Heating and Water Services Design in Buildings

KEITH J. MOSS

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Contents

Preface

Heating and Water Services Design in Buildings has been written following 13 years in the industry and 27 years teaching and consultancy work. The author has been exposed to college students, university undergraduates and open learning candidates, ranging in age from 16 to 48 years.

Many of those people came from the industry as apprentices or trainees, with limited experience initially. Many of the older students were either experienced and wanting qualifications or had transferred from other engineering disciplines.

The book has therefore been written for such students, whether they are following a course of study or requiring to increase their knowledge of building services engineering.

The book does require some knowledge of the industry, and if this is lacking recourse should be made to manufacturers' literature identified in the text. Access is also needed to the Chartered Institution of Building Services Engineers *(CIBSE) Guide* to current practice. Student membership of the Institution currently qualifies the student to a free extract of the *Guide,* and application for this grade of membership is strongly recommended.

The author is only too well aware that this book cannot address all the queries that may arise during its study, and therefore the student is also encouraged to seek a mentor who can advise and assist when part of the text needs more explanation than is provided.

CIBSE will gladly advise the enquiring student of the name and number of the secretary for the local region, who will be quite prepared to discuss the matter of a suitable mentor.

Each chapter begins with the nomenclature used and an introduction. The chapter closure at the end of each chapter identifies the competences that will have been acquired after successful completion.

The text is written in a way that actively involves the reader by encouraging participation in the solutions to examples and case studies, with some examples being left for the reader to try. It is intended to be a learning text in practical design.

The last chapter is entitled 'Loose ends', partly because it deals with topics not covered elsewhere in the text. It also happens to be one of the author's favourite radio programmes.

Acknowledgements

I am indebted to Mr Shaw, Mr Sedgley and Mr Douglas, who were my principal teachers at what used to be called the National College in Heating, Ventilating, Air Conditioning and Fan Engineering, and is now integrated with the University of the South Bank.

Grateful thanks are also due to Tony Barton, who preceeded me at the City of Bath College and initially set up the courses in HVAC. He it was who introduced a raw recruit from industry to the art of enabling students to learn.

Finally I have to thank all those students who have had to suffer my teaching over the years, because among other things they have taught me that people learn in many different ways, and this makes the profession of teacher a humbling experience and a vocation, in which the teacher is frequently the learner.

Introduction

Welcome to the discipline of building services engineering. You will find that it extends beyond the confines of this text, which concentrates on some of the 'wet services' within the building envelope. Part of Chapter 1, however, looks at the building and the way it behaves when it is intermittently heated, as it is important for you to select the plant and design the system that will best suit the building and its use. It is assumed that you have some knowledge of building services. Where necessary you are directed to current manufacturers' literature so that the text can take on a fuller meaning. It is strongly recommended that you also have access to parts of the *CIBSE Guide* to current practice. They include: pipe-sizing tables and velocity pressure loss data, and properties of building materials including admittance values.

At present, the Institution offers a free student handbook to enrolling student members. This has useful extracts from the *Guide,* and will be an aid in gaining maximum benefit from this book.

1 Heat requirements of heated buildings in temperate climates

Nomenclature

A	surface area (m^2)
A_p	area of the heated plane (m^2)
A_u	area of the unheated surfaces (m^2)
C	specific heat capacity (kJ/kg K)
d	design conditions
dt	temperature difference (K)
dt_t	total temperature difference (K)
E	fraction of heat radiation
EAT	entering air temperature (°C)
f	fabric heat loss ratio $\Sigma\ (UA)/\Sigma A,\ \Sigma(YA)/\Sigma A$ (W/m^2K)
f_r	thermal response factor
F_1, F_2	temperature interrelationships
F_3	plant ratio
H	thermal capacity (kJ/m^2)
h_a	heat transfer between the air and environmental points (W/m^2K)
h_{ac}	heat transfer between the air and dry resultant points (W/m^2K)
h_c	heat transfer coefficient for convection (W/m^2K)
h_{ec}	heat transfer between the environmental and dry resultant points (W/m^2K)
HTHW	high-temperature hot water
K	constant
k	thermal conductivity (W m/m^2K)
L	thickness of a slab of material (m)
L/k	thermal resistance of the slab (m^2K/W)
LAT	leaving air temperature (°C)
M	mass flow rate (kg/s)
n	index
n	operating plus preheat hours
N	number of air changes per hour
p	prevailing conditions

Q_f	conductive heat loss through the external building fabric (W)
Q_p	plant energy output (W)
Q_{pb}	boosted plant energy output (W)
Q_t	total heat loss=Q_f+Q_v (W)
Q_v	heat loss due to the mass transfer of infiltrating outdoor air (W)
R_a	thermal resistance of the air cavity (m^{2}K/W)
R_b	thermal resistance of brick (m^{2}K/W)
R_i	thermal resistance of insulation (m^{2}K/W)
R_p	thermal resistance of plaster (m^{2}K/W)
R_{si}	inside surface resistance (m^{2}K/W)
R_{so}	outside surface resistance (m^{2}K/W)
R_t	total thermal resistance (m^{2}K/W)
t_a, t_{ai}	indoor air temperature (°C)
t_b	balance temperature (°C)
t_c	dry resultant, comfort temperature (°C)
t_d	datum temperature (°C)
t_e	environmental temperature (°C)
t_{ei}	environmental indoor temperature (°C)
t_{eo}, t_{ao}	outdoor temperature (°C)
t_f	flow temperature (°C)
t_i	indoor temperature (°C)
t_m	mean surface temperature (°C)
t_o	outdoor temperature (°C)
t_p	temperature of the heated plane (°C)
t_r	mean radiant temperature (°C)
t_r	return temperature (°C)
$(t_r)_u$	mean radiant temperature of the unheated surfaces (°C)
t_x	temperature of the unheated space (°C)
U	thermal transmittance coefficient (W/m^{2}K)
v	ventilation heat loss ratio=$NV/3\Sigma A$ (W/m^{2}K)
V	volume (m^3)
VFR	volume flow rate (m^3/s)
Y	admittance (W/m^{2}K)
α	coefficient of linear thermal expansion (m/mK)
ρ	density (kg/m^3)
Σ	sum of

1.1 Introduction

Heat flow into or out of a building is primarily dependent upon the prevailing indoor and outdoor temperatures. If both are at the same value heat flow is zero, and the indoor and outdoor climates are in balance with no heating required.

During the heating season (autumn, winter and spring), when outdoor temperature can be low, the space heating system is used to raise indoor temperature artificially to a comfortable level, resulting in heat losses through the building envelope to outdoors. The rate of heat loss from the building depends upon:

- heat flow into or through the building structure, Q_f in watts;
- rate of infiltration of outdoor air, resulting in heat flow Q_v in watts to outdoors as the warmed air exfiltrates;
- building shape and orientation;
- geographical location and exposure.

STRUCTURAL HEAT LOSS

This occurs as heat conduction at right angles to the surface, and is initially expressed as **total thermal resistance** R_t, where

$$R_t = R_{si} + \Sigma\left(\frac{L}{k}\right) + R_a + R_{so} \qquad (\mathrm{m^2\ K/W}) \tag{1.1}$$

The **thermal transmittance coefficient** $U=1/R_t$ W/m^2 K, and is the rate of conductive heat flow through a composite structure (consisting of a number of slabs of material, which can include air cavities) per square metre of surface, and for one degree difference between indoor and outdoor temperature.

It follows that structural heat loss is given by

$$Q_f = \Sigma\ (UA)\mathrm{d}t \qquad (\mathrm{W}) \tag{1.2}$$

INFILTRATION

Consider Figure 1.1, which shows a section through a building. The prevailing wind infiltrates one side of the building, where the space heating appliances must be sized to raise the temperature of the incoming outdoor air. The heated air moves across the building to the leeward side, where it is exfiltrated. Here the heating appliances are not required to treat the air. Clearly, on another day the wind direction will have changed, and therefore all heating appliances serving the building perimeter must be sized to offset the infiltration loss Q_v as well as the structural heat loss Q_f.

Theoretically, the boiler plant is sized on the basis of the total structural heat loss plus only half of the total loss due to infiltration, as only half of the appliances are exposed to cold infiltrating air at any instant. In practice the full infiltration heat loss is employed in plant selection.

It follows therefore that the rate of infiltration—and hence the heat loss due to infiltration of outdoor air are dependent upon air temperature and wind speed. Further factors that relate to the building design may also influence infiltration

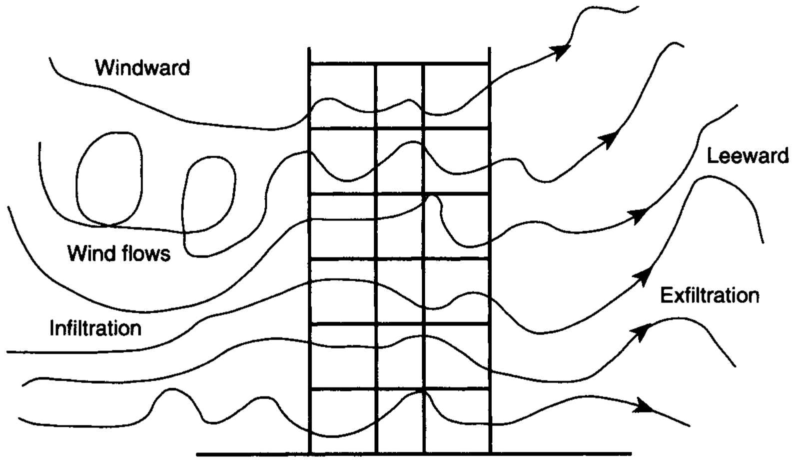


Figure 1.1 Section through a multistorey building showing prevailing wind pattern.

rates, and include stack effect in the building resulting from stairwells, lift shafts, unsealed service shafts and atria, and how well the building is sealed.

Thus from $Q=M.C.\mathrm{d}t$, $Q_v=(V.\rho.N.C.\mathrm{d}t)$ 3600, and if standard values of air density and specific heat capacity are taken,

$$Q_v=0.33\ NV\ \mathrm{d}t \tag{1.3}$$

BUILDING SHAPE AND ORIENTATION

This affects the way indoor temperature control is achieved to offset the effects of solar irradiation on and through the building envelope (see Chapter 6).

GEOGRAPHICAL LOCATION AND EXPOSURE

The location and elevation of the site are accounted for in the choice of outdoor design temperature. Information and data are offered in Section A2 of the *CIBSE Guide,* in the absence of local knowledge.

Exposure relates to the effect of wind speed, the increase of which gradually destroys the outside surface resistance R_{so}, increasing the thermal transmittance and infiltration.

1.2 Heat energy flows

There are two generic observations that apply to the natural world:

- Heat energy will always flow from a high-temperature zone to zones at lower temperature.

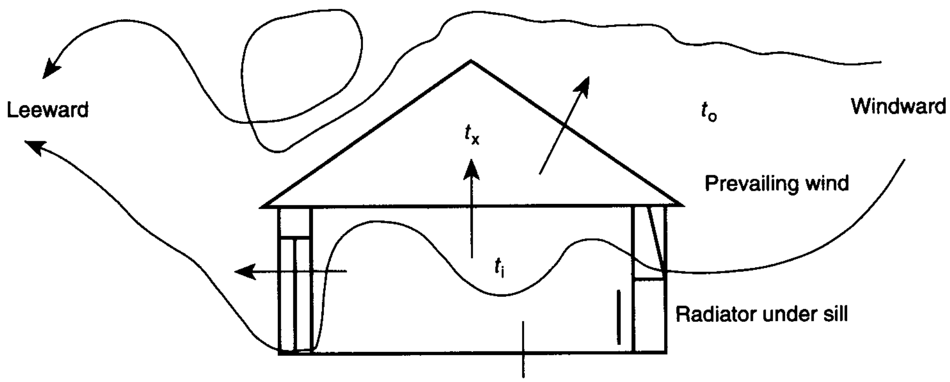


Figure 1.2 Section through building: $t_i > t_x > t_o$.

- The rate of heat flow is dependent upon the magnitude of the temperature difference between zones.

For example, if the heat flow from a building is 100 kW when t_i is 20 °C and t_o is -3 °C, where the magnitude of the temperature difference is 23 K, it is clear that heat flow will increase when outdoor temperature falls to -10 °C and the temperature difference is now 30 K.

The revised heat loss=100×30/23=130.4 kW.

