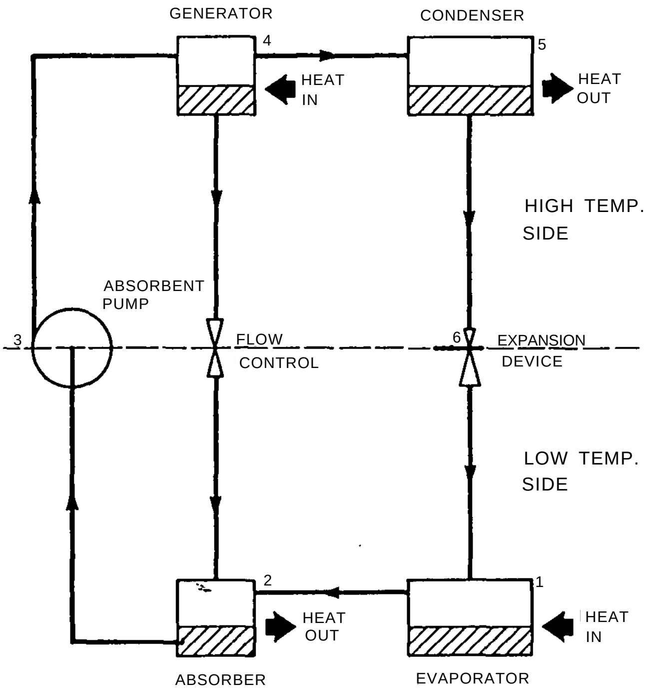



**Fig. B14.4.** Absorption cycle.

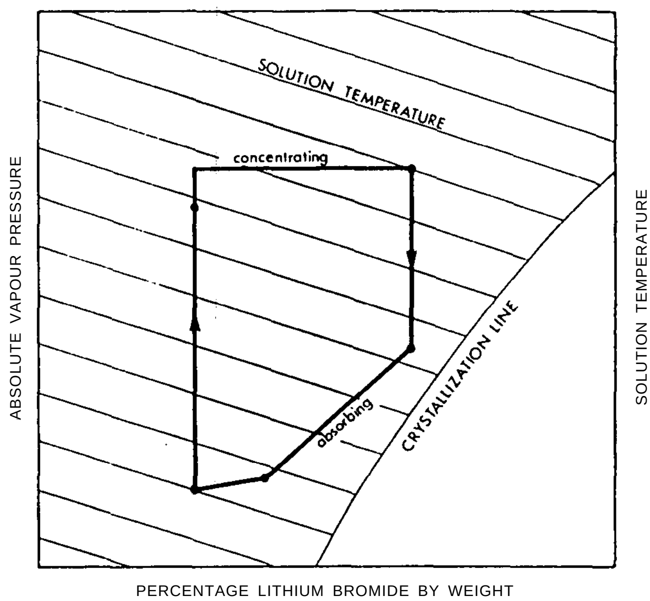



**Fig. B14.5.** Concentration diagram of an absorption cycle.

The heat generated due to the absorption of the refrigerant is usually removed by passing the cooling water through a cooling coil in the absorber before it passes through the condenser.

Care must be taken to ensure that the operation of the equipment does not allow the Lithium Bromide solution to crystallize otherwise refrigeration will cease. If this happens, depending upon the degree and reason for crystallizing, external heating by, for example, a steam spray may be necessary to decrystallize the Lithium Bromide.

### Thermoelectric Refrigeration

Thermoelectric refrigeration involves the practical application of the Peltier effect, which is the absorption and evolution of heat at the junctions of two dissimilar metals when an electric current is passed, see Fig. B14.6. In order to obtain an effective capacity, many such junctions or thermocouples are needed in parallel. Currently only small duty commercial plants using this principle are available.

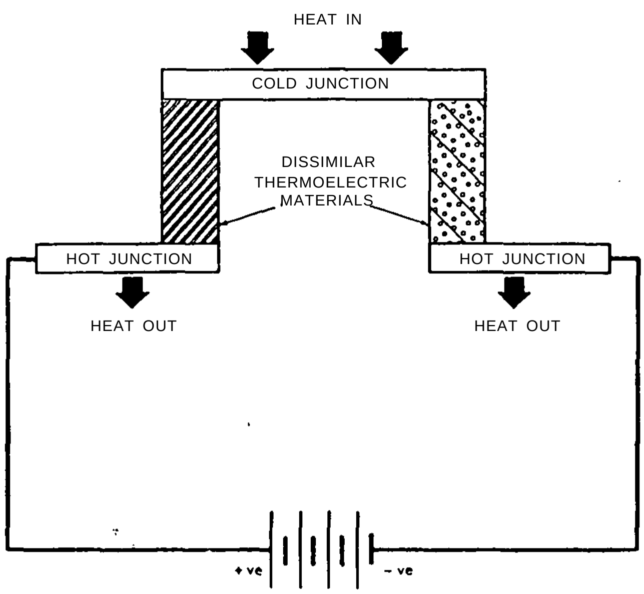



**Fig. B14.6.** Principle of thermoelectric refrigeration.

### Steam Jet Refrigeration

In the steam jet cycle, water is used as a refrigerant; the simplest form of this cycle is shown in Fig. B14.7. The temperature of the chilled water is reduced by evaporation at low pressure, the water vapour thus formed being induced continuously into the ejector by the injection of steam, so maintaining the low evaporating pressure.

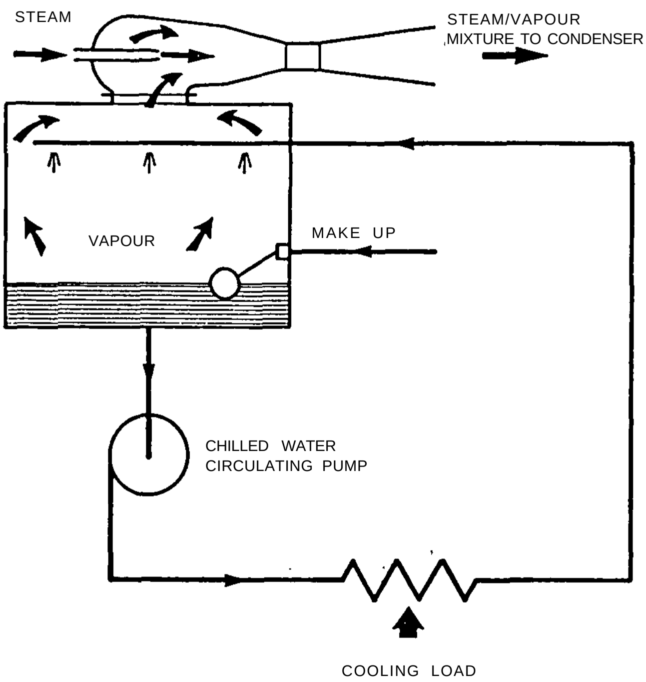



**Fig. B14.7.** Steam jet refrigeration.

The steam jet is most commonly employed in process applications where direct vaporization is used for the concentration or drying of food and chemicals, as in the freeze drying process, although with the advent of total energy plants its use may become more common.

### Air Cycle Refrigeration

The air cycle is used mainly in aircraft cabin air conditioning where its relative inefficiency is offset by the low weight of the component parts In its simplest form the system works by expanding cooled compressed air through a turbine, as a result of which its temperature is further reduced. The air is then injected directly into the cabin.

## COMPONENTS AND THEIR APPLICATION

Tables B14.1 to 5 list the types and applications of condensers, evaporators, compressors, expansion devices and systems.

### Component Balance

In choosing the individual components of a refrigeration system, it is possible to vary the size of each component while still producing the same final duty for a given condition. To provide the most economical selection it is necessary to consider the effect of part load operation on component balance.

During design the matching of components is normally made against the full load operation condition in the knowledge that the system will find a natural balance at part load. It is necessary to ensure that this natural balance is viable and safe in the varying conditions met in practice.

The principles involved are very similar to those employed in matching a fan to a ductwork system and similarly a graphical solution is generally considered to be the most suitable. The co-ordinates most often used are the compressor suction condition as the independent variable and refrigeration capacity as the dependent variable.

A typical solution is shown in Fig. B14.8 where a compressor and air-cooled condenser have been matched to form a condensing unit and the combined performance plotted together with the performance of an evaporator and expansion device. Typical duties are shown for the evaporator at design and part load conditions, assuming a constant flow rate for the medium being cooled (in this case air). The condensing unit performance is shown for four ambient air conditions, the operating points being where the condensing unit and evaporator curves intersect.

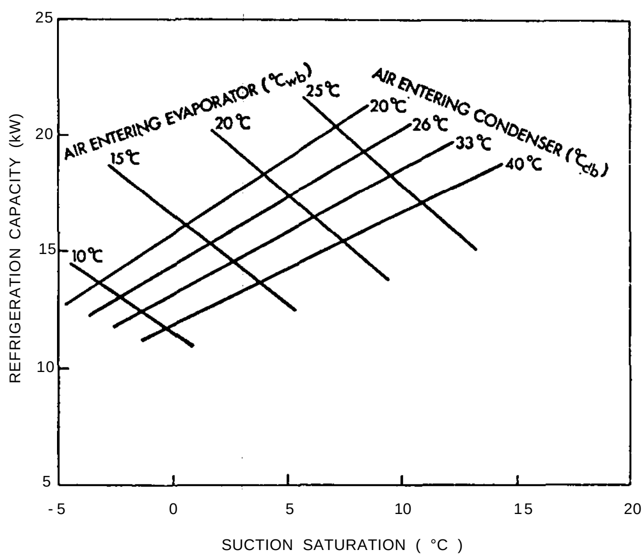



**Fig. B14.8.** Component matching.

A number of conclusions can be drawn from this example.

(*a*) As the temperature of the ambient cooling air increases, the condensing unit performance and consequently the evaporator performance decreases.

(*b*) If the load on the evaporator is reduced while the temperature of the air entering the condensing unit is maintained constant, a lower compressor inlet pressure will result.

(*c*) A lower compressor inlet pressure will also result if the temperature of the air entering the condenser falls while the load on the evaporator remains constant.

The safe operation of the plant must take into consideration the control of the condensing pressure, this being particularly important if the plant is expected to operate in cold weather, together with the varying of evaporator performance at changing plant loads, see Fig. B14.8.

**Table B14.1.** Condensers and their application.

| Type | Description | Usual range of application |
|---|---|---|
| Air-cooled. | Usually finned or gilled copper tube construction. Fins of aluminium or copper; latter preferable in corrosive atmospheres.<br>Constructed in single units or in banks.<br>Running costs of system higher due to condensing temperature being related to dry-bulb temperature of cooling air.<br>Condensing-pressure control by motorized dampers or other means often necessary. | Usually small, but in special cases in banks up to about 1 MW. |
| Water-cooled:<br>Shell and Coil. | Welded pressure vessel containing condensing surface in grid or coiled tube formation. Water in tubes, and water side chemically cleanable only.<br>Pressure control by means of water regulating valve.<br>Frequently used with water cooling towers. | Up to approximately 50 kW. |
| Water-cooled:<br>Shell and Tube. | Welded pressure vessel containing condensing surface in form of plain or finned straight tubes located between end tube plates and having removable end covers for access to inside of water tubes. Large sizes often fitted with side connection water boxes to facilitate cleaning without dismantling water connections.<br>Water side mechanically or chemically cleanable.<br>Water tubes normally copper with special alternatives for sea or bad water conditions.<br>Pressure control by means of water regulating valve. See Table B14.7 for cooling water requirements.<br>Frequently used with water cooling towers. | All duties. |
| Evaporative. | Similar to induced (or forced) draught water cooling tower in which the outside of the condenser surface is employed as packing.<br>Condenser surface normally galvanized steel tubes in grid formation.<br>Location in immediate vicinity of compressors desirable in order to avoid long refrigerant pipe runs.<br>Pressure control as for air-cooled condensers.<br>Water treatment required.<br>Frost protection should be considered. | Up to about 1 MW. |

**Table B14.2.** Evaporators and their application.

| Type | Description | Usual range of application |
|---|---|---|
| Baudelôt. | Assembly comprising nest of water-chilling evaporator coils located over water storage tank (usually divided into 'warm' and 'cold' sections). Pump recirculation delivers warm water from load over evaporator coils and thus to cold section.<br>Occupies considerable space in relation to its duty and may be expensive in capital cost as compared with shell-and-tube evaporator but gives water temperatures as low as 2°C. Ice formation is unlikely to damage tubes. | Up to about 1 MW when used in banks. |
| Submerged Coil. | Nest of bare pipe coils submerged in chilled water storage tank. Requires water agitation except sometimes in small units. Suitable for water temperatures down to 2°C and icebank applications. | Up to about 1 MW. |
| Shell and Tube:<br>Flooded. | Construction as for S/T (Shell and Tube) condenser. Water through tubes, refrigerant in shell giving flooded operation and high rate of heat transfer.<br>Control by expansion device sensitive to liquid level.<br>Can accommodate extreme fluctuations in load.<br>Large refrigerant charge.<br>Suitable for water temperatures down to about 4°C.<br>Also used for chilling brines and other fluids. | All duties. |
| Shell and Tube:<br>Dry Evaporation (D.X.). | Construction as for S/T condenser. Water through shell, refrigerant through tubes.<br>Control by thermostatic expansion valve limits flexibility to between about one-third and full load unless arranged with multiple valves and appropriate circuitry.<br>Small refrigerant charge.<br>Usually for water temperatures down to 4°C.<br>Also used for chilling brines and other fluids.<br>In many designs the water side can only be chemically cleaned. | Up to approximately 750 kW. |
| Air Cooling:<br>Dry Evaporation (D.X.). | Construction as for air-cooled condenser.<br>Control by thermostatic expansion valve.<br>Limited flexibility at light loads: and therefore not recommended for cooling fresh air.<br>Step control of capacity may be obtained by use of multiple coil assemblies with individual thermostatic expansion valves and liquid shut-off valves actuated from thermostatic devices. | Up to 1 MW when used in banks. |

**Table B14.3.** Compressors and their application.

| Type | Description | Usual range of application |
|---|---|---|
| Reciprocating. | Normally single acting; up to 16 cylinders and arranged in V or VW formation except in small sizes. Compressors approximately 25 kW duty and above may be arranged for capacity variation by unloading of cylinders. | Up to approximately 350 kW using R.12 or 600 kW using R.22. Speeds up to 47 rev/s (50 Hz supply). Direct coupling preferable for 100 kW and above. |
| Centrifugal. | Single or multi-stage. Single-stage and larger size multi-stage compressors often driven through gears to obtain necessary rotative speed. | From approximately 300 kW (depending on manufacture) to 15 MW. Over 3 MW normally multi-stage. |
| Rotary. | Medium-speed, jacketed, positive displacement compressor having large displacement against low-pressure differential. | |
| Screw-Type. | High-speed positive displacement compressor, compression being obtained by interaction of two screw cut rotors or monoscrews with toothed wheels. Capacity control down to 10% in oil injected version. Suitable for all refrigerants. | Up to approximately 3 MW. |

**Table B14.4.** Expansion devices.

| Type | Operation | Application |
|---|---|---|
| Thermostatic expansion valves. | Maintains a predetermined degree of superheat at the evaporator outlet. | For ‘dry’ evaporation up to about 400 kW. |
| Constant pressure expansion valve. | Maintains evaporator pressure. | For ‘dry’ evaporation up to about 20 kW. |
| Capillary tube. | Fixed restriction, the drop in pressure being determined by the length and diameter of the tube. | For ‘dry’ evaporation up to about 20 kW. |
| Float valve (low pressure) | Controls the liquid level in a flooded evaporator. | For ‘flooded’ evaporation from about 400 kW. |
| Float valve (high pressure) | Drains the condensed refrigerant into the evaporator at the same time maintaining the liquid seal between the condenser and evaporator. | For ‘flooded’ evaporation from about 400 kW. |

*Note:*
These are devices located between the condenser and the evaporator to separate the high and low pressure sides of the system, their prime function being to meter the flow of refrigerant into the evaporator.

**Table B14.5.** Systems and their description.

| Type | Description |
|---|---|
| Packaged liquid chillers | Reciprocating, screw, centrifugal and absorption machines are utilized in the construction of self-contained packaged liquid chillers. |
| Direct expansion systems | Reciprocating and screw machines in either packaged or custom built form. |

## REFRIGERANTS

### Classification

Refrigerants are normally classified in accordance with BS 4580.

### Selection

The choice of refrigerant for a given application is governed by a number of aspects:

(*a*) *Equipment size:* capacity required per machine since the capacity of a compressor may be varied by the use of different refrigerants.

(*b*) *Type of equipment:* reciprocating, centrifugal, screw compressor or absorption machine.

(*c*) *The application:* whether it is for comfort air conditioning or process cooling.

(*d*) *Economics:* first cost of refrigerant and equipment, and also of running.

(*e*) *Ease of handling and safety consideration:* remembering that while some refrigerants may be classified as non-toxic, they may kill by oxygen starvation.

*Reciprocating Machines*

The refrigerants most commonly used in reciprocating machines are R12, R22 and the R500 series of azeotropic mixtures. Ammonia, though toxic is relatively cheap but the system must be manufactured from different materials to those used for the fluorinated hydro-carbon compounds.

*Centrifugal Machines*

Refrigerant R11 is most frequently used while R12 may be used in large capacity machines and R113 is occasionally employed where a smaller duty is required.

*Absorption Systems*

The most usual refrigerant is water with Lithium Bromide salt as the absorbent.

### Refrigerant Characteristics

The major physical and thermal characteristics of the commonly used refrigerants are outlined in Table B14.6. Manufacturer’s tables and the pressure-enthalpy diagrams in Fig. B14.10 to 13 should be consulted for further information.

**Table B14.6.** Comparative performance of refrigerants.

| Refrig-erant | Chemical formula | Boiling point at pressure of 1·013 bar (°C) | Evapora-ting pressure at 5°C (bar) | Conden-sing pressure at 40°C (bar) | Compres-sion ratio | Refrigera-ting effect (kJ/kg) | Suction specific volume $m^2$ /kg | Theoreti-cal displace-ment (l/s kW) | Coeffi-cient of per-formance | Percen-tage Carnot C.O.P. | Relative cost (taking Rll as standard) |
|---|---|---|---|---|---|---|---|---|---|---|---|
| R11 | $CCl_3F$ | 23·8 | 0·496 | 1·747 | 3·52 | 157·0 | 0·332 | 2·12 | 7·17 | 90·2 | 1·00 |
| R12 | $CCl_2F_2$ | —29·8 | 3·626 | 9·607 | 2·65 | 115·0 | 0·047 | 0·409 | 6·66 | 83·8 | 1·57 |
| R22 | $CHClF_2$ | —40·8 | 5·838 | 15·34 | 2·63 | 157·8 | 0·040 | 0·255 | 6·59 | 82·9 | 2·36 |
| R113 | $CCl_2F—CClF_2$ | 47·6 | 0·188 | 0·783 | 4·16 | 129·5 | 0·652 | 5·03 | 6·62 | 83·3 | 2·26 |
| R114 | $CClF_2—CClF_2$ | 3·8 | 1·062 | 3·373 | 3·18 | 106·2 | 0·122 | 1·14 | 6·74 | 84·8 | 2·30 |
| R502 | $CHClF_2—CClF_2CF_3$ | —45·6 | 6·678 | 16·77 | 2·51 | 101·0 | 0·026 | 0·259 | 6·13 | 77·1 | 3·34 |
| R717 | $NH_3$ | —33·4 | 5·160 | 15·55 | 3·01 | 1088 | 0·243 | 0·214 | 6·87 | 86·4 | 1·46 |
| R718 | $H_2O$ | 100 | 0·009 | 0.074 | 8.46 | 2370 | 147 | 62·0 | 7·38 | 92·9 | — |

*Note:*
It has been assumed in preparing this table that this vapour enters the compressor in a saturated condition at 5°C. For R113 and R114, however, where saturated suction gas would result in condensation during compression enough super heat has been assumed to ensure saturated discharge gas.

## HEAT REJECTION AND COOLING WATER REQUIREMENTS

### Power Input and Heat Rejection

Table B14.7 gives a guide to the power input required for various methods of refrigeration together with the heat rejection which may be expected when evaporating at 2·0°C and condensing at 35·0°C.

**Table B14.7.** Approximate power input and heat rejection.

| Type | Power input per kW cooling (kW) | Heat reiection per kW cooling (kW) |
|---|---|---|
| Reciprocating compressor | 0·22 | 1·22 |
| Centrifugal compressor | 0·20 | 1·20 |
| Screw compressor | 0·19 | 1·19 |
| Absorption machine | 1·43 | 2·43 |

### Cooling Water Requirements

The optimum cooling water temperature range, based on capital and running costs can be taken as between 5·0°C and 9·0°C for most systems. Knowing the heat rejection and the desired temperature range the cooling water requirement may be obtained from the following equation:

$$Wc = \frac{H}{s\Delta t} \quad \text{B14.1}$$

where :

$Wc$ = cooling water required .. .. kg/s
$H$ = heat rejection .. .. .. .. kW
$\Delta t$ = temperature range .. .. .. °C
$s$ = specific heat capacity .. .. .. kJ/kg °C
(4·18 kJ/kg °C for water at 2°C)

With a power input of 0·25 kW per kW cooling at the evaporator, the condenser cooling water requirement for a 6·0°C temperature range is thus 0·05 kg/s.

### Methods of Water Cooling

Devices for the cooling of water fall into a series of groups as follows, not all of which are directly applicable to air-conditioning application (see also Table B14.8).

*Group 1* Ponds
(*a*) cooling (*b*) spray
*Group 2* Natural draught and atmospheric frames
(*a*) spray filled (*b*) wood filled
*Group 3* Natural draught chimney towers
*Group 4* Mechanical draught towers
(*a*) forced draught (*b*) induced draught
*Group 5* Ornamental fountains and special aesthetic treatments

Of the five groups mentioned above only group 4, mechanical draught towers, is now in general use on air-conditioning applications. Ponds, pools and fountains etc., can provide useful cooling capacity under certain conditions and may be considered as an alternative to a cooling tower, which is defined as an enclosed device for the evaporative cooling of water by direct contact with air.

### Cooling Towers (see Appendix 1)

*Design Considerations*

As in all heat transfer applications the performance of a cooling tower must be matched to the operating requirements and theoretical analysis necessitates that this performance and the operating conditions be matched in terms of the same variables using graphical solution as a basis.

By manipulation of the fundamental information on a series of charts representing various operating conditions it is quite simple to analyse the performance of a cooling tower at any off design condition, so long as the design data is available.

*Selection of Tower Type*

Table B14.8 describes the main characteristics of each group of the general classification. The basic information required by the designer which should accompany any enquiry is as follows:

(*a*) The design water-flow rate
(*b*) The design temperature range through which the water is to be cooled
(*c*) The design ambient wet-bulb temperature
(*d*) The operational height above sea level

**Table B14.8.** Characteristics of water cooling equipment.

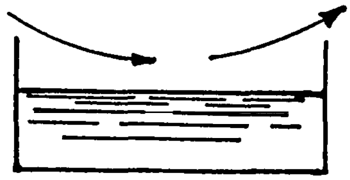
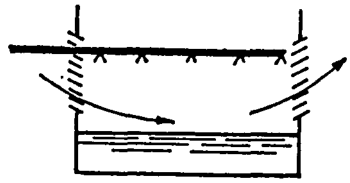
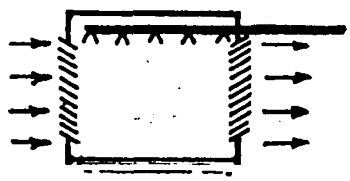
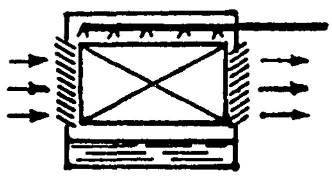
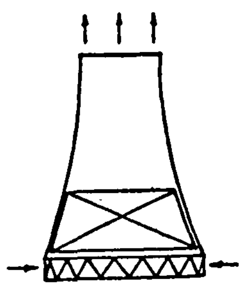
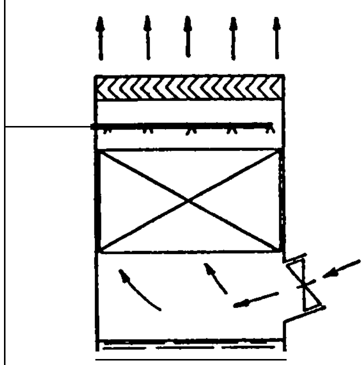
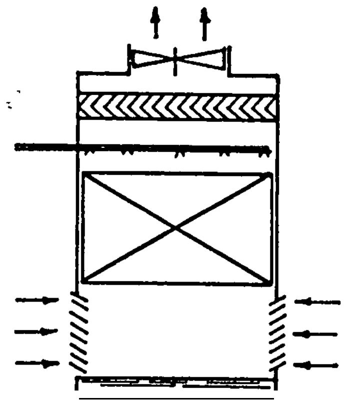
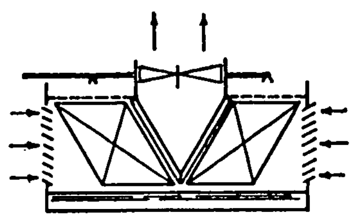
| Group | Type | Description and remarks | Sketch |
|---|---|---|---|
| 1 | (*a*) Cooling Pond | Simplest and cheapest form of water cooling. Cooling occurs partially from contact between air and water at the surface, with resultant evaporation, and partially from transfer of heat to pond walls and floor. Cooled water is drawn off at one end and return water enters at the other.<br>Simple to construct but very space-consuming and may involve expensive excavation.<br>Little make-up necessary–but greater than with a mechanical draught tower.<br>Serious contamination likely but this is readily visible.<br>Very low rate of heat transfer.<br>Poor approach to air wet-bulb temperature.<br>Precise cooling performance not readily determined. |  |
| | (*b*) Spray Pond | A pond with sprays mounted several feet above the surface, surrounded by louvred fencing to reduce loss through carry-over.<br>More compact than cooling pond–may be installed on roof-tops.<br>Low installation and maintenance costs.<br>Better performance than ponds without sprays but limited performance compared with towers since water/air time of contact is small.<br>High water loss in winds, with consequent nuisance.<br>Tends to collect foreign matter.<br>Unreliable performance results from fouling of spray nozzles. |  |
| 2 | Atmospheric Towers | Air movement through tower is mainly dependent on wind; water falls in cross flow to horizontal movement of air; partial counter flow effect from convection currents produced by warm water.<br>Long low-maintenance life due to lack of mechanical parts.<br>Recirculation of used air does not occur.<br>Tower of tall and narrow construction, therefore a high pumping head is necessary.<br>Unobstructed location is necessary.<br>Tower must face the prevailing wind.<br>Secure anchorage required against winds when of tall construction.<br>Risk of fog to leeward of tower.<br>Water temperature varies with wind velocity and direction.<br>Close approach to wet-bulb temperature is not practicable.<br>Cost almost as high as for mechanical draught tower. | |
| | (*a*) Spray-filled Frame | Contains no packing, only numerous spray nozzles at the top; air flows crosswise to vertical fall of water.<br>Low maintenance costs.<br>High pump head required to atomize water through nozzles.<br>Clogging of nozzles leads to unbalance and reduced performance.<br>High losses from windage.<br>Suitable only for relatively small cooling duties. |  |
| | (*b*) Wood-filled Frame | Contains baffling to increase the wetted surface for air/water contact, and longer contact time, the water dropping from deck to deck. Performance much better than that of spray-filled frame. |  |
| 3 | Natural Draught Hyperbolic Tower | Makes use of the stack effect of a chimney above the packing to induce air flow up through the packing in counter-flow to the water.<br>Advantages are low maintenance costs, much better performance than cooling frames but not suitable for high dry-bulb air temperatures since water inlet temperature must be higher than air dry-bulb.<br>Seldom applicable to air-conditioning. Close approach cooling not possible. Capital cost may be high owing to great height necessary to produce the draught.<br>Exact outlet water temperature control is difficult.<br>Used mainly for large cooling duties, e.g. power stations.<br>Generally only justified economically where water flow rate and temperature approach are large. |  |

**Table** ***B14.8.****—continued.*

| Group | Type | Description and remarks | Sketch |
|---|---|---|---|
| 4 | Mechanical Draught Tower | Makes use of fans to move the air through the tower, thus providing absolute control over the air supply.<br>Compact–low ground area.<br>Close control over water temperature.<br>Low pumping head.<br>Orientation of tower is not restricted by prevailing wind direction.<br>With efficient packing, approach temperatures of l-1·5°C are practicable though 3-4°C is usually preferred.<br>Fan horsepower can be high, increasing the running cost.<br>Mechanical failure substantially reduces performance.<br>Recirculation of used air into intake must be avoided or performance will suffer.<br>Higher operating and maintenance costs than natural draught towers.<br>Vibration and noise from the fans may be objectionable in certain locations. | |
| | (*a*) Forced Draught Tower | Is a mechanical draught tower in which the fans are situated at the air intake and blow ambient air into the tower across the packing. Mechanical equipment is near the ground on a firm foundation, thus vibration is kept to the minimum; fractionally more efficient than induced draught since velocity pressure converted to static pressure does useful work, while the fan handles inlet cold air, and thus the weight of air per unit volume is greater than in the induced draught arrangement. Mechanical equipment is situated in a comparatively dry airstream. Mechanical equipment readily accessible for maintenance.<br>Limited fan size, thus a larger number of smaller fans of higher speed compared with induced draught arrangement results in more noise but tower itself provides some attenuation.<br>Tendency for ice to form on the fans in winter and block or throttle the intake. Some types can be prone to recirculation of used air into the accessible low pressure fan inlet, and resulting reduction in performance may be substantial; this occurs if outlet air velocities are low. The air may be ducted away at high velocity, however. |  |
| | (*b*) Induced Draught Tower | Is a mechanical draught tower with the fans situated in the air outlet from the tower, usually on the top, sometimes in the side or even in ducting.<br>Large fans possible, thus low speed and low noise level. High inlet velocities can draw in rubbish; air filters can be fitted.<br>Recirculation of used air improbable due to high outlet velocity.<br>More prone to vibration since fan is mounted on a superstructure.<br>Mechanical parts less readily accessible for maintenance.<br>Mechanical parts in a hot humid airstream.<br>More compact ground plan than forced draught design due to absence of fans on the side. |  |
| | Counterflow Induced Draught Tower | An induced draught tower in which the fans create vertical air movement up the tower across the packing in opposition to the water flow. Thus the coldest water contacts the driest air.<br>Maximum performance arrangement.<br>Mechanical parts and water distribution are not always easily accessible for maintenance. Location can be such that up to three sides of the tower are completely obstructed by adjacent buildings, provided that the remaining air inlet(s) are suitably increased in size. | |
| | Cross Draught Tower | Usually an induced draught tower in which the fans create horizontal airflow as the water falls across the airstream.<br>Greater ground area than the counterflow tower, but the air intakes can be full height of tower which is consequently of low silhouette, blending well with architectural requirements.<br>Some risk of recirculation of saturated vapour if sited in confined space.<br>Low pumping head.<br>Convenient and accessible water distribution system.<br>An uncovered distribution basin will collect rubbish and a cover should be provided unless installation is indoors.<br>Location demands unobstructed air flow towards each end of tower if of the double inlet type. |  |
| 5 | Ornamental Devices | Occasionally, for small systems, it has been found convenient to make use of ornamental fountains and pools (such as in a restaurant or hotel forecourt). The main disadvantages are difficulties in performance prediction, and the danger of undersizing and causing excessive evaporation into the environment. Such systems are prone to fouling from algae, biological slime, airborne dirt and debris. Wide approach temperature. | |

(*e*) Any limitations on height, floor plan, weight, noise or appearance

(*f*) Preferably also a drawing showing the tower location on site

*Selection of the Site*
The location of the cooling tower should receive careful consideration. Obstructions in the tower vicinity must be avoided because space around the tower must be sufficient to allow free flow of air both to the inlet and from the discharge outlet.

Recirculation of the hot discharge air back into the inlet must be avoided as it will substantially reduce performance. Discharge ducting or extended fan casings may be necessary to do this. The siting should not be such that the discharge air may produce condensation upon nearby buildings and in the surrounding area.

The presence of exhaust heat from other equipment or of contaminated air from process plant will reduce tower performance and will encourage the setting up of corrosive conditions. The tower should be sited as far away as possible, upwind of smoke stacks and other potential sources of pollution. Where local atmospheric air pollution is unavoidable, filters may be provided for cooling tower air inlets. The location should be carefully studied in relation to the noise created by the air and water.

The local authorities should always be consulted on byelaws relating to the connection of mains water supplies to tanks and pumping circuits. In general it will be found that it is not permissible to connect pumps direct to the main and that a break tank must be interposed. Local fire regulations should be consulted when a timber tower is to be installed, particularly if any hazard or opportunity for ignition of the tower is likely.

*Water Treatment*
Every water cooling tower requires the application of a good standard of water treatment and reference should be made to Section B7.

*Testing*
Since it is rarely possible to test a cooling tower at its design stage, a code which lays down the essential conditions and acceptable tolerance so that the test data may be calculated back to the design condition is necessary and the relevant British Standards, such as BS 4485, should be used.

## CONTROLS

### Capacity Controls—Mechanical Refrigeration

*Reciprocating Compressors*
Capacity control of reciprocating refrigerating equipment is generally achieved in one of four ways:

(*a*) *Speed variation.* The output of a reciprocating compressor is directly proportional to the speed of shaft rotation and this may be varied by changing the speed of the prime mover. A certain minimum speed must be maintained for lubrication to be effective.

(*b*) *Cylinder unloading.* While several methods are available it is most common for the suction valve on one or more cylinders to be maintained in a raised position by hydraulic pressure so allowing the refrigerant gas to pass back and forth without check and thereby reducing the mass flow through the compressor. A minimum gas flow must be maintained to prevent overheating.

(*c*) *Hot gas bypass.* The load on the compressor in maintained while the evaporator capacity is varied. The most effective arrangement is to bypass the condenser with the hot gas and inject it into the system down stream of the expansion valve and up stream of the evaporator.

(*d*) *Evaporator pressure regulator.* This is a means of maintaining the evaporator pressure by throttling the flow of gas to the suction of the compressor.

Speed variation and cylinder unloading are more economical due to the greater reduction in power consumption arising at part load compared with the little or no reduction arising from using hot gas bypass or evaporator pressure regulation.

*Centrifugal Compressors*
There are three simple methods of capacity control; firstly by speed variation of the prime mover, secondly, and by far the most common, the use of adjustable prerotation vanes in the compressor suction and thirdly by the incorporation of adjustable blades in the diffuser. It is possible, by using these methods of control, to vary the plant output down to about 10% of full capacity, although the prime degree of unloading will depend upon the condenser water temperature available at part load.

Hot gas bypass may be used to reduce the plant output to near zero and also to reduce the risk of compressor surge.

*Helical Rotary Compressors (Screw)*
Control of capacity is normally obtained by varying the compressor displacement using a sliding valve to retard the point at which compression begins and at the same time reducing the size of the discharge port to obtain the desired volume ratio.

### Capacity Controls—Absorption Refrigeration

Two types of control are usual, both varying the capacity of the solution to absorb the refrigerant and so the evaporation rate in the evaporator.

(*a*) Slowing down the rate of boiling in the generator by throttling the steam or hot water input and so limiting the strength of the solution returned to the absorber.

(*b*) Bypassing some of the weak solution being pumped to the generator back to the absorber thus diluting the strong solution returning from the generator.

### Safety Controls

It is essential that safety devices should not be used to operate the plant under normal working conditions. Safety controls are provided to ensure that the plant fails safe in such a way that, in the event of a fault developing, no persons in the vicinity risk injury and the equipment is protected from damage. To provide maximum protection against frequent recycling, safety devices should be of the manual reset type. Table B14.9 lists types of safety controls and their function.

**Table B14.9.** Safety controls and their function.

| Type | Function |
|---|---|
| MECHANICAL REFRIGERATION | |
| High refrigerant pressure cut-out | To break circuit on excessive refrigerant pressure rise. |
| Low refrigerant pressure cut-out | To break circuit on fall in refrigerant pressure. |
| Low oil pressure cut-out | To protect against failure of lubricating system. |
| Low refrigerant temperature cut-out | To protect against low evaporating temperatures. |
| Fusible plug | To protect against high refrigerant temperatures. |
| Pressure relief device | To protect against high refrigerant pressure under static conditions. |
| Low water temperature cut-out | Used in water chillers to prevent the evaporator freezing. |
| Flow switches | To protect against reduced liquid flow through evaporator or condenser. |
| ABSORPTION REFRIGERATION | |
| Low refrigerant temperature cut-out | To prevent the evaporator freezing. |
| Low chilled water temperature cut-out | To prevent the evaporator freezing. |
| High solution temperature cut-out | To prevent over-concentration of the solution and consequent crystallization. |
| Low cooling water temperature cut-out | To prevent over-concentration of the solution and consequent crystallization. |
| Flow switches | To protect against reduced liquid flow through evaporator or condenser. |

## MULTIPLE WATER CHILLERS

In large air-conditioning systems, it is a common practice to split the refrigeration capacity between multiple water chilling units which can present some connection and control problems. It is also essential to co-ordinate the design of the control of the air-handling equipment with that of the refrigeration machines; the choice being between a constant-flow and a variable-flow chilled water system.

### Connection of Evaporators and Condensers

Three water-flow combinations are illustrated in Fig. B14.9.

(i) Evaporators and condensers piped in parallel.
(ii) Series connected evaporators, parallel condensers.
(iii) Series connected evaporators and condensers in counterflow.

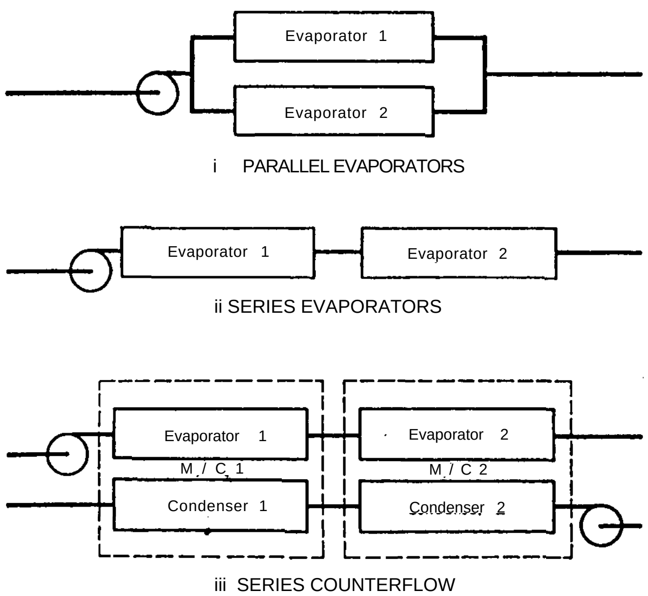



**Fig. B14.9.** Connections of evaporators and condensers.

*Parallel Evaporators*

Parallel circuits allow multi-pass heat exchangers at a relatively low water pressure drop, consequently a lower pump power is required than for a series circuit. However, a slightly higher compressor power is required than for a series circuit, due to both machines having the same evaporating temperature.

Care must be taken in the design of the controls with this arrangement as there is the possibility of short cycling under partial load conditions and, because of bypassing, the danger of freezing one evaporator when the other is switched off and control vested in a common downstream thermostat.

*Series Evaporators*

Compared to the parallel arrangement, this system uses a higher chilled water pressure drop and, therefore, a higher pump power, is required. Consequently, a single pass evaporator may well be necessary, hence component design suffers. The compressor power is slightly lower than for a parallel arrangement, as the upstream machine will have a higher, evaporating temperature than the downstream machine.

*Parallel and Series Condensers*

Parallel condenser water cooling circuits are sometimes an advantage if more than one cooling tower is to be used. Generally speaking, the compressor power will be greater, but the water pump power less, when a parallel arrangement is used, based on a similar argument to that used above for the evaporators.

*Series Counterflow*

This arrangement of evaporators and condensers is generally the most economical for multiple machine installations particularly for heat reclaim schemes.

*The Combination System*

The combination system is a special case which is particularly suitable for total energy installations and is used to obtain maximum economy in the use of steam to operate refrigeration machines.

Steam passes in series through a back pressure steam turbine driving a centrifugal water chiller, and then at the back pressure, into a pair of absorption machines, each equal in capacity to the centrifugal unit. The evaporators are connected in series parallel, the chilled water passing first through the pair of parallel connected absorption machines and then through the centrifugal machine. The three condensers are usually, though not necessarily, connected in parallel.

Table B14.10 gives typical values of the steam consumptions which can be achieved, based on nominal steam flow rates for a typical air-conditioning application.

**Table B14.10.** Typical steam consumptions.

| Type of system | Steam consumption per kW cooling (g/s) |
|---|---|
| Steam turbine driven centrifugal | 1·8 |
| Steam jet refrigeration plant | 1·3 |
| Steam driven absorption machines | 0·9 |
| Combination system | 0·7 |

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BS 5304: 1975. Code of practice for safeguarding of machinery.

BS 5500: 1985. Specification for unfired fusion welded pressure vessels.

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BS 5970: 1981. Code of practice for thermal insulation of pipework and equipment (in the temperature range of -100° C to +870° C).

Minimising the risk of Legionnaires' Disease, Technical Memorandum 13, CIBSE, London, to be published.


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## APPENDIX 1

### Legionnaires' Disease

Evaporative cooling towers operate at around 30°C and therefore provide temperature conditions in which the bacterium *legionella pneumophila* can multiply. Water treatment (see Section B7) is needed to inhibit multiplication and the design and siting of towers should minimise the likelihood of inhalation of the fine water aerosol which is constantly ejected.

The main requirements for reducing the risk of Legionnaires' Disease are as follows.

(*a*) Minimise aerosol generation by providing drift eliminators and matching their performance to the working air flow velocity within the tower.

(*b*) Minimise human exposure to the aerosol by ensuring that the location of the tower and the prevailing wind direction will not cause the rejected aerosol to be blown into occupied rooms or into the air intake of the building or nearby buildings.

(*c*) Ensure that the tower is operated and maintained correctly. Water treatment is required for corrosion control, scale prevention and to inhibit fouling. Blowdown or continuous bleed will be required and an allowance should be made for the resulting water loss. Cleaning and disinfection, at least once a year, is recommended for towers in constant use.

Towers taken out of service should be drained and cleaned on shutdown and checked and treated before recommissioning. Chlorination of the water system, including standby facilities, prior to cleaning will minimise risk to cleaning/maintenance staff. Note that chlorination loses much of its effectiveness for pH values greater than 7.

Clear, detailed operating and maintenance instructions should be provided both for the tower itself and the associated water circuit. Operating and maintenance actions should be systematically recorded by the operating staff, along with any additional comments.

Air cooled condensers do not present the problem of aerosols but care must be taken in design to ensure prompt drainage of water resulting from rainwater penetration.

## APPENDIX 2

### Refrigerant Charts

In the following charts it will be noticed that those for R11 and R12 use the critical point as the base from which numerical values of enthalpy and entropy are measured, whilst the charts for R22 and R502 use 0°C. This latter reference point is likely to have increasing use on future charts, since the critical point cannot always be used.

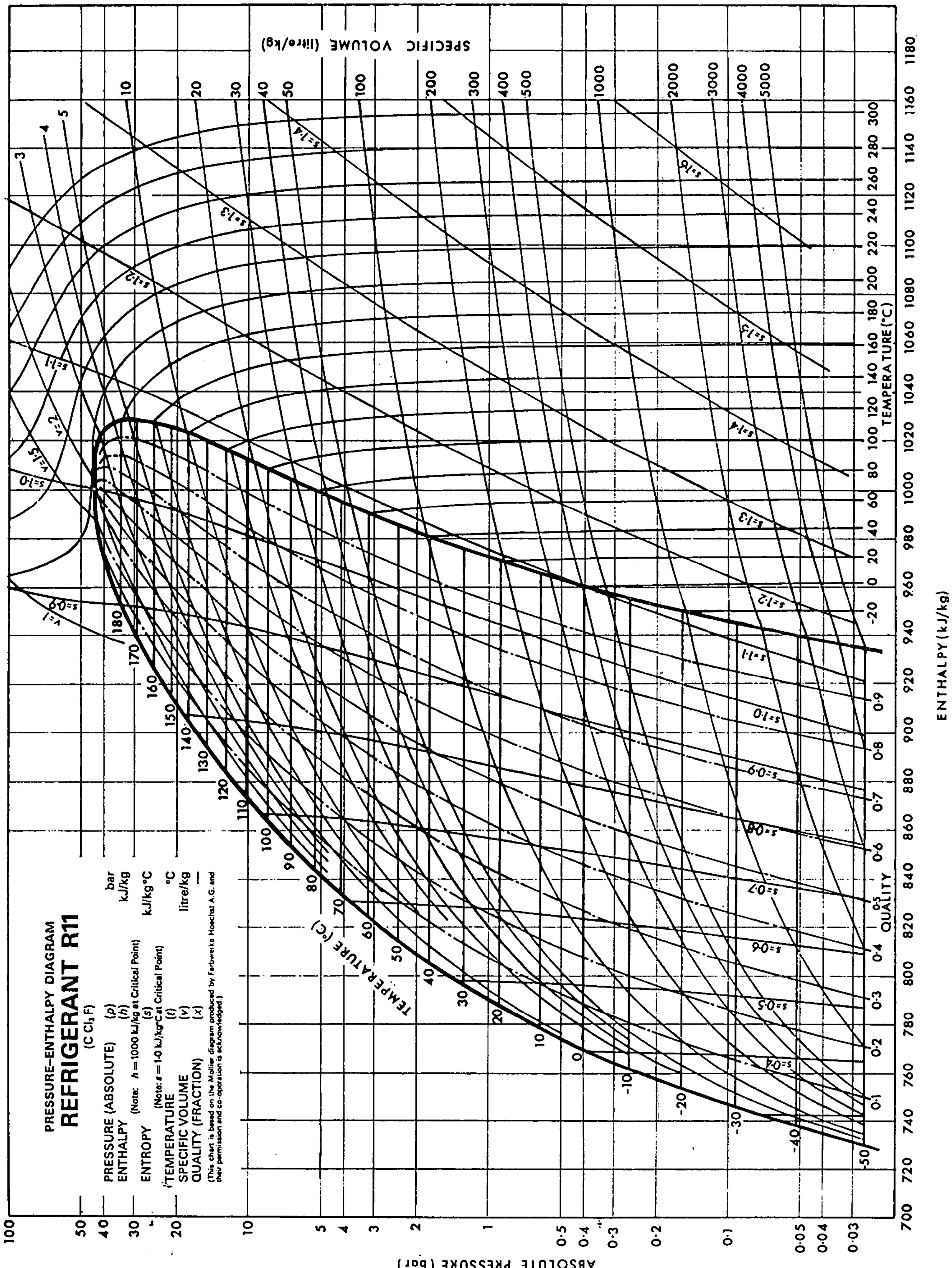



Fig. B14.10. Pressure-enthalpy diagram for R11.

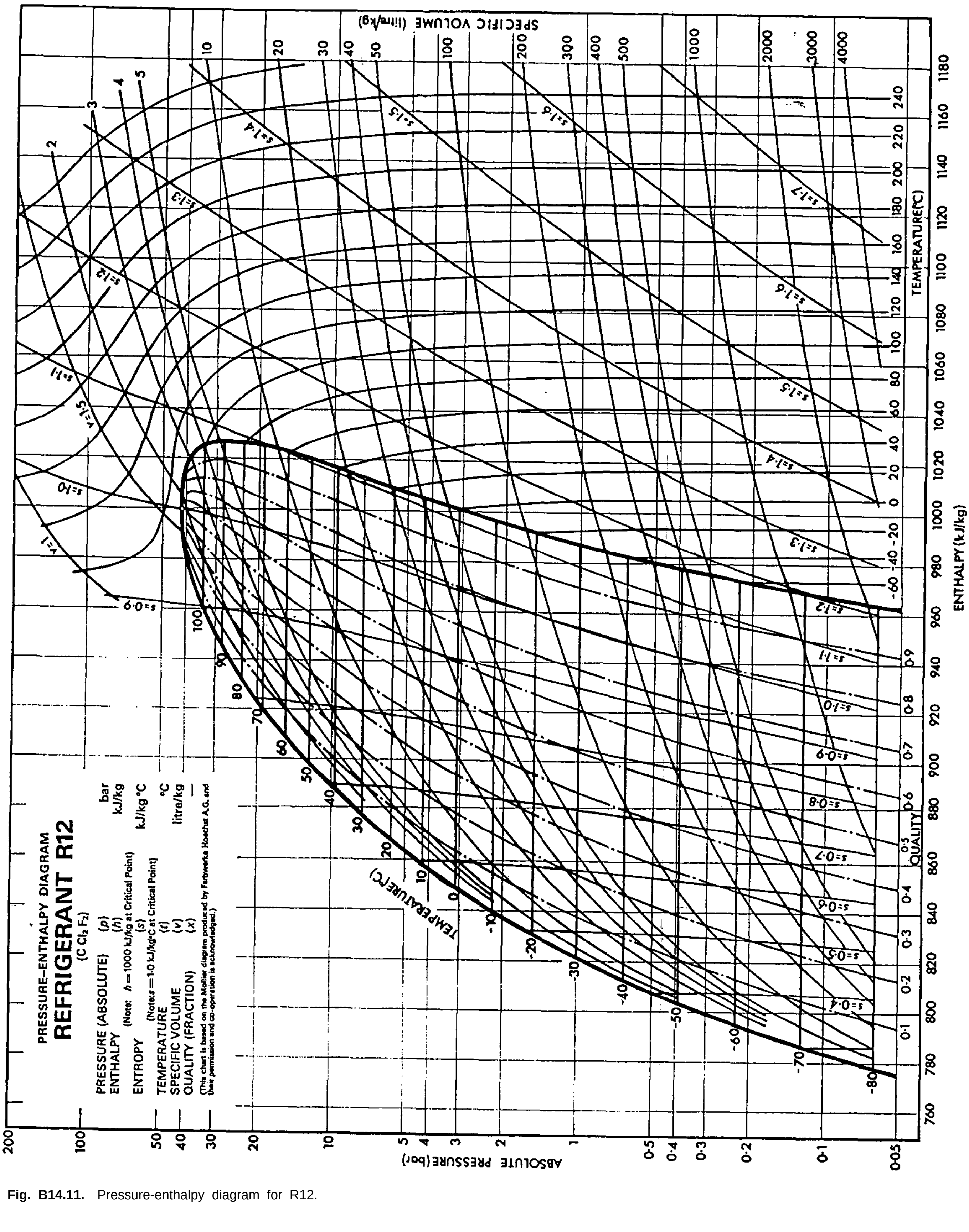



Fig. B14.11. Pressure-enthalpy diagram for R12.

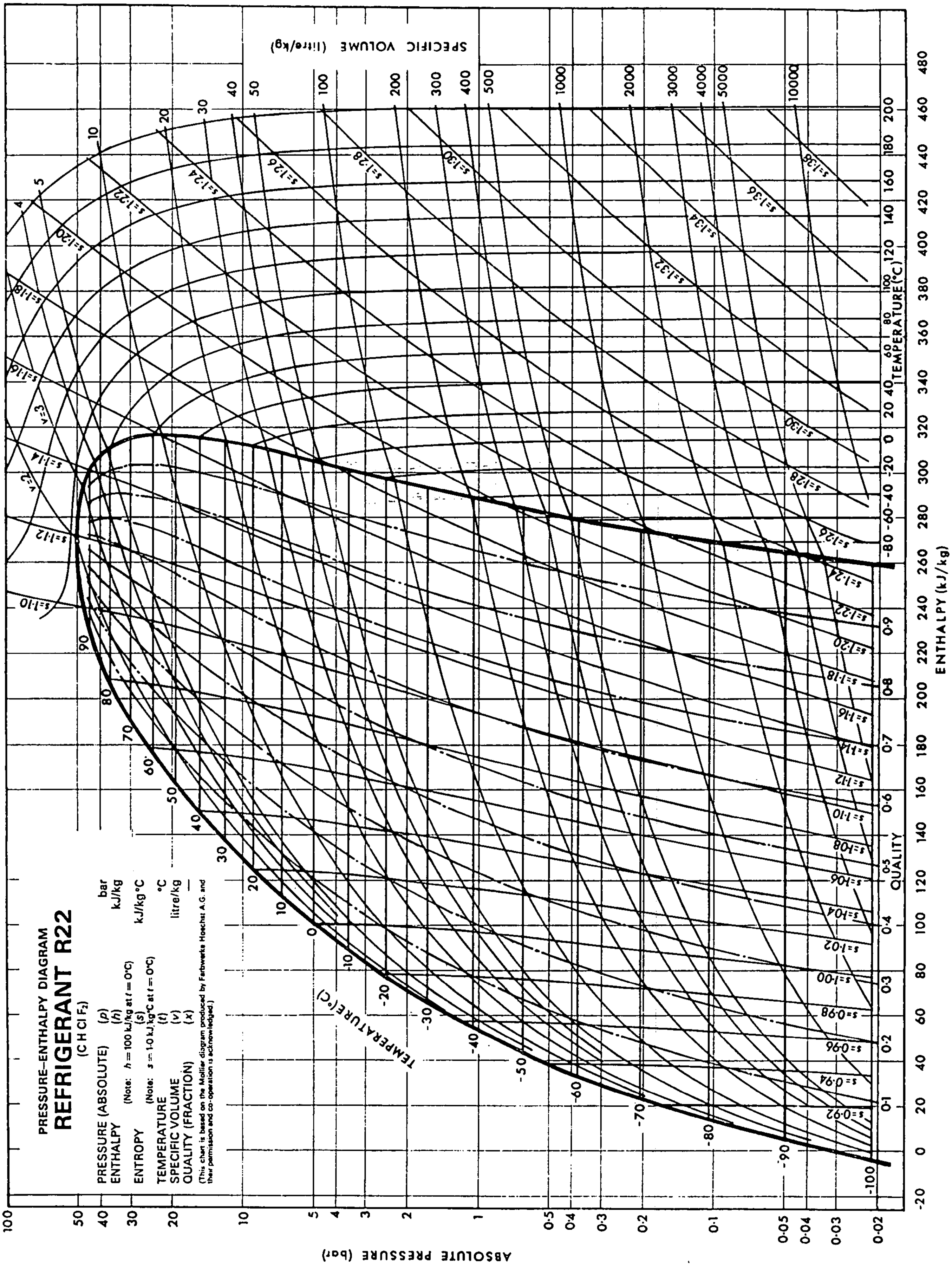
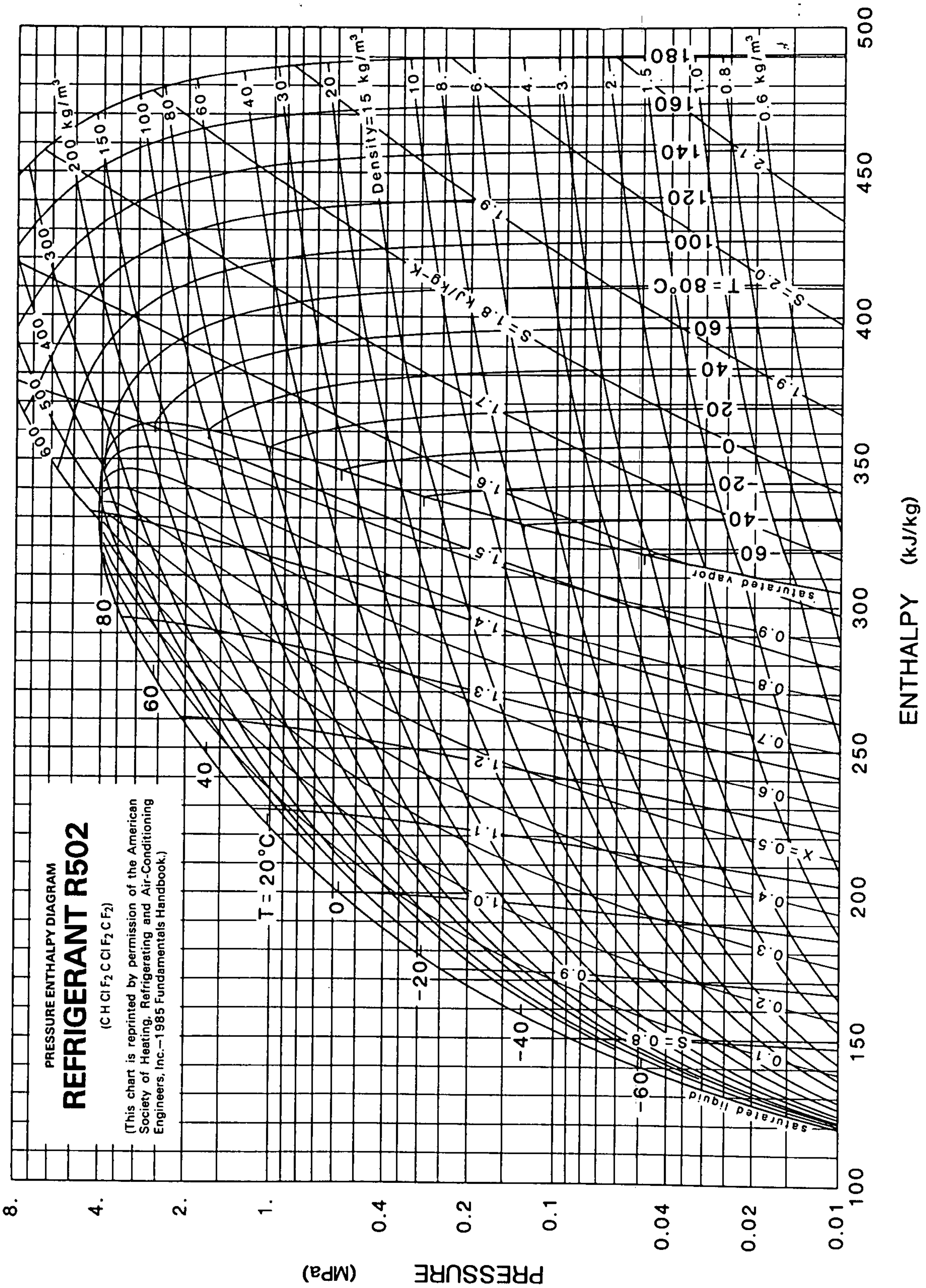



Fig. B14.12. Pressure-enthalpy diagram for R22.




Base values of enthalpy and entropy are as follows.
Enthalpy: $h$ = 200 kJ/kg at 0°C
Entropy: $s$ = 1·0 kJ/kg K at 0°C

**Fig. B14.13.** Pressure-enthalpy diagram for R502.

# SECTION B15 VERTICAL TRANSPORTATION




*This edition of Section B15 first published: 1986*

# SECTION B15 VERTICAL TRANSPORTATION

**Introduction**

The second half of this century has seen a great increase in the number of tall buildings, and a corresponding increase in mechanical means for transporting people vertically inside them. Such equipment must now be considered as part of normal building services. This Section provides a general description of the most common types of equipment in use, indicates how an economic system may be planned, discusses some of the factors where the transportation system affects other building services and enumerates aspects of system operation.

BS 5655: Lifts and service lifts contains much information pertinent to the detailed design of lifts. This has not been repeated in this Section of the *Guide* and reference should be made to the British Standard itself.

**PASSENGER LIFTS**

The capacity of any given lift equipment can be assessed fairly accurately. The difficulty in planning a lift system therefore lies not in calculating the performance but in estimating what the demand is likely to be before the building is occupied and frequently before any knowledge of the population or even, in the case of speculative developments, what type of tenant or even what tenancy arrangement will exist. It is therefore essential that a clear understanding be established between all parties concerned as to the basis to be used for planning. Decisions as to the most suitable number of lifts, their loads and their speeds for any project depend upon the space availability for the lift system and the mobility and space requirements of the building occupants.

The location of lifts within a building should take account of the proximity of entrances, stairs and the distribution of the users about the floors served. In general terms, lifts should be centralised but walking distance therefrom to the most remote occupied area should in no case be more than about 60 m; 45 m is to be preferred. If the entrance to the building is not central, there is still a case for centralising the lifts since usage during the day may outweigh the comparative inconvenience at morning arrival and evening departure.

**Assessment of Demand**

The maximum peak demand which is likely to be imposed on the lift system is normally expressed as the number of persons who will be carried during a given time period: a five-minute period is normally selected as being sufficient to show averaged peaks in traffic flow.

For office buildings the peak demand is usually heaviest in the down direction but the up peak normally puts the heaviest sustained load on the lift equipment. Fig. B15.1 shows a typical flow histogram for an office building

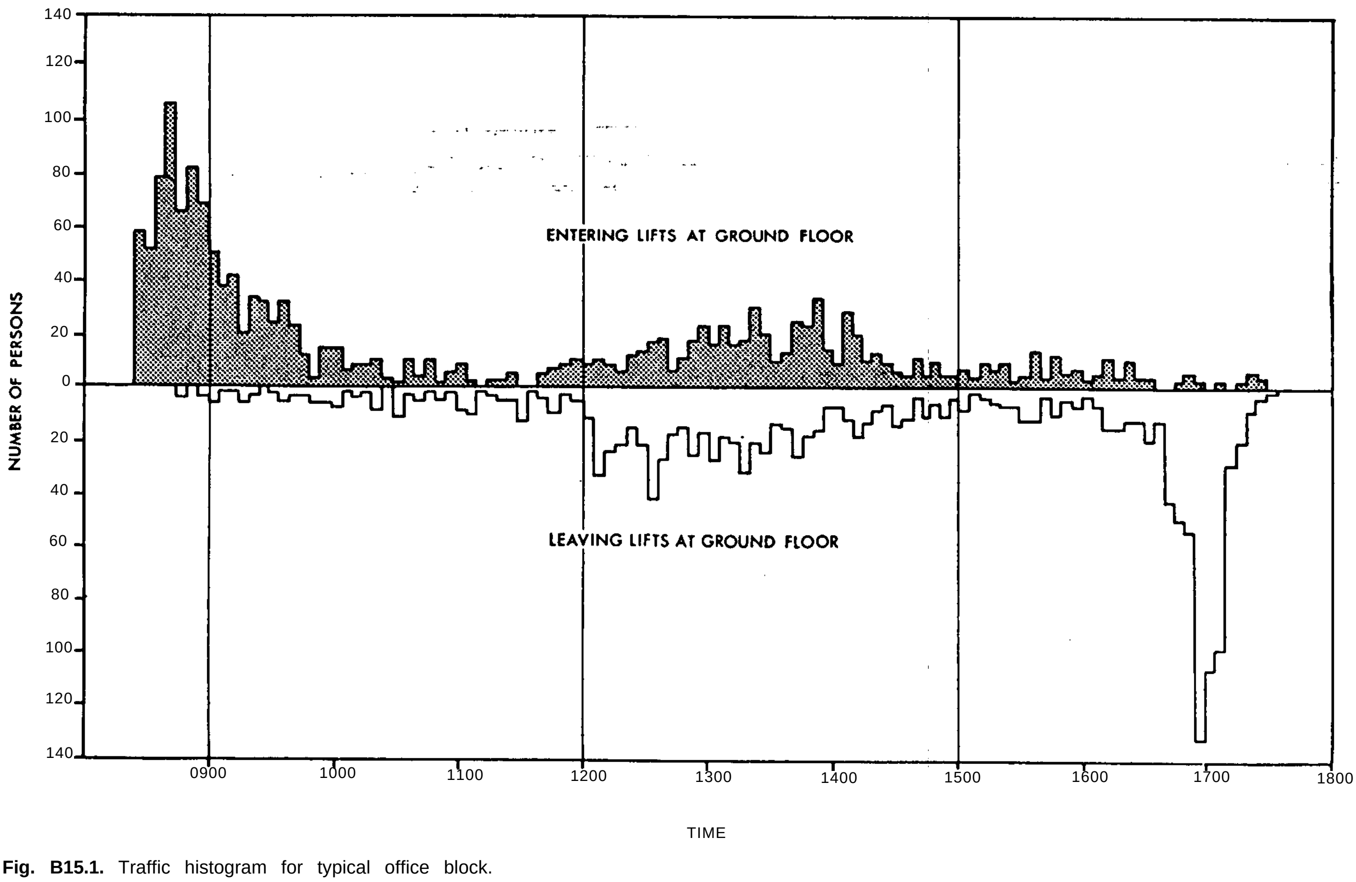



**Fig. B15.1.** Traffic histogram for typical office block.

occupied by a single tenant. In hotels and hospitals, as in office buildings during lunch periods, the traffic flow is two directional and peak two-directional traffic flow may similarly be assessed as the number of persons that will require to be carried in a five-minute period.

Particular features of the building may need to be taken into account. For instance, if an office building has a restaurant situated on an upper floor, this can place a very heavy demand on the lifts. If the building is served by two or more groups of lifts, one serving the lower floors with others serving the upper floors, then people going to the restaurant may have to change lifts at an interchange floor.

Since it is frequently impossible to decide what the vertical circulation will be before a building is occupied, several factors must be assumed in order to anticipate this figure.

*Assessment of the Building Population*

For purpose designed buildings, the accommodation and population will form part of the design brief but on speculative developments, for letting, it is only possible to make an assessment of the population from consideration of the lettable area. Surveys have shown that space standards may vary from 7·5 to 19 $m^2$ per person, the former usually for all-clerical buildings with conference rooms, boardrooms and showroom areas. The published minimum of 3·7 $m^2$ per person under the Offices, Shops and Railway Premises Act is unrealistic for overall population assessment and experience from existing buildings shows 9·5 to 11·25 $m^2$ as a good average range.

*Transportation During the Peak Period*

Surveys show that between 10 and 25% of the total building population will require transportation during a five-minute peak period according to whether they start or finish work at different times, as when a building is let to many tenants following different business interests, or whether all the staff start or finish at the same time as in a purpose designed building or in a single tenancy building. For purpose designed buildings, the assessment of the peak traffic can be obtained by traffic surveys taken from existing buildings populated by the owner but for buildings designed for letting, data can only be accumulated from similar buildings in the area.

Staff punctuality may be affected by the efficiency of *horizontal* transportation in the vicinity of the building: for instance, an office building adjacent to a railway station may have over half its population arrive on perhaps two trains thus producing a concentrated lift load for a very short period. If, however, the office were situated at half a kilometre walking distance from a station, then the load would be spread over a longer period due purely to the varying walking speeds of individual members of the staff.

If no other data are available for speculative developments, the general standard used should be based on an assumed peak requirement of 12% of the population per 5 minutes and, for single purpose buildings, an average peak requirement of 17% of the population per 5 minutes may be used. For hotels, hospitals, etc. peak requirements must be individually analysed. Having established the likely demand, simple performance calculations should be carried out on several combinations of numbers of lifts, lift capacity, speed, etc. The selection of the final installation will depend upon a balance between the quality of service demanded and the capital sum available.

*Economic Considerations*

The number of lifts provided will obviously have an effect on the quality of service offered e.g. four 16-person lifts may provide the same carrying capacity as three 24-person lifts for a given building, but the waiting time will be approximately twice as long with the three-car group than with the four cars. It is, therefore, necessary to assess the quality necessary to compare with similar buildings in the locality in order to ensure the maximum rental-potential. A building with a high rental-potential will clearly demand a better quality of service than a building with a low rental-potential. Similarly with high rental-potential buildings it is essential to consider the provision of higher cost, higher speed equipment in order to keep lift space to a minimum. When a low rental-potential exists, the lower cost equipment with lower speed and efficiency may be economically sound even though more lift space is required.

**Table B15.1.** Minimum number of lifts.

| Installation | Quality of service* |
|---|---|
| One lift per 3 storeys | Excellent |
| One lift per 4 storeys | Acceptable |
| One lift per 5 storeys | Below average |

*See Table B15.4 for definitions.

*First Approximations*

Without detailing car sizes or speeds, a rough guide to the minimum number of lifts required in one interconnected group for an average office building may be taken from Table B15.1, but a different standard would operate for hotels and blocks of flats. Where large numbers of people have to be moved however, cars smaller than 13 persons capacity are not satisfactory and the travelling time for a passenger in a lift car from the ground up to the top floor can be very long in instances where many intermediate stops are possible. For this reason, as well as for those of architectural planning and economics generally, it may be advisable to split a vertical transportation system into two or more zones with a group of lifts for each zone, thereby keeping the number of floors served within each zone to acceptable limits. Generally, 10 to 15 floors are suitable within each zone, provided sufficient lifts are included in each group and that acceptable waiting times can be achieved. In practice less than 4 lifts in each group can rarely provide an acceptable interval with this arrangement.

**Calculation of Lift Performance**

The following notes outline a typical basic procedure for the calculation of lift performance. Performance tables in BS 5655 Part 6 provide typical values for different combinations of numbers of lifts in a group, speed, rated load and numbers of floors served. Individual lift manufacturers use simple computer programmes for the calculation.

The actual calculation of lift performance depends upon characteristics such as acceleration, retardation, speed of door operation, switch timing and brake speed on starting, the degree of advance door opening offered, the performance of the lift over short travel distances, the stability of speed and the performance with variations of car load. It is essential to establish design criteria in addition to load and speed with the lift manufacturer in order to establish actual lift performance.

The probable up-peak performance and hence the round trip time of a lift may be calculated from consideration of the following:

(*a*) Total running time
(*b*) Total door operating time
(*c*) Total passenger transfer time (entry and exit).

*Running Time*

When a lift departs from the lobby at the time of peak demand, it will stop at several floors on its upward journey. the number of stops made may be estimated from the probability expression:

$$S_1 = S - S\left(\frac{S-1}{S}\right)^n \quad \ldots \quad \text{B15.1}$$

where:

$S_1$ = probable number of stops

$S$ = maximum possible number of stops

$n$ = number of passengers (usually assumed as 80% of the contract load)

Table B15.2 lists a limited number of solutions for six standard car sizes: intermediate values may be found by interpolation.

**Table B15.2.** Probable number of stops at 80% load.

| Contract load (persons and kg) | | Probable number of stops, $S_1$, for the stated maximum number of stops, S | | | | | |
|---|---|---|---|---|---|---|---|
| | | 5 | 10 | 15 | 20 | 25 | 30 |
| 5 | 400 | 4 | | | | | |
| 8 | 630 | 4 | 5 | | | | |
| 10 | 800 | 4 | 6 | 6 | | | |
| 13 | 1000 | 5 | 6 | 7 | 8 | | |
| 16 | 1250 | 5 | 7 | 9 | 10 | 10 | 11 |
| 21 | 1600 | 5 | 8 | 10 | 11 | 12 | 13 |

Knowing the probable number of stops, the time for the upward journey may be calculated taking into account not only the contract speed, but also the effect of acceleration and retardation at each stop from:

$$T_{up} = S_1\left(\frac{L}{S_1 v} + 2v\right) \quad \ldots \quad \text{B15.2}$$

where:

$T_{up}$ = total upward journey time .. .. s

L = total lift travel .. .. .. .. m

$v$ = contract speed .. .. .. .. m/s

For the downward journey it may be assumed that, at the time of peak demand, the car will run non-stop from top to bottom, thus:

$$T_{down} = \left(\frac{L}{v} + 2v\right) \quad \ldots \quad \text{B15.3}$$

where:

$T_{down}$ = total downward journey time .. .. s

Solutions for the term in brackets in equations B15.2 and 3 may be read from Table B15.3 for a number of contract speeds.

**Table B15.3.** Travel times for lifts.

| Travel (m) $L$ or $\frac{L}{S_1}$ | Travel time (s), $T_{down}$ or $\frac{T_{up}}{S_1}$ for following contract speeds (m/s) | | | | | |
|---|---|---|---|---|---|---|
| | 0·75 | 1·0 | 1·5 | 2·5 | 3·5 | 5·0 |
| 3 | 6 | 5 | 5 | 6 | 8 | 11 |
| 6 | 10 | 8 | 7 | 7 | 9 | 11 |
| 9 | 14 | 11 | 9 | 9 | 10 | 12 |
| 12 | 18 | 14 | 11 | 10 | 10 | 12 |
| 15 | 22 | 17 | 13 | 11 | 11 | 13 |
| 18 | 26 | 20 | 15 | 12 | 12 | 14 |
| 21 | 30 | 23 | 17 | 13 | 13 | 14 |
| 24 | 34 | 26 | 19 | 15 | 14 | 15 |
| 27 | 38 | 29 | 21 | 16 | 15 | 15 |
| 30 | 42 | 32 | 23 | 17 | 16 | 16 |
| 35 | 49 | 37 | 27 | 19 | 17 | 17 |
| 40 | 55 | 42 | 30 | 21 | 18 | 18 |
| 45 | — | 47 | 33 | 23 | 20 | 19 |
| 50 | — | 52 | 36 | 25 | 21 | 20 |
| 55 | — | 57 | 40 | 27 | 23 | 21 |
| 60 | — | 62 | 43 | 29 | 24 | 22 |
| 65 | — | — | 46 | 31 | 26 | 23 |
| 70 | — | — | 50 | 33 | 27 | 24 |
| 75 | — | — | 53 | 35 | 28 | 25 |
| 80 | — | — | 56 | 37 | 30 | 26 |
| 85 | — | — | — | 39 | 31 | 27 |
| 90 | — | — | — | 41 | 33 | 28 |
| 95 | — | — | — | 43 | 34 | 29 |
| 100 | — | — | — | 45 | 35 | 30 |

*Door Operating Time*

Because much of the round trip time of any lift is spent stationary at floor levels whilst passengers are getting into or out of the lift car, the time for this operation becomes critical and the shape and entrance width to permit rapid passenger transfer are therefore vital factors in efficient utilisation of lift well area. Clearly a lift car of wide and shallow proportions will permit faster passenger transfer than a car which is narrow and deep. Similarly, centre opening automatic doors are much more efficient than single panel or two panel two-speed side opening doors, as the door panels have only to move half the horizontal distance. Passengers will start toward a centre opening entrance as soon as the door panels part, whereas with single panel or two-speed side opening doors, the panel will have to move halfway across the entrance before passengers attempt to move.

A further consideration for lift installations with a door safety device of the ‘leading edge’ type, where a safety rail is projected past the edge of the car doors, is the time taken for this to operate. When the leading edge is interrupted by an obstruction, it causes the doors to reverse their movement without damaging the obstruction. Once this has happened, a time delay incorporated into the mechanism is activated to allow passengers time to enter or leave the car. The time delay may be set to be capable of being overridden by the controls inside the car, but it will always be used in its entirety in the case of all passengers leaving the car. Setting the time delay to a very short time will lead to complaints by passengers.

On other installations with modern forms of group controls, the leading edge safety device is omitted and detector doors are fitted. These incorporate an electric bridge arranged in close proximity to the car doors, which senses when there is an obstruction and prevents door closure. This has the advantage that doors can be closed as soon as no obstruction exists.

The time taken for door operation during a round trip is, therefore, a function of the width of the opening, the door type and the operating speed. This may be estimated from:

$$T_{do} = 2\ (S_1 + 1)\,\frac{W}{v_d} \quad \ldots \quad \text{B15.4}$$

where:

$T_{do}$ = door operating time, open and close .. s

$W$ = width of opening .. .. .. .. m

$V_d$ = door operating speed .. .. .. m/s

For centre opening doors, an operating speed of 0·4 m/s may be taken: for two-speed doors this is, in effect, reduced to 0·2m/s.

*Passenger Transfer Time*

Passenger transfer time can vary considerably and is affected by the shape of lift car, size and type of entrance, environment, e.g. office or hotel, etc.

For efficiently planned cars of correct proportions, with centre opening doors similar to those specified in BS 5655: Part 5, the average total time taken for each person to get into and out of a lift car may be taken as 2 seconds and thus:

$$T_t = 2n \quad \ldots \quad \text{B15.5}$$

where:

$T_t$ = total passenger transfer time .. .. s

$n$ = number of passengers

*Round Trip Time*

The round trip time, therefore, is the sum of the components of equations B15.2 to 5:

$$T_r = T_{up} + T_{down} + T_{do} + T_t \quad \ldots \quad \text{B15.6}$$

where:

$T_r$ = round trip time .. .. .. .. s

*Capacity and Interval*

From the round trip time, the capacity of a lift or a group of lifts during a five-minute peak may be calculated. Similarly, the interval may be determined: this latter applies to a group of lifts and is a measure of the quality of service potentially available, as quantified in Table B15.4.

**Table B15.4.** Intervals for lift groups.

| Quality of service | Interval (s) |
|---|---|
| Excellent | 25-35 |
| Acceptable–Office Blocks | 35-45 |
| –Hotels | 60 |
| –Residential | 90 |

*Example*

A group of four lifts, each car having a capacity of 21 persons (1600 kg) and a contract speed of 2·5 m/s serves a 20-storey office building (60 m travel). The lifts have a door width opening of 1·1 m and a door operating speed of 0·4 m/s. An average loading of 80% of full capacity is assumed. Determine the round trip time for any one lift, the capacity and the interval for the group.

1. Probable number of stops (Table B15.2) = 11

2. $T_{up} = 11\left(\dfrac{60}{11 \times 2{\cdot}5} + 2 \times 2{\cdot}5\right) = 79$ s

3. $T_{down} = \left(\dfrac{60}{2{\cdot}5} + 2 \times 2{\cdot}5\right) = 29$ s

4. $T_{do} = 2 \times 12 \times \dfrac{1{\cdot}1}{0{\cdot}4} = 66$ s

5. $T_t = 2 \times (21 \times 0{\cdot}8) = 34$ s

Thus round trip time = 208 s

$$\text{Capacity of group} = \frac{5 \times 60 \times 4 \times 21 \times 0{\cdot}8}{208}$$

= 97 persons per 5 minutes

$$\text{Interval for group} = \frac{208}{4} = 52 \text{ s}$$

## Miscellaneous

*Lift Speeds*

Recommended lift speeds for a number of applications and travels are set out in BS 5655 Part 6.

*Basements*

Local authorities frequently make the provision of more car parking space a condition of planning approval and the serving of basements by lifts can be significant. The addition of one basement, with fairly intensive use, can add 20 seconds to the round trip time of a lift. This can mean the addition of one lift to a 5 car group in order to maintain the required handling capacity if the facility for direct travel from this basement to upper floors is to be given. Consideration should always be given, therefore, to the installation of a separate lift to service basements and thus, for example, save the area of one lift through each of 20 floors.

*Lobbies*

Cul-de-sac lift lobbies are to be preferred and with groups of 4 lifts, 2 facing 2, a lobby provides the most efficient passenger circulation, as shown in Fig. B15.2. Due to the horizontal movement necessary by passengers when a landing call may be answered by any one of a group of lifts, central location of the landing call button is preferred. This horizontal movement, however, can cause delays to lifts if excessive, and for this reason, 3 lifts in a row is usually considered ideal for high efficiency but 4 lifts in a row should be considered the absolute maximum.

*Entrances*

Lifts with entrances on adjacent sides should be avoided wherever possible, as this greatly increases lift costs and can never give the same quality of performance as cars open at the front only. Whilst lift cars can be arranged to serve both back and front for slow-speed equipment, any arrangement other than entrances at the front only is likely to be less efficient and economic for lifts having a speed of 1–6 m/s and over.

*Counterweights*

The location of a lift counterweight can affect both lift equipment cost and structural design. The counterweight of an average passenger lift has a mass of approximately 2000 kg or more and, from the safety viewpoint, it is essential that no occupied space exists below this mass. In the event of a fire or major disaster severing the main lifting ropes, such a counterweight free falling over say 30 m would break through any structure in its path. If therefore building design does provide space underneath a counterweight then it is necessary to provide safety equipment to the counterweight similar to that provided for the lift car and this can increase installation costs considerably.

Counterweight safety gear may also require an increase in shaft size, without any increase in lift capacity.

**Control Systems**

*Automatic Push Button*

This is the simplest form of automatic lift control system and is capable only of accepting a single call at a time. Preference is given to accepting a call from the lift car control buttons over the call buttons on the landings, but the disadvantages of this system are obvious.

*Collective Control*

This system is the basis upon which all modern single, duplex or group supervisory control arrangements operate. It is arranged to accept a call upon a single pressure of the landing button and to memorise the call until such a time as a lift is capable of answering it. With such a system it would be undesirable to stop the upwards travelling lift in response to a passenger who wishes to travel down. Both up and down buttons are, therefore, provided on the landings and it is necessary for waiting passengers to indicate in which direction they wish to travel so that the lift car will stop only when travelling

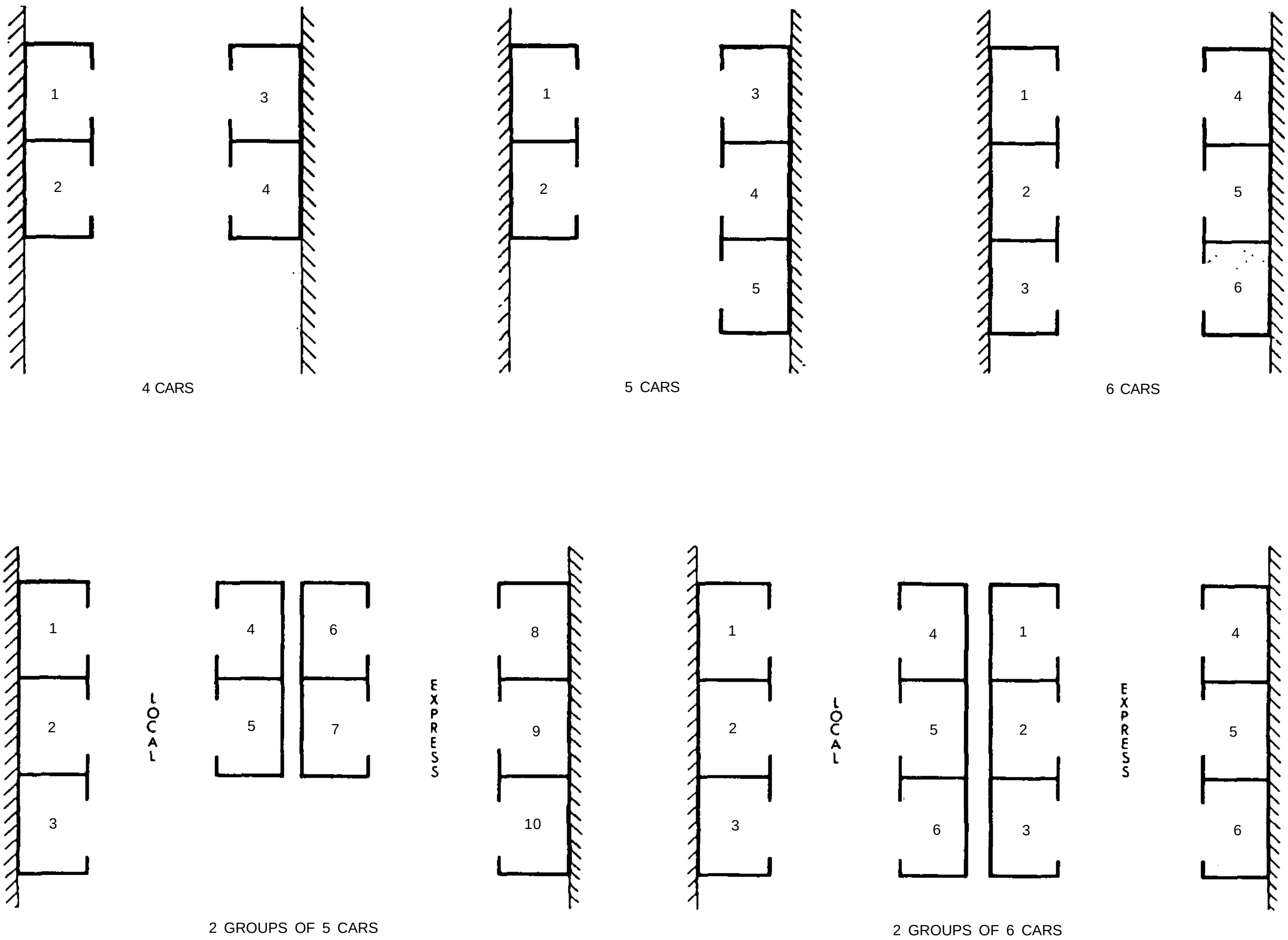



**Fig. B15.2** Lobby arrangements for lift groups.

in the correct direction. The control system, therefore, causes the lift car to stop as many times as necessary to pick up passengers at landings and will stop for car passengers as floors are reached. Refinements in control are possible such as weighing the 1ift car load to avoid stopping for additional passengers when the lift car is full, but maintaining the landing call until such time as the car is capable of accepting the passenger. When traffic from landing calls is mainly in the down direction, a 'down collective' system may be used which provides all the advantages of the collective control system but has only one button (DOWN) at each upper floor. This system is particularly suitable for flats, and hotels where no roof attractions exist.

The basic collective system is used for groups of lifts in both the simplified grouping systems, such as duplex and triplex, where two or three lifts are interconnected to operate from a common landing call system. It is also used for the various advanced forms of group supervisory systems where lift cars answer calls to an analytical supervisory system which computes the best methods of moving cars to deal with the overall group load at any instant in time and may also pre-position cars in anticipation of such a demand.