Likewise the output of a radiator varies with the magnitude of the temperature difference between its mean surface temperature t_m and room temperature t_i. Consider Figure 1.2, which shows a section through a building and the heat flow paths expected during the winter season. When temperatures are steady a heat balance may be drawn:

heat loss from the building=heat output from the space heater

Using appropriate equations:

$$(\Sigma(UA)+0.33NV)\ (t_i-t_o)=KA(t_m-t_i)^n \tag{1.4}$$

Index n is approximately 1.3 for radiators and 1.5 for natural draught convectors, and is found empirically.

Example 1.1

(a) A room has a heat loss of 6 kW when held at a temperature of 20 °C for an outdoor temperature of -1 °C. Find the required surface area of a radiator to offset the room heat loss, given that the manufacturer's constant K=13 W/m^{2}K, index n=1.3 and the flow and return temperatures at the radiator are 80 °C and 70 °C.

(b) If the outdoor temperature rises to 5 °C, find the required mean surface temperature of the radiator to maintain the room at a constant 20 °C.

Table 1.1 Example 1.1

Conditions	*Heat loss* (W)	*Radiator output* (W)	$(t_m - t_i)$ (K)	*Outdoor temperature* (°C)
Design	6000	6000	55	−1
Prevailing	4286	4286	42	+5

Solution

(a) Adopting equation (1.4), $6000=13A\ (75\text{-}20)^{1.3}$.

From which

$$A=2.52\text{m}^2$$

(b) Prevailing heat loss:

$$Q=\frac{6000\times(20-5)}{20+1}=4286\text{ W}$$

Substituting into equation (1.4):

$$4286=13\times 2.522(t_m\text{-}20)^{1.3}$$

From which

$$t_m=62\ ^\circ\text{C}$$

Do you agree?

These results are summarized in Table 1.1.

The system controls must vary the radiator mean surface temperature as the outdoor temperature varies. However, the rate of response required to changes in outdoor climate is dependent upon the thermal capacity of the building envelope, and this varies from lightweight structures, which have a short response, to heavyweight structures. No margin has been added to the radiator; a figure of 10% is frequently used in practice.

Example 1.2

A natural draught convector circuit has design conditions of: t_i=20 °C, t_o=-1 °C, t_f=82 °C and t_r=70 °C. Determine the required mean water temperature and the circuit flow and return temperatures to maintain a constant indoor temperature when outdoor temperature rises to 7 °C. Take index n as 1.5.

Solution

Here the energy balance may be extended, if it is again assumed that temperatures remain steady:

heat loss=convector output=heat given up by the heating medium

The last part of the heat balance is obtained from

$$Q=MC(t_f-t_r) \quad \text{(W)}$$

assuming the heating medium is water.

Under operating conditions the constants in each part of the heat balance can be ignored, and:

$$(t_i-t_o)\propto(t_m-t_i)^n\propto(t_f-t_r)$$

If design (d) and prevailing (p) temperatures are put together:

$$\frac{(t_i-t_o)_p}{(t_i-t_o)_d}=\frac{(t_m-t_i)_p^n}{(t_m-t_i)_d^n}=\frac{(t_f-t_r)_p}{(t_f-t_r)_d} \qquad (1.5)$$

Equating heat loss with heat output:

$$\frac{20-7}{20+1}=\frac{(t_m-20)^{1.5}}{(76-20)^{1.5}}$$

from which

$$t_m=60.7\ ^\circ\mathrm{C}$$

Do you agree?

Equating heat loss with heat given up:

$$\frac{13}{21}=\frac{\mathrm{d}t}{82-70}$$

from which

$$\mathrm{d}t=7.4\ \mathrm{k}$$

For two-pipe distribution:

$$t_f=t_m+\frac{1}{2}\mathrm{d}t=60.7+3.7$$

and

$$t_r=t_m-\frac{1}{2}\mathrm{d}t=60.7-3.7$$

thus t_f=64.4 °C and t_r=57 °C.
The results are summarized in Table 1.2.

Table 1.2 Example 1.2

	Condition								
	t_i	t_o	d_t	t_f	t_r	dt	t_m	$(t_m - t_i)$	dt
Design	20	–1	21	82	70	12	76	(76 – 20)	56
Prevailing	20	+ 7	13	64.4	57	7.4	60.7	(60.7 – 20)	40.7

CONCLUSION

A series of flow temperatures may be evaluated for corresponding different values of outdoor temperatures, and a plot of outdoor temperature versus circuit flow temperature produced. This provides the basis for calibrating the controls. See Figure 1.3.

At an outdoor temperature of 15 °C it is assumed here that there are sufficient indoor heat gains to keep indoor temperature at design of 20 °C without the use of the space-heating system. If this is the case, the **balance temperature** t_b is 15 °C.

The balance temperature can be calculated from:

$$t_b = t_i - \frac{\text{heat gain in watts}}{\text{design heat loss in W/K}}$$

Circuit flow temperature of 30 °C is an arbitrary value, and depends upon the type of control device.

It is worth noting that a change in outdoor condition will not require an immediate response from the controls to maintain a constant indoor temperature. The response time will depend upon the thermal capacity of the building envelope.

THERMAL CAPACITY OF THE BUILDING ENVELOPE

The thermal capacity H of the building envelope is normally measured in kJ/m^2 of structural surface on the hot side of the insulation slab, which may consist of a proprietary material, or it may have to be taken as the

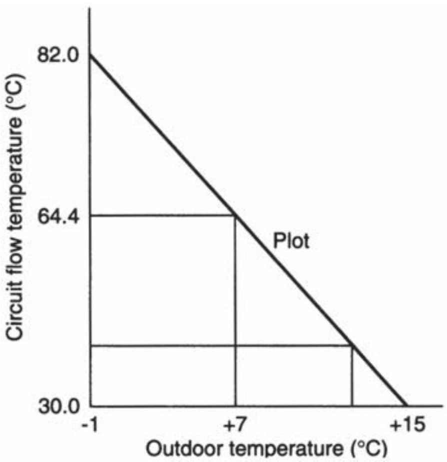


Figure 1.3 Calibration of temperature controls.

air cavity if there is no identifiable thermal insulation slab within the structure.

When the space heating plant starts up after a shutdown period of, say, a weekend, the building envelope is cold, and heat energy is absorbed into the structural layers on the room side of the insulation slab until optimum temperatures are reached in the layers of material. At this point the rooms should begin to feel sufficiently comfortable to occupy. The more layers of material there are on the room side before the insulation slab is reached, the greater will be the thermal capacity of the building envelope and the longer the preheat period. Conversely, the longer is the cooldown period after the plant is shut down.

The energy equation is

$$H=\text{slab thickness } L\times\rho\times C\times(t_m-t_d) \quad (\text{kJ/m}^2) \tag{1.6}$$

In section A3 of the *CIBSE Guide* a table lists the thermal conductivity, specific heat capacity and density of various building materials. These terms are properties of the materials listed, which vary with temperature and moisture content. Values of surface resistances and cavity resistances are included.

Example 1.3
Consider the composite walls (a) and (b) detailed in Figure 1.4.
From the data, determine the wall thermal capacity on the hot side of the insulation slab for each case and draw conclusions from the solutions.

Data
Use of table of properties of materials from the *CIBSE Guide*. Wall elements are: 10 mm lightweight plaster, 25 mm mineral fibre slab, 100 mm brick, air cavity, 100 mm brick.

Note that the thermal insulation slab is located differently in each case. Indoor temperature 20 °C, outdoor temperature -1 °C and datum temperature is taken as 12 °C.

Solution
For wall (a):

$$R_t=R_{si}+R_p+R_i+R_b+R_a+R_b+R_{so}$$
$$R_t=0.12+0.0625+0.7143+0.1613+0.18+0.119+0.06$$
$$=1.4171 \text{ m}^2\text{K/W}$$

As thermal resistance $R\propto dt$:

$$\frac{R}{R_t}=\frac{dt}{dt_t} \tag{1.7}$$

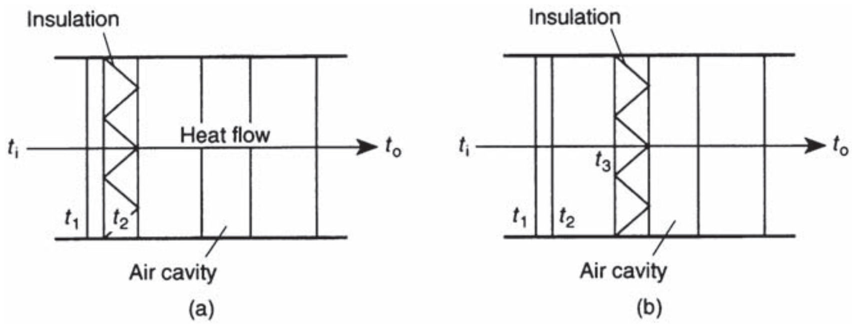


Figure 1.4 Two similar walls with insulation in different locations.

Thus

$$\frac{R_{si}}{R_t} = \frac{t_i - t_1}{t_i - t_o} \tag{1.7}$$

Substitute:

$$\frac{0.12}{1.4171} = \frac{20 - t_1}{20 + 1}$$

from which

$$t_1 = 18.22 \text{ °C}$$

Similarly

$$\frac{0.12+0.0625}{1.4171} = \frac{20 - t_2}{20 + 1}$$

from which

$$t_2 = 17.3 \text{ °C}$$

The mean temperature of the plaster, t_m, is therefore given by

$$t_m = 17.76 \text{ °C}$$

and the thermal capacity of the plaster, which is the only element on the hot side of the insulation, can now be determined from equation (1.6) and knowledge of the density and specific heat capacity of lightweight plaster. Thus:

$$H = 0.01 \times 600 \times 1.0 \times (17.76 - 12)$$

from which, for wall (a)

$$H = 34.6 \text{ kJ/m}^2$$

For wall (b), total thermal resistance R_t remains the same, but the slabs of material are arranged so that now the material on the hot side of the insulation includes the plaster and the inner leaf of the wall.

Inside surface temperature t_1 and temperature t_2 at the interface will have the same value. Interface temperature t_3 needs calculation, and from equation (1.7):

$$\frac{t_i - t_3}{t_i - t_o} = \frac{R_{si} + R_p + R_b}{R_t}$$

Substituting:

$$\frac{20 - t_3}{20 + 1} = \frac{0.12 + 0.0625 + 0.1613}{1.4171}$$

from which

$$t_3 = 14.9\ ^\circ\mathrm{C}$$

The mean temperature of the inner brick leaf of the wall therefore will be

$$t_\mathrm{m} = 16.1\ ^\circ\mathrm{C}$$

The thermal capacity on the hot side of the insulation now includes the plaster plus the inner brick leaf, for which density and specific heat capacity are required, and:

$$\begin{aligned} H &= 33.6 + (0.1 \times 1700 \times 0.8) \times (16.1 - 12) \\ &= (34.6 + 557.6) \quad (\mathrm{kJ/m^2}) \\ &= 592\ \mathrm{kJ/m^2} \end{aligned}$$

Do you agree?

There may be some anxiety over what exactly is datum temperature, taken here as 12 °C. It is, in this context, the point above which heat energy is measured. The indoor frost thermostat on a temperature control to a building might be set at 12 °C, this being the temperature below which it is not desirable for the building envelope to go when the plant is normally inoperative. This therefore seems a reasonable datum temperature to adopt.

Conclusion

The effect on the thermal capacity in wall (b) is considerable. This implies that before comfort levels are reached, the external wall will need to absorb 591 kJ/m^2 of heat energy from the space heating plant. The following analyses may be made.

1. Slab density has a significant effect on thermal capacity on the hot side of the thermal insulation slab.
2. It will take longer for comfort conditions to be reached in wall (b) than in wall (a). Thus the preheating period will need to be longer.

Conversely, cooling will take longer, allowing the plant to be shut down earlier.

3. It will take longer for the inside surface temperature of 17.3 °C to be reached in wall (b).
4. The thermal transmittance coefficient (U) is the *same* for both wall (a) and wall (b).
5. The thermal admittance (Y) is *lower* in wall (a) than in wall (b) (see Example 1.4).
6. The location of the thermal insulation slab in the structure dictates the **thermal response** f_r for that structure (see Example 1.4). It can effectively alter a sluggish thermal response (heavyweight) structure to a rapid thermal response (lightweight) structure when the insulation slab is located at or near the inside surface.
7. The inner leaf of the wall in composite wall (b) should consist of lightweight block if a faster response is required.
8. The air cavity may be taken as the insulation slab in the absence of insulation material in a composite external structure when identifying the slabs on the hot side.
9. If insulated lining is located on the inside of an external wall during refurbishment, the inside surface temperature of the wall is raised, thus improving comfort; the *U* and *Y* values are each reduced, thus saving energy; preheat and cooldown times are reduced; and the wall is behaving like a lightweight structure.
10. Internal walls and intermediate floors, which are not exposed to outdoor climate and which are constructed from dense material like concrete and blockwork, will have a flywheel or damping effect upon the rise and fall in temperature of the building structure for an intermittently operated plant. They in effect act as a heat store.