### Firefighting Lifts

In certain circumstances firefighting lifts may be required by a local authority. These must have certain special features (see also BS 5588):

(*a*) The effective platform area and contract load must not be less than 1·1 x l·4 m and 630 kg respectively.

(*b*) The speed of the lift must be such that it will run its full travel in not more than 60 seconds.

(*c*) The lift must have power operated doors giving a clear opening of not less than 0·8 m, and arranged to remain open whilst the car is at any floor, when under 'firefighting' conditions.

(*d*) An overriding 'Firefighting Lift Switch' must be provided at the service entrance floor level such that the firemen can obtain immediate master control of the lift. The operation of the switch must bring the lift to the fire control floor level without delay and with the doors parked open. Similarly, when under firefighting control all landing control buttons etc. must be inoperative and sole control vested in the car operating devices.

Provided that the siting meets with approval and that the necessary other features are incorporated in the control system, the firefighting lift may in normal circumstances be available for use as a passenger lift alone, or under group control.

## GOODS LIFTS AND SERVICE LIFTS

The design of goods lifts is dependent entirely upon the articles to be carried. Special care should be taken when lift cars are loaded by fork-lift trucks since, in these circumstances, the front wheels of the truck may move onto the lift platform whilst loading and unloading, giving a static loading condition far greater than that which the lift is designed to move. This can give rise to high loadings on machines, brakes, ropes, platforms etc., which must be allowed for in the initial design. Data on such loading should always be given to lift manufacturers as part of any performance specification.

BS 5655 Part 6 lists recommendations regarding speeds etc. for goods lifts, these generally being lower than those chosen for passenger lifts having the same travel. Traffic conditions and loading or unloading times are such that high speeds are neither necessary nor economic.

Where trolleys or trucks are to be carried or used for loading or unloading, levelling accuracy may be essential. Where this is required to within ±5 mm, variable voltage or hydraulic equipment should be used.

Service lifts are small goods lifts having car floor areas not exceeding 1·2 m$^2$ and compartment heights not exceeding 1·4 m. The serving level in many instances is at 0·85 m above floor level. Table B15.5 lists details of common classifications and speeds and it should be noted that in many cases such lifts are of unit construction.

**Table B15.5.** Service lift types.

| Duty | Load (kg) | Speed (m/s) |
|---|---|---|
| Document | 10 | 0·4 |
| Ledger etc. | 35 | 0·4 |
| Food service | 50 | 0·5 |
| Small goods | 100 | 0·25-0·5 |
| Canteen service | 110-150 | 0·25-0·5 |

## OBSERVATION LIFTS (WALL CLIMBERS)

In some situations, particularly for atria and where, for design purposes a lift is to be installed to the outside of a building, an observation type of lift is required.

Whilst these require all the mechanical and electrical items of a conventional passenger lift, a major difference is that the shaft walls which carry the guides and counterweight are kept to a physical minimum. Also the cars, which project past the side walls, are usually of glass construction to provide 'observation' both to the occupants of the lift car and those outside.

The handling capacity, speeds and controls are similar to those for conventional passenger lifts of similar size, but observation lifts are generally more expensive and require longer manufacture and installation times.

## ESCALATORS AND PASSENGER CONVEYORS

Escalators and passenger conveyors are suitable for applications where large numbers of passengers are to be moved and, particularly, to situations where 'tidal flow' traffic conditions occur such as in underground railway stations, airports etc. since they can be reversed in direction to suit varying demands throughout the day. Tread widths are normally between 0·6 and 1·05 m.

Refer to BS 5656: Escalators and passenger conveyors.

### Escalators

Escalators, for a given quality of service with dense traffic flow, will require a lesser horizontal space allocation than a similarly rated lift installation. Fig. B15.3 shows alternative arrangements for banks of escalators. The angle of inclination is normally 30° but, for a vertical rise of not more than 6 m and a speed not exceeding 0·5 m/s, a 35° angle is permissible.

Calculation of the theoretical capacity of an escalator or passenger conveyor is covered in BS 5656 and is summarised in Table B15.6.

**Table B15.6.** Theoretical capacity of escalators and passenger conveyors in persons/hour.

| Nominal width/ m | Rated speed/ m/s | | |
|---|---|---|---|
| | 0·5 | 0·65 | 0·75 |
| 0·6 | 4500 | 5850 | 6750 |
| 0·8 | 6750 | 8775 | 10125 |
| 1·0 | 9000 | 11700 | 13500 |

Since an escalator operates continuously, the driving motor normally runs at constant speed, although two-speed change-over operation is sometimes used to deal with variations in loads. Control equipment is, therefore, extremely simple.

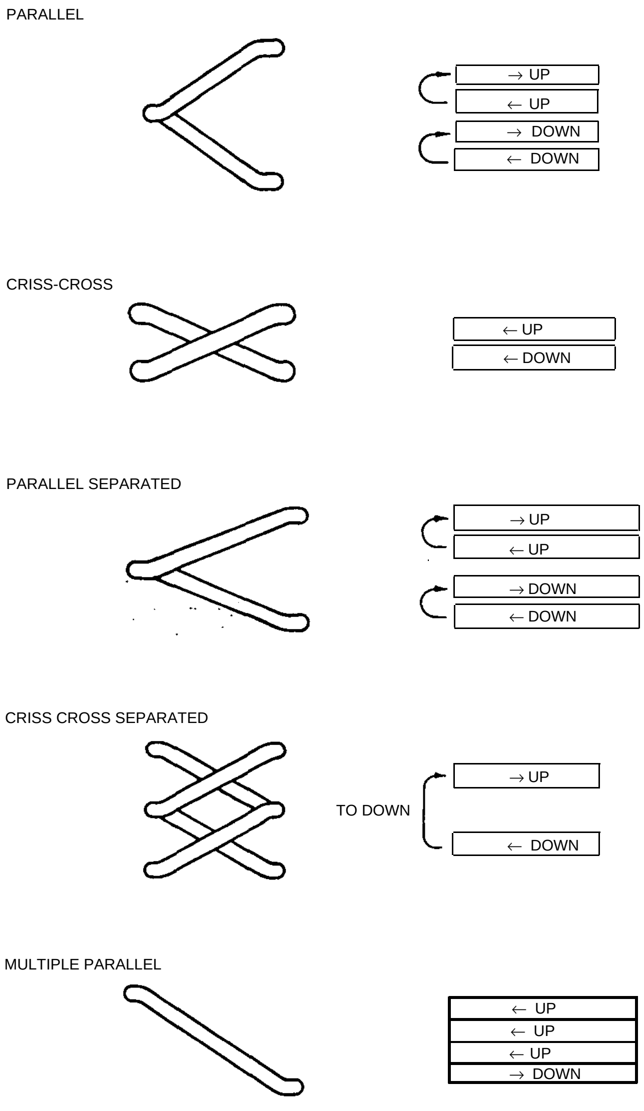



**Fig. B15.3.** Escalator arrangements.

### Passenger Conveyors

Passenger conveyors or moving pavements are adaptable installations for small vertical distances or for horizontal distances generally limited to about 300 m. Treadway speeds up to 0·75 m/s are permitted for slopes up to the limiting slope of 12°.

Conveyors are normally arranged to carry two persons side by side, as for the larger escalators, and are suitable for use by the infirm, for wheelchairs and for perambulators, wheeled shopping baskets etc. The treadways can be arranged to undulate to pass under or over obstacles e.g. a passenger underpass below a roadway.

As for escalators, control equipment is extremely simple.

## ENVIRONMENTAL FACTORS

For satisfactory operation, both in respect of the machinery and the occupants of the building, the installation of transportation systems has effects upon other building services.

### Heating and Cooling

Since most transportation installations are in heated buildings, no additional heating of any equipment is required.

However, with the increased use of static electrical equipment, many manufacturers now require the temperature in the motor room to be maintained between 5° and 35°C. This may necessitate some form of cooling in roof motor rooms.

Motor room temperature control to similar limits may be required for side ram hydraulic passenger lifts to assist the dissipation of heat built up in the oil, as most installations have the pump motor suspended in the oil tank for quietness. This may be particularly important for motor rooms internal to the building.

In exposed, unheated areas i.e. warehouses, engineering plant etc., heating to maintain a motor room temperature of 10° C is desirable to ensure satisfactory operation of control gear etc., by minimising condensation.

### Ventilation

For smoke extract purposes, a ventilation area of at least 1% of the plan area of the well must be provided at the top of the lift shaft. Individual Fire Officers may require this to be increased.

Most lift group installations providing continuous vertical circulation generate sufficient heat to require motor room ventilation, but for smaller lift groups a natural high and low louvered convection ventilation arrangement is normally satisfactory. For larger groups, mechanical ventilation may be necessary and the lift manufacturer should be consulted: an average figure for ventilation requirements can be calculated, although this may vary for different types of equipment, from the expression:

$$Q = 1{\cdot}3\ M\ v \qquad \text{B15.7}$$

where:

$Q$ = quantity of heat to be dissipated .. .. W

$M$ = contract load .. .. .. .. .. kg

### Noise

Noise from transportation systems can arise from a number of sources and although modern door operating devices and machine room equipment are relatively quiet, it is essential to consider the ambient noise level of surrounding accommodation when planning. Generally, therefore, lift equipment should be treated with the same respect with regard to airborne noise transmission as any other piece of machinery and should not be located adjacent to areas where ambient noise levels are low and where even relatively low levels of intermittent noise could be irksome. This is particularly important in flats and hotels where bedrooms are to be considered.

*Windage*

Every lift shaft is, of course, a chimney and in high rise buildings this can cause windage noises around lift entrances particularly at the top and bottom of the lift shaft. The location of lobbies in relation to building entrances and building external wind pressures and air turbulence can also affect this noise. A simple structural expedient may, however, be adopted in order to avoid aggravation of these problems by internal shaft pressure built up by the air-pump action of a lift car running in a structural shaft. Where groups of lifts are to be provided it is essential with high speed equipment, and always advantageous, to install the cars in open common lift wells, i.e. 2 or 3 lifts in a common structural well with dividing steel work for lift guide fixings. If, for structural reasons, this is not possible, then by-pass air holes as large as possible and frequent as possible should be provided in structural dividing walls between lift wells to allow air to flow freely from one shaft to the other in front of a moving lift car.

*Structural*

Most 1ift manufacturers take steps to isolate lift machines, motor-generators etc. from the building structure by suitable flexible mountings, and when these simple precautions are taken rarely are structure-borne noise problems encountered.

**Emergency Power Supplies**

The use of emergency power units for maintaining lift service during power failures has increased in recent years and this particularly applies to hospitals. The emergency supply normally caters for running all vital services and only a restricted lift service is available since normal economics dictate that it is not practical to supply an emergency power unit capable of running all the lifts in the building.

Arrangements can be made on the 1ift equipment for a simple type of emergency service, provided that the customer can deliver sufficient emergency power at the main 1ift control panel. When a power failure takes place then all the lifts will stop and the switchover to emergency power has to be put into operation. If only one lift is involved this is quite a simple matter but where several lifts are to be run from an emergency power unit, then it would probably be necessary to arrange for sequence starting of the installation. This would ensure that only the starting full load current of one lift would be applied at any time.

When an emergency power unit is installed, however, it is essential to provide means for absorbing power regenerated by the elevator motor generator set when this is running with over hauling loads such as a full load down.

**REGULATIONS**

Until recently there were no mandatory safety regulations affecting lift equipment other than where the building came under the jurisdiction of the Factories Act. The Shops, Offices and Railway Premises Act now, however extends the range of mandatory safety regulations similar to those previously required under the Factories Act and is at present administered by local authority Health Departments.

**Fire**

The Building Regulations 1985 give fire ratings of doors etc., which are applicable to lifts as a protected shaft. In addition to fire ratings however, local Fire Authorities reserve the right to establish special fire requirements for a particular building. As a general rule, whenever the building height is in excess of the extended height of local fire fighting apparatus, a fire cell is required from the fire access to the building, complete with escape staircase and a Firefighting Lift.

**Petroleum**

When a lift serves a basement car park, local authority Petroleum Regulations may apply if a ventilated lobby is not provided between the lift and the car parking space. This Regulation requires special arrangement of the electrical equipment which must be arranged for during manufacture. It is essential, therefore to notify the lift manufacturer when this condition exists.

**Maintenance and Inspection**

The main regulations covering the mandatory inspection or maintenance of lifts exist in the following documents:

(*a*) The Offices, Shops and Railway Premises (Hoists and Lifts) Regulation 1968. Statutory Instrument 1968 No. 849 Shops and Offices.

(*b*) The Health and Safety at Work Act 1974.

Insurance companies may insist on regular inspection either by their own inspectors or approved manufacturers' inspectors. Most lift manufacturers offer various types of maintenance contracts, the most popular being a fully comprehensive contract whereby a fixed annual charge covers all servicing and repairs.

**Compliance with BS 5655 Part 1**

Some of the more important provisions of Part 1: Lifts and service lifts: safety rules for the construction and installation of electric lifts of BS 5655 are outlined below:

*Shaft Lighting*

Shaft lights of the bulkhead type are required in the shaft and the pit at intervals of 7 m, and under the motor room floor. Lights should be two way switched from the lift motor room and from within the shaft adjacent to the entrance at the lowest landing served.

*Stop Pushes*

A 'stop' push should be mounted in the lift pit, adjacent to the entrance and positioned so that it can be operated from the landing with the doors open. There should also be a stop push in any top wheel room, again situated so that it can be safely operated from outside the lift shaft.

If the lift machinery is remote from or obstructed by other equipment from the main motor room switch then a separate stop push must be located adjacent to the moving parts.

*Car Top Control*

Suitable car top control for use by maintenance engineers should be fitted on the car roof.

This should consist of a stop push and constant pressure up and down push buttons, and should include a switch to cut out all normal controls to allow only control from the car roof. It should not interrupt the car door controls. Car top controls should also include an inspection lamp and socket outlet. The changeover switch, however, should automaticaly allow the lift to travel at 'inspection speed' which should not exceed 0·5 m/s.

*Motor Rooms*

There should not be any other equipment in lift motor rooms except that associated with the lifts. Rubber mats should be provided in front of and behind controllers.

Rope holes should be fitted with upstands. Handwinding or lowering procedures should be adequately shown on suitable notices and an 'Electric Shock Treatment' instruction should be mounted in the lift motor room.

A 'Danger' notice should be mounted outside the motor room door, which must be adequately and safely locked.

In the event of access to lift motor rooms by access ladders and trap doors, the access ladders should, where necessary, have hand holds and safety hoops, and trap doors should be counter-balanced.

## EQUIPMENT

### Traction Drive

The most common form of lift car drive is by traction where the car is moved by a traction rope drive using a counterbalance weight to maintain rope tension and to minimise the power necessary to move the car load by counterbalancing the dead weight of the car and a proportion of the contract load. The roping arrangement may be directed from car to counterweight, i.e. 1:1 or with muliplying pulleys to give 2:1 or 3:1 all as illustrated in Fig. B15.4(a), (b) and (c). For very high-speed lifts, 4 m/s and over, a double wrap roping arrangement is frequently used to provide adequate traction, whereby the ropes are passed from the car over the traction sheave, wrapped round the deflector sheave and passed back over the traction sheave for the second time before attachment to the counterweight as shown in Fig. B15.4(d).

Power may be transmitted to the traction sheave by one of two principal methods:

A *geared machine* driven by either:

(*a*) a single speed a.c. motor for contract speeds up to about 0·63 m/s;

(*b*) a two speed a.c. motor for contract speeds up to about 1 m/s, where high quality performance and floor levelling accuracy are not of prime importance. Pole changing a.c. motors are generally used with a 4:1 or 6:1 speed ratio;

(*c*) an a.c. variable speed motor controlled by static equipment for speeds of up to 1·6 m/s and in some instances 2 m/s, where high quality performance and levelling accuracy are important.

A *gearless machine* directly coupled to a d.c. shunt motor and controlled by the variable voltage system. Gearless equipment is generally used for all car speeds of 2 m/s and upwards.

### Hydraulic Drive

Some of the very early passenger lifts, many installed before 1900, were operated from hydraulic water power and in recent years the use of hydraulic power has been reintroduced, but using oil as the pumping medium with pressure supplied by a motor driven pump.

When used for passenger lifts, this equipment is generally restricted to 5 floors or less, with a maximum speed of 1 m/s, and is often of the side ram type requiring no well boring. The advantages are lower headrooms, less builder's work, and the ability to locate the motor room away from the lift, although it is generally recommended to be within 10 m of the lift shaft.

There are, however, some disadvantages, namely:

(*a*) loads are usually restricted to 1000 kg (13 persons) for single side acting rams;

(*b*) speeds are restricted to a maximum of 1 m/s;

(*c*) the number of journeys per hour are restricted to 60-80;

(*d*) sound insulation may require particular attention.

The main types of hydraulic lift arrangements are direct acting, side acting or indirect acting via a multiplying rope or chain, all illustrated in Fig. B15.5. For direct acting hydraulic lifts, well-boring is required.

### Drum Drive

Lift cars raised by winding the hauling rope round or down and descending under gravity are rarely used other than for service lifts up to 200 kg capacity. As no counterweight is provided, they suffer from inefficiency but some space is saved. Such an arrangement is shown in Fig. B15.6.

### Doors

Sliding lift doors or shutters are now generally used and fall into four main categories, as shown in Fig. B15.7.

*Centre Opening*

These allow speedy passenger transfer and are generally to be preferred to other types.

*Side Opening*

This is the cheapest form of door and the mechanism is simple and robust. It needs a disproportionate space within the well when open.

*Side Opening–two speed*

Has two leaves, both of which move in the same direction, one at twice the speed of the other.

**Table B15.7.** Types of electric lift: application data.

| Type | Application | Entrances | Power system and speeds | Machine |
|---|---|---|---|---|
| Light traffic passenger and passenger/ perambulator lifts | 5 and 8 person units for small offices and hotels. 8 and 10 person units for residential buildings and wheelchair access | Single panel sliding doors on car and on landings, power operated | Single speed for $v$ = 0·63 m/s<br>Two speed for $v$ = 0·75 and 1·0 m/s<br>Variable voltage for $v$ = 1·0 m/s | Geared |
| Stretcher/ passenger lifts | Residential buildings and homes for aged. Car for 8 person unit has a cupboard extension with a reduced height for stretchers | Single panel sliding doors on car and on landings, power operated | Single speed for $v$ = 0·63 m/s<br>Two speed for $v$ = 0·75 and 1·0 m/s | Geared |
| Bed/passenger lifts | 21 person unit for small hospitals, clinics, nursing homes and institutions. 26 and 32 person units, for general hospitals | Two panel side opening doors on car and on landings, power operated | Two speed for $v$ = 0·63 and 0·75 m/s<br>Variable voltage for $v$ = 0·75, 1·0 and 1·5 m/s | Geared |
| General purpose passenger lifts | Banks, office buildings, hotels etc. 8 and 10 person units have car depths suitable for wheelchairs | Two panel centre opening doors on car and on landings, power operated | Two speed for $v$ = 0·75 and 1·0 m/s<br>Variable voltage for $v$ = 1·0 and 1·5 m/s | Geared |
| Intensive traffic passenger lifts | Travels normally over 30 m for banks, offices, hotels etc. | Two panel centre opening doors on car and on landings, power operated | Variable voltage for all speeds | Gearless |
| General purpose goods lifts | Factories, industrial plants, warehouses | Collapsible sliding shutter doors on landings. Collapsible sliding shutter doors or midbar picket gates on car | Single speed for $v$ = 0·25 and 0·50 m/s<br>Two speed for $v$ = 0·25, 0·50, 0·63 and 1·0 m/s<br>Variable voltage for $v$ = 0·75 and 1·0 m/s | Geared |
| Heavy duty goods lifts | Factories, industrial plants, warehouses | Vertical bi-parting doors on landings. Vertical sliding panel door car | Single speed for $v$ = 0·25 m/s<br>Two speed for $v$ = 0·25, 0·50, 0·63 and 1·0 m/s<br>Variable voltage for $v$ = 0·50, 0·63 and 1·0 m/s | Geared |

*Bi-parting*

These part vertically and consist of two panels so interconnected that they move simultaneously downwards and upwards into the lift well.

Other arrangements of sliding doors which are available are two-speed, centre opening and three-speed side opening. These are used where a larger entrance width for a given shaft width is imperative, and can be very expensive due to the more complicated door gear and the increase in door panels.

**Application Data and Dimensions**

Table B15.7 gives a summary of application data for lifts, and BS 5655: Part 5 gives detailed dimension data. Table B15.8 and BS 5656 give typical dimensions for escalators.

**Table B15.8.** Dimensions for escalators.

| Details: Item | Details: Key to Fig. B15.8 | Typical leading dimensions (m) |
|---|---|---|
| Width, tread | A | 0·61 |
| tread | A | 0·81 |
| tread | A | 0·91 |
| tread | A | 1·01 |
| balustrade | B | A+0·31 |
| overall | C | B+0·38 |
| Balustrade height | D | 0·9 |
| | E | 1·2 |
| | F | 2·5 |
| | G | 2·0 |
| | J | 1·8 |
| Recommended clearance | K | 2·3 |
| | L | 1·8 |

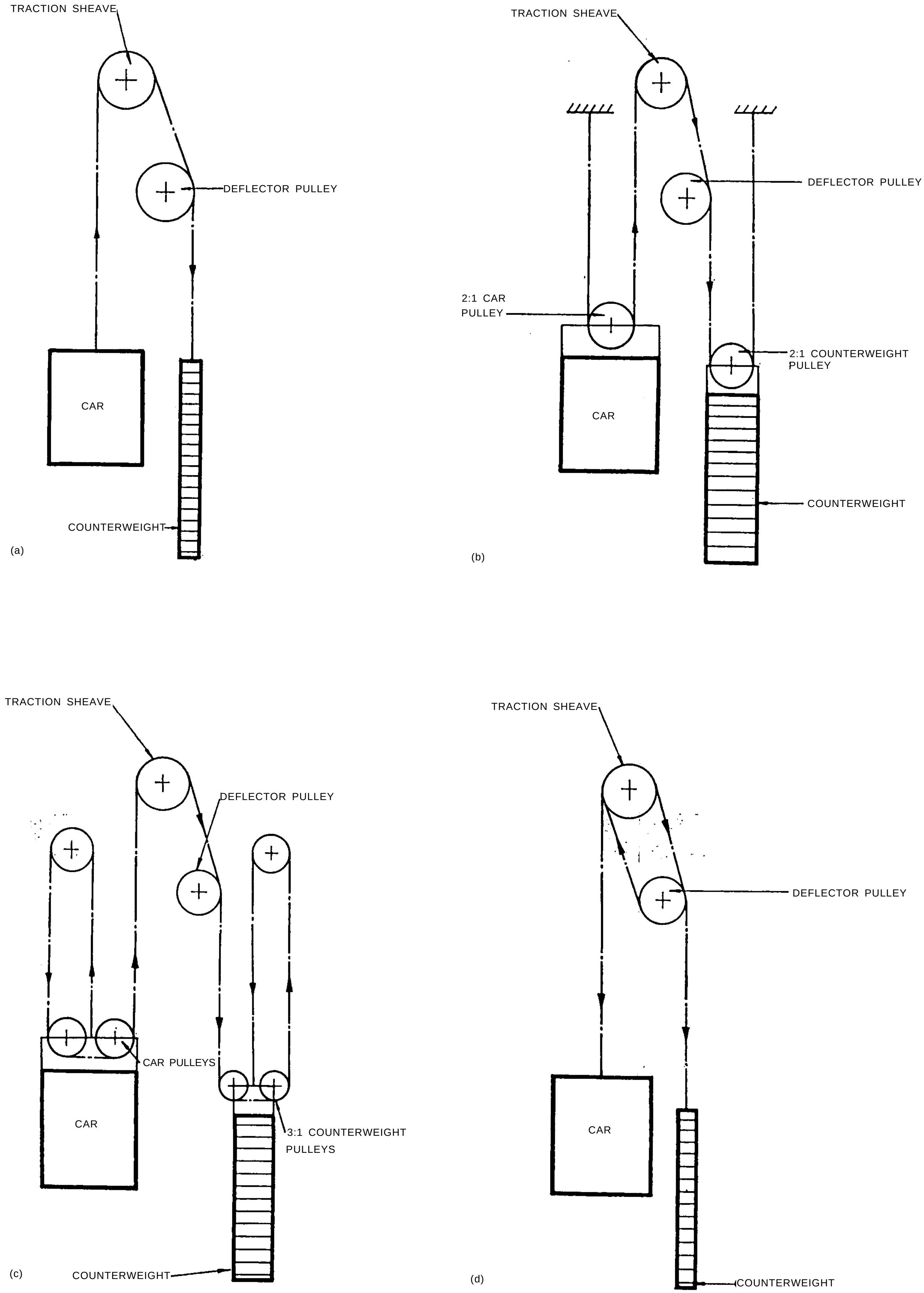



**Fig. B15.4.** Drive arrangements, traction.

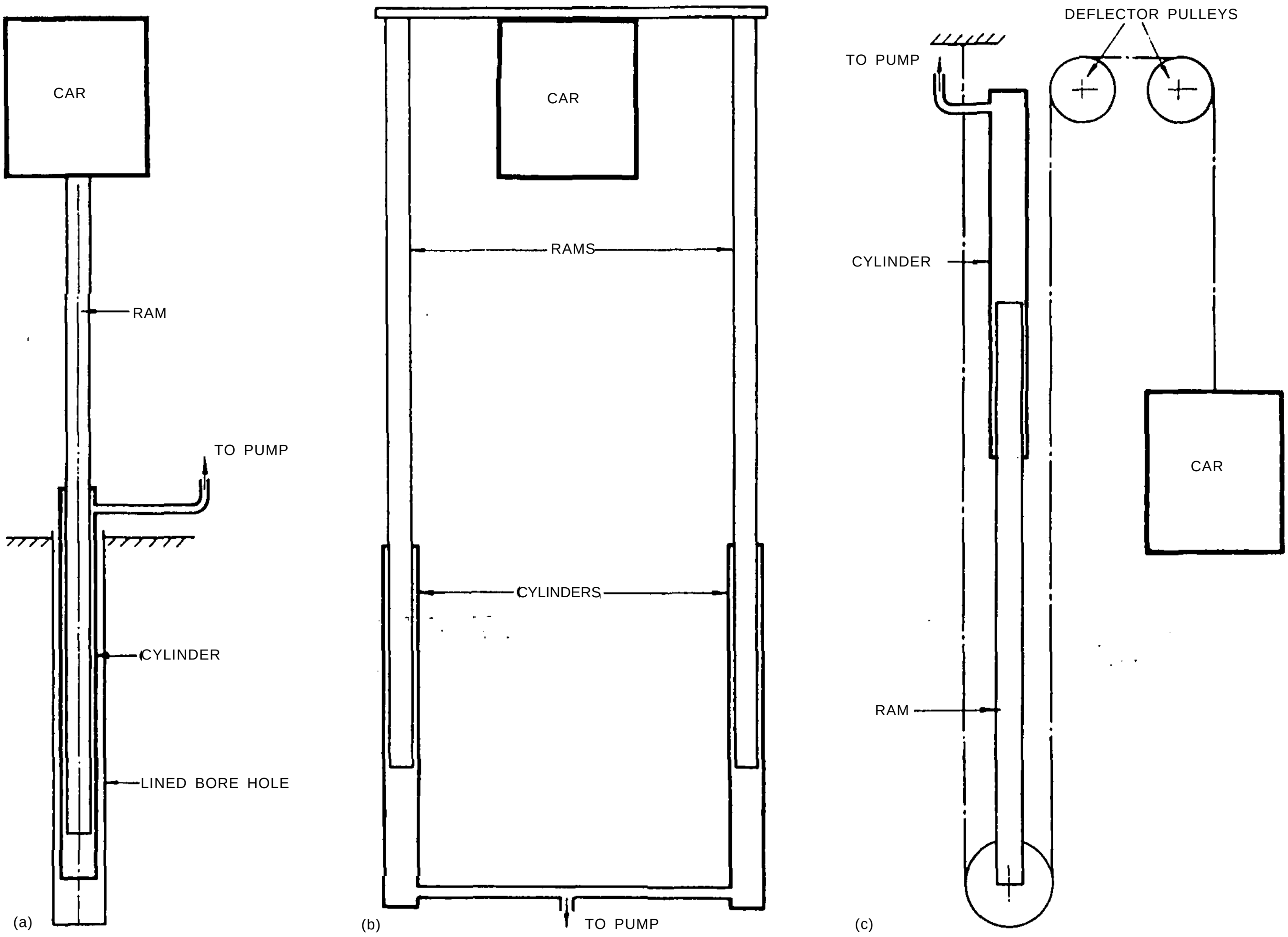



**Fig. B15.5.** Drive arrangements, hydraulic.

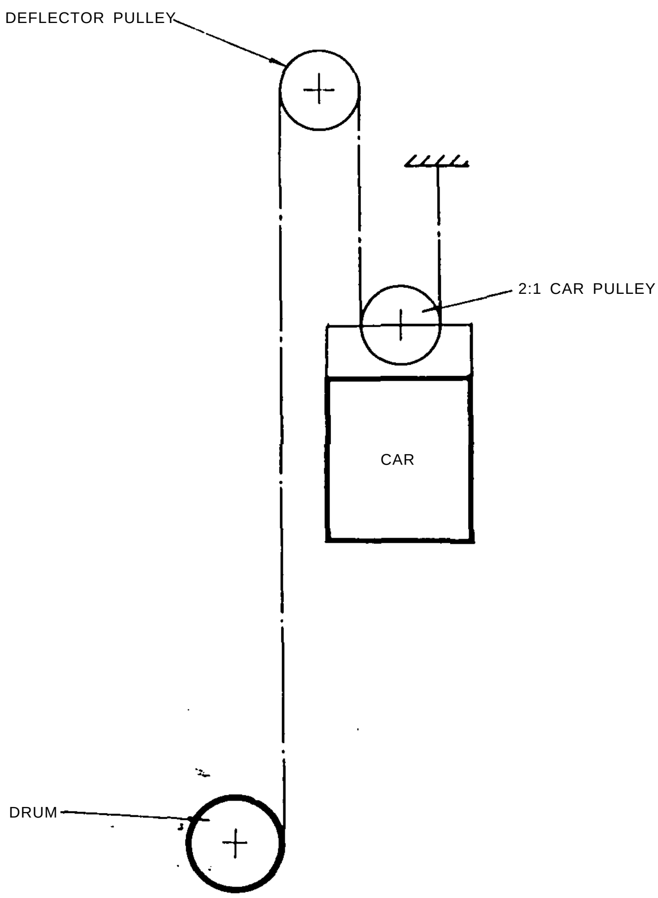



**Fig. B15.6.** Drive arrangements, drum.

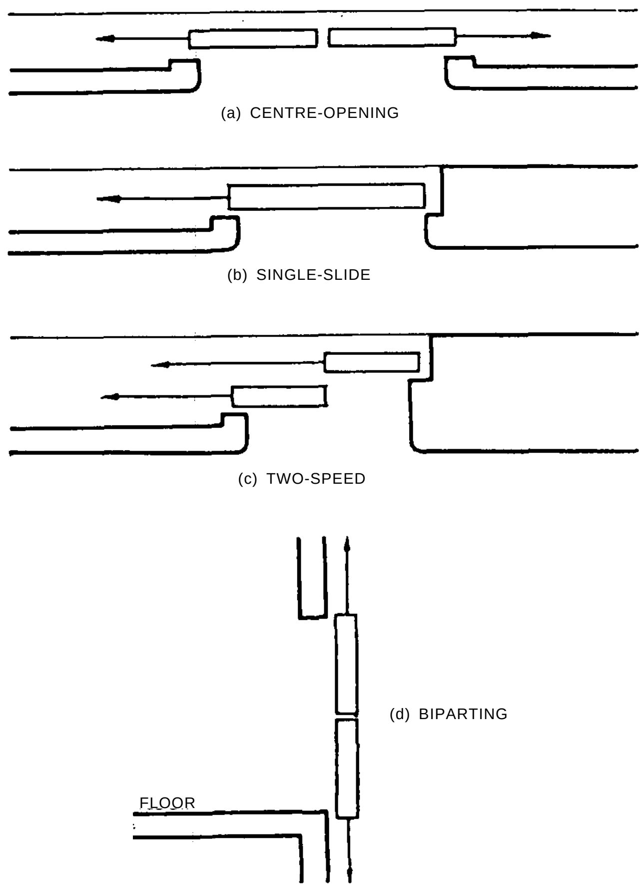



**Fig. B15.7.** Types of lift door.

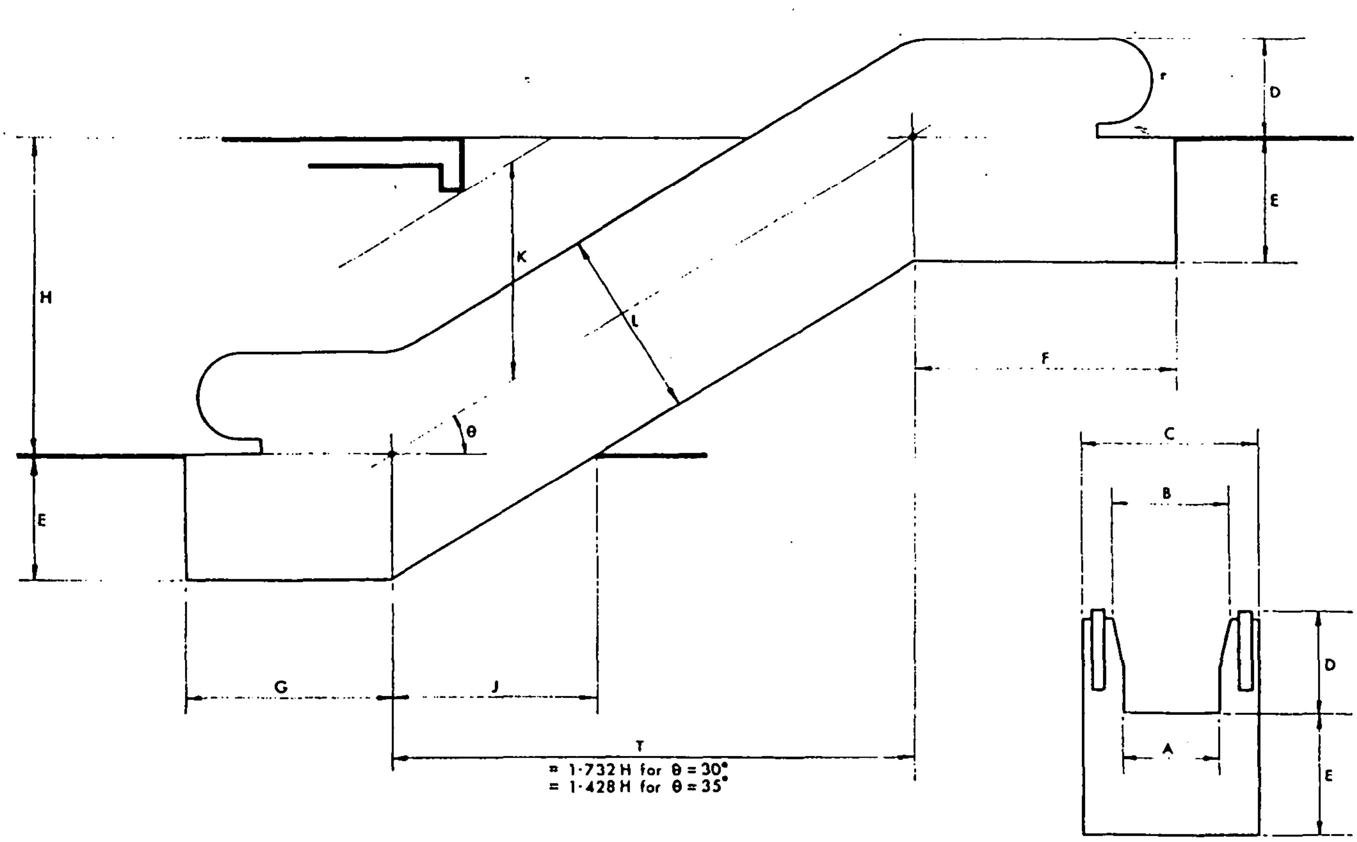



**Fig. B15.8.** Escalators, key to Table B15.8.

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The Health and Safety at Work etc. Act. HMSO, 1974.

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# SECTION B16 MISCELLANEOUS EQUIPMENT




*This edition of Section B16 first published: 1986*

# SECTION B16 MISCELLANEOUS EQUIPMENT

## Introduction

This section contains details of equipment which is either of such general application that it is relevant to several other Sections, or of such specialised interest that it would be incongruous elsewhere. By the nature of the subject matter this Section is overall fragmentary, but the items are arranged to give several sequences, viz.: piping, steam traps, valves, ductwork, dampers, pumps, compressed air.

## PIPEWORK

### Materials

Materials for service piping installations are commonly selected from tabulated data such as that included in BS 3202, summarised here as Table B16.1. Such listings are inevitably inadequate in scope and must continually be reviewed to take into account the emergence of new materials or specifications.

Table B16.2 lists the principal properties of a selection of plastics currently used in the manufacture of piping: parallel details of a few pipe metals are included for purposes of comparison.

**Table B16.1.** Materials recommended for service piping.

| Service | Material |
|---|---|
| Cold water | Copper, polythene, uPVC |
| Hot water | Copper, galvanised steel |
| Heating (hot water or steam) | Black steel, steel, copper |
| Steam | Black steel, steel, copper |
| Condensate | Copper |
| Gas (natural) | Black steel, steel, copper, aluminium, polypropylene |
| Liquefied petroleum gas | Copper |
| Compressed air | Galvanised steel, copper, aluminium |
| Vacuum | Copper, aluminium, black steel (medium), chemical lead for corrosive conditions |
| Distilled water | Stainless steel, glass, uPVC, polythene, aluminium |
| Demineralised water (hot or cold) | Stainless steel, glass, uPVC, polythene, aluminium |
| Drains and wastes | Stainless steel, chemical lead, cast iron, silicon iron (spun), chemical stoneware, vitreous enamel, rubber lined, glass lined, hard rubber, glazed fireclay, copper, glass, polythene, uPVC, polypropylene, chloroprene canvas hose |
| Fuel oils | Black steel, steel, copper |

To avoid external corrosion of steel pipework for buried or outdoor applications security bonded coatings may be used. The coating colour should be specified in accordance with BS 1710 Identification of Pipelines.

**Table B16.2.** Comparative properties of piping materials.

| Pipe material | Specific gravity | Specific heat kJ/kgK | Softening point (°C) | Heat distortion point at 1·82 MN/m² (°C) | Safe working temperature (°C) | Coefficient of linear expansion per °C × $10^{-6}$ | Thermal conductivity W/m²K | Weathering properties | Abrasion resistance |
|---|---|---|---|---|---|---|---|---|---|
| PVC: Normal impact | 1·40 | 1.10 | 75 | 70·0 | 60 | 55·1 | 0·179 | Excellent | Good |
| High impact | 1·37 | 1·10 | 73 | 65·0 | 49 | 75·1 | 0·189 | Fair | Good |
| Flexible | 1·37 | – | 70 | – | – | – | – | – | – |
| Polythene: Low density | 0·92 | 2·2 - 2·7 | 80 | 30·0 | 46 | 225·0 | 0·335 | Fair | Moderate |
| High density | 0·95 | 2·5 | 110 | 45·0 | 49 | 140·2 | 0·490 | Fair | Fair |
| Polypropylene | 0·91 | 1·9 | 130 | 52·8 | 85 | 140·2 | 0·149 | Good | Fair |
| ABS (acrylonitrile butadiene styrene) | 1·05 | 1·45 | 93 | 85·0 | 71 | 110·2 | 0·221 | Fair | Good |
| Nylon | 1·11 | 1·7 – 2·1 | 210 | 55·0 | 66 | 125·3 | 0·241 | Moderate | Very good |
| CAB (cellulose acetate butyrate) | 1·20 | 1·45 | 80 | 67·2 | 60 | 140·2 | – | – | – |
| PTFE | 2·18 | 0·96 | – | – | – | 120·1 | – | – | – |
| Polyester/glass fibre | 1·80 | 1·25 | – | 141·1 | 141 | – | – | Excellent | Good |
| Mild steel | 7·85 | 0·485 | – | – | – | 11·3 | 47·6 | Poor | – |
| Cast iron | 7·40 | 0·405 | – | – | – | 10·2 | 47·6 | Poor | Excellent |
| Copper | 8·91 | 0·390 | – | – | – | 16·9 | 172 | Excellent | – |
| Aluminium | 2·70 | 0·880 | – | – | – | 25·6 | 211 | – | – |
| Lead | 11·21 | 0·135 | – | – | – | 29·0 | 34·6 | Excellent | – |
| Stainless steel | 7·93 | 0·522 | – | – | – | 15·3 | 16·0 | Excellent | Excellent |

**Physical Dimensions**

The data listed in Table B16.3 have been derived from the appropriate British Standards, as follows:

| Column | | | Units |
|---|---|---|---|
| 1–6 | Directly from the appropriate British Standard | | |
| 7 | Internal area | $= \dfrac{\pi d^2}{4}$ .. .. | $mm^2$ |
| 8 | Metal area | $= \dfrac{\pi}{4}(D^2 - d^2)$ .. | $mm^2$ |
| 9 | Surface area/m | $= \dfrac{\pi D}{10^3}$ .. .. | $m^2$ |
| 10 | Moment of inertia | $= \dfrac{\pi}{64}\dfrac{(D^4 - d^4)}{10^4}$ .. | $cm^4$ |
| 11 | Section modulus | $= \dfrac{\pi}{32D}\dfrac{(D^4 - d^4)}{10^3}$ .. | $cm^3$ |
| 12 | Directly from the appropriate British Standard | | |
| 13 | Water content | $= \dfrac{\pi d^2}{4 \times 10^3}$ .. | litre/m |

The following definitions apply to the symbols:

$D$ = mean outside diameter .. .. .. mm

$d$ = mean inside diameter .. .. .. mm

**Application of Pipe Supports**

Data for pipe support spacing may be calculated, alternately, for conditions of known stress or known deflection. The former argument was chosen for the preparation of column 14 of Table B16.3, the stress value chosen being 10% of the total permissible. The values were derived from the expression:

$$\text{Centres of supports} = \left(\frac{10^{-4}\, S I}{W(D + d)}\right)^{1/2} \quad .. \quad \text{m}$$

where:

$I$ = moment of inertia .. .. .. .. $cm^4$

$W$ = total mass/m, pipe and water .. .. kg/m

$S$ = bending stress due to support spacing .. $N/m^2$

## FLEXIBILITY IN PIPEWORK

**Amount of Expansion in Pipework**

Table B16.4 gives the amount of expansion for a range of pipe materials for lengths of 1 m, at various temperature differences from that at which the pipe is assumed to be unstressed.