ENVIRONMENTAL CONTROL BY STRUCTURE

The building services industry is increasingly exposed as having a major role to play in limiting nitrous oxide and CO_2 emissions through correct plant selection and maintenance.

It can be seen from the foregoing analysis that the type of structure and its composition used in a building have a significant effect upon the way in which plant and space heating systems need to operate to maintain indoor design temperature. The concept extends further to the levels of CO_2 emission generated as a result of the building construction process, including manufacturing of materials and their transport to site.

The consumption of primary energy in the form of gas, coal or oil to run the plant is governed largely by how well the building is thermally insulated and how efficiently ventilation is controlled. Thus a well-insulated building, while consuming comparatively low amounts of primary fuel with consequent low flue gas emissions, may have generated excessive levels of CO_2 during the building process. There are clearly environmental penalties and benefits here, which lead to a consideration of an environmental cost-benefit analysis. This extends beyond the scope of this book, but environmental control by building structure is nevertheless an important associated topic.

VAPOUR FLOW

Air at an external design condition of 3 °C during precipitation (rainfall) can be at saturated conditions (relative humidity of 100%). If it is then sensibly heated to 20 °C dry bulb by passing it through an air heater battery, its relative humidity drops to 32%. At both dry bulb temperatures the vapour pressure remains constant at 7.6 mbar. This is because moisture in the form of latent heat has not been absorbed from or released to the air.

The partial pressure of the water vapour in the air is therefore altered only by adding or removing latent heat through the process of evaporation or condensation. Indoor latent heat gains are incurred immediately the building is occupied, owing to involuntary evaporation from the skin surface, exhalation of water vapour from the lungs, and sweating. In the winter, therefore, latent heat gain indoors is inevitable during occupancy periods. Cooking, dishwashing and laundering add to the latent heat gains. If the building is heated, the air is usually able to absorb the vapour production, with a consequent rise in vapour pressure. In an unheated and occupied building condensation may occur on the inside surface of the external structure because the air is unable to absorb all the vapour being produced.

Vapour pressure in heated and occupied buildings is inevitably higher than the vapour pressure in outdoor air. Vapour therefore will migrate from indoors to outdoors. In highly ventilated buildings or indoor locations this may well take place via the ventilating air.

Otherwise it will migrate through the porous elements of the building envelope. If the temperature gradient in the external structure reaches dew-point, the migrating vapour will condense. It is important to ensure that, when it does, it occurs in the external leaf of the structure, which is usually capable of saturation from driving rain. The use of vapour barriers in the building envelope is also common practice. It is important to ensure that the vapour barrier is located as near to the hot side of the external structure as possible. It is also important to know that vapour barriers only inhibit the migration of water vapour unless materials like glass, plastic or metals are used, and even here migration can occur around seals and through the smallest puncture. However, if vapour migration is largely inhibited, the likelihood of interstitial condensation (that occurring within the external structure) is rare.

The occurrence of condensation and dampness on the inside surface of the building envelope is avoided only by adequate ventilation and thermal insulation.

There are software programs that can identify the incidence of surface and interstitial condensation for given external composite structures with and without the use of the vapour barrier. The first part of this chapter has introduced you to simple thermal modelling, in which you now have the skills to analyse the thermal response of a building envelope to continuous and intermittent space heating. Later in this chapter this topic is introduced again.

1.3 Plant energy output

The determination of building heat loss, which forms the basis for the calculation of plant energy output Q_p, is dependent upon the mode of plant operation and hence the mode of occupancy. It can be divided into three categories: continuous, intermittent and highly intermittent.

It has long been known that building envelopes for factories and workshops having traditional transmittance coefficients (average U value approximately 1.5 W/m^2 K) and relatively high rates of infiltration (above 1 air change per hour) require higher levels of convective heating than radiant heating to maintain indoor design conditions. This is based on the knowledge that, living in the natural world as we do, we are quite comfortable in outdoor climates of relatively low air temperature and velocity if solar radiation is present with sufficient intensity.

A similar response is to be found indoors when a significant proportion of appliance heat output is in the form of heat radiation.

The calculation of plant energy output Q_p using temperature ratios F_1 and F_2 accounts for the varying proportions of radiant and convective heating offered by different heating appliances in such building envelopes.

However, it will be shown that for buildings subject to current thermal insulation standards (average U value 0.5 or less) and with infiltration rates below 1 per hour, plant energy output is about the same value for both highly radiant and highly convective systems.

The question then arises as to which system is appropriate for a particular application. This has in the past generated considerable discussion in the technical press. The answer for modern factories and workshops is not now to be found in the building type and shape but in the use to which the building will be put. High-tech dust-free environments will generally benefit from radiant systems. Processes producing dust and fumes will require heated make-up air, dictating an air-heating system. Areas of high occupancy will require the introduction of tempered fresh air to maintain indoor air quality unless the building is lofty.

Temperature ratios F_1 and F_2 allow the determination of indoor air temperature t_{ai} and environmental temperature t_{ei} for a given building envelope and space heating system. Along with dry resultant temperature t_c (the design index) and mean radiant temperature t_r, which can also be determined, we have four thermal indices by which to judge whether a heated building is comfortable to occupy.

The decision whether a heated building is comfortable for at least 80% of the occupants is found in the closeness or otherwise of air, environmental and mean radiant temperature to the design dry resultant or comfort temperature t_c.

The temperature ratios F_1 and F_2 are given by

$$F_1 = \frac{t_{ei} - t_{ao}}{t_c - t_{ao}} \tag{1.8}$$

$$F_2 = \frac{t_{ai} - t_{ao}}{t_c - t_{ao}} \tag{1.9}$$

The value of F_1 and F_2 can be obtained from tables, so that the unknown temperatures t_{ei} and t_{ai} can be determined for identified proportions of convective and radiant heating.

Alternatively the values can be calculated from

$$F_1 = \frac{18 + 6Ev}{18 + f + K(3v - f)} \tag{1.10}$$

$$F_2 = 4 - 3F_1 \tag{1.11}$$

These equations for F_1 and F_2 provide solutions that are approximate but within the limits of 2.5% error.

Factor K is a variable, and primarily depends upon the proportion of radiant to convective heat output from the space heating system. Table 1.3 gives values of K for use in solutions of temperature ratio F_1.

When air velocity is around 0.1 m/s, dry resultant temperature, commonly called **comfort temperature** t_c, which is the design comfort index, is obtained from

$$t_c = \frac{1}{2}t_a + \frac{1}{2}t_r \tag{1.12}$$

Plant energy output Q_p for continuous heating is obtained from

$$Q_p = \Sigma(UA)(t_{ei} - t_{eo}) + 0.34NV(t_{ai} - t_{ao})$$

Adopting equations (1.8) and (1.9) for F_1 and F_2 and substituting:

$$Q_p = [F_1\Sigma(UA) + 0.33F_2NV](t_c - t_{ao}) \tag{1.13}$$

Table 1.3 Values of factor K

Heat radiation proportion (%)	K
0	0
10	0.15
20	0.32
30	0.40
50	0.72
67	1.10
90	1.35

This equation is effectively the sum of the fabric heat loss grand the heat loss due to natural infiltration, Q_v. Q_f and Q_v can be determined separately if required.

CONTINUOUS HEATING

There now follows the determination routine for plant energy output for continuously heated buildings.

Example 1.4
Determine the plant energy output for a fully exposed workshop measuring 30×15×5 m high having a 'flat' roof, where the anticipated air change rate is 2.5 per hour:

(a) for a system of unit heaters (100% convective);
(b) for a system of gas-heated radiant tubes (10% convective, 90% radiant).

Data
t_c=18 °C, t_{ao}=-2 °C, area of wall glass=135 m^2.
Thermal transmittance coefficients: U_g=5.7, U_w=1.3, U_r=1.5 and U_f=1.0 W m^{2}K (the suffixes refer to glass, wall, roof and floor respectively).

Solution for continuous heating
The structural heat loss is tabulated in Table 1.4.

(a)
From equation (1.10):

$$F_1=0.91$$

Table 1.4 Example 1.4: tabulation of structure heat loss

Element	*Dimensions (m)*	*Area, A*	*U*	*UA*
Wall glass		135	5.7	769.5
Wall	(90 × 5) − 135	315	1.3	409.5
Roof	30 × 15	450	1.5	675
Floor	30 × 15	450	1.0	450
		ΣA 1350		$\Sigma(UA)$ 2304

From equation (1.11):

$$F_2=1.26$$

From equation (1.13):

$$Q_p=[(0.91\times2304)+(0.33\times1.26\times2.5\times2250)]\ (18+2)$$
$$=88710\text{ W}$$

(b)
From equation (1.10):

$$F_1=1.11$$

From equation (1.11):

$$F_2=0.68$$

from equation (1.13):

$$Q_p=[(1.11\times2304)+(0.33\times0.68\times2.5\times2250)]\ (18+2)$$

$$=76394\text{ W}$$

A comparison of the results shows an increase in plant energy output for the system of unit heaters over the radiant system of 15%.

However, a comparison of temperatures is also revealing.

(a)
From equation (1.8):

$$0.91=\frac{t_{ei}+2}{18+2}$$

from which

$$t_{ei}=16.2\ ^\circ\text{C}$$

From equation (1.9)

$$1.26=\frac{t_{ai}+2}{18+2}$$

from which

$$t_{ai}=23.2\ °C$$

From equation (1.12):

$$18=\frac{1}{2}\times 23.2+\frac{1}{2}t_r$$

From which

$$t_r=12.8\ °C$$

(b)
From equation (1.8):

$$1.11-\frac{t_{ei}+2}{18+2}$$

from which

$$t_{ei}=20.2\ °C$$

From equation (1.9):

$$0.68=\frac{t_{ai}+2}{18+2}$$

from which

$$t_{ai}=11.6\ °C$$

From equation (1.12):

$$18=\frac{1}{2}\times 11.8+\frac{1}{2}t_r$$

From which

$$t_r=24.2°C$$

The solutions are tabulated for analysis in table 1.5. Note the variations in the indoor thermal indices in both cases. For a thermally comfortable environment these indices should be approaching comfort temperature t_c. Note the slightly increased fabric heat loss for the radiant system due to the

Table 1.5 Example 1.4: comparison of temperatures

System	*Q_p (kW)*	*Q_f (kW)*	*Q_v (kW)*	*t_c (°C)*	*t_{ei} (°C)*	*t_{ai} (°C)*	*t_r (°C)*	*t_{ao} (°C)*
Warm air	89	42	47	18	16.2	23.2	12.8	–2
90% radiant	77	50	27	18	20.2	11.6	24.2	–2

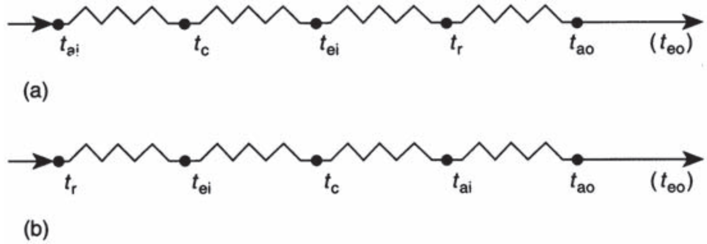


Figure 1.5 Heat flow paths for (a) convective and (b) radiant heating.

higher surface temperatures that will result with this type of heating system. The heat flow path from indoors to outdoors is different in each case and identifies from where the heat energy originates, namely from the air, t_{ai}, in (a) and from the surface (t_r being the mean value of the surfaces) in (b), as shown in Figure 1.5.

Example 1.5

If the same workshop is now considered to have an envelope constructed to current thermal insulation standards a further calculation and analysis may be undertaken. Revised thermal transmittance coefficients:

U_g=3.0, U_w=0.6, U_r=0.45 and U_f=0.45 W/m^{2}K.

Air change rate N=0.75 per hour.

Solution

(a)

From equation (1.10):

$$F_1=0.96$$

From equation (1.11):

$$F_2=1.12$$

From equation (1.13):

$$Q_p=31655 \text{ W}$$

(b)

From equation (1.10):

$$F_1=1.04$$

From equation (1.11):

$$F_2=0.87$$

From equation (1.13):

$$Q_p=30469 \text{ W}$$

Clearly the difference between the two outputs is insignificant, although an analysis of the constituents Q_f and Q_v is of interest: see Table 1.6.