**Pipeline Sets in Steel Pipe**

Tables B16.5 to 8 may be used to determine the flexibility that can be built into a pipeline, by means of bends and loops, to absorb expansion. These tables give the proportions that the legs of a pipe loop or set must have for a given amount of expansion. The data are based on the pipeline starting in the unstressed cold condition. Twice the amount of expansion can be asborbed if 50% cold draw is applied on installation.

**Table B16.4.** Expansion of pipes.

| Material | Expansion (mm/m) | | | | |
|---|---|---|---|---|---|
| | Mild/ carbon steel | Copper | Cast iron | Stain-less steel | Poly-propylene HD Poly-ethylene CAB |
| ø Coefficient of linear expansion per °C × $10^{-6}$ / * Operating temperature difference (°C) | 11·3 | 16·9 | 10·2 | 15·3 | 140·2† |
| 5 | 0·055 | 0·085 | 0·051 | 0·077 | 0·701 |
| 10 | 0·113 | 0·169 | 0·102 | 0·153 | 1·402 |
| 15 | 0·170 | 0·254 | 0·153 | 0·230 | 2·103 |
| 20 | 0·227 | 0·338 | 0·204 | 0·306 | 2·804 |
| 25 | 0·284 | 0·429 | 0·256 | 0·382 | 3·505 |
| 30 | 0·340 | 0·508 | 0·307 | 0·459 | 4·206 |
| 40 | 0·454 | 0·677 | 0·409 | 0·611 | 5·608 |
| 50 | 0·567 | 0·846 | 0·511 | 0·765 | 7·010 |
| 60 | 0·680 | 1·105 | 0·613 | 0·917 | 8·412‡ |
| 70 | 0·794 | 1·184 | 0·715 | 1·069 | 9·814‡ |
| 80 | 0·907 | 1·354 | 0·818 | 1·224 | 11·216‡ |
| 90 | 1·021 | 1·523 | 0·920 | 1·377 | 12·618‡ |
| 100 | 1·134 | 1·692 | 1·022 | 1·530 | 14.020‡ |
| 110 | 1·247 | 1·861 | 1·124 | 1·682 | 15·422‡ |
| 120 | 1·361 | 2·030 | 1·226 | 1·833 | 16·824‡ |
| 130 | 1·474 | 2·200 | 1·329 | 1·988 | 18·226‡ |
| 140 | 1·588 | 2·369 | 1·431 | 2·140 | 19·628‡ |
| 150 | 1·701 | 2·538 | 1·533 | 2·294 | 21·030‡ |

ø Values derived from Table B16.2
* Temperature of medium - ambient temperature
† Factors may be used for other plastics (see Table B16.2)
‡ Temperatures may exceed the limiting level for the material

The symbols in Tables B16.5 to 8 apply to lengths of pipes between guides, or between anchors if there are no guides, that are otherwise unrestrained. The pipeline must be supported either on hangers of lengths that do not cause constraint, or on sliding supports which allow both axial and sideways movement. The amount of expansion relates in all cases to the maximum total expansion to be absorbed in any one direction between anchors. Screwed or flanged fittings should not be used in making up the bends taking expansion. Fig. B16.1 to 3 show the pipeline configurations referred to below.

*Pipe with one or Two Pulled Bends*

Using Table 16.4, the total expansion between the anchor and bend may be calculated. The relationship between $L^2/x$ and $L/h$ can then be read from Tables B16.5 and 6 for any pipe size. Where 50% cold draw is to be applied, the value of expansion for use in calculations is half the total expansion. Tables B16.5 and 6 apply to sets where $L$ is greater than $h$. The pipeline must be designed for $L/h$ not greater than the value obtained from Tables B16.5

and 6. If $L/h$ is too large, either the offset must be increased or, if this is not possible, the other anchor must be brought closer to the bend to reduce $L$ and $x$.

If the pipeline is such that no value is shown for $L/h$ (within the appropriate limit lines), this means that the pipeline would be liable to buckle and the amount of expansion to be taken up must be reduced by shortening for the bend and anchors to give smaller values to $L$ and $x$, so reducing the corresponding value of $L^2/x$ until a tabulated value for $L/h$ is obtained.

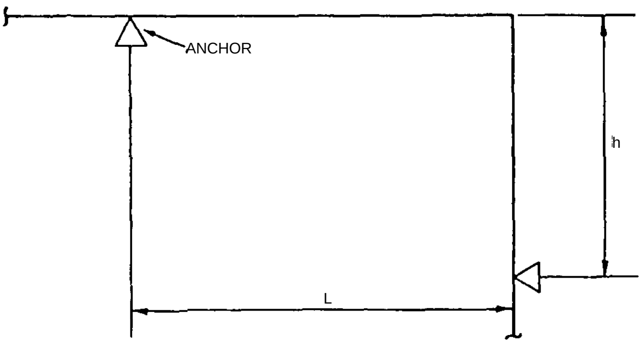



**Fig. B16.1.** Pipe with one pulled bend.

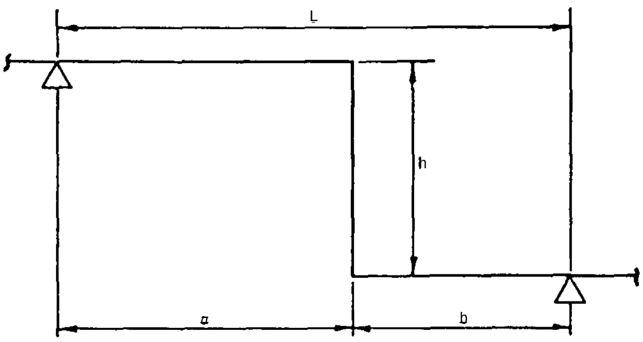



**Fig. B16.2.** Pipe with two pulled bends.

*Pipe with Expansion Loop*

If the loop is not symmetrically placed in the pipeline higher stresses are set up than if it is. To allow for this, the value of $L^2/x$ for the loop must be multiplied by the factor given in Table B16.7 before being used with Table B16.8. Note that the most economical loop, having smallest offset and producing the least anchor load, is obtained when $a = b$ and $g/L = 0{\cdot}5$.

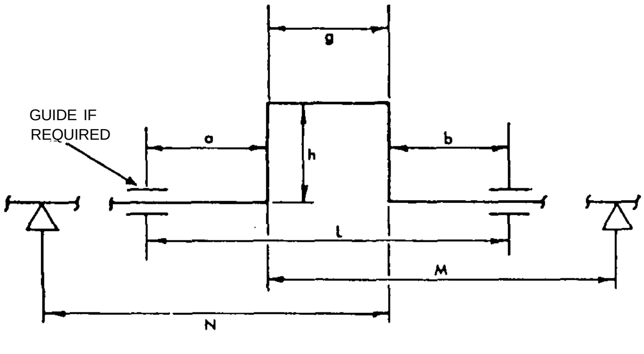



**Fig. B16.3.** Pipe with expansion loop.

*Guides for Pipe with One or Two Pulled Bends*

It is not possible to use guides with these pipeline configurations to eliminate a tendency to buckle as the reduction in effective length that they would give (without reducing the amount of expansion to be absorbed) is cancelled by the loss of flexibility.

*Guides for Pipes with Expansion Loops*

If the loop is such that no value is shown for $L/h$ (within the appropriate limit lines) this means that guides are required, preferably symmetrically, between the anchors and the loop. Try positions of these to give a smaller value of $L$, so reducing the corresponding value of $L^2/x$ (and increasing $x/L$) until a tabulated value of $L/h$ is obtained.

Further guides may be required on the expanding pipeline before the anchors, depending both on the size of the pipe and the axial thrust it carries.

**Anchor Loads**

The total anchor load comprises the force necessary to deflect the pipe bend(s) plus the total force required to overcome the friction at the supports between the bend on the longest leg and its anchor.

In the case of the pipe loop, the total anchor load comprises the force necessary to deflect the pipe loop plus the total force required to overcome the friction at all the guides and supports over the distance $M$ or $N$ shown in Fig. B16.3.

*Calculation of Anchor Loads*

The anchor load is calculated from the data for the longest leg of the offset or loop. The anchors at both sides of the offset or loop are then designed to carry this thrust both axially and laterally.

In friction and sliding supports, a safe figure to use is 30 N/m of pipe for every 25 mm of pipe diameter. For pipes on hangers, the frictional force is reduced and 15 N/m of pipe for every 25 mm of pipe diameter can be used provided drop rod lengths are sufficient not to cause constraint.

The following equations can be used to calculate anchor thrusts:

*Steel Pipes on Supports*

(*a*) One bend
$T = F + 1{\cdot}2\ LD$ .. .. .. .. .. B16.1

(*b*) Two Bends
$T = F + 1{\cdot}2\ aD$ .. .. .. .. .. B16.2

(*c*) Expansion loop
$T = F + 1{\cdot}2\ MD$ .. .. .. .. .. B16.3

*Steel Pipes on Hangers*

(*a*) One bend
$T = F + 0{\cdot}6\ LD$ .. .. .. .. .. B16.4

(*b*) Two bends
$T = F + 0{\cdot}6\ aD$ .. .. .. .. .. B16.5

(*c*) Expansion loop
$T = F + 0{\cdot}6\ MD$ .. .. .. .. .. B16.6

**Table B16.3.** Piping data.

| Nominal size† (mm) | Material* | Wall thickness (mm) | Diameters (mm) External Maximum | Diameters (mm) External Minimum | Diameters (mm) Mean internal | Areas Internal cross-sectional (mm$^3$) | Areas Metal cross-sectional (mm$^3$) | Areas Surface per metre length (m$^3$) | Moment of inertia (cm$^3$) | Section modulus (cm$^3$) | Weight of pipe (kg/m) | Water content (litre/m) | Centres for supports (m) |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 10 | Heavy steel | 2·95 | 17·4 | 16·8 | 11·2 | 98·7 | 130 | 0·054 | 0·33 | 0·393 | 1·02 | 0·098 | 1·7 |
| | Medium steel | 2·35 | 17·4 | 16·8 | 12·4 | 121 | 108 | 0·054 | 0·29 | 0·344 | 0·85 | 0·121 | 1·7 |
| 12 | Copper (Table X) | 0·60 | 12·045 | 11·965 | 10·8 | 91·7 | 21·6 | 0·038 | — | — | 0·19 | 0·092 | 1·0 |
| | Copper (Table Y) | 0·80 | 12·045 | 11·965 | 10·4 | 85·0 | 28·2 | 0·038 | — | — | 0·25 | 0·085 | 1·0 |
| | Copper (Table Z) | 0·50 | 12·045 | 11·965 | 11·0 | 95·1 | 18·2 | 0·038 | — | — | 0·16 | 0·095 | 1·0 |
| 15 | Heavy steel | 3·25 | 21·7 | 21·1 | 14·9 | 175 | 186 | 0·067 | 0·79 | 0·736 | 1·45 | 0·175 | 2·0 |
| | Medium steel | 2·65 | 21·7 | 21·1 | 16·2 | 205 | 155 | 0·067 | 0·71 | 0·656 | 1·22 | 0·205 | 2·0 |
| | Copper (Table X) | 0·70 | 15·045 | 14·965 | 13·6 | 145 | 31·6 | 0·047 | — | — | 0·28 | 0·145 | 1·4 |
| | Copper (Table Y) | 1·00 | 15·045 | 14·965 | 13·0 | 133 | 44·1 | 0·047 | — | — | 0·39 | 0·133 | 1·4 |
| | Copper (Table Z) | 0·50 | 15·045 | 14·965 | 14·0 | 154 | 22·9 | 0·047 | — | — | 0·20 | 0·154 | 1·4 |
| 20 | Heavy steel | 3·25 | 27·2 | 26·6 | 20·4 | 326 | 243 | 0·085 | 1·75 | 1·29 | 1.90 | 0.326 | 2·4 |
| | Medium steel | 2·65 | 27·2 | 26·6 | 21·6 | 367 | 203 | 0·085 | 1·50 | 1·11 | 1·58 | 0·367 | 2·4 |
| | Copper (Table X) | 0·90 | 22·055 | 21·975 | 20·2 | 321 | 59·6 | 0·069 | — | — | 0·52 | 0·321 | 1·4 |
| | Copper (Table Y) | 1·20 | 22·055 | 21·975 | 19·6 | 302 | 78·3 | 0·069 | — | — | 0·69 | 0·302 | 1·4 |
| | Copper (Table Z) | 0·60 | 22·055 | 21·975 | 20·8 | 340 | 40·2 | 0·069 | — | — | 0·35 | 0·340 | 1·4 |
| 25 | Heavy steel | 4·05 | 34·2 | 33·4 | 25·7 | 518 | 380 | 0·106 | 4·29 | 2·54 | 2·97 | 0·518 | 2·7 |
| | Medium steel | 3·25 | 34·2 | 33·4 | 27·3 | 586 | 312 | 0·106 | 3·70 | 2·20 | 2·44 | 0·586 | 2·7 |
| | Copper (Table X) | 0·90 | 28·055 | 27·975 | 26·2 | 540 | 76·7 | 0·085 | — | — | 0·68 | 0·540 | 1·7 |
| | Copper (Table Y) | 1·20 | 28·055 | 27·975 | 25·6 | 516 | 101 | 0·085 | — | — | 0·89 | 0·516 | 1·7 |
| | Copper (Table Z) | 0·60 | 28·055 | 27·975 | 26·8 | 565 | 51.7 | 0·085 | — | — | 0·46 | 0·565 | 1·7 |
| 32 | Heavy steel | 4·05 | 42·9 | 42·1 | 34·3 | 927 | 490 | 0·134 | 9·16 | 4·31 | 3·84 | 0·926 | 2·7 |
| | Medium steel | 3·25 | 42·9 | 42·1 | 35·9 | 1016 | 461 | 0·134 | 7·74 | 3·64 | 3·14 | 1·016 | 2·7 |
| | Copper (Table X) | 1·20 | 35·07 | 34·99 | 32·6 | 837 | 128 | 0·110 | — | — | 1·12 | 0·837 | 1·7 |
| | Copper (Table Y) | 1·50 | 35·07 | 34·99 | 32·0 | 806 | 158 | 0·110 | — | — | 1·39 | 0·806 | 1·7 |
| | Copper (Table Z) | 0·70 | 35·07 | 34·99 | 33·6 | 889 | 75·5 | 0·110 | — | — | 0·67 | 0·889 | 1·7 |
| 40 | Heavy steel | 4·05 | 48·8 | 48·0 | 40·2 | 1272 | 566 | 0·152 | 13·98 | 5·79 | 4·43 | 1·271 | 3·0 |
| | Medium steel | 3·25 | 48·8 | 48·0 | 41·9 | 1376 | 461 | 0·152 | 11.78 | 4·87 | 3·61 | 1·376 | 3·0 |
| | Copper (Table X) | 1·20 | 42·07 | 41·99 | 39·6 | 1234 | 154 | 0·132 | — | — | 1·36 | 1·234 | 2·0 |
| | Copper (Table Y) | 1·50 | 42·07 | 41·99 | 39·0 | 1197 | 191 | 0·132 | — | — | 1·69 | 1·197 | 2·0 |
| | Copper (Table Z) | 0·80 | 42·07 | 41·99 | 40·4 | 1284 | 104 | 0·132 | — | — | 0·91 | 1·284 | 2·0 |
| 50 | Heavy steel | 4·50 | 60·8 | 59·8 | 51·3 | 2070 | 784 | 0·189 | 30·8 | 10·2 | 6·17 | 2·070 | 3·4 |
| | Medium steel | 3·65 | 60·8 | 59·8 | 53·0 | 2205 | 651 | 0·189 | 26·2 | 8·70 | 5·10 | 2·205 | 3·4 |
| | Copper (Table X) | 1·20 | 54·07 | 53·99 | 51·6 | 2095 | 199 | 0·170 | — | — | 1·76 | 2·095 | 2·0 |
| | Copper (Table Y) | 2·00 | 54·07 | 53·99 | 50·0 | 1965 | 327 | 0·170 | — | — | 2·88 | 1·965 | 2·0 |
| | Copper (Table Z) | 0·90 | 54·07 | 53·99 | 52·2 | 2145 | 150 | 0·170 | — | — | 1·33 | 2·145 | 2·0 |
| 65 | Heavy steel | 4·50 | 76·6 | 75·4 | 67·0 | 3530 | 1005 | 0·239 | 64·5 | 17·0 | 7·90 | 3·530 | 3·7 |
| | Medium steel | 3·65 | 76·6 | 75·4 | 68·7 | 3700 | 831 | 0·239 | 54·5 | 14·3 | 6·51 | 3·700 | 3·7 |
| | Copper (Table X) | — | — | — | 64·3 | 3245 | — | — | — | — | — | 3·245 | 2·0 |
| | Copper (Table Y) | — | — | — | 63·1 | 3125 | — | — | — | — | — | 3·125 | 2·0 |
| | Copper (Table Z) | — | — | — | 64·7 | 3285 | — | — | — | — | — | 3·285 | 2·0 |
| 80 | Heavy steel | 4·85 | 89·5 | 88·1 | 79·0 | 4905 | 1285 | 0·279 | 114 | 25·6 | 10·1 | 4·905 | 3·7 |
| | Medium steel | 4·05 | 89·5 | 88·1 | 80·7 | 5115 | 1080 | 0·279 | 97·0 | 21·8 | 8·47 | 5·115 | 3·7 |
| | Copper (Table X) | 1·50 | 76·3 | 76·15 | 73·2 | 4210 | 352 | 0·239 | 24·4 | — | 3·11 | 4·210 | 2·4 |
| | Copper (Table Y) | 2·00 | 76·3 | 76·15 | 72·2 | 4100 | 467 | 0·239 | 31·9 | — | 4·11 | 4·100 | 2·4 |
| | Copper Table Z) | 1·20 | 76·3 | 76·15 | 73·8 | 4280 | 283 | 0·239 | 19·9 | — | 2·50 | 4·280 | 2·4 |
| 100 | Heavy steel | 5·40 | 114·9 | 113·3 | 103·3 | 8380 | 1840 | 0·358 | 272 | 47·7 | 14·4 | 8·380 | 4·1 |
| | Medium steel | 4·50 | 114·9 | 113·3 | 105·1 | 8680 | 1540 | 0·358 | 231 | 40·6 | 12·1 | 8·680 | 4·1 |
| | Copper (Table X) | 1·50 | 108·25 | 108·0 | 105·1 | 8680 | 504 | 0·340 | 71·4 | — | 4·45 | 8·680 | 2·7 |
| | Copper (Table Y) | 2·00 | 108·25 | 108·0 | 103·1 | 8355 | 832 | 0·340 | 115 | — | 7·33 | 8·355 | 2·7 |
| | Copper (Table Z) | 1·20 | 108·25 | 108·0 | 105·7 | 8780 | 405 | 0·340 | 71·2 | — | 3·57 | 8·780 | 2·7 |
| 125 | Heavy steel | 5·40 | 140·6 | 138·7 | 127·7 | 13 050 | 2270 | 0·438 | 520 | 73·4 | 17·8 | 13·05 | 4·4 |
| | Medium steel | 4·85 | 140·6 | 138·7 | 129·8 | 13 250 | 2065 | 0·438 | 470 | 67·4 | 16·2 | 13·25 | 4·4 |
| | Copper (Table X) | 1·50 | 133·5 | 133·25 | 130·4 | 13 350 | 621 | 0·419 | 134 | — | 5·47 | 13·35 | 3·0 |
| | Copper (Table Y) | — | — | — | — | — | — | — | — | — | — | — | 3·0 |
| | Copper (Table Z) | 1·50 | 133·5 | 133·5 | 130·4 | 13 350 | 621 | 0·419 | 134 | — | 5·47 | 13·35 | 3·0 |
| 150 | Heavy steel | 5·40 | 166·1 | 164·1 | 154·3 | 18 700 | 2700 | 0·518 | 862 | 105 | 21·2 | 18·70 | 4·8 |
| | Medium steel | 4·85 | 166·1 | 164·1 | 155·3 | 18 950 | 2455 | 0·518 | 787 | 95·4 | 19·2 | 18·95 | 4·8 |
| | Copper (Table X) | 2·00 | 159·5 | 159·25 | 155·4 | 18 950 | 988 | 0·501 | 304 | — | 8·71 | 18·95 | 2·7 |
| | Copper (Table Y) | — | — | — | — | — | — | — | — | — | — | — | 2·7 |
| | Copper (Table Z) | 1·50 | 159·5 | 159·25 | 156·4 | 19 200 | 743 | 0·501 | 230 | — | 6·55 | 19·20 | 2·7 |
| 200 | Steel | 4·88 | — | — | 209·3 | 34 400 | 3280 | 0·689 | 1880 | 172 | 25·9 | 34·42 | 5·1 |
| 250 | Steel | 6·35 | — | — | 260·4 | 53 250 | 5320 | 0·859 | 4745 | 347 | 42·0 | 53·24 | 5·8 |
| 300 | Steel | 7·14 | — | — | 309·6 | 75 300 | 7080 | 1·018 | 8865 | 547 | 55·8 | 75·30 | 6·1 |

* Materials are to the following British Standards: Heavy steel –BS 1387, Medium steel–BS 1387 } Steel stubes and tubulars suitable for screwing to BS 21 pipe threads.
Copper –BS 2871
Steel –BS 806 For dimensions see BS 3601 to 5. Steel pipes and tubes for pressure purposes.

† Nominal size generally approximates to bore of steel pipes and O.D. of copper pipes.

**Table B16.5.** Flexibility of single pipe set (Fig. B16.1).

| $\frac{L^3}{x}$ | Maximum permissible values of $\frac{L}{h}$ for steel pipelines of stated nominal bore (mm) | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | 15 | 20 | 25 | 32 | 40 | 50 | 65 | 80 | 100 | 125 | 150 |
| 0·5 | 1·6 | 1·4 | — | — | — | — | — | — | — | — | — |
| 1·0 | 2·3 | 2·0 | 1·8 | 1·6 | 1·5 | 1·4 | — | — | — | — | — |
| 2·0 | 3·4 | 3·0 | 2·7 | 2·4 | 2·2 | 2·0 | 1·8 | 1·7 | 1·4 | — | — |
| 4·0 | 5·0 | 4·4 | 3·9 | 3·5 | 3·2 | 2·9 | 2·6 | 2·4 | 2·1 | 1·9 | 1·7 |
| 6·0 | 6·3 | 5·5 | 4·9 | 4·4 | 4·0 | 3·6 | 3·3 | 3·0 | 2·6 | 2·4 | 2·1 |
| 8·0 | 7·4 | 6·5 | 5·8 | 5·1 | 4·7 | 4·2 | 3·8 | 3·6 | 3·0 | 2·7 | 2·5 |
| 10·0 | 8·3 | 7·3 | 6·6 | 5·8 | 5·3 | 4·8 | 4·3 | 4·0 | 3·4 | 3·0 | 2·8 |
| 15·0 | 10·5 | 9·2 | 8·2 | 7·2 | 6·6 | 6·0 | 5·4 | 5·0 | 4·3 | 3·8 | 3·4 |
| 20·0 | B 12·0 | 11·0 | 9·7 | 8·5 | 7·8 | 7·0 | 6·3 | 5·8 | 5·0 | 4·4 | 4·0 |
| 30·0 | A 15·0 | 13·5 | 12·0 | 10·5 | 9·8 | 8·6 | 7·8 | 7·1 | 6·2 | 5·5 | 5·0 |
| 40·0 | 17·5 | 16·0 | 14·0 | 12·5 | 11·5 | 10·0 | 9·2 | 8·4 | 7·3 | 6·4 | 5·9 |
| 50·0 | 20·0 | 18·0 | 16·0 | 14·0 | 13·0 | 11·5 | 10·0 | 9·5 | 8·2 | 7·3 | 6·6 |
| 75·0 | 25·0 | 22·5 | 20·0 | 17·5 | 16·0 | 14·0 | 13·0 | 12·0 | 10·0 | 9·0 | 8·2 |
| 100·0 | | | 23·0 | 20·5 | 18·5 | 16·5 | 15·0 | 13·5 | 12·0 | 10·5 | 9·6 B |
| 200·0 | | | | | | 24·0 | 21·0 | 19·5 | 17·0 | 15·0 | 14·0 A |
| 400·0 | | | | | | | | | 25·0 | 22·0 | 20·0 |

*Notes:*

If $\frac{x}{L} < 0{\cdot}4$, use all values of $\frac{L}{h}$.

If $0{\cdot}4 < \frac{x}{L} < 1{\cdot}2$, only use values of $\frac{L}{h}$ above line A.

If $1{\cdot}2 < \frac{x}{L} < 2{\cdot}4$, only use values of $\frac{L}{h}$ above line B.

$L$ and $h$ are measured in metres, $x$ is measured in millimetres.

**Table B16.6.** Flexibility of double pipe set (Fig. B16.2).

| $\frac{L^2}{x}$ | Maximum permissible values of $\frac{L}{h}$ for steel pipelines of stted nominal bore (mm) | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | 15 | 20 | 25 | 32 | 40 | 50 | 65 | 80 | 100 | 125 | 150 |
| 0·5 | — | — | — | — | — | — | — | — | — | — | — |
| 1·0 | 3·4 | 2·9 | — | — | — | — | — | — | — | — | — |
| 2·0 | 6·6 | 5·6 | 4·4 | 3·6 | 3·2 | 2·6 | 2·1 | — | — | — | — |
| 4·0 | 13·0 | 11·0 | 8·6 | 7·0 | 6·1 | 5·0 | 4·0 | 3·5 | 2·8 | 2·3 | 1·9 |
| 6·0 | B 18·5 | 16·0 | 12·5 | 10·0 | 9·0 | 7·3 | 5·8 | 5·1 | 4·0 | 3·3 | 2·8 |
| 8·0 | 24·5 | 21·0 | 16·5 | 13·5 | 12·0 | 9·6 | 7·7 | 6·7 | 5·3 | 4·4 | 3·7 |
| 10·0 | A 30·0 | 25·5 | 20·5 | 16·5 | 14·5 | 12·0 | 9·5 | 8·2 | 6·6 | 5·4 | 4·5 |
| 15·0 | 45·0 | 38·0 | 30·0 | 24·5 | 21·5 | 17·5 | 14·0 | 12·0 | 9·8 | 8·0 | 6·7 |
| 20·0 | | 49·0 | 39·0 | 32·0 | 28·0 | 23·0 | 18·0 | 16·0 | 13·0 | 10·5 | 8·8 |
| 30·0 | | | | 47·0 | 41·0 | 33·5 | 27·0 | 23·0 | 19·0 | 13·0 | 15·0 |
| 40·0 | | | | | | 44·0 | 35·0 | 30·0 | 25·0 | 20·0 | 17·0 |
| 50·0 | | | | | | | 43·0 | 37·0 | 30·0 | 25·0 | 21·0 B |
| 75·0 | | | | | | | | | 45·0 | 36·0 | 31·0 A |
| 100·0 | | | | | | | | | | 47·0 | 40·0 |
| 200·0 | | | | | | | | | | | |
| 400·0 | | | | | | | | | | | |

*Notes:*

If $\frac{x}{L} < 0{\cdot}4$, use all values of $\frac{L}{h}$.

If $0{\cdot}4 < \frac{x}{L} < 1{\cdot}2$, only use values of $\frac{L}{h}$ above line A.

If $1{\cdot}2 < \frac{x}{L} < 2{\cdot}4$, only use values of $\frac{L}{h}$ above line B.

$L$ and $h$ are measure in metres, $x$ is measured in millimetres.

**Table B16.7.** Multiplying factors for asymmetric loop placings.

| Ratio of $\frac{a}{b}$ | Multiplying factor |
|---|---|
| 1·0 to 2·0 | 0·7 |
| 2·0 to 4·0 | 0·5 |
| Over 4·0 | 0.3 |

**Table B16.8.** Flexibility in steel pipe loop (Fig. B16.3).

Maximum permissible values of $\frac{L}{h}$ for steel pipelines of stated nominal bore (mm) and for stated values of $\frac{g}{L}$

| $\frac{L^3}{x}$ | 15: —/1·0 | 15: 0·1/0·9 | 15: 0·3/0·7 | 15: 0·5 | 20: —/1·0 | 20: 0·1/0·9 | 20: 0·3/0·7 | 20: 0·5 | 25: —/1·0 | 25: 0·1/0·9 | 25: 0·3/0·7 | 25: 0·5 | 32: —/1·0 | 32: 0·1/0·9 | 32: 0·3/0·7 | 32: 0·5 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0·5 | 2·7 | 3·3 | 4·6 | 6·7 | 2·3 | 2·8 | 4·0 | 5·5 | 2·1 | 2·6 | 3·6 | 4·5 | 1·9 | 2·1 | 2·8 | 3·6 |
| 1·0 | 3·8 | 5·2 | 8·5 | 13 | 3·3 | 4·4 | 7·0 | 10 | 3·0 | 3·9 | 6·2 | 8·8 | 2·6 | 3·3 | 4·9 | 6·7 |
| 2·0 | 5·4 | 8·3 | 16 | 25 B | 4·5 | 6·6 | 12 | 18 | 4·4 | 6·0 | 10 | 17 | 3·8 | 5·2 | 8·8 | 13 |
| 4·0 | 7·7 | 14 | 32 | 51 A | 6·8 | 12 | 25 | 40 B A | 6·2 | 10 | 19 | 33 AB | 5·5 | 8·4 | 17 | 27 B |
| 6·0 | 9·5 | 19 | 46 | 80 | 8·4 | 16 | 37 | 62 | 7·6 | 13 | 25 | 49 | 6·7 | 12 | 25 | 39 |
| 8·0 | 11 | 24 | 62 | | 9·8 | 20 | 50 | | 9·0 | 17 | 31 | 64 | 7·8 | 14 | 32 | 52 A |
| 10·0 | 13 | 30 | 85 | | 11 | 23 | 64 | | 10 | 20 | 37 | | 8·7 | 17 | 40 | 66 |
| 15·0 | B 15 | 38 | | | 14 | 32 | | | 12 | 27 | 51 | | 11 | 22 | 59 | 96 |
| 20·0 | 18 | 48 | | | B 16 | 41 | | | B 14 | 34 | 63 | | 13 | 28 | 83 | |
| 30·0 | A 22 | | | | 19 | | | | 18 | 46 | | | B 16 | 39 | | |
| 40·0 | 25 | | | | A 20 | | | | A 20 | | | | 18 | | | |
| 50·0 | 28 | | | | 25 | | | | 23 | | | | A 20 | | | |
| 75·0 | 34 | | | | 31 | | | | 28 | | | | 24 | | | |
| 100·0 | | | | | | | | | 32 | | | | 28 | | | |
| 200·0 | | | | | | | | | | | | | | | | |
| 300·0 | | | | | | | | | | | | | | | | |

| $\frac{L^3}{x}$ | 40: —/1·0 | 40: 0·1/0·9 | 40: 0·3/0·7 | 40: 0·5 | 50: —/1·0 | 50: 0·1/0·9 | 50: 0·3/0·7 | 50: 0·5 | 65: —/1·0 | 65: 0·1/0·9 | 65: 0·3/0·7 | 65: 0·5 | 80: —/1·0 | 80: 0·1/0·9 | 80: 0·3/0·7 | 80: 0·5 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0·5 | 1·7 | 2·0 | 2·5 | 3·3 | 1·5 | 1·7 | 2·2 | 2·7 | 1·4 | 1·5 | 1·8 | 2·2 | 1·2 | 1·3 | 1·7 | 2·0 |
| 1·0 | 2·5 | 3·0 | 4·3 | 6·0 | 2·2 | 2·6 | 3·7 | 5·0 | 2·0 | 2·3 | 3·1 | 4·0 | 1·8 | 2·1 | 2·9 | 3·6 |
| 2·0 | 3·5 | 4·8 | 7·4 | 11 | 3·1 | 4·1 | 6·7 | 9·4 | 2·8 | 3·5 | 5·4 | 7·5 | 2·6 | 3·3 | 4·9 | 6·6 |
| 4·0 | 5·1 | 7·8 | 14 | 22 | 4·5 | 6·5 | 12 | 18 | 4·0 | 5·6 | 9·8 | 14 | 3·7 | 5·2 | 8·5 | 12 |
| 6·0 | 6·2 | 11 | 20 | 34 | 5·6 | 8·6 | 18 | 27 B | 5·0 | 7·4 | 14 | 21 | 4·5 | 6·7 | 12 | 18 |
| 8·0 | 7·2 | 13 | 27 | 46 B | 6.4 | 11 | 24 | 37 A | 5·7 | 9·0 | 19 | 29 | 5·3 | 8·1 | 16 | 25 |
| 10·0 | 8·1 | 15 | 35 | 59 A | 7·2 | 13 | 29 | 46 | 6·4 | 11 | 23 | 36 B | 5·9 | 9·6 | 20 | 31 B |
| 15·0 | 10 | 21 | 53 | 91 | 9·0 | 18 | 43 | 72 | 8·0 | 15 | 34 | 57 A | 7·3 | 13 | 29 | 48 A |
| 20·0 | 12 | 26 | 72 | | 11 | 22 | 58 | | 9·2 | 18 | 45 | 76 | 8·4 | 16 | 39 | 64 |
| 30·0 | 14 | 35 | | | 13 | 30 | 86 | | 11 | 25 | 68 | | 11 | 22 | 58 | 98 |
| 40·0 | B 17 | 45 | | | B 15 | 38 | | | 13 | 31 | | | 12 | 27 | 77 | |
| 50·0 | A 19 | | | | A 17 | 45 | | | B 15 | 40 | | | 14 | 33 | | |
| 75·0 | 23 | | | | 21 | | | | A 18 | 52 | | | B 17 | 46 | | |
| 100·0 | 27 | | | | 24 | | | | 21 | | | | A 19 | | | |
| 200·0 | 38 | | | | | | | | 30 | | | | 27 | | | |
| 300·0 | | | | | | | | | | | | | | | | |

| $\frac{L^3}{x}$ | 100: —/1·0 | 100: 0·1/0·9 | 100: 0·3/0·7 | 100: 0·5 | 125: —/1·0 | 125: 0·1/0·9 | 125: 0·3/0·7 | 125: 0·5 | 150: —/1·0 | 150: 0·1/0·9 | 150: 0·3/0·7 | 150: 0·5 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 0·5 | — | — | — | — | — | — | — | — | — | — | — | — |
| 1·0 | 1·5 | 1·8 | 2·2 | 2·8 | 1·4 | 1·6 | 2·0 | 2·4 | 1·3 | 1·4 | 1·7 | 2·0 |
| 2·0 | 2·2 | 2·7 | 3·9 | 5·2 | 2·1 | 2·4 | 3·3 | 4·4 | 1·8 | 2·1 | 2·9 | 3·7 |
| 4·0 | 3·2 | 4·3 | 6·8 | 9·8 | 3·0 | 3·7 | 5·8 | 8·2 | 2·6 | 3·4 | 5·0 | 6·9 |
| 6·0 | 3·9 | 5·6 | 9·5 | 14 | 3·6 | 4·9 | 8·3 | 12 | 3·3 | 4·4 | 7·0 | 10 |
| 8·0 | 4·6 | 6·8 | 12 | 19 | 4·2 | 6·0 | 11 | 16 | 3·8 | 5·3 | 9·0 | 13 |
| 10·0 | 5·2 | 8·0 | 15 | 24 | 4·7 | 7·0 | 14 | 22 | 4·2 | 6·2 | 11 | 16 |
| 15·0 | 6·4 | 11 | 23 | 37 B | 5·8 | 9·3 | 19 | 30 | 5·4 | 8·4 | 16 | 26 |
| 20·0 | 7·4 | 13 | 31 | 49 A | 6·8 | 11 | 24 | 39 B | 6·1 | 10 | 20 | 33 B |
| 30·0 | 9·2 | 18 | 48 | 75 | 8·3 | 16 | 36 | 60 A | 7·6 | 14 | 31 | 50 A |
| 40·0 | 11 | 23 | 65 | | 9·6 | 20 | 49 | | 8·8 | 17 | 41 | 67 |
| 50·0 | 12 | 29 | 80 | | 11 | 24 | 62 | | 9·8 | 20 | 51 | 84 |
| 75·0 | B 15 | 37 | | | 13 | 32 | | | 12 | 28 | 76 | |
| 100·0 | A 17 | 46 | | | B 15 | 40 | | | B 14 | 34 | | |
| 200·0 | 25 | | | | A 22 | | | | A 19 | | | |
| 300·0 | | | | | 28 | | | | 25 | | | |

*Notes:*

If $\frac{x}{L} < 0{\cdot}4$, use all values of $\frac{L}{h}$.

If $0{\cdot}4 < \frac{x}{L} < 1{\cdot}2$, only use values of $\frac{L}{h}$ above line A.

If $1{\cdot}2 < \frac{x}{L} < 2{\cdot}4$, only use values of $\frac{L}{h}$ above line B.

$L$, $g$ and $h$ are measured in metres, $x$ is measred in millimetres.

**Table B16.9.** Values of A for equation B16.7.

| $\frac{L}{h}$ | Pipe with one pulled bend | Pipe with two pulled bends* | Pipe with expansion loop with stated values 01 g/L | | | | | |
|---|---|---|---|---|---|---|---|---|
| | | | — 1·0 | 0·05 0·95 | 0·1 0.9 | 0·2 0·8 | 0·3 0·7 | 0·5 |
| 1·0 | 0·2 | 0·64 | 0·050 | 0·048 | 0·046 | 0·042 | 0·041 | 0·040 |
| 1·2 | 0·3 | — | 0·078 | 0·074 | 0·070 | 0·066 | 0·064 | 0·062 |
| 1·4 | 0·4 | — | 0·12 | 0·11 | 0·10 | 0·096 | 0·093 | 0·090 |
| 1·5 | — | 0·67 | — | — | — | — | — | — |
| 1·6 | 0·54 | — | 0·15 | 0·14 | 0·13 | 0·12 | 0·11 | 0·10 |
| 1·8 | 0·7 | — | 0·24 | 0·22 | 0·20 | 0·18 | 0·17 | 0·16 |
| 2·0 | 0·9 | 0·89 | 0·35 | 0·30 | 0·27 | 0·22 | 0·21 | 0·20 |
| 2·5 | 1·5 | 1·38 | 0·60 | 0·55 | 0·50 | 0·46 | 0·41 | 0·36 |
| 3·0 | 2·8 | 1·95 | 0·98 | 0·82 | 0·70 | 0·60 | 0·55 | 0·50 |
| 3·5 | — | 2·7 | 1·5 | 1·2 | 1·1 | 0·85 | 0·78 | 0·70 |
| 4·0 | 5·2 | 3·5 | 2·1 | 1·7 | 1·5 | 1·2 | 1·0 | 0·90 |
| 5·0 | 9·4 | 5·6 | 4·0 | 3·0 | 2·5 | 1·9 | 1·7 | 1·5 |
| 6·0 | 15 | 8·2 | 6·7 | 5·0 | 4·0 | 2·9 | 2·5 | 2·2 |
| 7·0 | — | 11 | — | — | — | — | — | — |
| 8·0 | 35 | 15 | 15 | 10 | 7·7 | 5·4 | 4·7 | 4·0 |
| 9·0 | — | 19 | — | — | — | — | — | — |
| 10 | 65 | 24 | 27 | 17 | 13 | 9·0 | 7·2 | 6·0 |
| 12 | 110 | 35 | — | — | — | — | — | — |
| 14 | 170 | 48 | — | — | — | — | — | — |
| 15 | — | — | 80 | 50 | 34 | 22 | 17 | 14 |
| 16 | 240 | 62 | — | — | — | — | — | — |
| 18 | 340 | 80 | — | — | — | — | — | — |
| 20 | 460 | 100 | 180 | 100 | 62 | 40 | 30 | 25 |
| 22 | — | 121 | — | — | — | — | — | — |
| 24 | — | 140 | — | — | — | — | — | — |
| 25 | 880 | — | 325 | 175 | 105 | 66 | 48 | 40 |
| 26 | — | 170 | — | — | — | — | — | — |
| 28 | — | 200 | — | — | — | — | — | — |
| 30 | — | 230 | 530 | 270 | 160 | 100 | 67 | 58 |
| 32 | — | 260 | — | — | — | — | — | — |
| 34 | — | 290 | — | — | — | — | — | — |
| 35 | — | — | 800 | 400 | 230 | 140 | 94 | 80 |
| 36 | — | 330 | — | — | — | — | — | — |
| 40 | — | 410 | — | 550 | 300 | 180 | 120 | 100 |
| 45 | — | 540 | — | 750 | 400 | 240 | 150 | 140 |
| 50 | — | 650 | — | — | 500 | 300 | 190 | 170 |
| 60 | — | — | — | — | — | 450 | 280 | 240 |
| 70 | — | — | — | — | — | 640 | 380 | 330 |
| 80 | — | — | — | — | — | — | 490 | 430 |
| 90 | — | — | — | — | — | — | 620 | 550 |
| 100 | — | — | — | — | — | — | — | 680 |

*The values of *A* in this column apply for values of *a/b* from 1 to 8. For values of *a/b* greater than 8, use the values for a pipe with one pulled bend.

where:

$T$ = total anchor thrust .. .. .. .. N

$F$ = load to deflect the pipe bend(s) .. .. N

$L$ = distance, from Figs. B16.1 to B16.3 .. m

$a$ = pipe length between set and anchor were $a > b$ .. .. .. .. .. m

$M$ = pipe length to the anchor including the the width of the loop where $M > N$ .. m

$D$ = pipe nominal bore .. .. .. .. mm

$F$ may be found from:

$$F = 120\,AI\frac{x}{L^3} \qquad \text{B16.7}$$

where:

$A$ = values given in Table B16.9

$I$ = moment of inertia .. .. .. .. $cm^4$

$x$ = expansion .. .. .. .. .. mm

**Flexibility of Branches off Expanding Pipes**

*Steel Branch Off Main Pipeline*

Table B16.10 may be used to determine the unrestrained length needed on a branch off an expanding main pipeline. The branch may be welded into the main or a screwed tee used and the pipes may be horizontal or vertical. The end of the branch away from the main pipeline is taken to be guided and the main pipeline to be laterally rigid. Alternatively, the branch could be anchored at the guide position shown, provided the main were flexible laterally. In either case the axial expansion of the branch must be considered separately using the data in Table B16.10.