The temperature interrelationships in (a) are as follows:
From equation (1.8):

$$0.96 = \frac{t_{ei} + 2}{18 + 2}$$

from which

$$t_{ei} = 17.2\ °C$$

From equation (1.9):

$$1.12 = \frac{t_{ai} + 2}{18 + 2}$$

from which

$$t_{ai} = 20.4\ °C$$

From equation (1.12):

$$18 = \frac{1}{2} \times 20.4 + \frac{1}{2} tr$$

from which

$$t_r = 15.6\ °C$$

The temperature interrelationships in (b) are calculated and found to be:

$$t_{ei} = 18.8\ °C,\ t_{ai} = 15.4\ °C \text{ and } t_r = 20.6\ °C$$

The solutions are tabulated for analysis in Table 1.6. Note that for both warm air and radiant heating the thermal indices are closer together, so comfort should be achieved with either system. The total value for Q_p in each case is very close, indicating that the proportions of convective and radiant heating are not important for a well-insulated building with low infiltration.

However, the constituents of fabric and ventilation loss do vary in each case, as they do more significantly for a poorly insulated building (Example 1.4).

Table 1.6 Example 1.5: solutions tabulated for analysis

System	*Q_p (kW)*	*Q_f (kW)*	*Q_v (kW)*	*t_c (°C)*	*t_{ei} (°C)*	*t_{ai} (°C)*	*t_r (°C)*	*t_{ao} (°C)*
Warm air	31.7	19.2	12.5	18	17.2	20.4	15.6	–2
Radiant	30.6	20.8	9.8	18	18.8	15.4	20.6	–2

Conclusion
The conclusion that may be drawn from these analyses is that for well-insulated buildings with air change rates less than 1 per hour the plant energy output is approximately the same for any system of heating regardless of the proportions of radiant to convective output.

However, for poorly insulated buildings and buildings with high air change rates the type of space heating assumes importance in the determination of plant energy output Q_p.

INTERMITTENT HEATING

The foregoing Examples 1.4 and 1.5 are based on the assumption that the heating plant operates continuously throughout the heating season. This is the case for factories working a system of three shifts, hospitals, airport terminals, residential homes etc. However, there many examples where plant operates intermittently, and this is likely to include night time and weekend shutdown.

Clearly equation (1.13) must be adjusted, as additional capacity will be required of plant and individual space heaters to provide a boosted output during the preheat period before occupation. Thus

$$Q_{pb}=F_3\times Q_p \tag{1.14}$$

F_3 is the **plant ratio,** and this is determined from

$$F_3=\frac{1.2(24-n)(f_r-1)}{24+n(f_r-1)}+1 \tag{1.15}$$

This may be described in words as

$$F_3=\frac{\text{design maximum heat requirement}}{\text{steady state design load}}$$

The **thermal response factor** f_r takes into account the thermal inertia of the building fabric. The greater the thermal storage capacity of the structure the higher will be the thermal inertia and the greater will be its mass. Thermal response is dependent upon the heat flow *into* the structure as well as the heat flow *through* the structure. These two characteristics are identified as **absorption** (admittance Y) and **transmission** (transmittance U). The admittance Y is determined from heat flow into the immediate layers on the inside of the structure. Its value is dependent upon the density and thermal storage capacity of these layers of fabric on the room side of the building envelope.

Admittance is not therefore easily calculated. Values of Y for different composite structures are given alongside the transmittance *(U)* values in section A of the *CIBSE Guide*.

Table 1.7 Thermal response factor f_r

Thermal response factor, f_r	Recommended preheat times (h)	
	Optimum start	Fixed start
<2.5	2	3
2.5–6.0	3	4
6.0–10.0	4	5
>10	5	6

The fact of the matter is that as heating plant becomes more intermittent in operation building heat loss becomes increasingly sensitive to heat flow *into* rather than heat flow *through* the fabric envelope.

Thus

$$f_r = \frac{\Sigma(AY) + 0.33NV}{\Sigma(AU) + 0.33NV} \tag{1.16}$$

High values of f_r indicate considerable thermal inertia and hence thermal storage capacity: see Table 1.7.

A building envelope having a thermal response factor of 3 indicates a lightweight structure. This would include timber frame with stud partitioning and lightweight infill panels in the envelope. A building constructed from concrete/brick/stone with lightweight infill panels in the envelope would be classified as medium weight and have f_r=6.

Values of f_r at around 10 indicate a structure built entirely with dense materials and having relatively small glazed areas.

As identified earlier, the location of the thermal insulation in the structure is decisive in putting a value on the thermal response factor f_r, and can have the effect of modifying a heavyweight building envelope with a high value of f_r.

It is therefore important to identify the location of the insulation slab in the structure, as it will also affect the admittance Y and hence plant ratio F_3. Look again at equations (1.14), (1.15) and (1.16).

Example 1.6

A heated building has a thermal response factor of 3 and is occupied continuously for 8 h per day. If the preheat period before occupancy is 3 h, determine the plant ratio F_3 that must be applied to the plant energy output Q_p for the building of 100 kW.

Solution

From equation (1.15):

$$F_3 = \frac{1.2\,(24-11)(3-1)}{24+11\,(3-1)} + 1$$

from which

$$F_3 = 1.2\times(26/46)+1$$
$$=1.68$$

From equation (1.14):

$$Q_{pb} = 1.68\times100 = 168 \text{ kW}$$

This represents the size needed for the boiler plant. Clearly, if the preheat period is extended to say 6 h, the plant ratio comes down in value and F_3 is determined as 1.46 and therefore Q_{pb}=146 kW.

It should also be remembered that this is the plant load required when outdoor temperature is at design.

The plant ratio (traditionally known as the plant margin or overload capacity) nevertheless does seem to be high but it should be borne in mind that this methodology has been developed with current insulation standards in mind. The effect of these standards has been to reduce significantly the net plant and space heater sizes from traditional values. The result is that on shutdown the building envelope will cool down and without the level of plant ratio F_3 the building envelope will take too long to reach optimum temperature after a shutdown period.

The exception is where thermal insulation is applied to the room-side surface, and here more traditional plant ratios of around 1.2 can be adopted with confidence.

Example 1.7

(a) A building measures 10×7×3.2 m high and has four windows each 2×1.5 m. Determine the response factor for the building and obtain the recommended preheat time for a fixed start.

Data

Coefficient	Wall	Roof	Floor	Glass	*N* air change
U	0.73	0.4	0.3	5.6	0.75
Y	3.6	2.8	2.5	5.6	

(b) Given the daily occupancy is 10 h determine the plant ratio F_3

Solution

(a) See Table 1.8.

Q_v per degree$=0.33NV=0.33\times0.75\times(7\times10\times3.2)=55.44$ W/K

From equation (1.16):

$$f_r = \frac{787+55.44}{187+55.44} = 3.47$$

Table 1.8 Example 1.7: determination of response factor

Element	*Dimensions*	*A*	*AU*	*AY*
Glass	$4 \times 2 \times 1.5$	12	67.2	67.2
Wall	$(34 \times 3.2) - 12$	96.8	70.7	348.5
Roof	10×7	70	28	196
Floor	10×7	70	21	175
		$\Sigma(AU)$	187 ΣAY	787

From Table 1.3 preheat will be about 4 h.

(b)
From equation (1.15):

$$F_3=1.2\times(24.5/58.3)+1$$
$$=1.5$$

HIGHLY INTERMITTENT HEATING

For systems operating for less than 12 h including preheat the criterion for heat flow changes. Heat flow *into* the structure (admittance Y) takes over from heat flow *through* the structure (transmittance U) in the plant energy output equation, thus:

$$Q_p=[F_1\Sigma(AY)+0.33NVF_2](t_c\text{-}t_{ao}) \tag{1.17}$$

Note that $f=\Sigma(AY)/\Sigma A$ now in the nomenclature for the calculation of F_1 and not $\Sigma(AU)/\Sigma A$, which applies when the thermal transmittance U is used in the plant energy output equation for Q_p.

A FINAL COMMENT ON ADMITTANCE

Admittance is determined primarily by the characteristics of the materials in the layers adjacent to the internal surface up to 100 mm. Moving the insulation slab from within the structure to its inner surface will have the effect of reducing the admittance and hence the thermal response f_r. The structure's thermal transmittance, however, will remain unchanged unless slabs of material are added or removed.

CONCLUSIONS

There are three modes of operation for plant:

- continuous;

- intermittent;
- highly intermittent.

Plant operating modes or time scheduling will depend upon the occupancy levels in the building and upon the outdoor climate.

Clearly, in severe weather when outdoor temperature is below design for extended periods it may be prudent to ensure that the client is advised to run normally intermittently operated plant continuously or for longer periods than normal. The plant ratio F_3 allows the plant to be operated under boost conditions during preheat. That is to say, the system flow temperature is elevated to increase the output of the space heating appliances during the preheat period.

If boost conditions during preheat are not a control option the plant ratio allows a more rapid warm-up period, which will be required in severe weather. The important lesson here is that elevating the boiler thermostat will not achieve greater output if the boiler has been sized on plant energy output Q_p alone. If the plant ratio/plant margin/overload capacity is not part of the boiler rating, elevating the boiler thermostat will result in the plant running continuously and not necessarily reaching its thermostat setting or achieving a higher output. Figure 1.6 illustrates the point.

1.4 Heat flow paths and conductance networks

It is now appropriate to attempt to reconcile the heat flow paths and conductance networks used as aids to identifying the behaviour of the building envelope to changes in outdoor climate and type of heating system. Figures 1.7 and 1.8 are in common use.

Indoor environmental temperature t_{ei} was adopted as the comfort index. It has

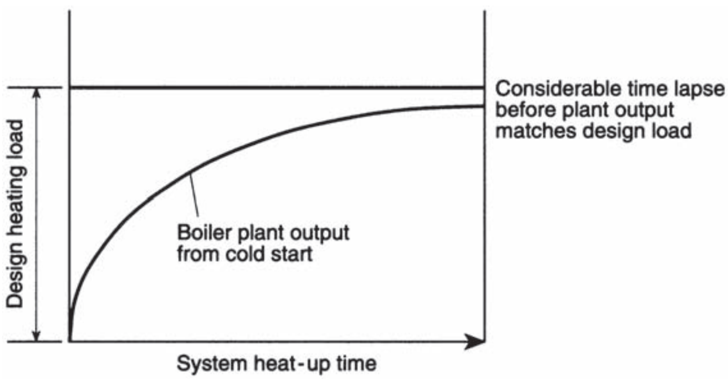


Figure 1.6 The effects of undersized plant on design load.

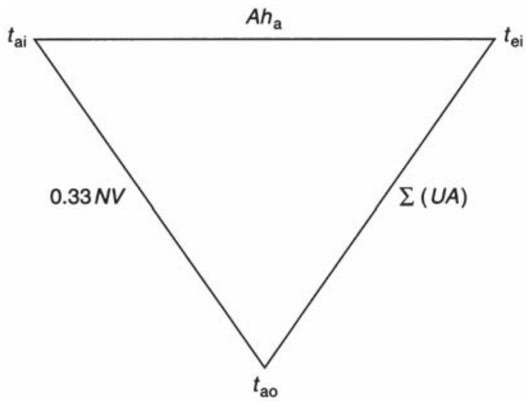


Figure 1.7 Basic conductance network.

fallen into disuse, as it is difficult to produce an instrument economically that measures environmental temperature. However, it is still used as a concept, and appears at the environmental point in Figure 1.7. For mainly radiant heating the heat flow path starts at the environmental point and moves anticlockwise to the indoor air point and then to the outdoor/environmental point.

Mean radiant temperature t_r will have the highest value: thus $t_r > t_e > t_a > t_{ao}$ because environmental temperature is weighted towards mean radiant temperature. Put simplistically:

$$t_e = \frac{2}{3}t_r + \frac{1}{3}t_a$$

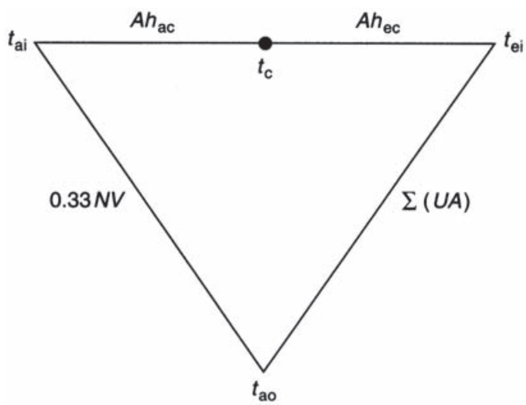


Figure 1.8 Modified conductance network.

A heat balance may be drawn for mainly radiant systems such that: heat flow from t_e to t_a=heat flow from t_a to t_{ao}
and

$$h_a \Sigma A(t_e - t_a) = 0.33NV(t_a - t_{ao}) = Q_v$$

Thus

$$t_a = t_e - \frac{Qv}{\Sigma A h_a} \tag{1.18}$$

For mainly convective heating the heat flow path commences at the indoor air point in Figure 1.7 and moves clockwise to the indoor environmental point and thence to the outdoor environmental point. Thus $ta > t_e > t_{eo}$.

The heat balance here is

$$\Sigma A h_a(t_a - t_e) = \Sigma(UA)(t_e - t_{eo}) = Q_f$$

from which

$$t_a = t_e + \frac{Qf}{\Sigma A h_a} \tag{1.19}$$

A similar approach is adopted for the modified conductance network (Figure 1.8). Here, with the introduction of comfort temperature t_c it is necessary to divide conductance h_a into h_{ac} and h_{ec} either side of the comfort point in the network. It can be shown that h_a=3/2h_c, where h_c is the average heat transfer coefficient for convection at normal room temperatures. Thus, h_a=3/2×3=4.5 W/m^{2}K.

Now

$$h_{ec} = 4h_a = 18 \text{ W/m}^2\text{K}$$

and

$$h_{ac} = \frac{4}{3}h_a = 6 \text{ W/m}^2\text{K}$$

From the modified conductance network (Figure 1.8), for radiant heating the heat flow path is clockwise from the environmental point and the heat balance will be

$$\Sigma A h_{ec}(t_e - t_c) = \Sigma A h_{ac}(t_c - t_a) \tag{1.20}$$

Similarly for convective heating the heat balance will be

$$\Sigma A h_{ac}(t_a - t_c) = \Sigma A h_{ec}(t_c - t_e) \tag{1.21}$$

It should be possible from these equations to determine t_a or t_e and achieve the same result as those obtained in Examples 1.4 and 1.5, where the temperature ratio equations for F_1 and F_2 were adopted.

Example 1.8
Determine the indoor environmental temperature for case (b) in Example 1.5 using equation (1.20) and the indoor air temperature for case (a) in Example 1.4 using equation (1.21).

Solution
Substituting the known data from Example 1.5 into the heat balance equation (1.20):

$$1350\times18\ (t_e\text{-}18)=1350\times6(18\text{-}15.6)$$

From which

$$t_e=18.87\ °C$$

This closely agrees with the solution for t_e from temperature ratio F_1 in Example 1.5.
Substituting the known data from Example 1.4 into the heat balance equation (1.21):

$$1350\times6\ (t_a\text{-}18)=1350\times18(18\text{-}16.2)$$

From which

$$t_a=23.4\ °C$$

This closely agrees with the solution for t_a from temperature ratio F_2 in Example 1.4, namely:

$$t_a=23.2\ °C$$

1.5 Practical applications

The methodology currently in use has now been sufficiently exposed here to allow solutions to practical applications. There is, however, one further equation to introduce at this point.

Mean radiant temperature t_r can be evaluated to a close approximation with the area-weighted enclosure temperature and adapted to suit the following case study.

$$t_r=\frac{A_p t_p + A_u (t_r)_u}{A_p + A_u} \qquad (1.22)$$

where

$$(t_r)_u = t_e - \frac{Q_f}{2h_a A_u} \qquad (1.23)$$

and

$$\Sigma A=A_p+A_u \qquad (1.24)$$

Case study 1.1

INTRODUCTION

A high-speed train (HST) shed is to be heated with radiant strip from high-temperature hot water having flow and return temperatures of 150 °C and 130 °C respectively. The comfort temperature is to be 16.5 °C when the outdoor temperature is -3 °C.

BRIEF

The HST shed is single storey and fully exposed, measuring 110 m× 17 m×8 m to eaves, 10 m to ridge. It has a total of 80 roof lights, each measuring 2×1.5 m. Thermal transmittance coefficients: wall 0.7, roof 0.5, floor 0.3, roof lights 5.7 W/m^2K.

Assume the end doors have the same transmittance coefficient as the wall and are normally closed.

Infiltration rate: 0.5 air changes per hour.

Radiant strip shall be by Dunham Bush or equal.

TASKS

1. Determine the plant energy output Q_p for continuous heating.
2. Calculate the area of radiant strip required for the flow and return temperatures specified.
3. Adopting the single-pipe configuration for the strip, determine the total length required and hence the number of lengths for the shed, assuming initially that the strip runs the full shed length.
4. Adopting a water velocity of 0.3 m/s in 32 mm black heavy-grade pipe (recommended by the strip manufacturer to achieve the quoted outputs), and using the specified temperature drop of 20 K, determine the maximum output of each strip.

   Using the appropriate output from the manufacturer's literature in W/m run for the mean water to comfort temperature difference, calculate the maximum length of each strip.
5. Assume initially that the radiant strip is located at eaves level. Determine the proportions of the heat loss above and below the heated plane (strip plane). Make an initial decision on the number of strips for the shed and determine the proportions of output above and below the heated plane.

   In the light of these solutions make a final decision on the number of strips.

6. Check with the manufacturer's literature on the strip spacing.
7. Draw a suitable radiant strip configuration for the shed and identify pipe connections, taking the main flow and return entry point at one corner of the building.
8. Recommend a method of temperature control assuming constant volume constant temperature in the flow main.
9. Determine the approximate load imposed on the roof by suspending the strip and connecting pipework from it.
10. Calculate the linear movement due to thermal expansion on each strip run and on the distribution pipework, and recommend how it might be accomodated.

SOLUTION

Clearly the solution relies on recourse to manufacturer's literature and pipe-sizing tables in the *CIBSE Guide*. You should obtain a copy of the manufacturer's brochure and have access to the pipe-sizing tables to assist in comprehending the solutions. Note that in this case the building is in continuous operation, and therefore plant energy output does not need correcting for intermittent operation. Heat loss will be by transmittance *through* the envelope, and hence values for admittance are not required.

Task 1

The surface area for the roof involves determination of the slope, which is 8.73 m. Surface areas of the envelope are:

$$\begin{aligned} \text{roof} &= 1921\ \text{m}^2 \\ \text{wall} &= 2032\ \text{m}^2 \\ \text{floor} &= 1870\ \text{m}^2 \\ \Sigma A &= 5857\ \text{m}^2 \end{aligned}$$

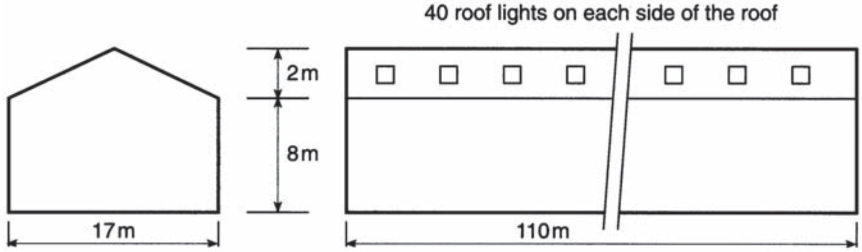


Figure 1.9 Case study 1.1: elevations of the high-speed train shed.

The plant energy output is determined from equation (1.13). Separating the fabric and ventilation losses:

$$Q_f=83 \text{ kW}$$

and

$$Q_v=52 \text{ kW}$$

Total:

$$Q_t=135 \text{ kW}$$

It will be seen from the manufacturer's literature that E=62%. K therefore has been taken from Table 1.3 as the same as 67% heat radiation, namely K=1.10, giving calculated values for F_1 as 1.014 and F_2 as 0.958.

Task 2

$$F_1 = \frac{t_e - t_o}{t_c - t_o}$$

Therefore

$$1.014 = \frac{t_e + 3}{16.5 + 3}$$

from which

$$t_e=16.8 \text{ °C}$$

$$F_2 = \frac{t_a - t_o}{t_c - t_o}$$

Therefore

$$0.958 = \frac{t_a + 3}{16.5 + 3}$$

from which

$$t_a=15.7 \text{ °C}$$

$$t_c = \frac{1}{2}t_a + \frac{1}{2}t_r$$

Therefore

$$16.5 = \frac{1}{2} \times 15.7 + \frac{1}{2}t_r$$

from which

$$t_r=17.3 \text{ °C}$$

Note the closeness of t_c, t_e, t_a and t_r.

Equation (1.23):

$$(t_r)u = t_e - \frac{Q_f}{2h_a(A)u} = 16.8 - \frac{83000}{2 \times 4.5(A)u}$$

Equation (1.24):

$$A_p = \Sigma A - A_u = 5857 - A_u$$

t_p=mean water temperature=140 °C
Substituting this data into equation (1.22):

$$17.3 = \frac{(5857 - A_u)140 + A_u(16.8 - 83\,000/(9\,A_u))}{5857}$$

from which

$$A_u = 5758\text{m}^2$$

Area of the heated plane:

$$A_p = 5857 - 5758$$
$$= 99\ \text{m}^2$$

Do you agree?

This is the required area of the heated plane at a mean temperature of 140 °C, namely the radiant strip, which is located at eaves level in the shed, to maintain a comfort temperature t_c of 16.5 °C when outdoor temperature is -3 °C.

Using Dunham Bush single strip of width 257 mm, the total length required will be

$$L = \frac{99}{0.257}$$
$$= 385\ \text{m}$$

This is equivalent to four lengths of strip running the length of the shed.

Task 4

The manufacturer uses 32 mm black mild steel tube to BS 1387. From the pipe sizing tables the mass flow rate of water at 0.3 m/s is 0.2611 kg/s. The maximum temperature drop for the HTHW system is given as 20 K: thus each strip output Q will be

$$Q = M.C.\text{d}t = 0.261 \times 4.2 \times 20 = 22\ \text{kW}$$

The manufacturer's data gives total output for 140 °C and 16.5 °C (say 120 K difference) as 434 W/m run.

Thus the maximum strip length

$$L = 22 \times \frac{1000}{434}$$
$$= 50 \text{ m}$$

This means that the radiant strip must be configured such that flow and return connections are made to each 50 m length to ensure that the output of 434 W/m run is achieved.

From the manufacturer's data, radiant strip comes in modules of 1.5 m: thus 33 modules make up each strip length at 49.5 m.

Task 5

The radiant strip will be insulated on its upper surface, and from the manufacturer's data 62% of the output is downward in the form of heat radiation, leaving 38% directed upwards into the roof void as heat convection. It is therefore necessary to determine the proportion of plant energy output above the heated plane and compare this with the upward emission from the strip. (Table 1.9). From equation (1.13):

$$Q_p = [(1.014 \times 2232) + (0.33 \times 0.958 \times 0.5 \times 1870)]\ (16.5 + 3)$$
$$= 49900 \text{ W}$$

From task 3, L=385 m; from task 4, maximum strip length=49.5 m. The number of strips=385/49.5=7.8.

Clearly for the sake of symmetry, eight strips would be appropriate in two banks of four. On this basis upward emission from the strip will be:

$$= (0.38 \times 434)\ (8 \times 49.5)$$
$$= 65308 \text{ W}$$

This is higher than the heat loss and the expected temperature above the strip plane can be calculated by equating this with the heat loss above the heated plane:

$$65308 = (2263 + 296)(t_c + 3)$$

from which

$$t_c = 22.5 °C$$

The heat loss from the roof void above the strip plane could therefore be equal to the upward emission from the strip, which is 66308 W.

Table 1.9 Case study 1.1, task 5

Element	*Area*	*U*	*UA*
Roof lights	240	5.7	1368
Roof	1681	0.5	840
Gable ends	34	0.7	24
			ΣUA 2232

Heat loss below the strip plane will be 135000-49900=85100 W. Strip output in the form of heat radiation downwards will be:

$$=(0.62\times434)(8\times49.5)$$

$$=106556\text{ W}$$

If the system had no room temperature control the temperature below the strip plane would also therefore exceed design value of 16.5 °C.

The solutions based upon 8×49.5 m strips are tabulated in Table 1.10. Analysing the results in the table and assuming that suitable room temperature control is provided (asterisked in Table 1.10), excessive roof void temperatures will be avoided if the ratio of heat loss to strip emission is the same both above *and* below the strip plane.