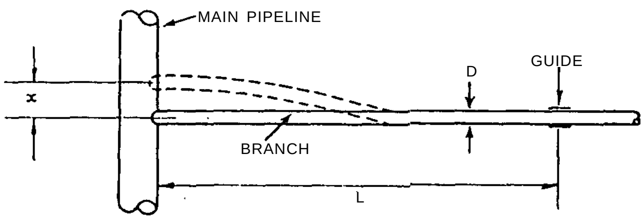



**Fig. B16.4.** Branch off expanding pipe.

The table is based on the branch being unstressed with the pipeline cold. Twice the amount of expansion can be accepted on the main pipeline if 50% lateral cold draw on the branch is applied on erection.

The branch pipe must be supported either on hangers of lengths that do not cause constraint, or on sliding supports which allow both axial and sideways movement.

For movement at the main greater than the values given in Table B16.10, use:

$$L = 0{\cdot}12\,\sqrt{xD} \qquad \text{B16.8}$$

where:

$L$ = unrestrained branch length .. .. m

$x$ = expansion .. .. .. .. .. mm

$D$ = nominal bore .. .. .. .. mm

*Guides and Anchors*

The lateral thrust at the guide or anchor, due to the sideways movement of the branch at the main is given by:

$$T = 24I\frac{x}{L^3} + 1{\cdot}5\,LD \qquad \text{B16.9}$$

for pipes on supports and:

$$T = 24I\frac{x}{L^3} + 0{\cdot}575\,LD \qquad \text{B16.10}$$

for pipes on hangers.

**Devices to Absorb Pipework Expansion**

Various proprietary devices are available to take up expansion or deflection in pipework systems subject to temperature changes.

In the past, mechanically sealed expansion devices have been used; for axial movement, telescopic sliding joints, and for deflection through an angle, ball joints. However, the need for attention or access to packing or seals has led to the wider use of bellows expansion joints, also called compensators, which do not need maintenance after installation.

Plain axial bellows, compressed and extended along their axis, can be used for small movements and are particularly advantageous in closely packed runs of pipe. With axial units, the action of the internal pressure will tend to open the bellows out lengthways. This must be resisted by strong pipe anchors at both ends of the pipe and by careful guiding of the whole pipe run.

**Table B16.10.** Minimum unrestrained branch lengths.

| Branch nominal bore (mm) | Minimum unrestrained branch length for stated pipeline expansion (m) | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | 8 mm | 15 mm | 20 mm | 25 mm | 40 mm | 50 mm | 65 mm | 80 mm |
| 15 | 1·2 | 1·5 | 1·8 | 2·1 | 2·7 | 3·1 | 3·4 | 3·7 |
| 20 | 1·5 | 1·8 | 2·4 | 2·7 | 3·4 | 3·7 | 4·3 | 4·6 |
| 25 | 1·5 | 2·1 | 2·7 | 3·1 | 3·7 | 4·3 | 4·9 | 5·2 |
| 32 | 1·8 | 2·4 | 3·1 | 3·4 | 4·3 | 4·9 | 5·5 | 5·8 |
| 40 | 1·8 | 2·7 | 3·4 | 3·7 | 4·6 | 5·2 | 6·1 | 6·4 |
| 50 | 2·1 | 3·1 | 3·7 | 4·3 | 5·2 | 6·1 | 6·7 | 7·3 |
| 65 | 2·4 | 3·4 | 4·3 | 4·9 | 6·1 | 7·0 | 7·6 | 8·2 |
| 80 | 2·7 | 3·7 | 4·6 | 5·5 | 6·7 | 7·6 | 8·2 | 9·1 |
| 100 | 3·1 | 4·3 | 5·2 | 6·1 | 7·6 | 8·5 | 9·8 | 10·7 |
| 125 | 3·4 | 4·9 | 5·8 | 7·0 | 8·2 | 9·8 | 10·7 | 11·9 |
| 150 | 3·7 | 5·5 | 6·4 | 7·6 | 9·1 | 10·7 | 11·9 | 12·8 |

Alternative systems using the bellows in bending have great advantages in cases of large expansion movements or where forces on the pipe or equipment or anchor loads on the structure must be kept very low. Singe bellows, spanned by tie-bar and hinge systems restraining the internal pressure effects, are referred to as angular compensators, which describes their movement. These units are used in sets of two or three to take up complex movement. A twin bellows unit with an intermediate pipe section spanned by a double-hinged tie-bar system is referred to as an articulated compensator and will absorb large movements at right angles to its axis.

Where any proprietary expansion device is used, the manufacturers' instructions for selection, installation, guiding and anchoring of pipework and maintenance of the unit must be followed carefully.

## STEAM TRAPS

Steam used for heating gives up its latent heat and changes to water. A steam trap is required at the outlet side of the heat exchanger (e.g. radiator, unit heater, calorifier, etc) to allow the removal of condensate from the plant and to prevent the escape of steam. Steam traps are also required to vent air which collects in every steam system particularly when it is shut down. In the case of large or complicated steam spaces it may be necessary to use additional automatic air vents.

### Classification and Selection

*Group 1 Traps* differentiate between steam and condensate mechanically, generally by the action of a float or bucket.

*Group 2 Traps* differentiate between steam and condensate by a temperature difference, which operates a thermostatic element carrying a valve.

*Group 3 Traps* differentiate between the force produced by the velocity of steam or condensate on a floating disc in the trap.

A trap must be chosen with characteristics which most nearly match those of the plant. Where several types of traps could be used the final decision should take into account noise, capacity, simplicity of installation, known reliability, ease of servicing and the desirability of minimising the differing types of traps on the plant. The trap must be of the correct capacity to handle the condensate load at all times.

Steam trap capacity depends on the size of the valve orifice, the pressure differential across it and the temperature of the condensate. The same trap body may be fitted with a number of different size valve orifices to suit the pressure range for which it is required and the capacity of the trap will change accordingly.

The pressure differential depends on the inlet pressure and the back pressure which may be imposed due to lifting the condensate, or pressure in the condensate return main. Both inlet and back pressure can be affected by the resistance of the pipework used to connect up the trap. It is the pressure differential across the trap orifice which determines its capacity.

Traps must be sized to handle the condensate load under all conditions. These include start-up from cold when the load may be increased but the pressure differential is reduced due to the heavy demand.

Thermostatic traps are wide open when handling cold condensate and consequently have a greatly enhanced capacity which is normally published by the manufacturer. With other trap types it is usual to allow a factor of say 2 to cover condensate loads on start-up. The actual figure will depend on the nature (thermal mass) of the plant, the frequency of start-up and the importance of prompt drainage under these conditions.

Oversized traps should be avoided. Initial costs, radiation losses, wear and maintenance costs are all likely to increase and steam losses in the event of trap failure will be higher.

### Applications

*Group Trapping*

Where multiple units are fed from a common steam header and drained into a common condensate return each unit should be trapped individually, see Fig. B16.5. This avoids preferential flow through one or more units and ensures the best performance.

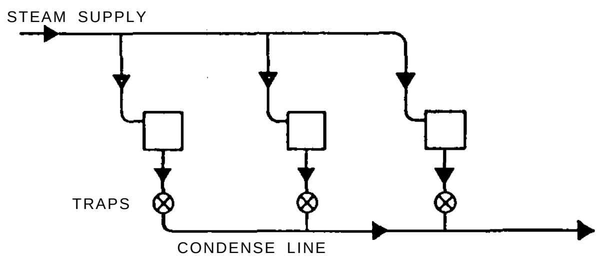



**Fig. B16.5.** Correct individual trappings.

*Lifting Condensate*

Wherever practicable, the condensate should drain away from a trap by gravity to a central collecting point and then be lifted by a steam or electric pump set to a high level main.

Where it is not possible to apply the above principle, condensate may be lifted directly to an overhead main by a trap, provided the steam pressure is sufficient. If the pressure is insufficient the steam space will waterlog. Where a steam trap is used for direct lifting of condensate it should be fitted at the bottom of the lift and close to the plant being drained to prevent steam locking. A check valve should be fitted after the trap to prevent condensate from running back and filling the steam space when steam is turned off.

When a modulating temperature control is used to change the steam pressure on a plant, waterlogging will occur if the pressure in the steam space falls below that required to operate the trap and lift the condensate, with the result that the controller will hunt and the plant output will be reduced. Hunting may well be accompanied by waterhammer.

*Superheated Steam*

Not all types of traps can be used on superheated steam and not all materials of construction are suitable for high temperatures. The maker's recommendations should be followed if a superheated steam supply has to be drained.

**Associated Equipment**

*Strainers*

Dirt or scale lodging between the valve and seat of a trap will cause the trap to pass steam continuously. A strainer is essential to prevent the passage of all but the finest dirt into the trap and assist in preventing trap failure. It must, of course, be cleaned periodically.

*Sight Glasses*

Any installation should include some means to enable the maintenance engineer to check steam trap performance. A sight glass fitted after every traps allows this to be done. With traps having a blast discharge feature it is recommended that the sight glass is fitted not less than 1 metre away from the trap to prolong the life of the glass.

Other proprietary arrangements are available for checking steam trap performance and combating steam wastage. These include the use of weir chambers installed upstream or downstream of each steam trap and provided with an adaptor to receive a plug-in sensor. This indicates whether steam leakage is taking place, sensing the absence of liquid in the weir by measuring the resistivity across the chamber.

The plug-in sensors are suitable for wiring connections to a remote reading station, arrangements for continuous monitoring or for incorporation in a building management system.

The above method provides a very simple check that can be carried out by any personnel. Performance checks on steam traps can also be made by the use of a stethoscope or an ultrasonic detector, but a measure of interpretive skill is required in both cases.

*Check Valves*

Where inverted bucket traps are used for superheated steam applications, the check valve should be fitted on the inlet to the trap to prevent loss of the water seal.

**Precautions**

*Frost*

Traps which contain water can be damaged by freezing. Because their valves are open wide, thermostatic traps are self-draining when cold so long as the discharge pipe drains freely. Mechanical traps normally contain some water when cold, so that they are readily damaged. Most thermodynamic traps can be frozen without coming to harm. Insulation helps to delay freezing but should not be relied on if the cold spell is prolonged.

*Water-hammer*

In a steam system, pockets of water will form in all low points, such as a sag in the steam main, a submerged heating coil, etc. When steam is turned on it picks up the water and at a change in direction the water will attempt to continue in a straight line, whilst the steam will change direction easily. The resultant forces produced by stopping the water are dissipated in stresses in the pipework, fittings, traps and ancillaries.

To prevent water-hammer there should be a continuous fall in the direction of the steam flow so that water pockets cannot form. Adequate provision should be made for drain points at natural water collecting situations offered by step relaying of mains, terminal positions, etc. In general steam lines should be drained at intervals of not more than 45 m.

Care must be taken if condensate has to be discharged into a flooded return line. If the condensate is discharged at steam temperature, the flash steam which forms as it passes through the trap will be injected into the cooler condensate in the main and cause water-hammer as it suddenly condenses. Water-hammer can be prevented if the condensate is cooled before entering the main. A bimetallic trap is suitable for this purpose, provided it is fitted with an adequate cooling leg. An alternative is to use a float trap. The continuous introduction of condensate to the main will reduce but may not eliminate water-hammer.

**Table B16.11.** Characteristics of steam traps and air vents.

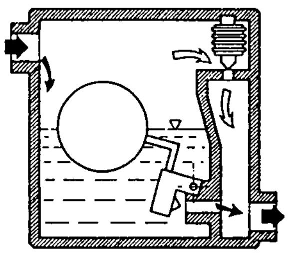
| Group | Type | Sketch | Remarks |
|---|---|---|---|
| 1 | Float traps |  | Advantages<br>Suitable for widely fluctuating loads and pressures. Easy to install and maintain. Removes condensate continuously as it forms. Patterns with balanced pressure air vents automatically discharge air.<br><br>Disadvantages<br>Can be damaged by water-hammer and corrosive condensate. Normally three or four differently sized valves and seats are required to cover the normal working range. |

**Table B16.11.**–*continued*

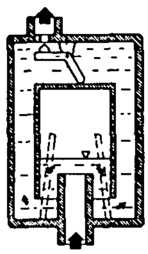
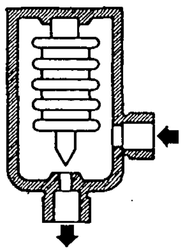
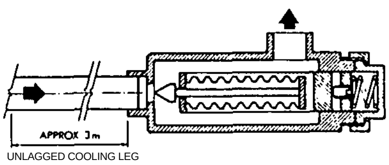
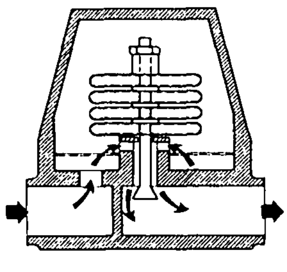
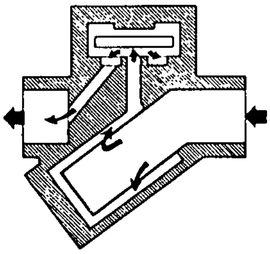
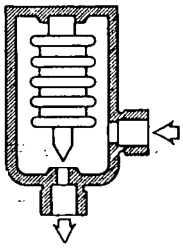
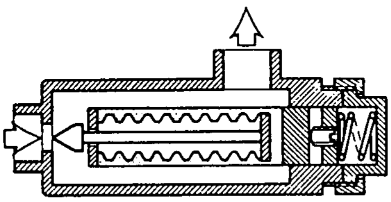
| Group | Type | Sketch | Remarks |
|---|---|---|---|
| 1<br>*(cont.)* | Inverted bucket trap |  | Advantages<br>Can be made for high pressure and superheated steam, will withstand water-hammer and can be made of corrosion resisting materials. When used on superheated steam a check valve should be fitted at the inlet. The working parts are simple.<br><br>Disadvantages<br>Wasteful of steam if oversized. Does not respond well to severe fluctuations of pressure and discharges air slowly. A thermostatic air vent fitted in a by-pass is recommended. When installed out-doors should be lagged.<br><br>Note. Open top bucket traps have similar advantages and dis-advantages. Although no longer manufactured some may still be found in service. |
| 2 | Thermostatic steam trap |  | Advantages<br>Small in size. Automatically discharges air. The valve is wide open on start-up so cool condensate and air discharge quickly. Capacity is high. Unlikely to freeze if condensate can run from the trap outlet. Maintenance is easy. Traditional elements have corrugated bellows of brass or phosphor bronze. Newer designs use stainless steel to form a bellows or a diaphragm type element.<br><br>Disadvantages<br>Older type elements are liable to damage by water-hammer, corrosive condensate, or superheated steam. (Stainless steel elements are more robust and some designs are suitable for use with superheated steam.) |
| | Liquid expansion steam trap |  | Advantages<br>Can be used on superheated steam and on higher pressures than the balanced pressure trap. Valve is wide open on start-up, so cool condensate and air discharge quickly. Capacity is high. A continuous discharge type, so is quiet in operation and not affected by vibration, steam pulsation or water-hammer. Auto-matically discharges air.<br><br>Disadvantages<br>Does not respond quickly to change in load or steam pressure. The element can be damaged by corrosive condensate.<br><br>Note. Because the element is on the discharge side of the valve orifice the trap will hold back condensate. This permits the use of some sensible heat from the condensate provided that water-logging of the steam space is acceptable. If this is not the case, a cooling leg must be fitted before the trap. |
| | Bi-metallic steam trap |  | Advantages<br>Usually small and robust. When cold the valve is wide open and air is freely discharged. The capacity is greatest when the con-densate is coolest. Some types are not damaged by freezing. They will withstand water-hammer and some are not affected by corrosive condensate. Suitable for use on high pressures and on superheated steam. Will work over a wide range of pressures without the need to change the size of the valve orifice, although the position of the valve may need adjusting in some cases. They will hold back condensate until cooling occurs thus using some of the sensible heat.<br><br>Disadvantages<br>Will not discharge condensate until it has cooled below the saturation temperature, so unsuitable for use where condensate must be cleared as soon as it forms, unless a cooling leg is provided. Responds slowly to changes in steam pressure and condensate load. |

**Table B16.11.**–*continued*

| Group | Type | Sketch | Remarks |
|---|---|---|---|
| 3 | Thermodynamic trap |  | Advantages<br>Very small yet has a large discharge capacity. Will work over the full range of pressures, without adjustment. It can be used on superheated steam and can also withstand vibration or severe water-hammer. Being normally made of stainless steel it can withstand corrosive condensate and is not damaged by being frozen.<br>Disadvantages<br>Normally requires a minimum pressure differential in order to function. If, on starting up, pressure at the trap builds up slowly it can discharge a lot of air but if the pressure builds up quickly the resultant high velocity air can shut the trap in the same way as steam and it will air bind. Trap operation can be noisy. Due to the blast discharge operation sight glasses and check valves should be fitted about 1 metre away from the trap. |
| Air vents | Balanced pressure air vent |  | This is similar to the balanced pressure steam trap. When the plant is cold the valve is wide open. As the temperature surrounding the element approaches the steam temperature, the internal liquid expands, generates a pressure within the element and closes the valve seat. |
| | Liquid expansion air vent |  | This air vent is similar to the liquid expansion steam trap. Changes in temperature cause the oil filled element to expand or contract and the valve moves towards or away from its seat. |

*Note:* Where water-hammer is present or the steam is superheated, the liquid expansion air vent is the better choice, as either of these conditions may damage balanced pressure units. Both the liquid expansion and balanced pressure vents are suitable for any pressure within their range without change of valve seat, but if conditions vary greatly the liquid expansion unit may require re-setting.

## VALVES

Valves are applied for the control and regulation of the flow of fluids in pipework systems. Fluids may be classed as gases or vapours, liquids and fluidised solids or solids in suspension. Each class presents particular problems of distribution or flow-rate regulation. Depending on the application, a valve may be subject to a wide range of fluid temperatures, velocities and operating pressures and the fluid itself may be corrosive or abrasive to some degree. Additionally, due to pressure or temperature variations within a distribution system, the fluid may undergo a phase change (e.g. a liquid may "flash" into a gas).

For a regulating application a valve requires to have a high resistance to offer control authority, whereas for an isolating application, tight shut-off is important. Each type of valve has its own special characteristics which determine its suitability for any particular application. The choice of materials for a valve may depend on the temperature and pressure at which it will be used. The manufacturer and the relevant British Standards should always be consulted if there is any doubt. The information in Table B16.12 may be supplemented by reference to catalogues and literature published by the British Valve Manufacturers' Association.

**Table B16.12.** Valve types.

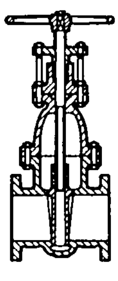
| Valve Type | Function | Sketch | Description |
|---|---|---|---|
| Gate valves -wedge | Opening and closing circuits |  | Widely used where uninterrupted flow is required. Not recommended for regulation and should be used fully open or fully shut. The solid wedge with matching tapered body seats has the advantages of strength, long life and not being subject to chatter. Rising stem types need clearance above hand wheel. Flexible wedge valves accommodate pipeline strains better, require less torque to seat/unseat and provide tighter sealing on inlet and outlet seats over a wide range of operating pressures. |

**Table B16.12.**–*continued*

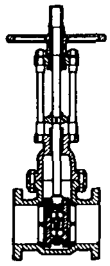
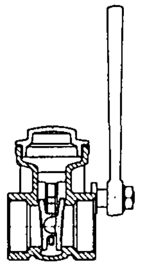
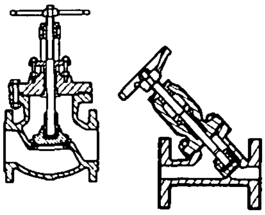
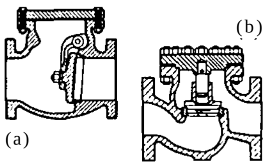
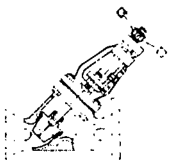
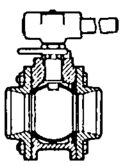
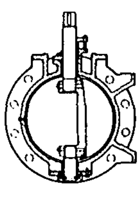
| Valve Type | Function | Sketch | Description |
|---|---|---|---|
| Parallel slide | Opening and closing circuits. Suitable for high tempera- tures and pressures |  | Unrestricted bore and very low pressure drop. Particularly suitable for demanding temperature and pressure applications but not for dirty fluids. Isolation is effected by means of two sliding discs which are traversed across the flow-path of the fluid. Seating enclosure forces not affected by hand-wheel effort. |
| Lever wedge | Quick (60°) opening and closing of circuits |  | With the wedge (either solid or split) pivoted on a spindle controlled by a hand operated lever this valve type is suitable for quick on/off operations e.g. laundry machines etc.<br><br>Note:<br>Care should be taken in design when using quick-acting valves in medium or high velocity flow applications. The sudden back pressure resulting from closure can set up water-hammer, which unless compensated for by an upstream air bottle or other damping device, can cause damage. |
| Globe valves | Screw- down, stop and regulation |  | Most reliable form of seating as stop valve for all fluids. By virtue of its resistance characteristics, may also be used for flow regulation. As well as manual hand-wheel operation, actuation by screw and nut mechanism through electric or air motor or hydraulic cylinder may be applied. |
| Check valves | One way flow | (b)  (a) | Also referred to as non-return valves (NRVs). A self-acting valve to prevent reversal of flow. Swing check type (a) has lower resistance and is suitable for liquids. Lift check type (b) is available in two patterns for horizontal and vertical pipe applications and is more suited for use with gases. |
| Flow measure- ment valve | Pressure differential measure- ment |  | A "Y" pattern globe valve with pressure test valves placed either side of the seat. The valve can also be used as a stop valve but with the disk with-drawn to the full open position. A pressure differential reading taken by manometer across the pressure test valves and related to the valves unique performance chart will give an accurate flowrate reading (see also Commissioning Code "W"). |
| Double regulating valve | Regulation and setting flow |  | A "Y" pattern valve for regulating the flow when measurement is being taken on a flow measurement valve for system balancing purposes. Once the required flow is achieved a locking device is applied which will allow the valve to be closed and re-opened to the original set position. This and the flow measurement valve are installed as a pair. |
| Ball valve | Quick action open/ close |  | A quarter turn valve utilising a centre pivoted ball with an orifice equal to or slightly less than the internal diameter of a standard pipe. Seated in a soft medium to give tight sealing, there are many versions covering the range from cold water and air to hydrocarbons and other fluids, usually in small sizes. |
| Butterfly valves | |  | A compact valve with good regulating characteristics. With wafer pattern units, disks must have freedom to enter adjacent pipes when in open position. Suitable for use with renewable synthetic elastomeric body lining and coated disks using a wide variety of materials. Frequently applied to larger diameter pipes. Operation by hand lever, hand gear or automatic electric or pneumatic actuation (see Note under lever wedge gate valve). |

**Table B16.12**.–*continued.*

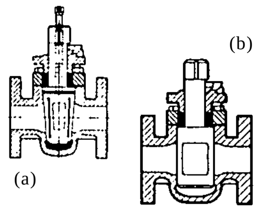
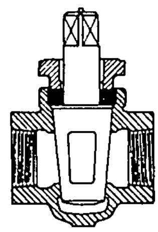
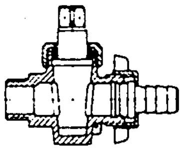
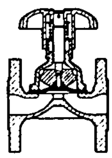
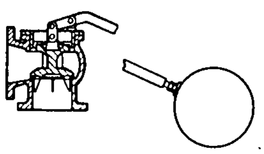
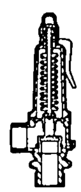
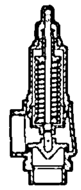
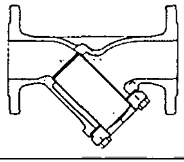
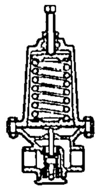
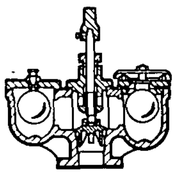
| Valve Type | Function | Sketch | Description |
|---|---|---|---|
| Plug valves | Quick action open/ close |  | Available in (a) taper or (b) parallel plug form with injected compound or lubricant for sealing. The modern equivalent of the cock, available in various patterns with circular or rectangular plug ports. Venturi patterns are available taking advantage of the low pressure drop (see Note under lever wedge gate valve). |
| Cocks | Quick action open/ close |  | Earliest and simplest form of closure device differing from a valve in that seating surfaces are always in sliding contact. Generally for low pressure applications unless specially prepared (see Note under lever wedge gate valve). |
| | Draining |  | Plug valve action but with one face open to atmosphere, and provided with a male tail for fitting a drain hose. Usually used to drain points in a system. Not suitable for drainage during system flushing where a line-size discharge is more appropriate (see also Commissioning Code “W”). |
| | Draw-off | | Usually known as a D/O cock and fitted to small boilers or as drain points on small systems. |
| Diaphragm | Aggressive fluid applications |  | Operating mechanism isolated from fluid by diaphragm such that body and diaphragm only are in contact with fluid. Diaphragm material can be varied to suit fluid. Smooth contour of body makes it suitable for lining with a variety of materials. Suitable for use with aggressive or abrasive fluids. |
| Equilibrium ball (float) valves | Automatic filling |  | Used to maintain constant liquid levels in make-up tanks. The ball follows the liquid level in a partially submerged state causing the valve to open and admit more water with falling water level and to close with a rising water level. Being in partial equilibrium, the float follows the water level over a wide range of inlet main’s pressures, making purpose designed units unnecessary. |
| Safety valves | Maximum pressure blow-off |  | The regulations in most countries demand that safety valves be installed on all pressure vessels to ensure that maximum design pressure is not exceeded. These regulations vary considerably and thus care should be exercised to ensure that such regulations are known and correctly interpreted.<br><br>Safety valves are available in several designs of which the following are common examples:<br>(*a*) An ordinary safety valve is one in which no use is made of the pressure discharged fluid to assist the lift of the valve which must be at least 1/12th of the internal diameter of the seating. Used for the protection of low-pressure vessels and pipe systems.<br>(*b*) A high-lift safety valve is one in which the lift is assisted by the pressure of discharged steam to give a lift of at least 1/24th of the internal diameter of the seating.<br>(*c*) A full-lift safety valve is one in which lift is assisted by the pressure of discharged steam to give a lift such as will provide an area round the edge of the valve seat to equal the net area through the seat orifice after deducting the area of guides and other obstructions. |

**Table B16.12.**–*continued.*

| Valve Type | Function | Sketch | Description |
|---|---|---|---|
| Relief valves | Maximum pressure stabilisa-tion |  | A relief valve is an automatic pressure relieving device actuated by the upstream static pressure. The valve opens in proportion to the increase in pressure over the design setting. It may be spring or direct weight loaded or pilot valve operated and used primarily for liquid services. |
| Strainers | Sediment arrest |  | A device incorporating a wire mesh ‘basket’ with a screen of varying mesh size. Despite system flushing, pipeline debris can still be present, and if not removed, can damage valve seats and other equipment. Properly valved, a strainer can be periodically cleaned. Also available as duplex units to permit continued operation while cleaning the basket. |
| Pressure reducing valves | Regulation of down-stream pressure |  | An automatic valve which reduces upstream system pressure to a pre-determined level downstream, maintaining this reduced pressure irrespective of variations in flow or pressure changes upstream. Valves may be direct acting or pilot operated, the latter using a small, direct-acting pilot to regulate the main valve. |
| Automatic air/vacuum relief valve | Automatic venting of air and/or release of vacuum |  | Also referred to as an Automatic Air Vent (AAV), Air Eliminator or Vacuum Break Valve. Float operated and hence automatic in operation, the valve is fitted to systems to release air pressure or vacuum during filling or emptying procedures. |

## DUCTWORK

### Application of Materials

Materials for ductwork should be chosen to suit the particular circumstances of environment, weight, appearance and cost. The most commonly used materials are galvanised sheet steel, black sheet steel, aluminium and stainless steel. Less frequently used materials are PVC, polypropylene, resin bonded glass fibre and moulded asbestos. Air ducts may also be constructed from brickwork, concrete, sheet asbestos, building board, and timber.

Specifications for the design, manufacture and site erection of ductwork systems are published by the Heating and Ventilating Contractors Association and are shown in Table B16.13.

By far the most widely used of these is DW/142 which is the current specification for sheet metal ductwork for low, medium and high pressure/velocity air systems.

**Table B16.13.** HVCA Specifications.

| HVCA Ref. No. | Title | Material | Scope | |
|---|---|---|---|---|
| | | | Velocity (m/s) | Static Pressure (Pa) |
| DW/142 | Specification for sheet metal ductwork for low, medium and high pressure/velocity air systems. | Galvanised sheet steel black sheet steel aluminium and stainless steel | 10 to 40 | Positive 500-2500<br>Negative 500-750 |
| DW/l43 | A practical guide to ductwork leakage testing. | — | Includes procedures, hints and a test sheet example. | |
| DW/151 | Specification for plastics ductwork for low-velocity, low-pressure air systems. | uPVC and polypropylene | Up to 10 | Positive or negative pressures not exceeding 500. |
| DW/191 | Code of practice for resin bonded glass fibre ducts for low-velocity, low-pressure air systems. | Resin bonded glass fibre. | Up to 10 | Positive or negative pressures not exceeding 500. |

## Sheet Metal Ductwork for Air Systems

*Materials*

Galvanised ductwork is fabricated from continuously hot-dip zinc coated and iron zinc alloy coated steel: wide strip, sheet/plate and slit wide strip to BS 2989.

The type of sheet normally used for ductwork is a bending and profiling quality designated Z2G.275. For black steel ductwork, the material used is rolled, close-annealed or strip mill cold-reduced black mild steel sheet.

Aluminium ductwork is fabricated from sheets and sections to BS 1470. Alloys 1200, 3101 and 5251 are the easiest to form and join.

Stainless steel is not a single specific material. The grade should be selected for its advantages in forming, welding, drilling and punching, its resistance to corrosion and surface finish.

*Internal Ductwork.*

A protective finish should be applied to mild steel sections (flanges, stiffeners, hangers, supports, etc) before fixing. Suitable protection for normal conditions would be two coats of red oxide or zinc chromate paint. The designer should stipulate any special finishes required, e.g. for black sheet steel ductwork.

Suitable protection for aluminium ductwork and sections used under normal conditions would be one coat etch primer and two coats zinc chromate paint. Mild steel sections used with aluminium ductwork should, before fastening to ducts, be protected as stated above. Special finishes appropriate to particular applications should be stipulated by the design engineer.

*External Ductwork*

Suitable protection for mild steel sections or black sheet ductwork for external use would be one coat of red oxide paint followed by two coats of bituminous paint. Suitable protection for galvanised ductwork would be one coat of mordant solution or calcium plumbate primer followed by two coats of bituminous paint.

*Galvanising After Manufacture*

The specification for galvanising after manufacture is to be to BS 729. To restrict the distortion encountered in galvanising, such ductwork should be manufactured from thicker sheet. Metal spray finishes may be applied to ductwork, zinc spraying and aluminium spraying both to BS 2569: Part 1 being the most commonly used.

*Anodising*

Anodising to BS 1615 provides an excellent finish for aluminium ductwork, offering a decorative appearance as well as giving protection.

## Ductwork Classification

HVCA Specification DW/142 classifies ductwork in accordance with its operating pressures and air flow velocities together with limits of air leakage rates. The essentials of this classification are shown in Tables B16.14 and 15.

**Table B16.14.** Ductwork Classification.

| Duct pressure class | Static pressure limit | | Mean air velocity (maximum) m/s | Air leakage |
|---|---|---|---|---|
| | Positive Pa | Negative Pa | | |
| Low | 500 | 500 | 10 | Class A |
| Medium | 1000 | 750 | 20 | Class B |
| High | 2000 | 750 | 40 | Class C |
| | 2500 | 750 | 40 | Class D |

**Table B16.15.** Air Leakage Limits.

| Air leakage | Leakage limit litre/s per m$^2$ of duct surface area |
|---|---|
| Low-pressure–Class A | $0.027p^{0.65}$ |
| Medium-pressure–Class B | $0.009p^{0.65}$ |
| High-pressure–Class C | $0.003p^{0.65}$ |
| High-pressure–Class D | $0.001p^{0.65}$ |

where $p$ is the differential pressure in Pa.

The static pressure in a ductwork distribution system reduces progressively from the fan to the terminal unit or outlet. The design engineer can, by considering the operational characteristics throughout the ductwork distribution, segregate the system into sections with duct leakage classifications appropriate to the static pressures and velocities.

For example, typical systems may be classified for leakage limits as follows:

| | |
|---|---|
| Plant rooms and main risers | Class C |
| Main floor distribution ducts | Class B |
| Low-pressure velocity ducts | Class A |

## Plant Connections

The sheet metal ducting used for the interconnection of air handling components or for the connection of an air handling unit to a ductwork distribution system is referred to as plant connections.

The sheet thickness used in fabrication of plant connections and the stiffening arrangements employed will be related to the operating pressure of the application. Similarly, permissible leakage rates will depend on operating pressures.

Maintenance staff may require to enter plant connection ducts to service air handling components. Where this is necessary, a floor plate should be provided in the plant connection, suitably supported from stiffeners so that loading is not transferred to the sheet metal ductwork.

**Duct Supports and Hangers**

The maximum support spacings shown in Tables B16.16 to 18 are for general ductwork applications. Closer spacings may be required to provide improved ducting rigidity or where there are limitations in the building structure.

**Table B16.16.** Supports for horizontal ducts – rectangular.

| Maximum duct size (longer side) | Hanger | | Bearing member | | | Maximum spacing of hanger |
|---|---|---|---|---|---|---|
| | Rod or studding (two) (dia.) | Flat strap (two) | Rolled steel angle (or flat) | Rolled steel channel section W H | Roll formed channel section profile W H | |
| mm | mm | mm | mm | mm | mm | mm |
| 400 | 6 | 25 × 0·8 (plain or perforated) | H W 25 × 25 × 1·5 (or 25 × 3 flat) (plain) | H W 25 × 25 × 1·5 | H W 20 × 25 × 1·5 | 3000 |
| 600 | 8 | 25 × 3 (plain) | 25 × 25 × 3 | 25 × 25 × 3 | 25 × 25 × 1·5 | 3000 |
| 1000 | 8 | 30 × 3 (plain) | 30 × 30 × 3 | 25 × 30 × 3 | 30 × 25 × 1·5 | 2500 |
| 1500 | 10 | 40 × 5 (plain) | 40 × 40 × 3 | 30 × 40 × 3 | 40 × 25 × 1·5 | 2500 |
| 2000 | 10 | 40 × 5 (plain) | 40 × 40 × 4 | 30 × 40 × 4 | 40 × 25 × 1·5 | 2500 |
| 3000 | 12 | 40 × 6 (plain) | According to circumstances | | | |

H = Height W = Width

**Table B16.17.** Supports for horizontal ducts – circular.

| Maximum duct diameter | Hanger | | Bearing member | | Maximum spacing | |
|---|---|---|---|---|---|---|
| | Drop rod or studding (two) (dia.) | Flat strap (two) | Stirrup | Wrap-round or split clip | Spirally-wound duct | Straight-seamed duct |
| mm | mm | mm | mm | mm | mm | mm |
| 305 | – | 25 × 0·8 (plain or perforated) | – | 25 × 0·8 (plain or perforated) | 3000 | 1800 |
| 813 | 8 | 25 × 3 (plain) | 30 × 4 | 25 × 3 | 3500 | 2500 |
| 1016 | 10 | 40 × 5 (plain) | 40 × 5 | – | 3500 | 2500 |
| 1524 | 10 | 40 × 5 (plain) | 40 × 5 | – | 3500 | 2500 |

**Table B16.18.** Supports for horizontal ducts – flat oval.

| Maximum length or major axis mm | Hanger | | Bearer | | Maximum spacing mm |
|---|---|---|---|---|---|
| | Drop rod or studding (two) (dia.) mm | Flat strap (two) mm | Flat strap mm | Rolled steel angle mm | |
| 400 | 8 | 25 × 0·8 (plain or perforated) | 25 × 3 (plain) | 25 × 25 × 3 | 3000 |
| 605 | 8 | 25 × 3 (plain) | 30 × 4 (plain) | 25 × 25 × 3 | 3000 |
| 1005 | 10 | 30 × 3 (plain) | 40 × 5 (plain) | 30 × 30 × 3 | 3000 |
| 1510 | 10 | 40 × 5 (plain) | – | 40 × 40 × 3 | 3000 |

Notes to Tables B16.16 to B16.18.

(1) The dimensions included in the tables are to be regarded as minima.

(2) The maximum spacings set out in the tables are related solely to duct weight considerations. Closer spacings may be required by reason of the limitations of the buildings structure or to achieve the necessary duct rigidity.

## Other Ducts

*Plastic Ducts for Low-velocity Low-pressure Air Systems*

The HVCA Specification DW/151 details standards for ductwork constructed from uPVC and polypropylene and gives information on mechanical, thermal, and chemical properties of the materials, types available, linear expansion, fire properties, etc.

Where ducts in excess of the sizes given are required, it is usual to reinforce these with a glass fibre/polyester resin laminate. Details of this are given in the HVCA Specification.

Both unplasticised polyvinylchloride (uPVC) and polypropylene are suitable for use for higher temperatures than those shown in the tables. However the design of ductwork would need to be modified. In some cases it may be necessary to reinforce the ducts with the laminate previously referred to and in addition to providing for higher temperatures, the ducts would then be suitable for use with higher pressures. Reference should be made to specialist manufacturers for design of ducts for these purposes.

*Resin Bonded Glass Fibre Ducts for Low-velocity Low-pressure Air Systems*

The HVCA Specification DW/191 is the current specification. Resin-bonded glass fibre is a lightweight rigid material with good thermal insulation and acoustic absorption properties. It has particular advantages for use as ductwork where those properties are required and in such circumstances it may be found to be cheaper than metal ductwork with external insulation and acoustic lining.

The material is available in the form of flat board and in circular pre-formed section. The board is easily fabricated in the workshop or on site, the main requirement being a large flat table on which to work. Special tools for cutting are readily available from the board suppliers.

Table B16.19 summarises construction details.

**Table B16.19.** Resin-bonded glass fibre ductwork.

| Length of side (mm) | Pressure (Pa) | Acceptable cross joint | Maximum spacing of joints or stiffeners (mm) | |
|---|---|---|---|---|
| | | | Transverse | Longitudinal |
| 300 | Up to 500 | A B C D | Unlimited | Unlimited |
| 301 to 500 | Up to 250 | A B C D | Unlimited | Unlimited |
| 301 to 500 | 251 to 500 | A B C D | 1200 | Unlimited |
| 501 to 900 | Up to 250 | D | 1200 | Unlimited |
| 501 to 900 | 251 to 500 | D | 600 | Unlimited |
| 901 to 1200 | Up to 250 | D | 600 | Unlimited |
| 901 to 1200 | 251 to 500 | D | 600 | One stiffener centrally located |
| 1201 to 1800 | Up to 250 | D | 600 | Two stiffeners equally placed |
| 1201 to 1800 | 251 to 500 | D | 600 | Two stiffeners equally placed |

*Notes:*

A–Butt C–Plain metal sleeve

B–Socket and spigot D–Reinforced metal sleeve

1 In all cases, the stiffening provisions are the minimum necessary to restrict deflection. For ducts over 500 mm, particularly in the 251 to 500 Pa range some further stiffening may be required and the installing contractor should provide for this.

2 In locating stiffeners, regard should be paid as to whether the duct is under positive or negative pressure. Stiffeners should be securely fixed to the duct as appropriate.

3 Where the larger side of the duct requires transverse stiffening, the shorter side must be similarly stiffened so that the stiffening is continuous around the duct.

4 Where longitudinal stiffening is required, the transverse stiffeners must be fitted first, then the longitudinal members bolted between the transverse stiffeners.

*Polypropylene*

This material is available as:

(*a*) Extruded sheet of size up to 3000 by 1500 mm, in a range of thicknesses between 1 and 9·5 mm.

(*b*) Convolute tube in sizes from 22 to 75 mm. Pipe in natural and black in a range of sizes up to 300 mm diameter. A comprehensive range of fittings is available for pipes up to about 150 mm diameter. Above this diameter it is usual to fabricate fittings as required or to weld side branches directly to the ductwork.

Polypropylene pipe should be used where available. If unavailable, circular and rectangular ducts must be constructed from sheet. The thicknesses, angles (where required) and stiffeners are set out in Tables B16.20 and B16.21.