Above the strip plane:

$$\frac{49\,900}{65\,308}=0.764$$

Below the strip plane:

$$\frac{85\,100}{106\,556}=0.799$$

As the ratio above the strip plane is lower than that below, the calculated roof void temperature of 22.5 °C should not be reached during plant operation, and the potential building heat loss of 150408 W will not occur.

If the ratio was reversed the temperature above the heated plane would rise above design, and it would be prudent to consider ways of avoiding the possibility. For example, the heated plane could be lowered from eaves level.

Table 1.10 Case study 1.1, task 5

Heat flow	Q_p	t_c	*Strip output*	*Potential heat loss*
Heat loss above plane	49 900	16.5		65 308
Emission above plane		22.5	65 308	
Heat loss below plane	85 100	16.5		85 100
Emission below plane		16.5*	106 556	
Totals	135 000		171 864	150 408

This would alter the proportions of upward and downward heat loss. Alternatively, destratification fans could be located in the roof void.

The final decision on the number of single radiant strips is therefore 8×49.5 m. The equivalent area of the heated plane is:

$$8\times49.5\times0.257=102 \text{ m}^2$$

This compares favourably with the theoretical area, A_p=99 m^2, calculated in task 2.

Task 6

As a general rule of thumb the spacing of radiant strip should be around half the mounting height, which is 8 m. The HST shed is 17 m wide, and with four strips the spacing is approximately 3.4 m, which is acceptable.

Task 7

The layout for the system needs consideration in two respects: temperature control, and the need for ease in hydraulic balancing. You should consider these two points yourself and attempt to lay out a pipe and strip system for the shed, then turn over the page and look at a potential solution.

Temperature control could be achieved via two-port modulating valves located in the flow to each pair of banks of radiant strip (not shown in the sketch plan), each responding to a thermostat suitably located in the shed. A reverse return pipe layout is shown to ease hydraulic balancing.

A balanced pressure valve would need to be considered in the event of all four modulating valves being closed if the pump is dedicated to this system. Refer to Chapter 3 for the application of balanced pressure valves.

Do you agree with the proposed layout and the method of temperature control for the banks of radiant strip?

Task 8

Temperature control is described in the solution to task 7. Each valve would be controlled by a suitably located thermostat with blackened sensing bulb.

Task 9

The load on the roof structure is left for you to undertake. You will find it is in excess of 3 t.

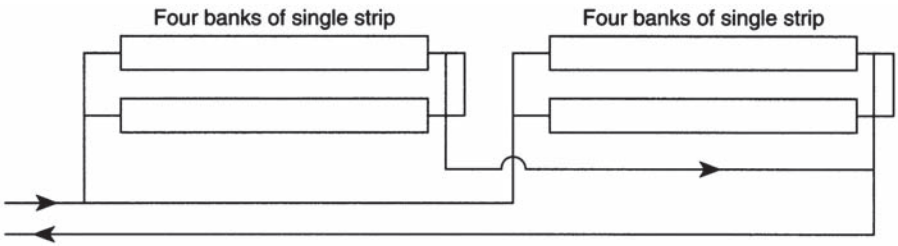


Figure 1.10 Case study 1.1: sketch in plan of the proposed layout for the eight banks of radiant strip.

Task 10

The coefficient of linear expansion for mild steel is 0.000012 m/m K. For the pipework running the length of the shed, linear expansion will be:

$$\text{coefficient} \times \text{length} \times (\text{mean temperature-fill temperature})$$

$$\text{expansion} = a \times L \times \mathrm{d}t \tag{1.25}$$

Thus

$$\begin{aligned} \text{linear expansion} &= 0.000012 \times 110 \times (140-10) \\ &= 0.172 \text{ m} \\ &= 172 \text{ mm} \end{aligned}$$

For the banks of strip:

$$\begin{aligned} \text{linear expansion} &= 0.000012 \times 50 \times (140-10) \\ &= 0.078 \text{ m} \\ &= 78 \text{ mm} \end{aligned}$$

If expansion devices like axial compensators or expansion loops are used, account must be taken of the anchor loads sustained by the building structure.

Alternatively, the pipework can be suspended in such a way as to allow free movement. However, if this approach is adopted particular attention must be given to venting the air from the pipework and radiant strip.

CONCLUSION

The tasks and solutions to the above assignment represent a practical approach to design. You will have noticed in both that there are two parts to the solutions for assessing the space heating requirements: first the calculation of the energy needs for the building and then the determination of the selected space heating output. In this case study the two closely agree. In other situations a compromise must be made between the theoretical energy need and the practical output from the chosen space heaters.

In the next case study you will find that such a compromise has to be made.

FURTHER STUDY

An alternative design using gas-heated radiant tube would be worthwhile. This is left for you individually or in a group to pursue as an assignment after acquiring appropriate manufacturers' literature. Remember to start by determining the area of the heated plane, A_p, using average tube temperature, as it will be different from the above solution.

Case study 1.2

INTRODUCTION

A workshop is to be heated from steam supplied at 2.0 bar gauge and 0.9 bar dry using unit heaters of the recirculation type. Design conditions are: comfort temperature of 18 °C when outdoor temperature is -3 °C.

The occupancy of the workshop will be intermittent with two consecutive 8 h shifts a day.

BRIEF

The workshop, which is fully exposed, measures 40×30×5 m to eaves, with two roof ridges running the length of the building to form two

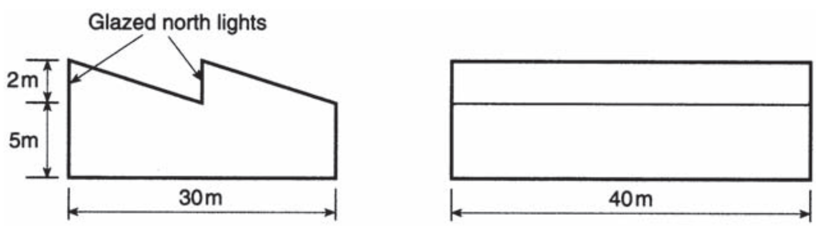


Figure 1.11 Case study 1.2: elevations of the factory.

continuous vertical roof lights 2 m in height. One of the roof lights is located along the workshop axis (Figure 1.11). The infiltration rate N=0.5 h^{-1} and the thermal transmittance coefficients are: U_g=6.0, U_r=0.5, U_w=0.7 and U_f=0.4 W/m^2K The admittance coefficients are specified in case they are required: Y_g=6, Y_r=2.0, Y_w =4 and Y_f=6 W/m^2K. Assume the doors have the same *U* and *Y* values as the walls. The unit heaters shall be by Dunham Bush or equal and of the recirculating type. As a general rule of thumb the air recirculated by the heaters should be equivalent to between 4 and 5 air changes per hour. This ensures adequate circulation of warmed air.

TASKS

1. Adopting the plant energy output routine, determine the heat loss from the building envelope at design conditions and then calculate the boosted plant energy output for the required level of occupation.
2. Determine the anticipated indoor air temperature. Comment upon the relationship between this and dry resultant, mean radiant and environmental temperature.
3. Select and determine the number of unit heaters required from the data given in the manufacturer's literature. Take account of the vertical temperature gradient, floor coverage/throw, EAT/LAT and air change rate of the selected heaters.
4. Undertake a comparison of the total heater output and the boosted plant energy output, and make a decision on the final number and type of unit heaters for the workshop.
5. Recommend a suitable temperature control for the unit heaters.
6. Prepare a sketch layout of the heater locations and steam and condense distribution mains, showing levels and steam traps.
7. Size the steam and condense mains back to a point at high level in the corner of the workshop.

Table 1.11 Case study 1.2, task 1 . Data relating to building heat loss:

Element	*Dimensions*	*A*	*U*	*UA*
wall	140 × 5	700	0.7	490
floor	40 × 30	1200	0.4	480
roof	2 × 15.133 × 40	1211	0.5	606
glass	2 × 2 × 40	160	6.0	960
		Σ*A* 3271		Σ(*UA*) 2536

ALTERNATIVE BRIEF/TASKS

3. Select and determine the number of indirect gas-fired air heaters, floor or roof mounted, from chosen manufacturer's literature.
6. Prepare a sketch layout of heater location and fuel supply lines.
7. Size the fuel line back to a point at low level in the corner of the workshop.
8. Prepare a typical flue detail.
9. Account for the air required for combustion.

SOLUTION

As with the first case study, recourse is needed here to CIBSE pipesizing tables and manufacturer's literature for you to participate fully.

Task 1

Data relating to building heat loss are tabulated in Table 1.11. As there is zero heat radiation, E=0 and K=0. From equation (1.10):

$$F_1 = \frac{18}{18 + f}$$

where f=(UA)/?A=2535/3271=0.775.
Thus:

$$F_1=0.96$$

and since (equation (1.11)): F_2=4-3F_1, then

$$F_2=1.12$$

Substituting into the plant energy output (equation (1.13):

$$Q_p=[(0.96\times2535)+(0.33\times1.12\times0.5\times7200)]\ (18+3)$$
$$=79044 \text{ W, say } 80 \text{ kW}$$

This now needs adjusting for intermittent heating, and from equation (1.16), thermal response

$$f_r = \frac{\Sigma(AY) + 0.33NV}{\Sigma(AU) + 0.33NV}$$

Heat flow *into* the structure is tabulated in Table 1.12. Substituting:

$$f_r=3.91$$

From Table 1.7, preheat period for a fixed start=4 h.

We can now determine the plant ratio F_3 from equation (1.15):

$$F_3 = \frac{1.2\,(24-20)(3.91-1)}{24+20(3.91-1)} + 1$$
$$= 1.17$$

Table 1.12 Case study 1.2, task 1. Heat flow into the structure

Element	*A*	*Y*	*AY*
Wall	700	4.0	2800
Floor	1200	6.0	7200
Roof	1211	2.0	2422
Glass	160	6.0	960
	Σ*A* 3271		Σ(*AY*) 13382

Thus boosted plant energy output

$$Q_{pb}=F_3\times Q_p=1.17\times 80=93.6 \text{ kW}$$

Note how the length of the heating period has a significant effect on the plant ratio F_3. Shortening the preheat to 2 h gives a plant ratio of 1.274, and Q_{pb}=102 kW. Nevertheless, the ratio is not as significant as in earlier examples. This is due to the length of the occupancy period, which here totals 16 h and not the 8 or 9 h for an office.

Task 2

Employing the temperature interrelationships for F_1 and F_2, t_e is evaluated as 17.16 °C and t_a as 20.52 °C. From the equation for comfort temperature t_r is evaluated as 15.48 °C. With t_c at 18 °C the thermal indices are within a range of 5 K, which is likely to be satisfactory.

There is no guidance in the form of comfort zones for non-sedentary occupations, owing to the variation in levels of activity and its effect on body heat loss.

With a range of 5 K and with the comfort temperature lying in mid position, comfort should be achieved as long as t_c is selected with the level of activity in mind. By this reasoning, Case study 1.1 with t_c at 16.5 °C implicitly has a higher level of activity than Case study 1.2 with t_c at 18°C.

Task 3

With reference to the manufacturer's literature you will see that there are two types of unit heater: the horizontal discharge model and the vertical discharge unit. Either model is acceptable, and it would be sensible for you to consider two proposals. This type of heating relies on the air as the heat-carrying medium, and therefore the heaters must be capable of discharging the air so that the whole of

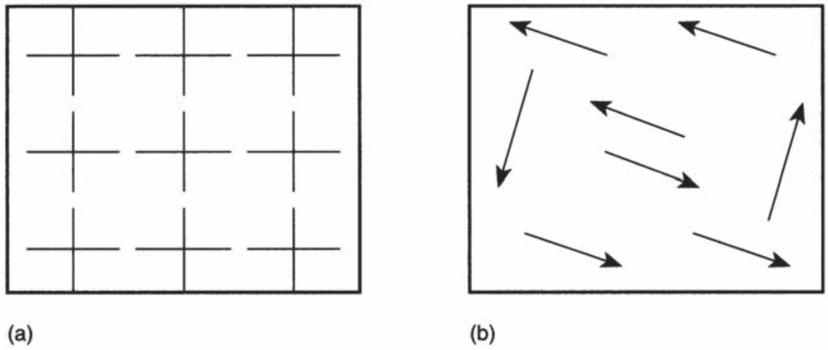


Figure 1.12 (a) Downward discharge unit heaters; (b) horizontal discharge unit heaters.

Table 1.13 Case study 1.2, task 3. Performance specification for the basic model.