**Table B16.20.** Rectangular low velocity/pressure polypropylene. Temperature range -10 to +40°C.

| Length of side (mm) | Minimum sheet thickness (mm) | Type of cross joint | Minimum size of flanges (mm) | Minimum size of stiffeners (mm) | Maximum spacing between flanges/ stiffeners (mm) | Maximum spacing between supports (mm) |
|---|---|---|---|---|---|---|
| Up to 400 | 3 | Spigot and socket | – | – | – | 2400 |
| 401 to 600 | 3 | Flange joints with mild steel backing flanges | 40 × 6 polypropylene with 30 × 4 mild steel backing | 40 × 6 polypropylene with 30 × 4 mild steel backing | 1200 | 2400 |
| 601 to 800 | 4·5 | Flange joints with mild steel backing flanges | 45 × 6 polypropylene with 40 × 6 mild steel backing | 45 × 6 polypropylene with 40 × 6 mild steel backing | 1200 | 2400 |
| 801 to 1000 | 4·5 | Flange joints with mild steel backing flanges | 45 × 6 polypropylene with 40 × 6 mild steel backing | 45 × 6 polypropylene with 40 × 6 mild steel backing | 1200 | 2400 |
| 1001 to 1200 | 6 | Flange joints with mild steel backing flanges | 45 × 6 polypropylene with 40 × 6 mild steel backing | 45 × 6 polypropylene with 40 × 6 mild steel backing | 800 | 2400 |
| With larger side lengths the polypropylene is glass reinforced | | | | | | |
| 1201 to 1500 | 3 with 1·2 $kg/m^2$ glass reinforcement | Flange joints with mild steel backing flanges | 55 × 6 polypropylene with 50 × 6 mild steel backing | 55 × 6 polypropylene with 50 × 6 mild steel backing | 800 | 2400 |
| 1501 to 2250 | 3 with 1·2 $kg/m^2$ glass reinforcement | Flange joints with mild steel backing flanges | 55 × 6 polypropylene with 50 × 6 mild steel backing | 55 × 6 polypropylene with 50 × 6 mild steel backing | 800 | 2000 |
| 2251 to 3000 | 3 with 1·2 $kg/m^2$ glass reinforcement | Flange joints with mild steel backing flanges | 70 × 10 polypropylene with 60 × 10 mild steel backing | 70 × 10 polypropylene with 60 × 10 mild steel backing | 600 | To suit loading |

**Table B16.21.** Circular low velocity/pressure polypropylene. Temperature range -10 to +4O°C.

| Diameter (mm) | Minimum sheet thickness (mm) | Type of cross joint | Minimum size of flanges (mm) | Minimum size of stiffeners (mm) | Maximum spacing between flanges/ stiffeners (mm) | Maximum spacing between supports (mm) |
|---|---|---|---|---|---|---|
| Up to 500 | 3 | Spigot and socket | – | – | – | 2400 |
| 501 to 750 | 4·5 | Flanges with mild steel backing flanges | 40 × 6 polypropylene with 30 × 4 mild steel backing | 40 × 6 polypropylene with 30 × 4 mild steel backing | 2400 | 2400 |
| 751 to 1200 | 6 | Fianges with mild steel backing flanges | 50 × 6 polypropylene with 40 × 6 mild steel backing | 50 × 6 polypropylene with 40 × 6 mild steel backing | 2400 | 2400 |
| With larger diameters the polypropylene is glass reinforced | | | | | | |
| 1201 to 1750 | 2 with 1·8 $kg/m^2$ glass reinforcement | Flanges with mild steel backing flanges | 55 × 3 polypropylene with 1·2 $kg/m^2$ glass reinforcement and 50 × 50 × 4 mild steel angle section | 40 × 40 × 4 mild steel with 1·2 $kg/m^2$ GRP or GRP section with moment of inertia = 140 $cm^4$ | 1200 | 2000 |
| 1751 to 2500 | 2 with 1·8 $kg/m^2$ glass reinforcement | Flanges with mild steel backing flanges | 55 × 3 polypropylene with 1·2 $kg/m^2$ glass reinforcement and 50 × 50 × 4 mild steel angle section | 40 × 40 × 4 mild steel with 1·2 $kg/m^2$ GRP or GRP section with moment of inertia = 140 $cm^4$ | 1200 | To suit loading |

*Unplasticised Polyvinylchloride (uPVC)*

uPVC is available as pressed or extruded sheet to BS 3757 in a wide range of colours and several grades. Sheet thicknesses from 0·13 to 9·53 mm are available with a maximum length of 300 mm.

Extruded uPVC sections are also available, both rectangular and circular to BS 3506. The maximum size for circular extrusions is 460 mm diameter and circular ducts in excess of this must be fabricated from flat sheet.

Sheet thicknesses and other constructional details are shown in Tables B16.22 and B16.23.

**Table B16.22.** Rectangular – Low-velocity/pressure uPVC. Temperature range –10 to +4O°C.

| Length of side (mm) | Minimum sheet thickness (mm) | Type of cross joint | Minimum size of angle flanges (mm) | Minimum section for intermediate stiffeners (mm) | Maximum spacing between joints/stiffeners (mm) |
|---|---|---|---|---|---|
| Up to 400 | 3 | Spigot and socket | – | – | – |
| 401 to 600 | 3 | uPVC angle flanges with mild steel backing flanges | 30 × 30 × 4·5 uPVC with 25 × 25 × 3 mild steel backing | 30 × 30 × 4·5 uPVC with 25 × 25 × 3 mild steel backing | 2400 |
| 601 to 800 | 4·5 | uPVC angle flanges with mild steel backing flanges | 30 × 30 × 4·5 uPVC with 25 × 25 × 3 mild steel backing | 30 × 30 × 4·5 uPVC with 25 × 25 × 3 mild steel backing | 2400 |
| 801 to 1000 | 4·5 | uPVC angle flanges with mild steel backing flanges | 30 × 30 × 4·5 uPVC with 25 × 25 × 3 mild steel backing | 30 × 30 × 4·5 uPVC with 25 × 25 × 3 mild steel backing | 1200 |
| 1001 to 1200 | 4·5 | uPVC angle flanges with mild steel backing flanges | 40 × 40 × 4·5 uPVC with 30 × 30 × 3 mild steel backing | 40 × 40 × 4·5 uPVC with 30 × 30 × 3 mild steel backing | 1200 |
| With larger side lengths the uPVC is glass reinforced | | | | | |
| 1201 to 2250 | 4·5 with 0·6 kg/m$^2$ glass reinforcement | uPVC angle flanges with mild steel backing flanges | 45 × 45 × 6 uPVC with 40 × 40 × 4 mild steel backing | 45 × 45 × 6 uPVC with 40 × 40 × 4 mild steel backing or GRP section with moment of inertia = 140 cm$^4$ | 800 |
| 2251 to 3000 | 4·5 with 1·2 kg/m$^2$ glass reinforcement | uPVC angle flanges with mild steel backing flanges | 55 × 55 × 6 uPVC with 50 × 50 × 4 mild steel backing | 55 × 55 × 6 uPVC with 50 × 50 × 4 mild steel backing or GRP section with moment of inertia = 325 cm$^4$ | 600 |

**Table B16.23.** Circular – Low-velocity/pressure uPVC. Temperature range –10 to +40°C.

| Diameter (mm) | Minimum sheet thickness (mm) | Type of circumferential joint | Minimum size of angle flanges (mm) | Minimum section for intermediate stiffeners (mm) | Maximum spacing between joint/stiffeners (mm) |
|---|---|---|---|---|---|
| Up to 500 | 3 | Spigot and socket | – | – | – |
| 501 to 750 | 4·5 | uPVC angle flanges with mild steel backing | 30 × 30 × 4·5 uPVC with 25 × 25 × 3 mild steel backing | 30 × 30 × 4·5 uPVC with 25 × 25 × 3 mild steel backing | 2400 |
| 751 to 1200 | 4·5 | uPVC angle flanges with mild steel backing | 40 × 40 × 4·5 uPVC with 30 × 30 × 3 mild steel backing | 40 × 40 × 4·5 uPVC with 30 × 30 × 3 mild steel backing | 2400 |
| With larger diameters the uPVC is glass reinforced | | | | | |
| 1201 to 1500 | 3 with 0·6 kg/m$^2$ glass reinforcement | uPVC angle flanges with mild steel backing | 45 × 45 × 6 uPVC with 40 × 40 × 4 mild steel backing | 45 × 45 × 6 uPVC with 40 × 40 × 4 mild steel backing or GRP section with moment of inertia = 140 cm$^4$ | 1200 |
| 1501 to 2500 | 3 with 0·6 kg/m$^2$ glass reinforcement | uPVC angle flanges with mild steel backing | 45 × 45 × 6 uPVC with 40 × 40 × 4 mild steel backing | 45 × 45× 6 uPVC with 40 × 40 × 4 mild steel backing or GRP section with moment of inertia = 140 cm$^4$ | 800 |

*Asbestos Cement*

For extract applications where the ductwork is exposed to high temperatures, asbestos cement ducts may be used. In constructing asbestos cement ducts particular attention is required to ensure that internal surfaces are smooth.

Constructional standards for asbestos cement ducts are given in Table B16.24.

**Table B16.24.** Dimensions of asbestos cement ducts.

| Size of duct (mm) | Thickness (mm) | Dimensions over socket (mm) | Maximum length (mm) |
|---|---|---|---|
| Square and rectangular ducts | | | |
| 102 × 102 | 9·5 | 149 × 149 | 1830 |
| 102 × 152 | 9·5 | 152 × 203 | 1830 |
| 152 × 152 | 9·5 | 203 × 203 | 1830 |
| 152 × 203 | 9·5 | 203 × 260 | 1830 |
| 203 × 203 | 9·5 | 260 × 260 | 1830 |
| 203 × 254 | 9·5 | 260 × 311 | 1830 |
| 254 × 254 | 9·5 | 311 × 311 | 1830 |
| 254 × 305 | 12·7 | 324 × 375 | 1830 |
| 305 × 305 | 12·7 | 375 × 375 | 1830 |
| 381 × 381 | 12·7 | 451 × 451 | 1120 |
| 457 × 457 | 12·7 | 527 × 527 | 1120 |
| 533 × 533 | 12·7 | 603 × 603 | 1120 |
| 610 × 610 | 12·7 | 680 × 680 | 1120 |
| Circular ducts (diameter) | | | |
| 178 | 9·5 | 235 | 1830 |
| 203 | 9·5 | 260 | 1830 |
| 229 | 9·5 | 286 | 1830 |
| 254 | 9·5 | 311 | 1830 |
| 305 | 12·7 | 375 | 1830 |
| 381 | 12·7 | 451 | 1830 |
| 457 | 12·7 | 527 | 1120 |
| 533 | 12·7 | 603 | 1120 |
| 610 | 12·7 | 680 | 1120 |

*Builder's Work Ducts*

Builder's work ducts are occasionally used for air distribution within buildings and to form plant connections for air handling components. Typical examples include the use of corridor ceiling voids and large vertical ducts in multi-storey buildings connecting to sheet metal ductwork systems on each floor. Plenum chambers and large plant connections are sometimes constructed in builder's work.

When considering the use of builder's work ducts it must be borne in mind that virtually all building materials are porous to some degree. Furthermore, most construction methods, particularly those using mortar joints, entail numerous leakage paths between components. The application of a sealant, even with top-class workmanship, is unlikely to be effective particularly at corners, and settlement expansion and contraction of the structure will create further problems.

Summarising, the use of builder's work ducts is not recommended unless very generous allowances can be made for leakage. Even with this proviso, it would be important to provide easy personnel access so that re-sealing may be undertaken to attempt to compensate for deterioration.

**Flexible Ductwork**

Flexible ductwork must meet the standards of air-tightness required by the specification, be within the frictional resistance limits specified by the designer, and comply with the fire regulations or requirements relevant to the application.

*Fabric Flexible Ducts*

Typical proprietary flexible ducts comprise a liner and cover of tough, tear-resistant, glass fibre, proofed fabric, reinforced with a galvanised spring wire helix between liner and cover, with an outer helix of glass fibre cord. Various types are available using a range of materials and care must be taken to ensure that the selection is appropriate to the application.

Lengths of flexible ductwork should be kept to a minimum. In normal circumstances, connections should not exceed six diameters in length.

*Metal Flexible Ducts*

Proprietary flexible ducts are available as spirally wound rigid section bendable ducts, manufactured from a variety of metals. Similar products, incorporating non-metallic linings, coverings, or layers between metal facings, are also available.

The selection of flexible metal ducts will depend upon the relative importance to the particular application of:

(*a*) resistance to corrosion

(*b*) resistance to high temperatures

(*c*) ability to withstand damage on site

(*d*) lightness and ease of handling

(*e*) adaptability to pre-insulation

(*f*) choice of lengths for economic use

(*g*) means of support

**Ductwork Cleanliness**

With commercial filtration standards dust can penetrate or by-pass the filter medium and enter the ductwork system. A proportion of this dust will fall out of entrainment, particularly where air velocities are low, and settle on the internal duct surfaces.

Similarly, with extract/recirculation applications, dust generated in the space(s) served by the system may be carried into the duct and subsequently settle out in the same way.

Undue accumulation of dust in ductwork systems may be viewed as posing a threat to the health of building occupants. For this reason, it is recommended that plant connection ducts and other parts of the system with easy access be kept clean. Other sections of ductwork, including small ducts, may be inspected where necessary using fibre optic techniques. Specialist companies can offer a comprehensive cleaning service for ductwork using purpose-designed specialist equipment. This includes the use of a mop on a long-lead helical semi-stiff wire with a rotating drive which in conjunction with a vacuum, can be used to clean all ducts down to very small sizes.

## DAMPERS

In air-conditioning and ventilation applications dampers are used for controlling the volume of air passing along a duct and for controlling the spread of fire and smoke.

Casings for proprietary dampers may be provided with shop fabricated flanges, spigot connections or proprietary duct jointing devices. Additionally, to meet current specifications for duct leakage rates, double-skin casings are used to reduce the number of bearing points or other leakage routes open to atmosphere. In many designs only the operating spindle itself penetrates the outer casing and the bearing for this may be sealed.

### Volume Control Dampers

*Single Leaf*

These consist of a single blade, usually pivoted at the centre (See Fig. B16.6). Normally these dampers are adjusted manually.

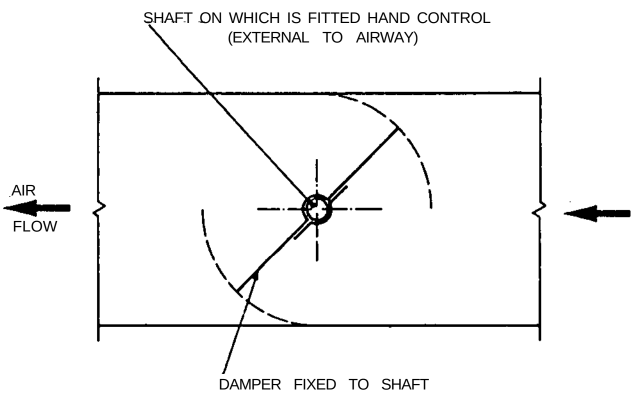



**Fig. B16.6.** Single leaf dampers – centre pivoted.

*Multi-leaf*

These consist of a number of rectangular blades mounted on spindles which are supported in bearings in a frame. There are two main types: parallel blades, where the blades rotate in one direction, and opposed blades, where adjacent blades rotate in opposite directions (see Fig. B16.7). The blades may be made from single sheets of metal suitably formed to provide adequate stiffness or they may be of the double-skinned, stream-lined or aerofoil type.

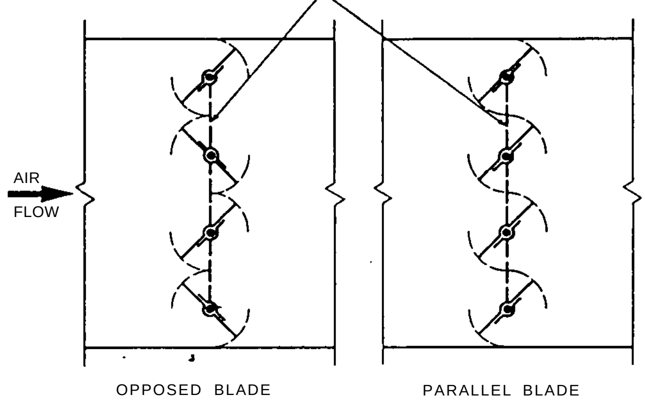



**Fig. B16.7.** Multi-leaf dampers.

The damper casing may be provided with spigots to adapt the unit for use in circular or flatted-oval ducts in addition to rectangular duct applications.

Where dampers are required to provide tight shut-off, the specification may call for seals at the sides and trailing edges of the blades. Stainless steel, felt and neoprene are typical materials used for these purposes.

For commissioning or balancing operations the damper setting would be adjusted manually from an operating quadrant, control knob or handwheel. A position indicator would be provided together with a locking device to fix permanently the damper setting required.

For automatic regulation applications the damper setting may be varied using a variety of remote actuators operating through a mechanical linkage. Direct electric drive arrangements are also available and damper status indicator lights may be provided. For ease of access for servicing it is recommended that electric direct drive units are always installed external to the damper casing.

Automatic spring return to a fail-safe damper setting may be arranged as a provision in the event of actuator power failure.

*Iris*

These are for use in circular ducts and use the ‘camera shutter’ principle (see Fig. B16.8).

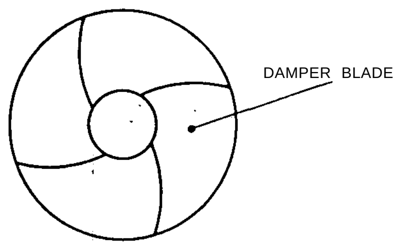



**Fig. B16.8.** Iris damper.

*Slide*

These consist of a single rectangular blade which is moved into the duct by a required amount (see Fig. B16.9). They have the disadvantage of requiring a lot of space outside the duct. Their use is mainly confined to process applications such as pneumatic conveying or ‘spot’ extraction systems, (e.g. welding booth applications).

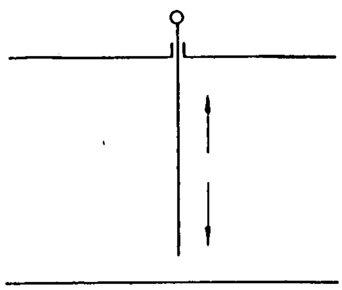


**Fig. B16.9.** Slide dampers.

*'Hit and Miss'*

These consist of perforated fixed and sliding blades, the latter being moved to adjust the air flow rate (see Fig. B16.10). Units of this type are also applied as flow-equalising devices to improve the flow profile across a duct.

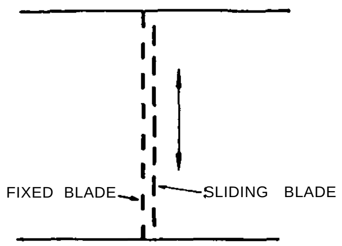



**Fig. B16.10.** 'Hit and Miss' damper.

**Automatic Volume Regulating Devices**

Proprietary units are provided to meet various volume regulating requirements for air conditioning and ventilating applications. These include constant volume units and devices which modulate air volume flow rates to meet changes in a control variable. Such units may be self-acting (see Fig. B16.11) or actuated from a suitable power source in response to an external signal (see Fig. B16.12). An air flow equalising device is often fitted within these units.

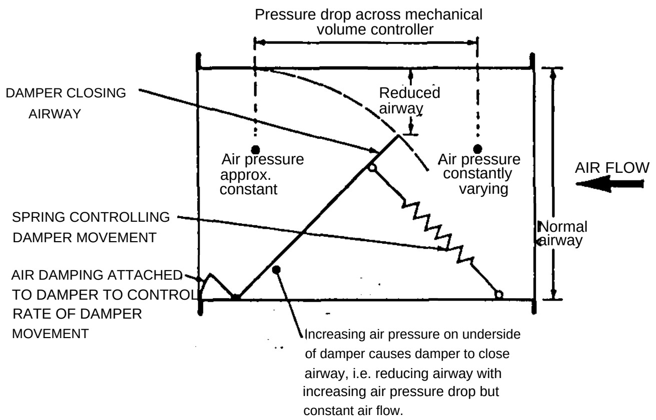



**Fig. B16.11.** Self-actuating volume regulating device.

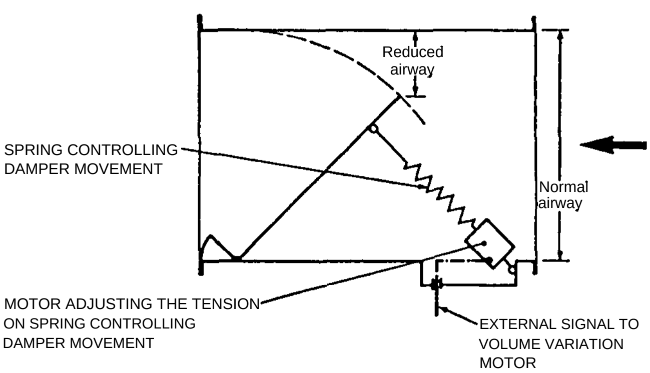



**Fig. B16.12.** Externally actuated volume regulating device.

Butterfly blade arrangements, where twin blades are pivoted in a 'V' assembly, may also be used for this device.

**Fire Dampers**

*General*

These units require to be type tested and rated to BS 476: Part 8 to maintain the integrity of a fire compartment for a specified minimum period. Where necessary, factory assembled units of two dampers in series may be applied to meet extended period requirements.

Account must be taken of the thermal expansion of a fire damper unit when heated. The use of the HVCA/HEVAC purpose designed installation frame allows for the expansion of the damper assembly under fire conditions without undermining the integrity of the surrounding structure.

As with all duct mounted components, there is a need for fire damper casings to meet current specifications for duct leakage.

*Interlocking Curtain*

A series of interlocking rollformed blades of stainless or galvanised mild steel are held in the open position by a device incorporating a fusible link or thermal actuator. At a preset temperature (normally 72°C) the curtain will be released to shut off the duct and to maintain the integrity of the fire compartment for the rated period of the damper. For interlocking curtain units this is normally 2 hours. Closure of the damper is effected by constant tension stainless steel springs which pull the curtain across the duct opening, positively locating the bottom blade into locking ramps (see Fig. B16.13).

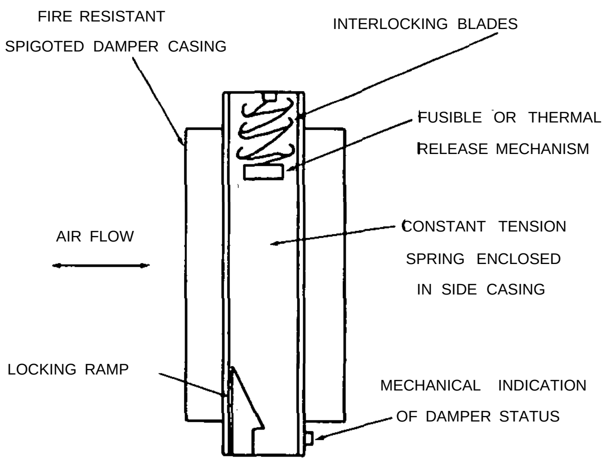



**Fig. B16.13.** Interlocking curtain fire damper (Suitable for use in both horizontal and vertical operating planes).

The spigoted casing connections may be made to suit rectangular, circular or flatted oval ducting.

The system can be designed with the folded interlocked blade assembly located outside the airstream, thus ensuring minimum air flow resistance. This facility is particularly useful in high velocity systems.

The damper release mechanism incorporating the fusible link or thermal actuator must permit manual testing of the damper operation to the fully closed position in accordance with BSCP 413: Ducts for building services.

The release mechanism should also be suitable for reloading and replacement without the use of tools. Replacement cassette mechanisms which meet this requirement are available.

Interlocking curtain fire dampers are provided with electromagnetic or solenoid actuators to offer the facility of interfacing with smoke detection or similar safety related systems. For ease of access actuators are positioned outside the damper casing.

*Offset Hinged Single Blade*

This type of fire damper comprises a single blade held in the open position by a fusible link, see Fig. B16.14. Excessive heat causes the link to part and the blade, in some instances assisted by weights, swivels into position to seal the ventilation duct. The unit is of heavy plate construction and horizontal and vertical patterns are manufactured, both being mounted within a framework suitable for forming part of the ducting.

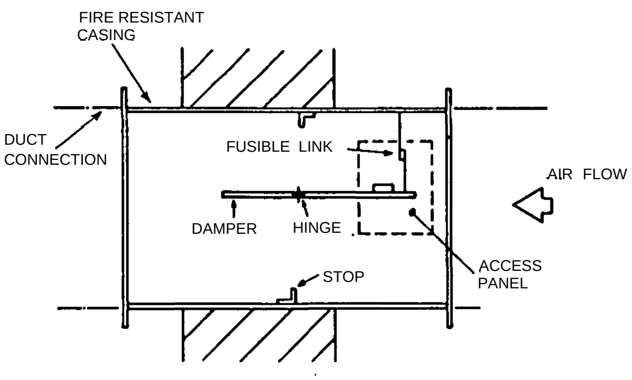



**Fig. B16.14.** Offset hinged single blade fire damper.

*Multi-leaf*

In construction and operation, multi-leaf type fire dampers are similar to the offset hinged single blade type. A mild steel tie-bar linkage arrangement is incorporated, however, to ensure that all blades move in unison whenever the fusible link parts, see Fig. B16.15.

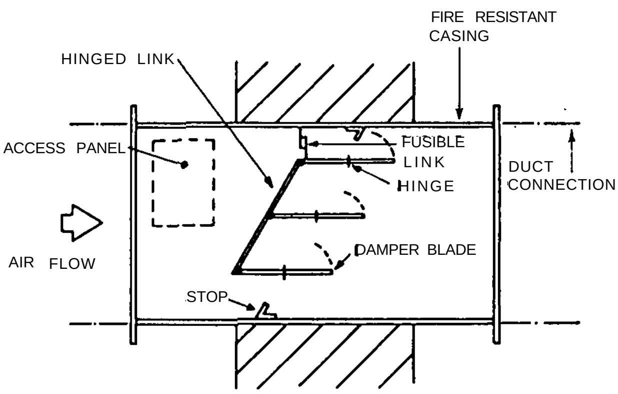



**Fig. B16.15.** Multi-leaf fire damper.

*Intumescent Coated Honeycombs*

Intumescent paint incorporates a foaming agent and, on being heated, it expands to form a thick insulating coat upon the material to which it has been applied. This characteristic is used to create a fire barrier within ductwork. The main advantages claimed for this type of fire damper are that it is cheap, has no moving parts, can be made resistant to corrosive vapours and can easily be replaced. However, it is not suitable for remote actuation or for smoke control.

*Combination Fire and Smoke Dampers*

These multi-bladed units are 2 hour fire rated and provided with a fusible link or thermal actuator to offer fail-safe closure under fire conditions. Pre-loaded integral closure springs ensure snap-action tight shutoff under dynamic airflow conditions.

In addition, the blade design and the non-degradable peripheral seals are arranged to enable the damper to provide tight closure to products of combustion. When used in this role the damper is powered open by remotely operated electric or electromechanical actuators.

This versatile unit can thus provide smoke control as well as complete fire protection. A system of units may be integrated with a smoke detection system or with a building management system. Remote indication of damper status can also be arranged.

*Smoke Release Dampers*

These units may be wall or roof mounted as well as installed in duct inlets/outlets.

The design of the metal blades and the provision of edge seals offer tight-sealing characteristics and ensure that leakage (and, hence, energy) losses are minimal. In an emergency, remote signal actuation allows the blades to be spring operated to the fail-safe, full-open position, thus permitting smoke venting to atmosphere. The units may be powered to the closed position by a remotely operated integral actuator.

## PUMP TYPES

According to their characteristic constructional features, pumps are divided into the following general classes:

(1) *Reciprocating pumps* containing a piston or plunger which is given a reciprocating movement in the pump cylinder.

(2) *Rotary pumps* in which liquid is forced through the pump by means of rotating screws or gears, etc., without the use of centrifugal force.

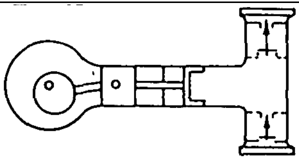
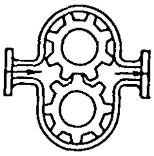
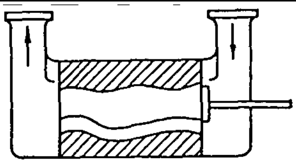
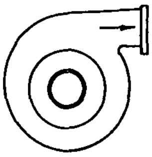
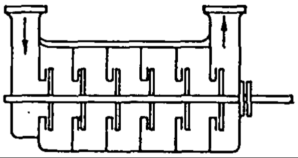
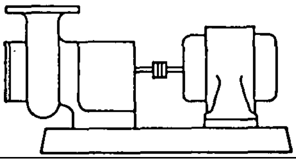
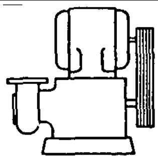
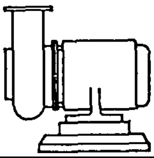
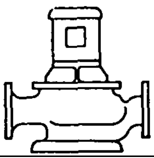
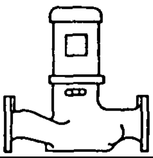
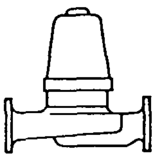
(3) *Centrifugal pumps* in which liquid flows through the pump due to the centrifugal force imparted by the rotation of one or more impellers.

Table B16.25 lists details of pump types.

### Reciprocating Pumps

A reciprocating pump generally comprises one or more pistons moving backwards and forwards in their respective cylinders, each stroke drawing in liquid through one valve which is then forcibly expelled through another. The amount of liquid pumped with each stroke is governed by the cylinder volume and the speed of operation. Since the pump is positive acting it will always attempt to expel the contents of the cylinder into the discharge pipe, so that the pressure developed by the pump depends entirely on the restriction encountered. To safeguard the system

**Table B16.25.** Types of pumps.

| Type | Sketch | Description |
|---|---|---|
| Reciprocating |  | Reciprocating pumps comprise one or more cylinders each having a piston. A pump having one discharge and one suction stroke per cycle is referred to as single-acting and a pump having two discharge and two suction strokes per cycle is referred to as being double-acting. When driven by an electric motor a reciprocating pump incorporates a crankshaft driven through a suitable speed reducing gear since these pumps are essentially of low speed operation. |
| Rotary gear |  | The gear type of pump shown has many variations in the field of rotary pumps. Liquid fills the spaces between the teeth on the suction side and is carried over to the discharge side, where it is forced out by the engaging teeth. One gear is keyed to the driving shaft, the other rotates as an idler. Primarily used for pumping oil or other liquids with sufficient lubricating properties. |
| Rotary moyno |  | The moyno screw pump consists of one single thread rotor running eccentrically in a double threaded rubber or similar casing. The length of the rotor is such that liquid is trapped in the stationary helices between two successive threads of the rotor. Since the rotor has an eccentric motion as it rotates, it is connected to the drive shaft by a connecting rod and two universal joints. |
| Centrifugal single-stage |  | The majority of applications involving the use of a centrifugal pump can be dealt with by a single stage pump, which has one impeller. This has backward curved vanes and is fitted in a casing usually of volute shape to provide maximum conversion of kinetic energy into pressure energy. |
| Centrifugal multi-stage |  | When higher pressures are required than can be reasonably obtained by the use of a single impeller, several impellers are fitted to a common shaft to form a multi-stage pump, see page B16.29. It is usual for multi-stage pumps to incorporate a hydraulic balancing device to relieve the pump bearings from the thrust produced by the high pressure generated at the pump outlet. |
| Direct coupled |  | Pump driven through flexible coupling from motor, both units mounted on steel or cast iron baseplate. The pump has its own bearings and duty is adjusted by varying the impeller diameter. |
| Belt drive |  | Pump with its own bearings driven by V-belts from motor fitted at side of pump on slide rails or mounted above pump. Pulleys can be selected to run pump at appropriate speed to suit duty required. |
| Close coupled |  | Motor has a specially extended shaft which enters pump casing and to which impeller is directly fitted. The pump does not have its own bearings, and the thrust developed has to be withstood by the bearings in the motor. Duty adjusted by varying the impeller diameter. |
| Inline (floor mounted) |  | An inline pump has its casing arranged to provide the inlet and outlet connections on the same centre line to facilitate ease of installation. |
| Inline (pipeline mounted) |  | A pipeline pump has its connections arranged in line, as above, but is of light enough construction to be fitted directly in a pipeline. |
| Wet rotor |  | Widely used for domestic central heating. The motor is designed to allow its rotor to run in water, thus no seal is required for the motor shaft which carries the impeller where it enters the pump waterway. The stator has to be protected from water, either by a stainless steel enclosure around the rotor, known as a can, or by embedding the stator windings in a heat resistant moulding. The small gap between the rotor and stator is thereby reduced with either method, making the unit susceptible to seizure if particles of matter such as rust or scale should become lodged in this gap. These pumps usually incorporate a means of adjusting output, either in the form of a restriction at the pump discharge or by increasing leakage loss internally from discharge to suction inlet. |

against excessive pressure being built up at no flow, it is usual to fit a relief valve after the pump. A reciprocating pump is limited to low flow rates and requires more maintenance than a centrifugal pump under continuous running. However it is more efficient and is ideally suited for low flow rates at high pressures, as in the case of boiler water feed. Reciprocating pumps are self priming and pump cold liquids satisfactorily under suction to pressures of approximately 70 kPa. The suction lift possible is dependent on the liquid temperature and its boiling point. When handling hot liquids as in the case of boiler feed, it may be necessary to ensure that there is a positive pressure on the pump to safeguard against flashing into steam. Steam driven reciprocating pumps are referred to as either simplex (one steam and water cylinder), duplex (two steam and water cylinders), or triplex (three steam and water cylinders).

**Rotary Pumps**

Rotary pumps provide their output by positive displacement, as do reciprocating pumps, but the difference is in the use of a rotational movement to achieve this. The types of rotary pumps are numerous but they all deliver a constant volume against a variable discharge pressure. Their field of application is primarily for pumping oil or other liquids having lubricating qualities and of sufficient viscosity to prevent excessive slip leakage within the pump. The operation of a rotary pump depends on the formation of practically fluid-tight enclosed spaces filled with liquid on the suction side of the pump and the displacement of this on the discharge side. They are therefore all self-priming and work satisfactorily under suction to pressures of approximately 70 kPa, although higher lifts are possible under certain conditions depending on the nature of the liquids, pump size and speed.

**Centrifugal Pumps**

Centrifugal pumps utilise centrifugal force which imparts a high velocity to the liquid being pumped. Pressure energy is obtained by the rotation of an impeller fitted within a casing which is usually of the volute type. Liquid is directed into the centre or eye of the rotating impeller vanes from which it gains a high velocity. The pump casing is designed to provide a maximum conversion of velocity into pressure energy, either by the uniformly increasing area of a volute shape or by the introduction of diffuser guide vanes. The centrifugal pump is ideally suited for almost all applications involving the circulation or the transfer of hot or cold water.

An exception to this is where low flow rates at high pressure are required, as in the case of boiler water feed, where a reciprocating pump may be preferred because of its higher efficiency. For boiler water feed applications requiring a large flow rate it is necessary to revert to a centrifugal pump of the multi-range type incorporating several impellers in series to generate a high output pressure.

*Suction Lift*

When fully primed with liquid a centrifugal pump can produce a lift through its suction pipeline of up to approximately 70 kPa dependent on the boiling point of the liquid. However, this type of pump is not inherently self-priming, i.e. if started with an empty suction pipe the centrifugal action of the pump will not remove the air and it will run dry. It is always preferable to fit a centrifugal pump where there is a positive pressure on the suction side to ensure a trouble free installation. Self-priming pumps are available, but usually there are limitations in their capacity or application. The maximum suction lift is related to the temperature of the liquid to be pumped since cavitation will occur if the pressure inside the pump falls below the vapour pressure corresponding to the temperature of the liquid. The liquid will vaporise on its way through the impeller, but as the vapour bubbles pass into the region of higher pressure they will collapse with an accompanying noise, which can damage the impeller. See Fig. B16.16.

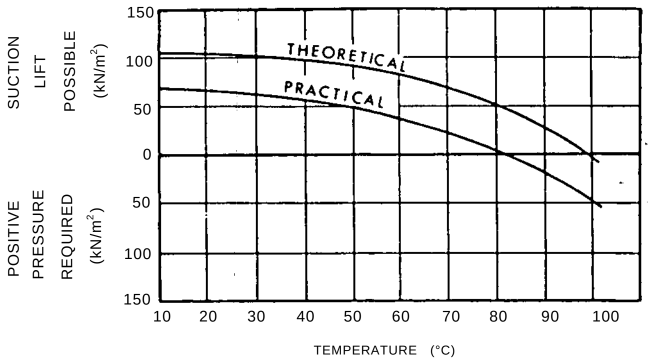



**Fig. B16.16.** Maximum theoretical and practical suction conditions for water at various temperatures, based on atmospheric pressure at sea level.

*Centrifugal Pump Construction*

The point at which the driving shaft enters the pump casing requires a liquid seal, usually a stuffing box packed with rings of graphited asbestos compressed by a gland. A slight drip of liquid is necessary to lubricate the packing and a tapping is usually provided on the pump for a drain pipe. Alternatively, a mechanical seal can be fitted, generally comprising a carbon ring which is spring loaded against a stationary seat with a rubber ring over the shaft.

Replacement of a worn mechanical seal requires a higher skill level than the addition of packing to a stuffing box. The maintenance attention available may, therefore, influence the choice of seal.

With water heating systems, the drip feed to the stuffing box represents an energy loss to the system. Again, this should be taken into account in selecting the type of seal.

*Temperature Considerations*

Most centrifugal pumps are constructed of a cast iron casing, gun-metal impeller and stainless steel shaft, and can handle up to approximately 120°C. For higher temperatures, up to approximately 150°C, it is usual for pumps to incorporate a jacket through which a small amount of cooling water passes to reduce the temperature in the stuffing box to below 100°C. For high temperature installations above approximately 150°C the pump construction changes considerably to cater for the stresses due to expansion, to enable it to resist external stresses due to thermal movement of the pipework connected to it, and to withstand the high system operating pressure involved. Casings of these pumps are usually of cast steel.

**Pump Laws**

*Change of Speed – Impeller Diameter Constant*

Volume varies directly as the speed.
Pressure developed varies as the square of the speed.
Power absorbed varies as the cube of the speed.
These laws may be written:

$$\frac{Q_2}{Q_1} = \frac{N_2}{N_1} \quad \text{B16.11}$$

$$\frac{P_2}{P_1} = \frac{N_2^2}{N_1^2} \quad \text{B16.12}$$

$$\frac{Z_2}{Z_1} = \frac{N_2^3}{N_1^3} \quad \text{B16.13}$$

where:

| | | |
|---|---|---|
| $Q_1$, $Q_2$ = discharge volumes | .. .. .. | litre/s |
| $P_1$, $P_2$ = pressures | .. .. .. .. | kPa |
| $Z_1$, $Z_2$ = powers absorbed | ... .. .. | W |
| $N_1$, $N_2$ = rotational speeds | .. .. .. | rev/s |

*Change of Impeller Diameter*

With constant speed of rotation the volume varies as the cube of the diameter of the impeller, i.e.:

$$\frac{Q_2}{Q_1} = \frac{D_2^3}{D_1^3} \quad \text{B16.14}$$

where:

$D_1$, $D_2$ = impeller diameter .. .. .. m

The pressure developed varies as the square of the diameter of the impeller, i.e.:

$$\frac{P_2}{P_1} = \frac{D_2^2}{D_1^2} \quad \text{B16.15}$$

The power absorbed varies as the fifth power of the diameter of the impeller, i.e.:

$$\frac{Z_2}{Z_1} = \frac{D_2^5}{D_1^5} \quad \text{B16.16}$$

**Centrifugal Pumps in Circuit**

The output of a centrifugal pump may be shown by its 'performance' curve, which falls, from a maximum pressure with no flow (outlet valve closed), to the reduced presure which will be produced under maximum flow conditions, see curve $C_1$ in Fig. B16.17.

A system of pipework into which a pump is fitted has its own characteristics which may be drawn as a 'system' curve, curve $R$ in Fig. B16.17. The total resistance of the system is calculated for one fluid flow rate, the curve being plotted from the relationships of flow rates to the square of the pressure loss.

The actual operation of the pump may be found at the intersection on its 'performance' curve with the 'system' curve, points $B$ in Fig. B16.17.

If the frictional resistance of a circuit has been under-estimated, it may be possible to overcome the consequent lower output from the pump by increasing its speed so that the revised 'performance' curve crosses the 'system' curve at the required design flow rate.

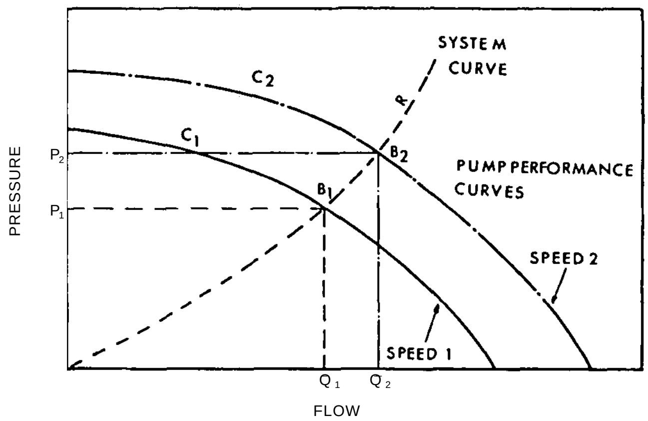



**Fig. B16.17.** Pump performance and system characteristics.

**Centrifugal Pumps in Series**

When two similar pumps are operated in series their combined pressure for a given flow rate will be double that of the single pump.

*Tandem or Multi-stage Pumps*

The limiting economical pressure for a single impeller pump is around 300 kPa. For higher pressures pumps may be used in tandem and consist of virtually independent pumps each with its own casing and impeller, but with a common shaft and with external connections to conduct the liquid from one stage to the next.

Multi-stage pumps comprise several impellers on a single shaft with integral passages for the liquid to pass from the delivery of one impeller to the suction of the next. Such pumps can generate pressures up to 12 MPa. For a given pressure, the greater the number of impellers the less peripheral velocity is needed and, within certain limits, the greater the efficiency of the unit. See Fig. B16.18.

Guide vanes are often employed to reduce shock losses in the passage of water from one pump to the next. The impellers and guide passages should be as smooth as possible to minimise friction. Design provision must be made for the axial thrust on the shaft due to the pressure exerted on each of the impellers. The efficiency of a multi-stage pump may be as high as 80 to 85%.

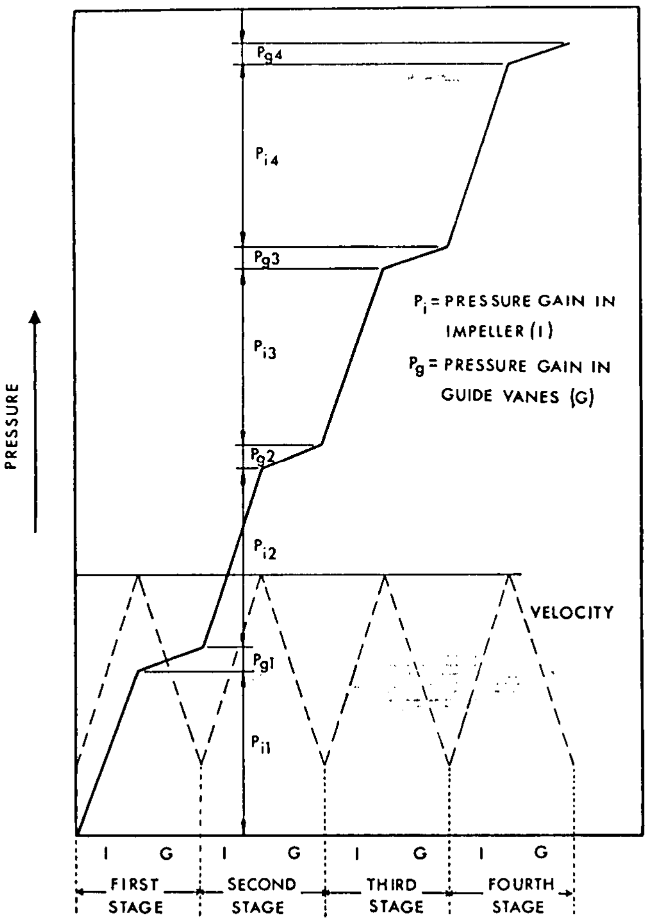



**Fig. B16.18.** Pressure/velocity changes in a multi-stage pump.

## Centrifugal Pumps in Parallel

*Pumps with Equal Diameter Impellers*

If two pumps are fitted with equal size impellers to act as main and standby they can be operated in parallel. Whilst the capacity available at any given pressure is twice that of the individual pumps the rising system-pressure characteristic will determine the actual flow of water through the system (see Fig. B16.19). Careful selection of pumps is required to ensure satisfactory operation in parallel without 'surging'.