Medium	*Model*	*Output (kW)*	*EAT (°C)*	*LAT (°C)*	*Air volume (l/s)*	*No. of heaters (l/s)*	*No. of heaters (kW)*
Steam @ 2bar.g	LH25	20.3	20	49	610	13	5
"	LH40	35.9	20	49	1055	8	3
"	LH63	62.4	20	54	1610	5	2
"	LH100	105.2	20	53	2835	3	1

the working plane is covered. There will otherwise be a tendency for cold spots to occur where the warmed air does not reach. The two sketches in Figure 1.12 should clarify the point.

In both proposals the greatest heat loss is at the perimeter of the building; this should be reflected in the location of the unit heaters. Downward discharge may incur potential cold spots outside the circular floor coverages. The manufacturer should be able to give you coverage against mounting height for their vertical discharge units. Horizontal discharge shows the throw to scale from each horizontal discharge unit. A substantial gap between the throw limit and the adjacent heater identifies the potential cold spots. Note too that the air movement is in one direction for the horizontal units: this will assist warmed air circulation.

From Dunham Bush data, the performance specification for the basic model is given in the first six columns of Table 1.13. Knowing that volume flow rate is given by

$$\mathrm{VFR} = NV/3600 \qquad (1.26)$$

and taking air recirculation as 4 per hour to satisfy the required air movement rule of thumb for air heating:

$$\mathrm{VFR} = 4 \times \frac{7200}{3600} = 8\ \mathrm{m^3/s}$$
$$= 8000\ \mathrm{l/s}$$

Dividing this by the air volumes in column 6 gives the minimum number of heaters in column 7 required to satisfy the rule of thumb.

Dividing the boosted plant energy output of 93.6 kW by the model outputs in column 3 gives the number of heaters required to offset the building heat loss. You can see at a glance that the last two columns do not agree, and we have not yet considered the throw/coverage requirements to prevent cold spots occurring in the workshop.

This is a prime example of design calculation and practical application appearing irreconcilable.

Clearly the first option might be to consider another manufacturer in the hope that a closer match will occur. It is likely, however, that the improvement will be marginal. It is apparent that if the number of heaters is based upon the boosted plant energy output of 93.6 kW the throw/coverage for the workshop will be inadequate. This can be ascertained from the manufacturer's literature.

It follows therefore that to satisfy the throw/coverage criterion and the rule-of-thumb air recirculation rate, a number of unit heaters will be needed, and it is likely that twelve LH25, eight LH40 or six LH65 models will have to be selected. If LH25s are employed, output will be 244 kW; if LH40s are selected output will be 287 kW; and if LH63s are chosen output will be 374 kW.

It is at this point that the young design engineer gets cold feet. The thought of oversizing the space heating system by a factor of 3 appears like suicide. However, for the reasons given, basing selection on the boosted plant energy output alone will not heat the workshop at the working plane, which is usually taken as 1 m from the floor. As long as the throw/coverage criterion is met, any one of the three options can be considered, although it may be argued that the first option (using the LH25s) gives rise to a densely populated system of heaters!

There are a few other issues worth mentioning here.

- EAT will be higher at the heater intake than at the working level: hence the use of 20 °C when using the manufacturer's literature. Warm-air heating gives rise to a vertical temperature gradient, and half a degree can be added for each metre of vertical height.
- LAT should be below 50 °C, as air is very buoyant at high temperature and is difficult to get down to the working level. This makes the selection favourite the LH40 model.
- Mounting heights are important and must be adhered to: refer to the manufacturer's literature. Do not be put off by the client wanting the units

located in the roof to get them out of sight! The warmed air will simply not reach the working level.
- Consider the use of destratification fans located above the unit heaters in the roof space. There will be a saving in running costs.

Task 4

The comparison of boosted plant energy output with total heater output using eight LH40s is as follows:

$$Q_{pb}=93.6 \text{ kW}$$

$$\text{Total heater output}=8\times 35.9=287 \text{ kW}$$

This is three times the boosted heat loss.

Task 5

As with any fan-assisted heater, a room thermostat controlling the fan provides adequate temperature control. Whether the unit heater fans are linked to two or four room thermostats, suitably located, or whether each heater is controlled by its own thermostat is a matter for discussion. When the fans are off heat, output is cut by about 90%.

Clearly, with such a high overload capacity the unit heaters will be operating intermittently.

The heating medium, which here is steam, does not require control. The steam trap on each unit heater return will respond to the variations in rates of condensate formed.

Task 6

This task is left for you to complete.

It is important to note that you cannot balance a steam pipe network as you can a water system. Regulating a steam supply with a valve simply causes erosion of the valve seat. The physical layout of the steam pipework is therefore important in order to achieve a balanced system.

Task 7

Design procedures for steam and condense pipe sizing are given in Chapter 5, which should be read and understood before proceeding with this task, which is left for you to undertake.

CONCLUSION

With this second case study the design heat loss for intermittent plant operation emphatically is not reconciled with the design requirements of the space-heating system, and a compromise in favour of the space-heating system output is taken in the interests of good engineering practice. It is hoped that you can now put a strong defence in favour of the final decision here.

FURTHER STUDY

The alternative brief refers to the use of indirect gas- or oil-fired heaters. This is left for you to do on your own or in a group, following acquisition of the appropriate manufacturer's literature.

It would also be useful to cost out both systems, assuming that an adequate supply of both steam at 2.0 bar gauge and natural gas is available on site at the entry point into the workshop.

It is likely of course that the indirect gas-fired system will prove cheaper to install, but it will not be making use of the steam that is available. The client would no doubt want to make a contribution to the proposals at this point.

Case study 1.3

Listed below are four values for the thermal response factor f_r for various building envelopes along with two occupancy levels. Identify the type of building envelope in terms of its thermal storage capacity in each case (light, medium, heavy) and determine the plant ratios $F3$. The preheat times given are based on optimum start control (See Table 1.7). Tabulate the solutions in a presentable manner. Analyse the solutions for $F3$ and determine the effect of halving the preheat times.

This case study is left for you to undertake.

DATA

(a) f_r=1.5, preheat time=2 h
(b) f_r=8, preheat time=4 h
(c) f_r=11, preheat time=5 h
(d) f_r=3, preheat time=3 h

Assume in each case an occupancy of 8 h and then an occupancy of 16 h.

1.6 Chapter closure

This completes the work on heat requirements of heated buildings in temperate climates. You will now be competent to undertake simple thermal modelling of the building envelope, space heater selection and boiler plant analysis with respect to the need for space heating.

Radiator selection is based upon plant energy output and does not require the detailed analysis adopted in Case Studies 1.1 and 1.2.

2 Low-temperature hot water heating systems

Nomenclature

C	specific heat capacity (kJ/kg K)
CTVV	constant temperature variable volume
CVVT	constant volume variable temperature
d	pipe diameter (m)
dh	difference in head (m)
dp	pressure difference (kPa)
dp_s	identified pressure difference around system (kPa)
dp_v	pressure difference across valve (kPa)
dt	temperature difference (K)
EL	equivalent length for fittings (m)
f	frictional coefficient
F & E	feed and expansion
g	gravitational acceleration (9.1 m/s^2)
HTHW	high- temperature hot water
k	velocity pressure loss factor
k_t	total velocity pressure loss factor
K_v	flow coefficient
L	straight length (m)
l_e	equivalent length when k=1.0 (m)
LTHW	low-temperature hot water
M	mass flow rate (kg/s)
MTHW	medium-temperature hot water
MWT	mean water temperature
N	valve authority
pd	pressure drop (Pa, kPa)
Q	output, load (kW)
TEL	total equivalent length (m)
u	mean velocity (m/s)
VFR	volume flow rate (l/s, m^3/s)
VLTHW	very low-temperature hot water
VTVV	variable temperature variable volume
ρ	density (kg/m^3)
Σ	sum of

2.1 Introduction

Water has been used as a liquid heat-carrying medium for a considerable time, owing to its availability and high specific heat capacity of approximately 4.2 kJ/kgK, compared with 2.2 for low-viscosity oil and 1.01 for air.

Organic and mineral liquids are available as liquid heat-transporting media having the advantage of higher evaporation temperatures than water at similar pressures, and minimal corrosion properties, although they have lower specific heat capacities. However, they are more expensive and are therefore rarely used in building services. Table 2.1 identifies arbitarily the distinctions between water-heating systems operating at different temperatures.

The decision over which level of temperature to use will depend on a number of factors. However, it is important to note that the output of the heat exchanger is dependent upon the magnitude of the difference between mean surface (system) temperature and room temperature, so the higher the system temperature the greater will be the output of the heat exchanger.

Conversely, the lower the system temperature the larger will be the heater exchanger surface for the same output. Low surface temperature radiators, for example, are therefore comparatively large.

2.2 Space heating appliances

It is assumed that you are aware of the common types of heat exchanger on the market and have a knowledge of their construction and operation. If this is not the case, consult manufacturers' literature.

Heating appliances will employ one or two modes of heat transfer in varying proportions, which you need to identify as this can have a significant effect in the design process. This is analysed in Chapter 1.

Table 2.2 lists some of the more common space-heating appliances.

It is important to know where to locate the different types of appliance and what applications are appropriate. Table 2.3 lists some examples.

Table 2.1 Water heating systems

System	*Flow temperature (°C)*	*dt (K)*	*Supporting pressure (bar gauge)*
VLTHW	50	10	–
LTHW	85/80	12/10	–
MTHW	120	15/20	2.7
HTHW	180	30/50	10 +

Table 2.2 Common space heating appliances

Indirect	*Direct*
Radiators	Oil- or gas-fired forced/draught heaters
Natural draught convectors	Radiant vacuum gas heaters
Unit heaters	Oil- or gas-fired radiant heaters
Forced draught convectors	Electric natural draught convectors
Pipe coils	Electric tubular heaters
Embedded pipe coils	Electric underfloor resistance cables
Radiant panels	Electric quartz heaters
Radiant strip	Electric ceramic heaters
Convector radiators	Electric forced draught heaters
Continuous convector	

DISTRIBUTION PIPEWORK

The heating medium requires transporting from the heat source to the space-heating appliances. The distribution pipework normally consists of two pipes: a flow and a return. However, there are two other forms of distribution, and a special application of two-pipe distribution. (Figure 2.1).

Two-pipe and reverse return distribution provide a constant water temperature throughout the system. Series and one-pipe do not, and the

Table 2.3 Typical applications

Radiators/natural draught convectors:	*Unit heaters:*
Offices	Factories
Schools	Kitchens
Private dwellings	Canteens
Hostels	Garage workshops
Hotels	
Restaurants	
Hospitals	
Forced draught convectors:	*Embedded pipe coils:*
Entrances/foyers	Entrance halls
Assembly halls	Residential
Dining rooms	Public areas
Rooms with limited wall space	Libraries
Stairwells	Museums
Committee/boardrooms	
Radiant strip/panels:	*Luminous heaters (e.g. quartz):*
Factories	spot heating (eg. churches,
Aircraft hangars	workshops)
Loading bays	

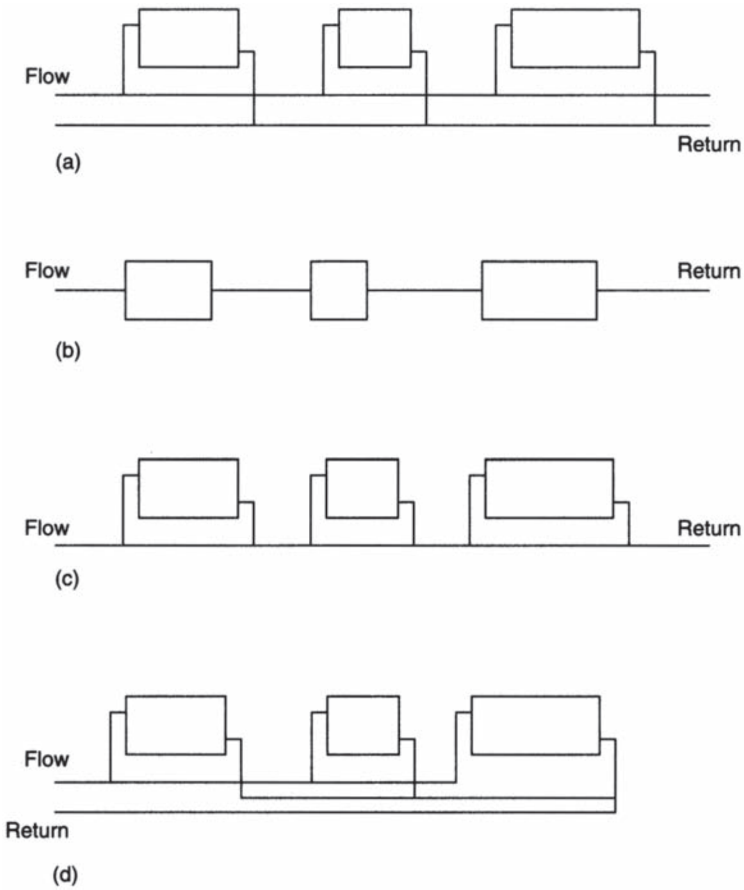


Figure 2.1 Heating distribution pipework: (a) two-pipe distribution; (b) series distribution; (c) one-pipe distribution; (d) reverse return or equal travel.

system mean water temperature reduces as one moves away from the heat source. This means that appliances must increase in size to maintain the same output.