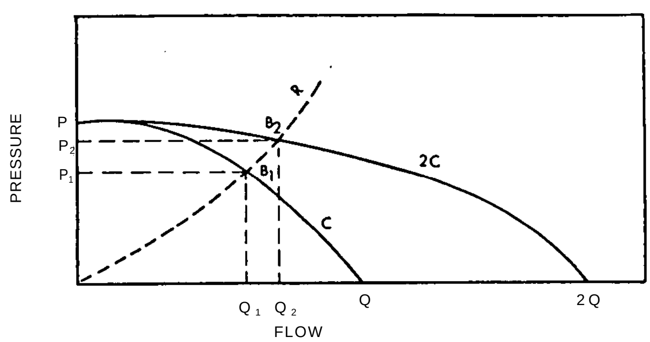



**Fig. B16.19.** Pumps in parallel – equal impeller diameter.

*Pumps with Impellers of Differing Diameters*

Where two pumps are provided with impellers of differing diameter, a choice of three working points $B_1$, $B_2$ and $B_3$ is available. Parallel operation is not possible if $B_3$ is above the zero flow discharge pressure of $P_1$. The characteristic curve for parallel operation is found by adding the capacities of pumps 1 and 2 at various pressures. The working points $B_1$, $B_2$ and $B_3$ can then be obtained by plotting the system-pressure characteristic and noting where it crosses the pump output curves, see Fig. B16.20. Again, careful pump selection is necessary for successful parallel operation.

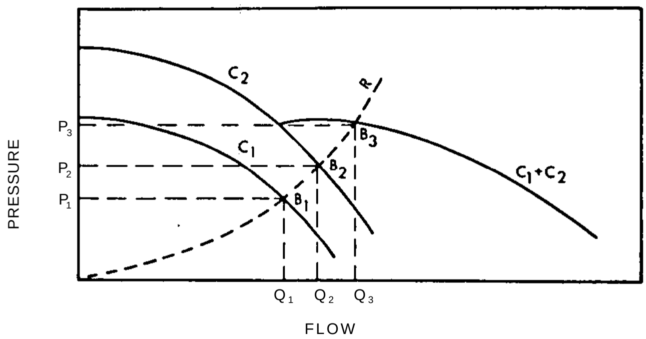



**Fig. B16.20.** Pumps in parallel – different impeller diameters.

## Distribution of Pump Pressure

The performance of a pump will be the same whether it is situated in the flow or the return of a heating system. However, because the system pressure at the point of entry of the cold feed is atmospheric (plus any static head due to the elevation of the feed and expansion tank) this may be taken as the system neutral point. Thus from this neutral point to the pump suction inlet, the effect of the pump pressure will range from zero to a negative value at the pump inlet corresponding to the frictional resistance of this part of the system. Similarly, from the neutral point to the pump outlet the pump pressure will range from zero up to a positive pressure corresponding to the friction loss in this section of pipework. The total pump pressure is the algebraic sum of the positive and negative pressure produced by the pump. It is, therefore, necessary to position the pump relative to the cold feed such that the section below datum pressure is minimal and that no part of the circuit is below atmospheric pressure when the pump is operating. At the same time, the pump pressure at the open vent point must not cause a discharge from the system.

## Applications and Installations

*Hot Water Service*

Circulating pumps sized to offset the heat losses from secondary pipework will normally require only a low circulation rate. The presence of the pump, or its associated check valve, in the flow connection may, therefore, restrict draw-off demand particularly when this is at a peak level. To overcome this the pump may be installed in the return connection but with its direction of flow towards the draw-offs. Thus, when a draw-off is opened both the flow and return secondary pipes will contribute to the outflow, the former being completely unrestricted, and the latter being pump assisted. Where this mode of pump installation is used, the return connection must be high enough in the cylinder to ensure that circulation does not run cold.

Normal materials for pump construction are cast iron casing and gun-metal impeller, but in areas known to have an aggressive water supply it may be necessary to avoid the use of certain materials, e.g. gun-metal may have to be zinc free to avoid de-zincification.

*Cold Water Pressurising Pumps*

For permanent cold water supply installations within buildings pumps are normally of the centrifugal type, being either single or multi-stage depending on the output pressure required. Reciprocating pumps are occasionally used to provide a temporary high pressure water supply, bearing in mind that they are limited to low flow rates. It is essential to fit a safety valve after a reciprocating pump to safeguard the installation against excessive pressures that would otherwise be built up if it is left running against closed end, a factor which does not apply to centrifugal pumps.

*Sewage Lifting*

When toilets etc., which would normally discharge into a sewer, are fitted at a low level which makes gravity flow impossible, then it is necessary to provide a collecting chamber for sewage together with mechanical means of lifting it to the sewer. This is usually achieved in one of two ways; by using pumps or a sewage ejector operated by compressed air.

Pumps are usually of the centrifugal type without a strainer but containing an impeller specially designed to avoid clogging, sometimes having a single ‘S’ shaped vane.

Vertical spindle pumps are used, suspended directly into a concrete sewage collecting chamber. Alternatively, pumps can be fitted in an adjacent dry chamber where they are easily accessible for maintenance. For smaller installations packaged units can be obtained comprising duplicate pumps mounted to a suitable collecting chamber.

The use of sewage ejectors is normally limited to applications requiring low flow rates and pressures. Sewage is allowed to collect in a chamber until it is nearly full. The rising level of sewage lifts a float which opens a valve admitting compressed air into the chamber. The compressed air then forcibly ejects the sewage from the chamber through a non-return valve and up the discharge pipe.

**Installation**

Pipework connections should be made in such a way as to avoid imparting stress to the pump, since the casing is not designed to support an external load. To minimise the transmission of vibration from the pump through the system it should be bolted down to a concrete block having a fairly large mass; a block 300 mm thick should suffice. Beneath the block should be laid compressed machinery cork, or equivalent, to minimise the transmission of pump vibration through the building structure.

**Control of Pumps**

Methods of controlling pumps can be sub-divided into three types; those which operate on water level, water pressure or water flow.

*Level Controls*

When a pump is being used to fill a high level tank it is possible to control the operation of the pump by a level switch fitted in this tank. These switches are adjustable in their cut in and cut out levels. A typical arrangement would be to set the switch to start the pump when the tank is approximately half empty and to stop the pump when the water has been raised to within approximately 50 mm of the maximum which the ball valve will allow into the tank. The two types of level switching arrangements most commonly used are as follows:

(*a*) *Float Switches*

These rely for their operation on a ball float rising or falling with the water level and at pre-set levels electrical contacts are operated. Float switches having a vertical rod provide a large differential between on and off, the setting being determined by fixing a collar on the rod above and below the ball float. On a rise in level the buoyancy of the float will cause it to engage against the collar and lift the rod, thus causing the switch contacts to break. Similarly on a drop in level the weight of the ball float, when falling to the position of the bottom stop, will cause the rod to fall and the contacts to make. Another form of float switch provides for horizontal fixing in the side of the tank and is operated by two permanent magnets, one fitted to the switch arm within the waterway and the other fitted to the contact head external to the tank. When the ball float is in its preset low position the magnet inside the tank repels the magnet outside causing it to change its position and make its contacts and start the pump. When the ball float rises with the water the position of these magnets is reversed causing the contacts to break. Although mention has been made of utilizing float switches to fill high level tanks they can, of course, be operated in reverse and used for starting pumps to empty sumps, etc.

(*b*) *Electrode Controls*

This form of level control has no moving parts and relies for its operation on the electrical conductivity of water or other liquid. Metallic tubes or rods are suspended into the tank, a long one being set for the level at which the pump is to cut in, and a shorter one at the level at which the pump is to cut out (in the case of tank filling operation). It is also necessary to have an earth return from the tank which can take the form of a direct connection where this is metal. In the case of a non-metallic tank an additional long electrode is required. While the water is in contact with the longer (cut in) electrode and the side of the tank (or earth return electrode) a very small current flows through the water across these low voltage electrodes. When the water falls below the cut in electrode this circuit is broken and the associated relay will close its contacts and cause the pump to start. The pump will then be retained in operation, an additional relay ensuring that this continues until the water level rises to the shortest (cut-out) electrode. Water containing impurities is a better conductor than pure water, e.g. demineralised water or condensate, and where the latter are to be pumped special variations of electrode relays are required. Reverse operation can be obtained from electrode controls to empty sumps.

*Pressure Switches*

It is not often possible to use a pressure switch alone to control a pump. Since water is almost incompressible there is a danger of rapid on/off operation of the pressure switch damaging the contacts of the starter and electric motor. This difficulty is overcome by introducing a volume of air, which is an easily compressed medium, the air forming the upper part of a pressure vessel connected to the pipework. This arrangement is often referred to as a pneumatic system. As water is drained off the air expands and its pressure falls, eventually starting the pump at a predetermined minimum pressure. The pump will then satisfy the demand taking place and also gradually increase the air pressure in the vessel until the pressure switch reaches its predetermined limit when it will stop the pump. Since some of the air will become absorbed into the water during the course of time a float switch is normally fitted in the vessel and used to operate an air compressor, thus restoring the air volume. Whereas with level controls it is necessary to provide wiring from the top of the building down to the pumps, this is not necessary with the pneumatic system since all the electrical equipment is confined to the pump room.

*Flow Switches*

In some water service systems the static head available is insufficient to provide a satisfactory draw-off rate at some points of use. To overcome this, sensitive flow switches are used which react to the initial flow taking place when a draw-off is first opened and start a pump to provide added pressure. For small bore applications (up to 50 mm) the switch will require to respond to flows of the order of 0·04 litres/second with correspondingly higher flows for larger diameter applications. When demand stops, the flow rate will drop below the sensitivity of the switch, break the electrical contact and stop the pump.

## COMPRESSED AIR PLANT

The major central plant components for a compressed air system, and their salient feature are summarised in the following notes. Pipework distribution systems for compressed air are dealt with in Section B6.

### Compressors

A wide range of compressor types are available. These may be divided into the following general classes according to their characteristic constructional features:

(*i*) *Reciprocating* – cylinder and piston assemblies with simple, duplex and radial drive arrangements; cylinders may be cross or tandem coupled for two stage compression with special arrangements for multi-stage compression.

(*ii*) *Diaphragm* – flexible diaphragm which reciprocates; single or multi-stage, mechanically or hydraulically operated.

(*iii*) *Rotary* – one or more parts rotate in a closely fitting circular or part circular casing, air being entrained and compressed by the resulting movement; these can be divided into five types:

(*a*) sliding vane – in which longitudinal vanes slide radially in a rotor mounted eccentrically in a casing.

(*b*) roots – in which two or more rotors, each having two or three lobes, revolve and intermesh with one another in a casing.

(*c*) screw – in which compression is obtained by the rotation of a single screw rotor or by the intermeshing of two screw rotors in a casing.

(*d*) liquid sealed – in which a liquid displaces air within a rotating element.

(*e*) aerodynamic – in which air is delivered centrifugally or axially, under pressure, by rotating vaned impellers.

Regulation of compressor output may be by the following methods:

(*i*) stop and start control

(*ii*) variable speed control

(*iii*) constant speed operation with unloading of the compressor (by throttling the suction inlet).

Indicative duty ranges for the above compressor types are shown in Table B16.26.

High pressure applications are normally served by reciprocating compressors. Where high output duties are required, two stage generation using different types of compressor may be used (e.g. a rotary low-pressure compressor unit with a reciprocating compressor as the second stage).

Most compressor types can be adapted to provide "oil free" operation (i.e. where little or no oil is discharged into the delivery air stream).

### Filtration

Suspended particulate matter in atmospheric air can cause abrasion damage within air compressors. Additionally, impurities in the compressed air may be undesirable. For this reason, intake air is filtered, the following filter types being typical:

(*i*) dry type – using fabrics such as glass wool, foam plastics, brushes etc.

(*ii*) viscous impingement type – using perforated metal or packed wire-mesh with viscous coatings.

(*iii*) oil bath type – where the intake air flows through a trough of oil and then through a wire-mesh screen or baffles.

(*iv*) self-cleaning type – using an endless travelling curtain on a continuous drive which passes through an oil bath, thus removing accumulated particles and re-coating the curtain with clean oil; provision is made for the periodic removal of sludge from the oil bath.

Frequent cleaning of filters is a vital requirement for air compressor operation. Filtration of compressed air delivery is necessary and may also be required at the point of use (see Section B6).

### Cooling the Air and Removing Moisture

Depending on their size, air compressors may be air or water cooled.

The moisture carrying capacity of air falls as the pressure is increased and as the temperature is decreased. Between compression stages, an intercooler may be used to reduce the air temperature and improve compression.

Excess moisture can be removed from compressed air by cooling it at the compressor delivery using an after cooler.

While mains water may be used for these cooling applications, running costs are likely to favour the use of closed cycle cooling using a fan-cooled radiator or evaporative cooling (e.g. a cooling tower) for heat rejection. Airtraps, protected by a strainer, would be necessary to remove water automatically from inter-coolers and aftercoolers.

To meet high standard requirements for dry air (e.g. hospital applications) the use of adsorption dryers will be necessary. These must be protected from contamination by an oil pre-filter.

### Compressed Air Storage

Most compressed air systems are provided with an air receiver, sized to store a volume of air appropriate to the consumption rate for the application. In the absence of details of consumptions, an empirical rule is to provide a receiver capacity equivalent to the compressor output for one minute. Such sizing would only be suitable for uniform demand. To meet peak consumption rates at least 3 times this storage capacity should be provided.

Receivers should be sited where the ambient temperature is low, to ensure that the stored air cools down thus promoting the condensation of moisture. Entrained oil will also tend to settle-out in the receiver.

Automatic drainage of water from an air receiver using an air trap, protected by a strainer, is an essential requirement. Where oil contamination in a receiver is likely, the drain outlet pipe should rise a short distance to the trap and a manual drain cock should be provided in the receiver just above the trap level. This will permit periodic manual draining-off of oil and scum and prevent the trap from becoming fouled.

**Table B16.26.** Indicative duties of air compressor types.

| Type | Maximum gauge pressure (bar) | Speed (rev/s) | Duty (litre/s) | Remarks |
|---|---|---|---|---|
| Reciprocating | | | | Control by cylinder unloading, suction throttling, varying the clearance volume using clearance pockets and adjustable compression stroke by valve control |
| Single stage | 10 | 25 | — | |
| Multi stage | 700 | — | — | |
| Diaphragm | | | | Single or multi-stage, inherently oil free |
| Mechanically actuated | 3 to 7 | — | 3 to 12 | |
| Hydraulically actuated | 12 to 2000 | — | 0·2 to 30 | |
| Rotary | | | | |
| Sliding vane | | | | |
| single stage | 3 | 5 to 60 | 25000 | |
| two stage | 10 | 5 to 60 | 25000 | |
| oil cooled | 8 | 50 | 25000 | Higher efficiency than dry type |
| Lobed rotors | | | | |
| single stage | 1 | 4 to 100 | 25000 | |
| two stage | 2 | 4 to 100 | 25000 | |
| Screw rotors | | | | |
| single stage | 4 | 50 to 400 | 25000 | |
| two stage | 10 | 50 to 400 | 25000 | |
| Liquid ring | | | | |
| single stage | 2 | 60 | 25000 | Low volumetric efficiency |
| two stage | 3 | 60 | 25000 | |
| Aerodynamic | | | | Control by speed variation or by throttling the discharge |
| centrifugal | 0·4 to 6 | up to 400 | 60000 | |
| axial | — | up to 400 | High | |

# SECTION B18 OWNING AND OPERATING COSTS




*This edition of Section B18 first published: 1986*

# SECTION B18 OWNING AND OPERATING COSTS

## INTRODUCTION

The selection of building services engineering systems, whether to purchase or to hire, will generally involve many options. The evaluation of these options to find the most economic and suitable choice requires the consideration of capital and operational costs which are likely to be incurred during the life of the plant, and the operational requirements such as safety, reliability, maintainability etc. This Section provides advice on the identification and assessment of costs and other economic and operational factors significant to the selection. Some statistical cost data on plant and building operation and maintenance is also provided.

A further choice faced by the engineer where there is an existing plant or an installation approaching the end of its economic life, is to decide whether it is better to continue to maintain the existing plant or to replace it. Essential factors that should be considered in such situations are also identified in this Section.

## ECONOMIC EVALUATION

There are many ways of evaluating engineering systems. The parameters most commonly assessed are:

(*a*) Capital costs.
(*b*) Costs in use.
(*c*) Investment analysis (e.g. pay back period).
(*d*) Total life cycle costs.

### Capital Costs

This is a simple method of cost comparison which, while appropriate for some investments, has serious limitations. The decision to select a particular configuration of plant is based purely on the capital expenditure and should only be taken when the investor is limited in the capital sum he can raise and consciously ignores operational costs and other behavioural aspects of the plant. The limitation of the approach is that plant which has high capital cost elements may result in long term operational benefits over the lifetime of the plant (e.g. higher reliability or lower running costs); a choice based on capital costs alone will disregard these potential benefits.

The approach to consider capital costs only has been common in speculative building projects where the developer was not responsible for maintenance and operating costs. However, over the last few years energy and operational costs of buildings have significantly increased and these have become an important parameter in the selection of a building by a prospective buyer or tenant.

It must be noted that the capital cost of an installation may comprise many components. Some of these are:

(*a*) Feasibility study, design and contracting costs.
(*b*) Capital cost of plant and equipment.
(*c*) Site acquisition (purchase or rent of land, plant room space etc.).
(*d*) Other services required (e.g. gas, electricity mains etc.).

Different designs and plant configurations generally yield different cost combinations of the above factors. For example the plant room space required depends on the number of units installed to provide the overall plant capacity, and the cost of a central plant room can vary with these different options (see Section A9 for methods of sizing heating and cooling plant).

A checklist of some of the many factors affecting capital and operating costs is given in Table B18.1.

### Costs in Use

Some organisations use the operating costs, or costs in use, as the deciding factor for a suitable installation option. This may become necessary where there are no capital restrictions but there is an allocation for revenue costs.

A checklist of the significant items requiring consideration in the assessment of costs in use is given in Table B18.1.

### Total Life Cycle Costs

Total life cycle costs are the total costs of both acquiring and using a physical asset such as a heating installation or electrical distribution system. They include the capital, or first cost of designing, purchasing, installing and commissioning; the operational cost during the life of the asset for energy, labour, and operational materials; the maintenance costs associated with the asset principally comprising labour, materials and spares storage; the failure cost of breakdowns and providing alternative facilities and the disposal cost when the asset reaches the end of its life.

Unfortunately, little guidance exists at present on which to make accurate assessments of these elements, many of which may change during the lifetime of the asset. To use total life cycle costing to compare alternatives, cost forecasts must be made in terms of current monetary values.

Typical applications for total life cycle costs are where:

(*a*) Energy costs of an asset are likely to be high throughout its life.
(*b*) An asset has a long life.
(*c*) Investment costs are high.
(*d*) Significant savings can be made by reducing maintenance or operational costs.

**Table B18.1.** Capital costs and costs in use – significant factors requiring consideration.

| Item | Influencing factors |
|---|---|
| CAPITAL COSTS | |
| Feasibility, design and contracting costs | Complexity and size of installation, level of design detail, total capital costs. |
| Plant, equipment and construction costs | Design options, plant configurations, installation complexity, site location, safety requirements, fire requirements, need for standby etc. |
| Site acquisition costs | Space requirement (plant room, service, storage, accommodation etc.), rent or purchase, local restrictions (e.g. smoke emissions), fuel storage. |
| Other service costs | Supply authority contributions, capacity of existing service mains, ducts and trenches etc. for mains. |
| Builders' work costs | Design details, type of plant and equipment, method of construction. |
| COSTS IN USE | |
| Operational costs | Method of plant operation (automatic, manual) and attendance, level of measurements, readings and logging. |
| Preventive maintenance costs | Level of preventive maintenance selected, method and level of inspections and logging, complexity of plant, statutory and mandatory requirements, level of reliability expected, requirement for specialist labour. |
| Corrective maintenance costs | Plant breakdown and tolerable downtime, requirement for specialist labour. |
| Stores for spares | Degree of plant uniformity, tolerable downtime, local availability of spares. |
| Overheads | Maintenance management and administration, vehicles, building management systems etc. |
| Energy costs | (Methods of estimating fuel costs are given later). |
| Insurance | |

Plant maintenance costs have two components: preventive and corrective. Preventive maintenance, also known as planned maintenance, is carried out to reduce the incidence of breakdown by careful inspection, adjustment and service of the plant.

Planned maintenance should reduce the need for corrective maintenance. However, a careful balance between the corrective and preventive elements must be established in order to ensure that the cost of planned maintenance does not exceed the benefits derived from it. Fig. B18.1 shows a typical example of corrective, preventive and total maintenance cost relationships.

It must be understood that economics are not the sole criterion for the level of maintenance. It is necessary to carry out a basic level of preventive maintenance to sustain essential plant operations, safety and to conform to statutory requirements.

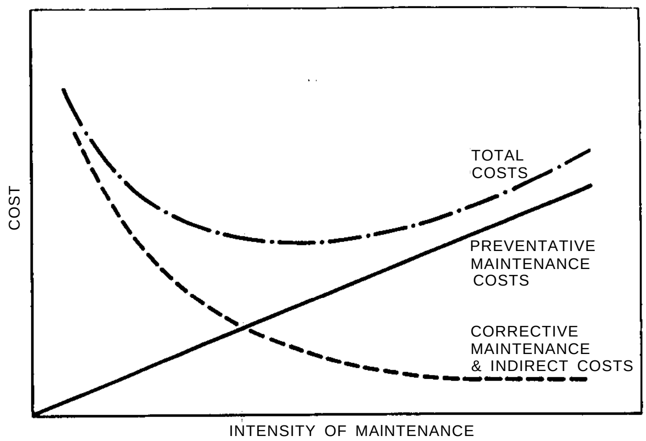



**Fig. B18.1.** Costs of plant operation related to maintenance.

## OTHER FACTORS

Other factors which should play a significant role in the decision making process for the determination of plant and its configuration include:

(*a*) Reliability.
(*b*) Maintainability.
(*c*) Safety.

These aspects of plant operation often involve subjective assessments which are difficult to quantify in terms of life cycle costs. The engineer is, therefore, faced with a decision which is a compromise between costs and other behavioural aspects.

### Reliability

Selecting an installation on a financial basis alone will not be satisfactory. The engineer will have to ensure that it will operate to satisfy the required performance criteria, including reliability. Techniques are available to evaluate the reliability that can be expected from plant and some guidance is available in Section A9.

Plant reliability can be maintained or improved without significant addition to the costs by simple qualitative considerations such as:

(*a*) Plant and equipment standardisation and stock of common spares.
(*b*) Provision of standby equipment commensurate with the downtime that can be tolerated.
(*c*) Concentrating maintenance on critical areas of the installation.

### Maintainability

Plant and equipment, however reliable, are bound to fail at some time. The plant selected, therefore, should be easily maintainable to avoid prolonged disruptions to the service it provides. Key factors that should be considered in the assessment of the plant include:

(*a*) Economic life of the plant (see Table B18.2).
(*b*) Availability and cost of spare parts.
(*c*) Requirement for and availability of specialist labour.
(*d*) Complexity of plant.
(*e*) Mean time between failures.

(*f*) Typical downtime and acceptable interruptions of service.
(*g*) Quality of maintenance manuals.
(*h*) Ease of maintenance.

**Safety**

Plant must be designed and maintained to a safety standard which at least satisfies statutory requirements. Further considerations may be required to satisfy the standards of a particular organisation.

## INVESTMENT APPRAISAL

In general the object of investment is to use resources in the most profitable way. Investment decisions, therefore, are based on a comparison of the costs incurred and the potential benefits. With investments in construction, benefits may not start to accrue until some years after the first investment.

A proper comparison of investment costs and potential benefits must take account of the time lapse. To do this, it is necessary to allow for the changes in the value of money with time. This entails not only a consideration of current interest rates and inflation levels but forecasts of these into the future. The basic aim of the comparison is to evaluate all investments and potential benefits against a common base in terms of value for money.

This technique is known as investment appraisal and a number of standard methods have been developed:

(*a*) Simple payback period.
(*b*) Net present value.
(*c*) Payback period.
(*d*) Index of profitability.

### Simple Payback Period

This is a form of investment appraisal mainly used where a scheme replacing an existing installation has to be justified by the savings generated by the replacement scheme. The period during which the capital cost of the investment is repaid by the savings is known as the payback period. Where factors affecting the future costs and benefits of the investment are likely to change rapidly it is preferable to select a short payback period to minimise the risk on return of the investment.

There are two methods of evaluating the payback period; one is known as the simple payback, the other discounted payback. In the former method the payback period is determined by dividing the capital cost by the annual savings generated by the replacement scheme. The following example shows a simple payback calculation.

*Example*

It is intended to expend £1,000 on energy conservation measures. The saving in energy is expected to be £333 per year from this expenditure. Therefore, the payback period is 1,000/333 = 3 years.

### Net Present Value

The net present value is the representation of future costs and benefits as a present day cost or benefit. In

**Table B18.2.** Life factors.

| Type | Item | Typical economic life (years) |
|---|---|---|
| Steam and HTHW boiler plant | Shell and tube boilers | 15 to 25 |
| | Water tube boilers | 25 to 30 |
| Medium and low pressure boiler plant | Steel boilers | 15 to 20 |
| | Sectional cast-iron boilers | 15 to 25 |
| | Electrode boilers | 30 to 40 |
| Boiler plant auxiliaries | Combustion controls | 15 to 20 |
| | Boiler electrodes | 5 to 10 |
| | Feed pumps | 15 to 20 |
| | Feed treatment plant | 15 to 20 |
| | Firing equipment | 15 to 20 |
| | Fuel handling plant (solid) | 10 to 15 |
| | Fans | 15 to 20 |
| | Instrumentation | 10 to 20 |
| Steel chimneys | | 8 to 15 |
| Heating installations | Calorifiers and heat exchangers | 20 to 25 |
| | Control equipment | 15 to 20 |
| | Pipework installations | 25 to 30 |
| | Pumps | 20 to 25 |
| | Radiators, cast iron | 20 to 25 |
| | Radiators, steel | 15 to 20 |
| | Suspended ceiling heating | 15 to 20 |
| | Tanks (depends on material and location) | 15 to 30 |
| | Valves | 20 to 25 |
| | Incinerators | 15 to 20 |
| Ventilation and air conditioning installations | Refrigeration plant, medium and large, compression and absorption | 15 to 20 |
| | Distribution systems | 25 to 30 |
| | Terminal units | 15 to 25 |
| | Cooler batteries | 15 to 20 |
| | Fans and washers | 15 to 20 |
| | Heater batteries | 15 to 20 |
| | Cooling towers (depending on materials) | 10 to 25 |
| | Package air handling units (under 35 kW) | 8 to 15 |
| Laundries | Laundry plant (washing machines) | 15 to 20 |
| Kitchens | Cooking equipment | 15 to 20 |
| Electrical distribution network | Mains cables | 25 to 30 |
| | Switchgear and distribution equipment | 25 to 30 |
| Electrical Installations in buildings | Final circuits and outlets | 20 to 25 |
| | Lighting installations | 20 to 25 |
| | Electric motors, etc. | 20 to 25 |
| Electrical generating plant | Generators | 25 to 30 |
| | Prime movers, diesel (continuously rated) | 15 to 20 |
| | Prime movers, steam (continuously rated) | 25 to 30 |
| | Standby prime movers, diesel | 25 to 30 |
| Miscellaneous electrical equipment | Batteries, lead acid, sealed | Up to 10 (static) |
| | Batteries, nickel alkaline, vented | 20 to 25 |
| | Call systems | 20 to 25 |
| | Clock systems | 20 to 25 |
| | Fire alarm systems | 20 to 25 |
| | Telephone systems | 20 to 25 |
| | Automatic temperature controls (including pneumatic) | 15 to 20 |

*Note:* The above life estimates are based on good standards of plant maintenance.

order to do this it is necessary to discount all future costs and benefits by the minimum rate of return expected from the investment. This rate of return is also referred to as the discount rate.

Consider an investment of £100 at a real rate of return (inflation adjusted) of 7%.

After the first year the sum would become £100 + (£100 × $^{7}/_{100}$) = £107.

After the second year the sum would become £107 + (£107 × $^{7}/_{100}$) or (£100 + (£100 × $^{7}/_{100}$)) + ((£100 + (£100 × $^{7}/_{100}$)) × $^{7}/_{100}$) = £114.49.

Alternatively we can say that:

£107 in 1 year is worth £100 today if discounted at 7% and £114.49 in 2 years is worth £100 today if discounted at 7%

The effects of inflation can be allowed for by considering the real costs (inflation adjusted) in these calculations.

Tables are available giving factors for various discount values for single years, and compounded for multiple years when they are known as the cumulative discount factor.

Two factors need to be known to calculate the cost in present terms of future single capital payments, *f*, or future regular payments, *c*.

The factor, *f*, for single year discounting is obtained from:

$$f = \frac{1}{(1 + r)^n} \quad \text{B18.1}$$

The cumulative discount factor *c* over *n* years is given by:

$$c = \frac{(1 - (1 + r)^{-n})}{r} \quad \text{B18.2}$$

where:

$r$ = discount rate (as a fraction)
$n$ = number of years

*Example*

An opportunity has arisen for the acquisition of new premises. It is necessary to determine whether it would be more advantageous to continue with the existing building and oil-fired boiler plant or move into the new premises with the additional expenditure involved in a new installation with gas-fired boiler plant. The costs involved are given in Table B18.3.

It is considered that the discount rate will be 5%. The respective calculations would be as follows:

For 20 years ($n$ = 20, $r$ = 0·05), from Equation B18.2:

$c = (1 - (1 + 0{\cdot}05)^{-20})/0{\cdot}05$
$c = 12{\cdot}4622$

This gives the factor *c* that items of annual expenditure need to be multiplied by to give the expenditure in terms of present value.

**Table B18.3.** Example costs for existing and proposed scheme.

| | Existing installation | New installation |
|---|---|---|
| BUILDING | – | £150 000 |
| Land cost | £50/m$^2$ year | £50/m$^2$ year |
| Land area | 500 m$^2$ | 500 m$^2$ |
| Building repairs | £5 000/year | – |
| INSTALLATION | | £300 000 |
| Fuel | £150 000/year | £60 000/year |
| Maintenance and operating | £20 000/year | £14 000/year |
| REPAIRS | | |
| 3 year | £15 000 | |
| 5 year | – | £2 000 |
| 8 year | £20 000 | – |
| 10 year | – | £4 000 |
| 12 year | £30 000 | – |
| 15 year | – | – |
| 16 year | £50 000 | £8 000 |
| Mains | – | £10 000 |
| Life span | 20 years | 20 years |

The cost of repairs at set intervals are single items of expenditure. The factor *f*, therefore, has to be calculated at these intervals.

For 3 years ($n$ = 3, $r$ = 0·05), from Equation B18.1:

$f = 1/(1 + 0{\cdot}05)^3$
$f = 0{\cdot}8638$

Similarly,

$n$ = 5, $f$ = 0·7835
$n$ = 8, $f$ = 0·6768
$n$ = 10, $f$ = 0·6139
$n$ = 12, $f$ = 0·5568
$n$ = 15, $f$ = 0·4810
$n$ = 16, $f$ = 0·4581

The present values of each component of both options are listed in Tables B18.4 and 5.

**Table B18.4.** Example net present value calculations for existing building.

| Existing building item description | Expenditure | *c* or *f* | Present value |
|---|---|---|---|
| Capital cost | – | – | – |
| Energy costs | £150 000/year | 12·4622 | £1 869 330 |
| Maintenance and operating costs | £20 000/year | 12·4622 | £249 244 |
| Repair costs: | | | |
| 3 year | £15 000 | 0·8638 | £12 957 |
| 8 year | £20 000 | 0·6768 | £13 536 |
| 12 year | £30 000 | 0·5568 | £16 704 |
| 16 year | £50 000 | 0·4581 | £22 905 |
| Other costs: | | | |
| Building repairs | £5 000/year | 12·4622 | £62 311 |
| Land charge | £25 000/year | 12·4622 | £311 555 |
| | | Net present value | £2 558 542 |

**Table B18.5.** Example net present value calculations for proposed building.

| New building item description | Expenditure | c or f | Present value |
|---|---|---|---|
| Capital cost (including costs for design etc.) | | | |
| plant | £300 000 | 1·0 | £300 000 |
| building | £150 000 | 1·0 | £150 000 |
| gas mains | £10 000 | 1·0 | £10 000 |
| Energy costs | £60 000/year | 12·4622 | £747 732 |
| Maintenance and operating costs | £14 000/year | 12·4622 | £174 471 |
| Repair costs: | | | |
| 5 year | £2 000 | 0·7835 | £1 567 |
| 10 year | £4 000 | 0·6139 | £2 456 |
| 15 year | £8 000 | 0·4810 | £3 848 |
| Other costs: | | | |
| Land charge | £25 000/year | 12·4622 | £311 555 |
| | | Net present value | £1 701 629 |

From the above calculations it can be seen that although there is an immediate capital outlay of £460 000 if moving to new premises, it would still be more economical over the life of the plant.

It should be borne in mind that in the comparison of different options, items of expenditure which are common to all schemes may be omitted from the calculation, e.g. land charges in the above case.

A full method for calculating energy investments by the discounted cash flow method is contained in the CIBSE Algorithm 1 – *Appraising Energy Conservation Investments using Discounted Cash Flow Techniques.*

*Taxable Allowances*

Where taxable allowances are provided for capital expenditure and/or depreciation then the appropriate adjustment must be made to all the calculations shown previously.

**Payback Period**

A previous example showed that an investment of £1 000 on energy conservation which yielded an annual energy saving of £333, gave a simple payback period of 3 years.

However, using net present value methods, at a discount rate of 5% the net present value of the savings will be:

| | | Cumulative total |
|---|---|---|
| Year 1 | £333 × 0·9524 = £317.15 | £317.15 |
| Year 2 | £333 × 0·9070 = £302.03 | £619.18 |
| Year 3 | £333 × 0·8638 = £287.65 | £906.83 |
| Year 4 | £333 × 0·8227 = £273.96 | £1180.79 |

It should be noted that the discounted payback period will always be greater than the simple payback period; in this case between 3 and 4 years.

**Index of Profitability**

Where funds are available the scheme with the lowest net present value (*NPV*) would normally be chosen. However, where there are a number of mutually exclusive proposals the scheme with the lowest *NPV* may require a higher capital investment which could restrain resources. A method of ranking the schemes to identify the most profitable ones (i.e. those providing the highest profit per pound spent), is needed. A ranking technique called the index of profitability, *IOP*, lists all schemes in ascending order of capital cost. The index of profitability for each scheme is determined by comparing each scheme with the scheme having the lowest capital cost, which is designated scheme 1:

$$IOP(n) = \frac{NPV(1) - NPV(n)}{CC(n) - CC(1)} \quad .. \quad .. \quad .. \quad \text{B18.3}$$

where:

$IOP(n)$ = index of profitability for scheme $n$

$NPV(n)$ = net present value for scheme $n$

$CC(n)$ = capital cost for scheme $n$

The scheme with the highest IOP is the preferred scheme under this method.

An example containing 5 hypothetical schemes is shown in Table B18.6.

**Table B18.6.** Example of index of profitability.

| Scheme | Capital cost | *NPV* | *IOP* | Comparison |
|---|---|---|---|---|
| 1 | 0 | £1 700 000 | – | – |
| 2 | £ 50 000 | £1 050 000 | 13 | 2 to 1 |
| 3 | £150 000 | £1 000 000 | 4·1 | 3 to 1 |
| 4 | £275 000 | £1 069 500 | 2·3 | 4 to 1 |
| 5 | £300 000 | £1 250 000 | 1·5 | 5 to 1 |

**Repair or Replace**

When plant is approaching the end of its economic life (see Table B18.2) it is necessary to consider whether to maintain the existing plant or to replace it. The factors which require consideration are:

(*a*) Cost of replacement.
(*b*) Costs in use (maintenance, energy, cost of breakdown etc.) of both new and existing plant.
(*c*) Value placed on reliability and safety.

The criterion to be followed in making the decision is to spread the capital expenditure of replacement plant over its life period as a discounted annual value. If the annual equivalent of the capital cost of the new plant and its annual cost in use is less than the annual cost in use of the existing plant then it will be cost effective to replace. The limitation of this analysis is that it ignores any significant future maintenance costs of the new plant. Nevertheless it provides a simple and easy method of comparison.

## ESTIMATION OF ENERGY CONSUMPTION FOR BUILDING SERVICES

Building services energy consumption accounts for a very significant proportion of the operating costs of most premises. With the likelihood of progressive increases in the real cost of energy, this significance will become even greater. Because of the importance of this factor the provision of an indicative estimate of annual energy costs to operate the services of a building should form an essential part of the design information provided for the client.

Engineers should recognise the complexity of the calculations involved and qualify their predictions accordingly. Generally it will be necessary to make assumptions, and these should be stated clearly.

Above all, it is necessary to bear in mind the influence on energy costs exerted by building occupants. Although a building may be designed to function at a conservative temperature with modest, natural ventilation the occupants may operate the building on an entirely different basis. Frugal operation of a building can result in complaints while profligate operation may never be noticed without monitoring. For these reasons, operational experience of the actual building concerned is invaluable in refining the prediction of energy consumption.

Techniques for estimating building energy usage are also shown in CIBSE Building Energy Code Part 2, *Calculation of Energy Demands and Targets for the Design of New Buildings and Services.*

**Space Heating**

A wide variety of techniques are available for the estimation of energy requirements for space heating. These range from relatively simple manual methods to very large computer program packages based on complex data banks.

Different methods produce different results. Also, different operatives using the same method can produce wide discrepancies.

All methods attempt to take account in some way of weather variations, the thermal properties and response behaviour of the building and the environmental system and the hours of use of the accommodation.

Computer programs for energy estimating are available from various commercial sources with comprehensive user manuals to give guidance on application. Additionally, computer based energy estimating techniques are made available by the public utilities, both gas and electricity, to assist their prospective customers to estimate their requirements.

Comparisons between typical program estimates and actual recorded consumption data for space heating suggest that errors may not exceed ±25% for traditional buildings on which most experience is based. It is necessary for the designer to assess merits of these programs and decide whether they are appropriate to his requirements. Some of the options may incur extensive data preparation and computing time, although the resulting level of accuracy is likely to be better. With well insulated buildings having low fabric heat losses and operated intermittently, the thermal capacity of the building and the environmental system becomes a much more dominant factor in the overall energy demand. Behaviour in this non-steady-state area is more difficult to predict and, hence, larger deviations from the estimate may occur.

*Manual Calculation Method*

The following method for estimating energy consumption for space heating was proposed by Billington[1]. The assessment of the thermal characteristics of the building will significantly influence the outcome of this method.

*Equivalent Hours*

Consider a heating plant whose calculated design-day duty (without margins) is given by:

$$Q_d = HD\ t_d \qquad \text{B18.4}$$

where:

$Q_d$ = calculated duty .. .. .. .. W
$H$ = load per K .. .. .. .. .. W/K
$Dt_d$ = design temperature difference .. .. K

Taking $D\,t_a$ as a mean seasonal inside to outside temperature difference, and the plant running continuously for the season at less than full load, then ignoring other energy gains, the fuel consumption will be:

$$F = 24\,N\,H\,D\,t_a\,K \qquad \text{B18.5}$$

where:

$F$ = fuel consumption .. .. .. .. kg
$N$ = period of continuous running .. .. days
$K$ = a factor .. .. .. .. .. kg/W

$$K = \frac{3{\cdot}6}{h\,CV} \qquad \text{B18.6}$$

where:

$h$ = plant seasonal efficiency
$CV$ = calorific value of fuel .. .. .. kJ/kg

and:

$$D\ t_a = t_i - t_o \qquad \text{B18.7}$$

where:

$t_i$ = indoor temperature .. .. .. .. °C
$t_o$ = average seasonal outdoor temperature .. .. .. .. °C

An ‘equivalent hours of operation’, *E*, can be defined where the plant operates at full load, but the fuel consumption, *F* is the same.

Hence, equation B18.5 can be written in terms of equivalent hours and the design temperature difference, thus:

$$F = E\,H\,D\,t_d\,K \qquad \text{B18.8}$$

Equating equations B18.5 and B18.8:

$$E = \frac{24\ N\ D\ t_a}{D\ t_d} \quad .. \quad .. \quad .. \quad .. \quad .. \quad \text{B18.9}$$

$E$ may be regarded as the 'equivalent operation load'. This can be written:

$$E = \frac{24\ D_o}{D\ t_d} \quad .. \quad .. \quad .. \quad .. \quad .. \quad \text{B18.10}$$

where:

$D_o$ = seasonal total of degree-days to the base $t_i$

Hence, from equations B18.4 and B18.8:

$$F = E\ K\ Q_d \quad .. \quad .. \quad .. \quad .. \quad .. \quad \text{B18.11}$$

*Internal Heat Gains*

For every building there are sources of heat gain other than the heating system (occupants, lighting, sunshine and equipment) and these represent a significant proportion of the heat input necessary to maintain comfort temperatures.

The average seasonal heat requirement is, therefore:

$$Q_s = H(t_i - t_o) - G \quad .. \quad .. \quad .. \quad .. \quad .. \quad \text{B18.12}$$

where:

$Q_s$ = seasonal heat requirement .. .. W
$G$ = miscellaneous heat gains .. .. W

The net annual fuel consumption is

$$F_a = 24\ N\ [H\ (t_i - t_o) - G]\ K \quad .. \quad .. \quad .. \quad \text{B18.13}$$
$$d = 24\ H\ N\ (t_i - t_o - d)\ K \quad .. \quad .. \quad .. \quad \text{B18.14}$$

where:

$F_a$ = the net annual fuel consumption .. kg
$d$ = the average temperature rise which can be maintained by the miscellaneous heat gains alone

= $G/H$ .. .. .. .. .. K

The temperature ($t_i$ - $d$) is also that outdoor temperature above which no heating is required. Therefore $N\ (t_i - t_o - d) = D_d$ is the annual total of degree-days to the base ($t_i$ - $d$). It is important to note that $D_d$ is mainly a climatic factor which depends on the base temperature selected. The latter is dependent on the design indoor temperature, the amount of energy released internally, $G$, and the rate of heat loss from the building, $H$.

The equivalent hours of operation at full load of the heating plant now become:

$$E_a = \frac{F_a}{H\ D\ t_d\ K} \quad .. \quad .. \quad .. \quad .. \quad .. \quad \text{B18.15}$$

$$= \frac{24\ D_d}{D\ t_d} \quad .. \quad .. \quad .. \quad .. \quad .. \quad \text{B18.16}$$

where:

$E_q$ = equivalent full-load annual-operation .. .. .. .. .. hours

It can be seen that $E_q$ is a function only of the climate and of the indoor design temperature; therefore, all buildings of similar type at a given location will have the same value of $E_q$.

This will be true irrespective of the method of heating or the fuel used. The fuel consumption will of course depend also on the plant seasonal efficiency. Buildings of different types and usage may well have different internal heat gains and it is necessary therefore to select appropriate values of the degree-day base if this approach is to be used to estimate values for $E_q$. This is best done by considering the magnitudes of $G$ and $d$. Suggested values of $d$ are given in Table B18.7.

**Table B18.7.** Values $d$ (equation B18.14).

| Class of building | Building structure | $d$ (K) |
|---|---|---|
| 1 | Building with large area of external glazing, much internal heat-producing equipment and densely populated. | 5 to 6 |
| 2 | Buildings with one or two of the above factors. | 4 to 5 |
| 3 | Traditional buildings with normal glazing, equipment and occupancy. | 3 to 4 |
| 4 | Sparsely occupied buildings with little or no heat-producing equipment and small glazed area. | 2 to 3 |
| 5 | dwellings. | 5 to 8 |

*Notes:* Unless separately allowed for in the design heat loss add 1 K for single-storey buildings.

*Degree-days*

Degree-days have for many years been used as a means of comparing, over different periods, the variations in load sustained by heating plants in different parts of the country. The standard method is to assess, for monthly periods, the daily difference in K between a base temperature of 15·5 °C and the 24-hour mean outside temperature. The monthly totals may then be used to compare monthly changes in the weather factor, or be added together for the heating season, enabling the severity and duration of the winter to be compared from year to year and from place to place.

Alternative bases of calculation of degree-days have been put forward[1] to take account of the fact that buildings are rarely heated continuously to design temperature. For simplicity it seems preferable to retain the character of the degree-day as a climatic parameter, rather than to extend it to include the effect of building structure, intermittency and so on. Separate arrangements are therefore suggested below to allow for non-continuous operation.

Table B18.8 lists average seasonal totals of degree-days for geographical areas.

**Table B18.8.** Degree-days 1st September to 31st May.

| Degree-day areas | Degree days |
|---|---|
| Thames Valley | 2034 |
| South Eastern | 2275 |
| Southern | 2130 |
| South Western | 1840 |
| Severn Valley | 2109 |
| Midlands | 2357 |
| West Pennines | 2233 |
| North Western | 2355 |
| Borders | 2464 |
| North Eastern | 2354 |
| East Pennines | 2243 |
| East Anglia | 2304 |
| West Scotland | 2399 |
| East Scotland | 2496 |
| North East Scotland | 2617 |
| Wales | 2094 |
| Northern Ireland | 2330 |

Averaged over a 20 year period to 1979 for a base outside temperature of 15·5 °C.

Degree-day data are published regularly by the Department of Energy. Separate figures are published for every month of the year and, in addition to the 20 year average data, shorter term current values are given to permit comparison of weather trends. In addition, the Department publishes 'Fuel Efficiency Booklet No. 7' describing the derivation and recommended usage of degree-days.

In order to evaluate the effects of $d$ it is convenient to quote the degree-day totals for a series of base temperatures. Table B18.9 gives ratios of $D_d/D_{15·5}$ for the average degree-day total. From these, the value of $D_d$ can be found for a specified indoor temperature in a given class of building.

**Table B18.9.** Ratio $D_d/D_{15·5}$

| Base Temperature °C | $D_d/D_{15·5}$ |
|---|---|
| 10 | 0·33 |
| 12 | 0·57 |
| 14 | 0·82 |
| 15 | 0·94 |
| 15·5 | 1·00 |
| 16 | 1·06 |
| 17 | 1·18 |
| 18 | 1·30 |

*Example*

Find $D_d$ for a continuously heated Class 2 building in the Thames Valley held at 20°C. The base temperature is 15°C to 16°C and the degree-day totals from Tables B18.8 and 9 are:

For 15°C = 2034 × 0·94 = 1912
For 16°C = 2034 × 1·06 = 2156

Using the above example and assuming an outdoor design temperature of -3°C, the equivalent full load hours per annum can be found. Thus taking the value for 16°C, from equation B18.16:

$$E_q = \frac{24 \times 2156}{20 + 3} = 2250 \text{ hours}$$

Values can be calculated along these lines for each building class and for varying base temperatures.

*Intermittent Operation*

It is stressed that the terms 'light', 'medium' and 'heavy' refer to the thermal capacity of the building fabric. Furthermore, it should be noted that although constructions may be similar, the location of thermal insulation in the fabric can influence strongly its thermal capacity. The assessment involves judgement as well as calculation and it may be appropriate to take account of building contents as well as the fabric in some cases. Many buildings will fall into the middle ranges, classes B and C of Table B.18.10, only very light shed-type structures and massive constructions belonging to either extreme.

**Table B18.10.** Classification of structures by thermal inertia.

| Class of building | $f_r = \frac{\Sigma (AY)+nv/3}{\Sigma (AU)+nv/3}$ | Building structure |
|---|---|---|
| A (very heavy) | Over 6·5 | Buildings of curtain walling, masonry or concrete, especially multi-storey, much subdivided within by solid partitions. |
| B (heavy) | 6·0 | Buildings with large window areas but appreciable area of solid partitions and floors. |
| C (medium) | 4·0 | Single-storey buildings of masonry or concrete with solid partitions; buildings, especially curtain walling with small areas of solid partition. |
| D (light) | 2·5 | Single-storey buildings of factory type with little or no solid partitions; top floors of multi-storey buildings when undivided. |

Where:

$f_r$ = building thermal response factor
$A$ = surface area of building element .. .. .. $m^2$
$Y$ = thermal admittance for the surface of a building element .. .. .. .. $W/m^2K$
$N$ = natural ventilation rate .. .. .. .. $h^{-1}$
$V$ = volume of treated space .. .. .. .. $m^3$
$U$ = thermal transmittance of a building element .. .. .. .. .. .. $W/m^2K$

*Occupancy Pattern and Response Factors*

In an HVRA Report[2] the effects of intermittent operation are separated from those of week-end shut-down. From the Report, the following correction factors can be derived for:

(*a*) The length of the working week, see Table B18.11.
(*b*) The response of the building and plant, see Table B18.12.
(*c*) Length of the working day, see Table B18.13.

**Table B18.11.** Correction factor for length of working week.

| Working week | Light building | Heavy building |
|---|---|---|
| 7 day | 1·0 | 1·0 |
| 5 day | 0·75 | 0·85 |

**Table B18.12.** Correction factor for response of building and plant.

| Type of heating | Light building | Medium building | Heavy building |
|---|---|---|---|
| Continuous | 1·0 | 1·0 | 1·0 |
| Intermittent–responsive plant | 0·55 | 0·70 | 0·85 |
| Intermittent–plant with long time lag | 0·70 | 0·85 | 0·95 |

**Table B18.13.** Correction factor for length of working day–intermittent operation only.

| Occupied period (hours) | Light building | Heavy building |
|---|---|---|
| 4 | 0·68 | 0·96 |
| 8 | 1·00 | 1·00 |
| 12 | 1·25 | 1·02 |
| 16 | 1·40 | 1·03 |

*Note:* The above factors apply only to intermittent heating. The stated occupied periods do not include preheating times but no separate allowance is necessary for this.

Some typical examples of the application of these factors are shown in Table B18.14. The wide range of equivalent hours of full load operation shown in this table illustrates the difficulty of providing accurate estimates of energy consumption for space heating. It also helps to explain the significant variations in available published data.

*Example*

Calculated heat loss from building .. .. 1000 kW
Building Class (Tables B18.7 and 10) .. .. 3B
Degree-days, Borders (Table B18.8) .. .. 2464
Indoor design temperature .. .. .. 19°C
Outdoor design temperature .. .. .. –1°C
Usage of heating plant .. .. .. Intermittent
Building occupancy .. 5 days/week, 8 hours/day

*Solution*

Equivalent hours full load operation

Base temperature (Table B18.7)
= 19 – 4 = 15°C

Ratio for 15°C (Table B18.9)
= 0.94

Thus, equivalent full load operation (uncorrected) from equation B18.16:

$$E_q = \frac{24\times 2464 \times 0{\cdot}94}{19 - (-l)} = 2779 \text{ hours}$$

From Tables B18.11 to 13 factors are as follows:

(*a*) 5 day week 0·85
(*b*) Intermittent (heavy building) 0·95
(*c*) 8 hour day 1·00

Thence, equivalent full load operation (corrected):

$$E_q = 2779 \times 0{\cdot}85 \times 0{\cdot}95 \times 1{\cdot}00 = 2244 \text{ hours}$$

*Net Annual Heat Requirement* ($Q_r$)

$$Q_r = \frac{2244 \times 1000 \times 3600}{1\,000\,000} = 8078 \text{ GJ/year}$$

*Annual Fuel Requirement* ($Q_n$)

Class D fuel oil (Table C5.9) *CV* = 45·5 MJ/kg, specific gravity 0·835, seasonal efficiency (Table B18.15) say 65%:

$$Q_n = \frac{8078 \times 1\,000\,000}{45{\cdot}5 \times 1\ 000 \times 0{\cdot}835 \times 0{\cdot}65} = 327\,000 \text{ litre/year}$$

Natural gas (Table C5.12) *CV* = 38·7 MJ/m$^3$, seasonal efficiency (Table 18.15) say 65%:

$$Q_n = \frac{8078 \times 1\,000\,000}{38{\cdot}7 \times 1\ 000 \times 0{\cdot}65} = 321\ 000 \text{ m}^3\text{/year}$$

**Table B18.14.** Equivalent hours of full load operation for Thames Valley area.

| Class of building | Equivalent hours* | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | Continuous heatingo | Continuous 5 days/week | | Intermittent 5 days/week | | | | |
| | | | | 8 hours/dry | | 12 hours/day | | |
| | | Correction factor† | Equivalent hours | Correction factor† | Equivalent hours | Correction factor† | Equivalent hours |
| 1C | 2000 | 0·80 | 1600 | 0·80×0·70×l·00 | 1120 | 0·80×0·70×1·12 | 1250 |
| 2B | 2290 | 0·85 | 1950 | 0·85×0·85×1·00 | 1650 | 0·85×0·85×l·02 | 1690 |
| 3B | 2590 | 0·85 | 2200 | 0·85×0·85×1·00 | 1870 | 0·85×0·85×1·02 | 1910 |
| 4D | 2880 | 0·75 | 2160 | 0·75×0·60×1·00 | 1300 | 0·75×0·60×l·25 | 1620 |
| 5A | 1700 | — | — | — | — | l·00×0·95×l·03 | 1660‡ |

*Notes:*
* Outdoor temperature: –1°C; indoor temperature: 19°C.
o Lower values have been taken for *d* except for building class 5A where 6°C has been assumed.
† Continuous heating values are multiplied by these factors.
‡ Value for 16 hours/day, 7 days/week.

Direct electricity, non-storage in ceiling seasonal efficiency (Table B18.15) say 95%:

$$Q_r = \frac{8078 \times 1\ 000\ 000}{0{\cdot}95} = 8503 \text{ GJ/year}$$

$$= 2\ 360\ 000 \text{ kWh/year}$$

*Year-round Heating*

For buildings requiring heating throughout the year (e.g. old peoples' homes) the equivalent hours of full load operation should be calculated from winter and summer degree day data.

*Hot Water Services and other Heating Requirements*

Separate calculations must be made to estimate the energy requirement for hot water service provision (see Section B4 for estimated consumption data) and any special heating arrangements. Due allowance must also be made for the heating requirements of mechanical ventilation systems (i.e. seasonal fresh air heat load).

*Annual Fuel Consumption of Office Buildings*

A useful check figure for the annual fuel consumption of centrally heated office buildings (intermittent heating) is 5·4 GJ per kW of installed boiler capacity including standby. Fig. B18.2 shows the annual energy input related to the installed boiler capacity for the various values of equivalent full load hours of operation. (The plots on this figure are taken from the BRS Survey of Office Buildings in South east England).

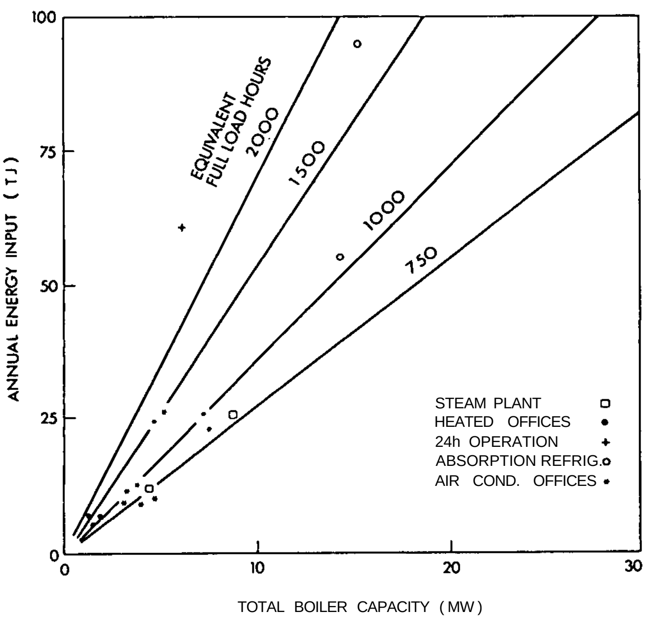



**Fig. B18.2.** Boiler fuel consumption.

*Domestic Heating Costs*

Guide to Home Heating Costs booklets are published by the Energy Efficiency Office for Southern, Midlands and North areas of England and for Scotland and Wales. The booklets are compiled in simple terms and although they are primarily intended to provide indicative guidance to house owners, they offer useful data for interpretive use by building services designers.

*Seasonal Efficiencies of Space Heating Systems and Plant*

It is customary for manufacturers to quote the operating efficiency of heat generators based on results obtained during controlled tests. The seasonal efficiency of such appliances when installed in the field will generally be less than the optimum efficiency due to the following factors:

(*a*) The thermal efficiency of a heat generator may vary under full load and part load conditions.
(*b*) The number land sizing of the heat generators in relation to the estimated system load varies.
(*c*) The seasonal load demand of the system.
(*d*) The method of control.

The above factors may be accounted for by the 'heat conversion efficiency' which may be expressed as:

$$\eta_h = \frac{Q_h}{CV_g} \quad \ldots \quad \text{B18.17}$$

where:

$\eta_h$ = heat conversion efficiency
$Q_h$ = heat delivered to the system .. .. W
$CV_g$ = heat potential in fuel used .. .. W

The manner in which heat emitted from the system is utilised may be defined as 'utilisation efficiency'. This would be dependent on a number of design factors including the method of control, the disposition and sizing of equipment, the size, construction and thermal behaviour of the building, the required design conditions and the method of operation.

$$\eta_u = \frac{Q_r}{Q_h} \quad \ldots \quad \text{B18.18}$$

where:

$_u$ = utilisation efficiency
$Q_r$ = estimated heat requirement .. .. W

The overall seasonal efficiency of the system will be the product of the heat conversion efficiency and the utilisation efficiency. A suggested list of efficiencies is given in Table B18.15. The above factors should be carefully evaluated for a particular system in the light of experience and judgement of the engineer.

*Fuel Oil Preheating*

The heat requirements for preheating fuel oils are dependent upon the following factors:

(*a*) grade of oil used,
(*b*) quantity and method of storage adopted,
(*c*) length and size of oil distribution mains,
(*d*) type of oil burner used,
(*e*) ambient temperature conditions.

It is therefore difficult to give overall figures for heat requirements which cover all cases. Table B18.16 indicates a range af overall heat requirements covering storage tanks, distribution, mains, line heaters, and burner pre-heater.

**Table B18.15.** Seasonal efficiencies of heating systems.

| Type of system | Heat conversion efficiency (%) | Utilisation efficientcy (%) | Seasonal efficiency of system (%) |
|---|---|---|---|
| INTERMITTENT | | | |
| Automatic centrally fired radiator or convector systems | 65 | 97 | 63 |
| Automatic centrally fired warm air ventilation systems | 65 | 93 | 60 |
| Fan-assisted electric off-peak heaters | 100 | 90 | 90 |
| Direct electric (non-storage) floor and ceiling systems | 100 | 95 | 95 |
| *District heating/warm air systems | 75 | 90 | 67·5 |
| CONTINUOUS | | | |
| Automatic centrally fired radiator or convector systems | 70 | 100 | 70 |
| Automatic centrally fired warm air ventilation systems | 70 | 100 | 70 |
| Electric storage radiator systems | 100 | 75 | 75 |
| Electric floor storage systems | 100 | 70 | 70 |
| Direct electric floor and ceiling systems | 100 | 95 | 95 |
| *District heating/ radiator systems | 75 | 100 | 75 |
| WATER HEATING | | | |
| †Gas circulator/storage cylinder | 65 | 80 | 52 |
| †Gas and oil fired boiler/ storage cylinder | 70 | 80 | 56 |
| Off-peak electric storage with cylinder and immersion heater | 100 | 80 | 80 |
| Instantaneous gas multi-point heater | 65 | 95 | 62 |
| *†District heating with local calorifiers | 75 | 80 | 60 |
| *†District heating with central calorifiers and distribution | 75 | 75 | 56 |

*Notes:*
1 Very high efficiency heat generators may raise the heat conversion efficiency.
2 Solid fuel appliances working in conjunction with intermittent systems may require allowance for re-kindling.
3 Heavier liquid fuels will require preheating allowance.
* Allowance should be made separately for mains heat losses on a seasonal basis.
† Dependent on the size of the heat generator, the summer conversion efficiency may deteriorate, thereby reducing the overall seasonal efficiency significantly.

**Table B18.16.** Fuel oil preheating.

| Type of oil | Power per litre of oil burned W |
|---|---|
| Class D | – |
| Class E | 30 to 85 |
| Class F | 55 to 85 |
| Class G | 65 to 100 |

## Estimation of Electrical Energy Consumption

The services covered under this heading include lighting, lifts, pump power for fluid distribution, heating and hot water service systems, and fan power for mechanical ventilation. Electricity usage for air conditioning and refrigeration is dealt with separately.

It should be noted that, with present day tariff levels the cost of electricity consumption for most buildings with typical services installations is about equal to energy costs for heating. For heavily serviced buildings such as those with full air conditioning, electricity costs are likely to be dominant.

### *Lighting*

Except for deep-plan buildings where much of the lighting installation may be operated throughout the hours of occupancy, the use of artificial lighting for many applications is related to the quality of daylight available.

Established methods of daylight design for buildings use the daylight factor concept based on the CIE overcast sky luminance distribution (see BRE Digests 41 and 42). Predictions of energy consumption for artificial lighting usage need to take account of the sky luminance under the whole range of weather conditions. The illumination requirements of the application, the location and plane of the lighting requirement, the shape and position of the window, room dimensions and the reflection of room surfaces are all relevant factors.

Newer methods for estimating daylight in buildings are contained in BRE Digests 309 and 310 and a forthcoming CIBSE window design guide, also a simplified graphical method is shown in CIBSE Building Energy Code Part2(a).

Table B18.17 shows the number of hours per year that the daylight level in a building falls below certain levels; the values assume occupation from 0900 to 1730 on a 5 day week. Thus, if the daylight factor on the working plane is 2%, the natural illuminance will be less than 485 lux for 1655 hours of the working year.

**Table B18.17.** Hours per year that natural internal lighting is below stated level.

| Daylight Factor (%) | Number of hours for stated illumination intensity lux | | |
|---|---|---|---|
| | 320 | 485 | 750 |
| 0·5 | 2186 | 2186 | 2186 |
| 1·0 | 2070 | 2186 | 2186 |
| 2·0 | 1050 | 1655 | 2186 |
| 3·0 | 720 | 1050 | 1700 |

Recent advances in lighting technology have resulted in improved lamp efficiencies and an array of control techniques to promote energy efficiency. These latter range from separately switched arrangements for lighting to completely programmed lighting sequences based on photo-electric sensing of illumination levels. Provisions such as automatic switching-off of a proportion of the lighting on a timed basis with recovery of illumination by re-set switches or by personnel sensors which may be acoustic, ultrasonic or infra-red have also been shown to be effective in promoting energy eficiency.

Details of lighting use in offices are given in Table B18.18 (see also CIBSE Energy Code Part 2(a)).

**Table B18.18.** Lighting use in offices.

| Illumination to be maintained (lux) | Daylight factor at wall opposite to window (%) | Approximate usage (h/year) | | |
|---|---|---|---|---|
| | | Fittings near inner wall | Fittings near middle of room | Fittings near window |
| 200 | 1 | 1600 | 550 | * |
| 200 | 2 | 800 | 280 | * |
| 400 | 1 | 2150 | 950 | * |
| 400 | 2 | 1400 | 470 | * |
| 600 | 1 | 2150 | 1450 | * |
| 600 | 2 | 2150 | 720 | * |

*Up to about 200, which is the number of office hours between one hour before sunset and 1730.

Office lighting may be used outside normal hours for cleaning, security or other purposes. Usage for these purposes will vary but a typical requirement would be between 2 and 6 hours per day. This represents an additional usage of between 520 and 1560 hours in the course of a working year.

Lights in circulation spaces, particularly where these are escape routes will probably be used for the whole period the building is occupied and this may be required for safety reasons. In other internal areas of the building such as toilets, lighting is likely to remain in use once it has been switched on unless energy economy controls have been applied.

*Office Equipment*

With the increasing use of electronic data processing and other office equipment account needs to be taken of the usage of electricity for these purposes.

Section A7 lists typical electricity consumptions for a range of such equipment. Although in many instances office equipment may be switched on continuously during working hours it will be necessary to give consideration to usage patterns and diversities appropriate to particular applications.

*Lifts*

The electricity consumption of lift equipment is small compared with the total electricity consumption of a building. It varies from approximately 0·5% for a low rise building to approximately 3% for a high rise building. Table B18.19 gives average daily consumptions.

Where lifts are used by security personnel outside of normal working hours it is necessary to ascertain the mode of usage adopted. If the lifts are used on independent services this could result in continuous running of the generator. This would have a very considerable influence on power consumption and, hence, operating costs.

**Table B18.19.** Energy consumption by lifts.

| Contract load kg | Energy consumption per day for the following contract speeds m/s MJ | | | | |
|---|---|---|---|---|---|
| | 0·75 | 1·0 | 1·5 | 2·5 | 3·5 |
| 600 | 60 | 80 | 100 | – | – |
| 900 | 70 | 90 | 120 | 180 | 240 |
| 1200 | 80 | 100 | 130 | 190 | 250 |
| 1500 | 90 | 110 | 140 | 200 | 260 |
| 1800 | – | – | 150 | 210 | 270 |

*Fan and Pump Power*

For the estimation of total installed loads for fan power at an early stage of a design, it may be convenient to arrive at a volume flow rate by using the product of floor area and a nominal air change ventilation rate. A nominal estimate of the fan pressure can also be made based on the resistance of typical ventilation plant components and an approximate resistance for the ductwork distribution system related to its scale and the air velocity. Power requirements can be obtained from:

$$q = \frac{V\,L}{\eta_f} \quad \text{.. .. .. .. .. ..} \quad \text{B18.19}$$

where:

| | | | |
|---|---|---|---|
| $q$ | = | power .. .. .. .. .. | W |
| $V$ | = | flow rate .. .. .. .. .. | $m^3/s$ |
| $L$ | = | pressure .. .. .. .. .. | Pa |
| $\eta_f$ | = | efficiency | |

At later design stages, or with existing installations, actual motor power may be used. Similarly, pump power estimates may be based on a nominal volume flow rate and the resistance of components in the pipework, the total length of pipework, and the typical resistance per unit length together with an allowance for fittings.

The efficiency in the above expression relates to the operating performance of the fan or pump installed in the distribution system. This may be significantly less than the type test efficiency quoted by the manufacturer, particularly in the case of centrifugal fans in air handling units.

When supply and extract fans are on the same circuit, their pressures should be added, see Fig. B18.3.

*Monitoring of Building Energy Consumption*

Any refinement of the estimation methods shown above can only come from accumulated operational experience. For this to be acquired it is necessary to monitor and record the following aspects of actual performance:

(*a*) Annual consumption of all energy forms (fuel, electricity).
(*b*) Indoor temperature during occupation.
(*c*) Duration of occupied period.
(*d*) Mode of operation of the heating installation.
(*e*) Degree-days for the locality concerned and the monitoring period or actual recorded outdoor temperatures.

Various reporting formats are available to rationalise the recording of these data. Fundamental data for the building are also necessary including:

(*a*) Floor area (total and net) and number of floors.
(*b*) General building details (e.g. percentage glazing) and thermal properties of various elements.
(*c*) Design heat losses and conditions.
(*d*) Illuminance levels.
(*e*) Internal heat gain sources.
(*f*) Number of occupants.

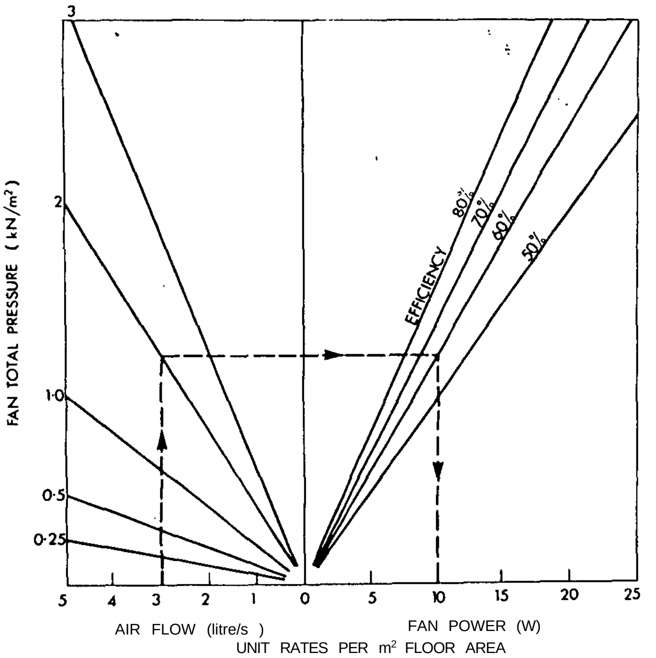



**Fig. B18.3.** Energy consumption by fans.

## UTILITY COSTS

Utility costs are the expenditure on fuel, electricity, gas and water. Of these, only water costs are likely to be unrelated to the quantity used; in some cases they are assessed as a percentage of the rateable value of the building. The other costs are based on consumption and tariff structures.

The effective electricity tariff must be carefully studied when annual costs are being calculated, as these vary from one part of the country to another. The tariffs of some supply boards discriminate between winter and summer so that a system having a comparatively high connected load, or which uses refrigeration for air conditioning, is disproportionately penalised.

Some gas tariffs carry rebates in each of the two summer quarters where at least a proportion of the building uses gas operated cooling equipment.

### Electricity Charges

A supply of electricity in the UK is obtainable from one of the area electricity boards. Supply tariffs vary, and are dependent mainly on:

(*a*) The class of consumer (e.g. domestic, industrial, commercial or agricultural).
(*b*) The size of the consumer's load and the pattern of the demand.

Almost invariably, large industrial supplies are provided on the basis of a maximum demand with a fuel price adjustment clause, while small industrial consumers are offered alternative tariffs such as flat rates or a two-part tariff with a fixed charge and block unit rates. Large commercial consumers may qualify for a maximum demand tariff similar to that offered to industrial consumers but an alternative block or two-part tariff is provided for small commercial premises, related to floor area or the electrical installed load. In some cases the tariffs offered for commercial and industrial supplies are identical.

For domestic supplies most area electricity boards offer a tariff consisting of a standing charge plus a unit charge irrespective of the size of the residence.

Special tariffs are available to all classes of consumer for restricted hour supplies during off-peak periods. The tariffs require separate circuits for metering purposes. The cheapest rates apply when supplies are taken at night (typically 2300 to 0700) and higher rates are charged for off-peak supplies requiring an afternoon or evening boost.

*Maximum Demand Tariffs*

For the larger consumer, a maximum demand tariff is usually the most economical. The overall cost per unit under this tariff and for a particular load factor is given in Fig. B18.4. The load factor may be calculated using equation B18.20:

$$L = \frac{100\,U}{D\,H} \quad \text{.. .. .. .. ..} \quad \text{B18.20}$$

where:

$L$ = load factor .. .. .. .. .. per cent
$U$ = number of units consumed during the period .. .. .. .. .. kWh
$D$ = maximum demand .. .. .. kW
$H$ = number of hours in the period .. .. h

It can be seen from Fig. B18.4 that the overall cost per unit varies with load factor for a typical maximum demand tariff and consumers can frequently reduce their electricity charges by concentrating on improvements to their electrical load factor. Typical load factors are given in Table B18.20.

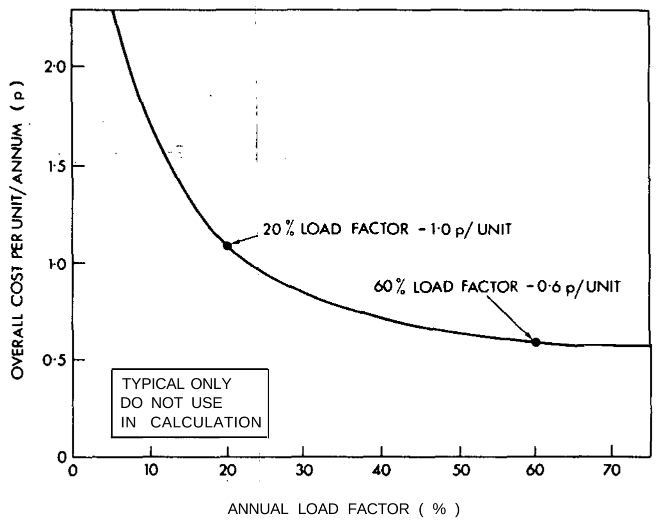



**Fig. B18.4.** Typical cost curve for maximum demand tariff.

## SOME USEFUL CAPITAL AND MAINTENANCE COSTS

The figures given below are average values and must be used accordingly. Actual costs are very dependent on size and usage of the building being considered. Figures quoted are based on 1985 costs.

**Table B18.20.** Typical load factors.

| Premises | Load factors (%) |
|---|---|
| Offices/commercial | 20 to 40 |
| Hospitals | 30 to 40 |
| Schools (day) | 20 to 35 |
| Schools (technical) | 30 to 45 |
| Industrial* | 30 to 85 |

*Depending on type of process and whether intermittent or continuous.

## Capital Costs

Capital Costs for some of the major systems in building services installations i.e. boiler plant, air conditioning, heating etc, are given below, and should be assessed independently in estimates for composite installations.

Costs for boiler plant includes:

(*a*) boiler/s
(*b*) pumps
(*c*) instrumentation and automatic controls
(*d*) electrical installation
(*e*) fuel storage and handling
(*f*) pipework and thermal insulation
(*g*) initial water treatment
(*h*) feed and expansion tank.

Allowance must be made for the following where appropriate:

(*a*) chimneys
(*b*) pressurisation
(*c*) calorifiers; storage and non-storage.

Costs for the following systems are based on offices. Heating costs are for a mix of radiator and convector installations. Ventilation installations are based on air change rates recommended in Section B2.

**Table B18.21.** Offices capital costs.

| Equipment | Capital cost £/m$^2$ |
|---|---|
| Heating | 10 |
| Ventilation low velocity | 82 |
| high velocity | 86 |
| Air conditioning | |
| unit air conditioners | 99 |
| all-air system | 132 |
| air/water system | 110 |

**Table B18.22.** Boiler plant capital costs.

| Plant load kW | Gas £ | Oil £ | Solid fuel £ |
|---|---|---|---|
| 100 | 9 000 | 11 000 | 15 500 |
| 250 | 15 000 | 17 500 | 22 500 |
| 500 | 22 500 | 25 000 | 30 000 |
| 750 | 28 500 | 31 000 | 37 000 |
| 1 000 | 33 500 | 35 500 | 42 500 |
| 1 500 | 41 500 | 42 500 | 51 500 |
| 2 000 | 47 000 | 47 500 | 58 500 |
| 3 000 | 57 500 | 61 500 | 75 000 |
| 4 000 | 69 500 | 74 500 | 91 000 |
| 5 000 | 76 500 | 87 000 | - |

## Maintenance Costs

The great variety of engineering systems renders impracticable the inclusion here of comprehensive maintenance costs. They are, however, generally significant to cost in use and should therefore be evaluated for each system.

The maintenance costs of boilers and firing equipment (cast iron and steel shell type) can be approximated on the basis of the total capital cost of the equipment (applicable to units in excess of 150kW capacity):

| | |
|---|---|
| handfired solid fuel | 5% p.a. |
| automatically fired solid fuel | 8% p.a. |
| automatically fired liquid or gas fuel | 6% p.a. |

*Maintenance Costs for Office Buildings*

In a study of operating costs for a number of office buildings, the breakdown of maintenance costs was found to be as shown in Fig. B18.5.

*Costs Related to Floor Area*

For budgeting purposes it may often be sufficient to estimate costs from the size of the building. For this purpose the best estimates are given by the index 'functional floor area', which is the combined areas of office accommodation, corridors and toilets; it is usually about 70 to 90% of the grosss floor area. Not unexpectedly, maintenance costs in air-conditioned buildings are greater than for those with heating only. The average annual charges for service maintenance related to functional floor area, inclusive of on-costs were found to be (1985 cost basis):

| | |
|---|---|
| heated offices | £11/m$^2$ |
| air-conditioned offices | £16/m$^2$ |

This method of estimation takes no account of the standards for the internal environment. This is a serious limitation where a cost prediction is required for environments much different from those in the survey. The type of system can also influence the values (for example the cost of filter replacements would be greater for an all-air system than would be the case for air/water systems).

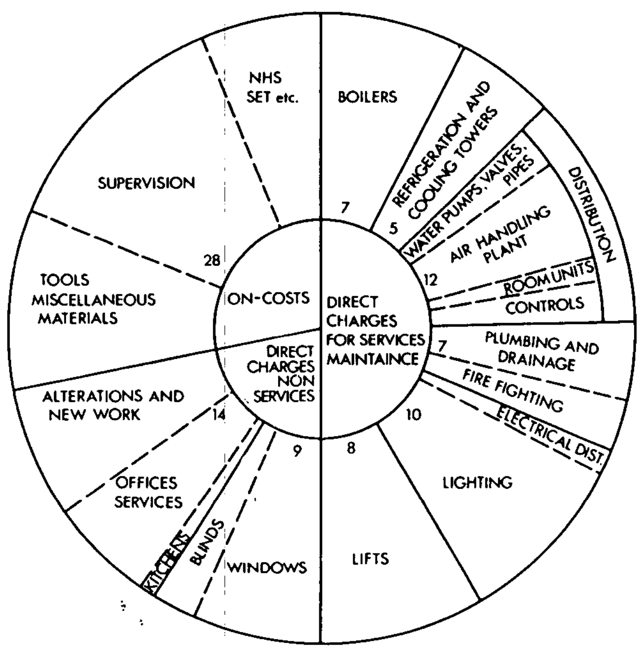



**Fig. B18.5.** Maintenance costs for air-conditioned offices.

## INSPECTIONS AND INSURANCE

### Inspection

Periodic examination of various types of plant is a statutory requirement in the UK. The requirements are embodied in The Factories Act 1961, and/or various individual statutes, depending on the type of plant, business, or environment.

The main types of plant requiring periodic examination are:

*(a)* steam boilers and receivers
*(b)* air receivers
*(c)* lifts, lifting machines and lifting tackle
*(d)* power presses
*(e)* certain types of ventilation and dust extraction equipment.

Possible changes are under consideration by the Health and Safety Commission for pressurised systems and ventilation plant. New regulations and codes of practice are expected in 1987 under the general heading of pressurised systems which will make fundamental changes to the inspections currently required by law. Many systems and vessels which are currently not subject to examination are expected to become so.

### Insurance

Insurance is available for many different types of risk, and an insurance broker wil be able to advise on specific costs.

Table B18.23 gives some typical figures for premiums related to plant cost.

**Table B18.23.** Guide to types of cover and approximate costs for some areas of plant insurance.

| Type of plant | Cover | Typical cost |
|---|---|---|
| Boilers, pressure vessels and other plant under steam pressure. | Indemnity against explosion and collapse:<br>*(a)* to the plant<br>*(b)* to other property belonging to the insured<br>*(c)* sudden and unforeseen damage<br>*(d)* cover for reinstatement costs and requirements of Local Authorities<br>*(e)* third party cover is available as an option if required. | For standard horizontal multitubular boiler:<br>For *(a)* & *(b)* & *(d)*: £90.<br>To include *(c)*: £185.<br>For inspection: £60.<br>(These costs allow for indemnity of £1m.) |
| Hot water boilers, calorifiers, pipes. | As steam, but *(b)* & *(e)* are not necessary as these are automatically covered by fire policy, and third party policies. | For standard 500 MJ cast iron boiler including 1 calorifier and pipes: £50.<br>For annual inspection (optional): £30.<br>(These costs allow for an indemnity of £50,000). |
| Electrical and mechanical plant i.e. pumps, refrigeration plant, compressors, motors, generators, transformers etc. | Against:<br>*(a)* breakdown<br>*(b)* other sudden unforeseen damage including costs associated with repair, such as dismantling and transit to repairers. | Typically for a small (10kW) electric motor *(a)* would be £13 while *(a)* & *(b)* would be £19.<br>For switchboards, sum insured determines price.<br>For a board worth £50,000 for *(a)*: £350.<br>For annual inspection (optional) of: motor: £7 switchboard: £35. |
| Wiring installation. | Not normally insured. Inspection to suit user needs can be arranged. Cost is dependent on prior survey. Work is generally to IEE Regulations standards unless otherwise specified. | Assessed on time spent, based on rate of about £150 per day. |
| Passenger and goods lifts. | Available for:<br>*(a)* breakdown<br>*(b)* sudden and unforeseen damage.<br>Inspection is required by law.<br>Third party is available as an option, if necessary. | Determined by capacity, total power of electric motors (or other power source) and the number of floors served.<br>For *(a)* and *(b)* for a 6 person lift serving 6 floors or less would be £120.<br>For two inspections per annum £70. |
| Crane, hoists and lifting machines. | Available as for lifts.<br>Inspection on the majority of lifting machines is required by law. | For *(a)* and *(b)*, 5 ton mobile crane: £600.<br>For annual inspection £65. |
| Lifting tackle, runways etc. | Insurance is not normally purchased.<br>Inspection is required by law. | For two inspections per annum of manual (1 tonne): chain block: £15; 6 m runway: £25. |
| Hired plant: | Provided against hirers legal liability for:<br>*(a)* damages/loss of the plant<br>*(b)* liability to pay continuing hire charges.<br>Optional cover available<br>*(c)* damage to goods being lifted<br>*(d)* damage to insureds, or third party surrounding property<br>*(e)* consequential loss.<br>Limits of indemnity would be determined by negotiation. | Determined on the type of plant hired and the amount of hiring charges.<br>Typical premium rate for a policy covering *(a)* and *(b)* on "general contractors plant" (excluding scaffolding and tower cranes) based on annual charges of £25,000 would be 3% to 4%. |
| Base date for all costs – second quarter, 1986 | | |

## REFERENCES


1 BILLINGTON, N. S., Estimation of annual fuel consumption, *JIHVE* 34, 253, Nov. 1966.

2 BILLINGTON, N. S., COLTHORPE, K. J. and SHORTER, D. N., Intermittent heating, *HVRA Laboratory Report No. 26,* BSRIA, Bracknell, 1964.

# INDEX to VOLUME B

The Chartered Institution of Building Services Engineers Delta House, 222 Balham High Road, London SW12 9BS

# Corrigenda to *CIBSE Guide*

Ref: GDE/3

February 1992

The back of this page is gummed to allow the user to insert some of the corrections in their rightful places within the *Guide*.

## Additional amendments to 1986 and 1988 (reprinted) versions:

### Volume A

Page A1-9
At the bottom of first paragraph, second column, this sentence should be added:

See also guidance based on more recent information given in *Guide* Section B2.

Page A4-4
Some early copies of Volume A give the term "$Q_v$" on the left hand side of equation A4.2. The left hand side of this equation should be "Q" (i.e. in metres per second). The definition of "$Q_v$" immediately following equation A4.2 should be deleted.

### Volume B

Page B3-4
Ventilation efficiency

Delete "efficiency" and substitute "effectiveness" in heading and subsequent text. Delete "%" from definitions of $E_c$ and $E_v$.

Page B3-34
Figure B3.52 Operation of reversing valve

In the diagram demonstrating cooling mode, arrow to the right of expansion device should point left.

The following equations should read:

Page B3-37
Table B3.6. The following equation should read:

Hot source, exposed sides and top

$$Q = 0.038\ A_s \sqrt[3]{\frac{h\, A_t\, D}{A_s}} + 0.5\,(A - A_s)$$

Page B3-38
Table B3.9. Replace equations with the following:

$$Q = lv(X\sqrt{\frac{2X}{W}} + W)$$

For R ≤ 2;

$$Q = v(5\sqrt{\frac{2}{R}}\, X^2 + A)$$

For 2 < R ≤ 5;

$$Q = v(5\sqrt{\frac{R}{2}}\, X^2 + A)$$

For X > 0.75 W; calculate flow rate *Q* for plain arrangement and multiply by 0.75.

Page B3-42
Equation B3.11 should read:

$$N = \frac{D \times 10^6}{3600\, v\, c\ (h/100)} \quad .. \quad .. \quad .. \quad B3.11$$

### Volume C

Page C3-12
Equation C3.51

$$f_c = 0.64\ \frac{(t_s - t_f)^{t.25}}{L^{0.25}} \quad .. \quad .. \quad .. \quad .. \quad C3.51$$

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Section B15: Vertical Transportation
This section has been superseded by
Guide D: Transportation Systems in buildings (1993)

### Volume C

Page C3-12
Equation C3.51

$$f_c = 0.64\ \frac{(t_s - t_f)^{1.25}}{L^{0.25}} \quad \ldots \quad \text{C3.51}$$