It follows that two-pipe distribution and reverse return, as well as being preferable in most cases, offer easier design procedure.

Reverse return uses the most pipework, as three pipes are associated with each appliance. However, it reduces the problems of hydraulic balancing, as each terminal is the same pipe distance from the pump. It is adopted when large numbers of inaccessible heat exchangers are connected, as in the case of a system of air conditioning employing ceiling-mounted fan coil units.

THE HEAT SOURCE

This of course refers to the generator or boiler, of which there are numerous types, and again you are expected to know some of the more common ones or to refer to manufacturers' literature. With fuels such as gas, coal and oil, the boiler provides the facility for combustion to take place with primary and secondary air and heat transfer by conduction, radiation and convection from the furnace and flue gases to the water via heat exchangers, which are specially designed and located in the boiler to promote the maximum thermal efficiency from maximum load to a turndown ratio of around 30% of boiler output.

Further consideration to boiler plant is given in Chapter 6.

PRIME MOVER

Circulating pumps are used to transport the water around the system: see Chapter 3.

2.3 Pipe sizing

Pipe-sizing procedures for hot pipes are complicated by the effect of pipe emission. This falls into two categories: useful, as in the case of low-level pipes under radiators; and unusable, as instanced by pipework distributed in enclosed ducts or at high level in a factory.

The effect of pipe emission in a pipe network is progressively to reduce the temperature drop between flow and return as one moves away from the heat source. Thus it might be 12 K at the generator and 7 K at the index terminal. As mass flow rate is determined from

$$M = \frac{Q}{C\mathrm{d}t}$$

this also effectively increases the rate of flow M progressively from what it would be if dt was constant.

Initially pipe emission is an unknown, as the pipe sizes are unknown. It therefore follows that an estimate of the sizes of pipes in the network must initially be made so as to assess the potential emission. A knowledge of which pipes are lagged and which are unlagged is also required.

Alternatively, pipe emission may be estimated as a percentage of terminal outputs and pipe sizing undertaken as a preliminary process, with system hydraulic balancing used to iron out the inaccuracies.

THE INDEX RUN

Although terminals may be connected in series, one-pipe or two-pipe configurations, the space-heating network will consist of circuits in parallel. The

index circuit therefore is that circuit initially having the greatest pressure drop: usually the longest circuit in the network.

ALLOWANCE FOR FITTINGS

The hydraulic losses sustained when water is forced around the network result from that in straight pipe, pipe fittings, control valves plant and the terminals. Losses are usually converted into equivalent lengths of straight pipe, at least for the fittings. This is achieved either by approximation using a percentage on straight pipe or by converting the fittings to equivalent lengths of straight pipe from

$$EL=(k_t \times l_e) \quad \text{(m)}$$

for each pipe section and total equivalent length from

$$TEL=L+EL \quad \text{(m)}$$

Values for l_e are obtained in the CIBSE pipe-sizing tables alongside those of mass flow rate M. Velocity pressure loss factors k for pipe fittings are also found in the CIBSE pipe-sizing tables.

PRESSURE DROP DUE TO HYDRAULIC RESISTANCE OF STRAIGHT PIPE AND FITTINGS

Pressure drop for each pipe section:

$$\mathrm{d_p}=\mathrm{pd/m} \times TEL \quad \text{(Pa)}$$

INDEX PRESSURE DROP

This is calculated from the sum of the pressure drops in each pipe section forming the index run and the hydraulic losses through control valves, plant and the index terminal on that circuit.

PUMP DUTY

This includes the calculated index pressure drop and the total system flow rate for which the pump is responsible. Further work is done on pumps in Chapter 3.

ECONOMIC PRESSURE LOSS FOR PIPE SIZING

For a given rate of flow, the choice of small pipe sizes results in high rates of pressure loss and hence pumping costs. The choice of large pipe sizes for the same rates of flow increases capital costs.

An 'economic' pipe size is said to be achieved at around 300 Pa/m. However, the index run is sized below this at around 250 Pa/m to avoid excessive balancing resistances on the branches. Branches may be sized up to the limiting velocity, which will give pressure drops in excess of 300 Pa/m.

MAXIMUM WATER VELOCITIES IN PIPES

These are up to 1.5 m/s for steel and 1.0 m/s for copper, or above 50 mm diameter they are 3.0 m/s for steel and 1.5 m/s for copper.

The limiting velocities are in response to the unwanted generation and transmission of noise at higher values. Noise generation may not be critical in applications with relatively high background noise levels, however.

Low water velocities give rise to poor signals on commissioning valves.

PIPE-SIZING PROCEDURE

Having calculated the mass flow rates in each pipe section of the network it is useful to identify each section with a number for reference purposes and then to identify the number of circuits in the system. The sizing procedure below is then followed.

1. Select the index run and index terminal.
2. Size the index circuit.
3. Assess the index pressure drop.
4. Assess the pressure drop at the branches.
5. Size the subcircuits.
6. Assess the balancing resistances.
7. Determine net pump duty.

APPROXIMATE PIPE SIZING

This is sometimes done as a short-cut procedure for pipe sizing by taking a pd of say 300 Pa/m throughout the pipe network and using a constant dt of say 10 K. Table 2.4 shows the heat-carrying capacities for different pipe sizes based on these criteria. Inaccuracies are accounted for in the process of hydraulic balancing.

If a more detailed analysis is required, the approximate pipe sizes can be identified from Table 2.4 and an estimate made of the pipe emission in each pipe section of the system when recourse is made to the appropriate pipe emission tables in the *CIBSE Guide*. The detailed analysis can then proceed by

Table 2.4 Approximate pipe sizes for transporting heat loads in kW using water at 75 °C and a temperature drop of 10 K

Pipe diameter, d (mm)	*Mass flow rate, M (kg/s)*	*Pressure drop, pd dp (Pa/m)*	*Velocity, u (m/s)*	*Load, Q (kW)*
15	0.092	300	0.5	3.9
20	0.213	300	0.7	9.0
25	0.393	300	0.75	16.5
32	0.855	300	0.9	36.0
40	1.3	300	1.0	54.6
50	2.47	300	1.2	104.0
65	5.0	300	1.5	210.0

apportioning the pipe emission to each pipe section in the network. Have a look at Case study 2.4.

2.4 Circuit balancing

If a heating system is left unbalanced the design flow rate will not be achieved in the index run, as most of the water will flow through the shortest circuit. The objective, with the assistance of regulating valves, is to make each circuit in the system equal in pressure drop to that in the index run. The pump is not then able to favour any one circuit. This will ensure that design flow rates are achieved in the index circuit and all the subcircuits in the system. This process is also called **hydronic balancing**.

Case study 2.1

Figure 2.2 shows diagrammatically an LTHW heating system serving five terminals. Mass flow rates are given as 0.63 kg/s at each terminal, and the pd across each terminal is 5 kPa. The pd through the boiler plant is 20 kPa, and an allowance for fittings on straight pipe of 25% is to be made. Adopting the pipe-sizing tables from the *CIBSE Guide* for black heavy-grade tube, follow the sizing procedure outlined above to size and balance the system.

SOLUTION

1. Identify the index run

The lengths of pipe in each section are given in the tabulated data. The index run will consist of sections 5, 6, 7, 8, 9. The other circuits consist

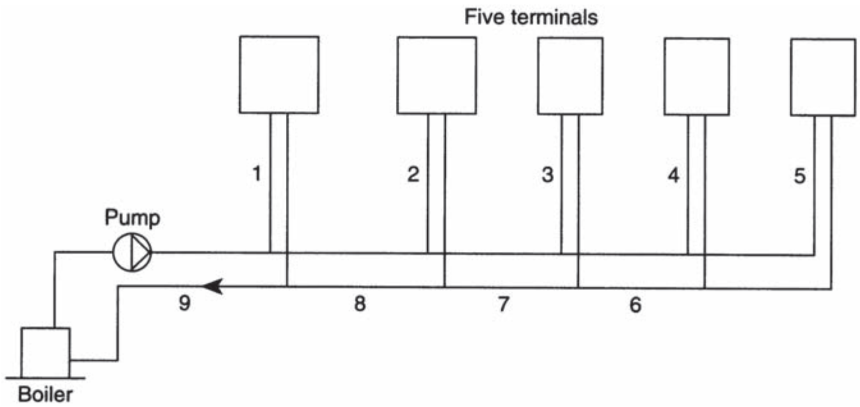


Figure 2.2 Case study 2.1: LTHW system. Note: cold feed, open vent and f&e tank omitted.

of sections: 4, 6, 7, 8, 9; 3, 7, 8, 9; 2, 8, 9; and 1, 9. There are therefore five circuits in the system.

2. Size the index circuit

The index circuit can now be sized at around 250/300 Pa/m. The results are shown in Table 2.5.

3. Assess the index pressure drop

The index pd can now be determined by summing up the section pd's and adding the plant and index terminal pd. The pd in each pipe section is calculated as shown above and included in the table.

Sum of section pd's=22561 Pa

Plant and index terminal pd=25000 Pa

Total index pd=47.561 kPa

4. Assess the pd at the branches

The pd available at the branches can now be calculated: considering circuit 4, 6, 7, 8, 9, only section 4 requires sizing, and the pressure available at branch 6/4 will be equivalent to that for section 5, namely 2730+5000=7730 Pa. As 5 kPa is required at the terminal, the pressure available for sizing section 4 will be

$$pd = \frac{pd}{TEL} = \frac{2730}{8 \times 1.25} = 273 \text{ Pa/m}$$

Now taking circuit 3, 7, 8, 9, similarly the pd at branch 7/3 will be equivalent to the sum of sections five and six, namely 2730+1750+ 5000=9480 Pa. As 5 kPa is required at the terminal, the pressure available for sizing section 3 will be

$$pd = \frac{2730 + 1750}{8 \times 1.25} = 448 \text{ Pa/m}$$

For circuit 2, 8, 9, the pd at branch 8/2 will be equivalent to the sum of those in sections five, six and seven, namely 2730+1750+1125+ 5000=10605 Pa. As 5 kPa is required at the terminal the pressure available for sizing section 2 will be

$$pd = \frac{2730 + 1750 + 1125}{8 \times 1.25} = 560 \text{ Pa/m}$$

For circuit 1, 9, the pd at branch 9/1 will be equivalent to the sum of those in sections 5, 6, 7, 8, namely 2730+1750+1125+1956+5000 =12561 Pa. Alternatively the pd at this branch will be: index pd *minus* section 9 pd *minus* plant pd, or 47561-15000-20000 =12561 Pa. Allowing for 5 kPa at the terminal in section 1 the pd available to size the pipe in this section is given by

$$pd = \frac{7561}{8 \times 1.25} = 756 \text{ Pa/m}$$

5. Size the subcircuits

The available pd's can now be entered in the table and the appropriate pipe sections sized accordingly. These pd's cannot be exceeded when selecting a pipe size, otherwise one of these subcircuits will exceed the pressure drop in the index run.

This approach to pipe sizing attempts to balance the subcircuits with the index run. If there were an infinite number of pipe sizes from which

Table 2.5 Tabulated results for Case study 2.1

Section	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*
M (kg/s)	0.63	0.63	0.63	0.63	0.63	1.26	1.89	2.52	3.15
Available pd (Pa/m)	756	560	448	273	250	250	250	250	300
Diameter *d* (mm)	**25**	**32**	**32**	**32**	**32**	**40**	**50**	**50**	**50**
Actual pd (Pa/m)	740	168	168	168	168	280	180	313	480
Actual length *L*	8	8	8	8	13	5	5	5	25
TEL (+ 25%)	10	10	10	10	16.25	6.25	6.25	6.25	31.25
Index pd (kPa)					**2.73**	**1.75**	**1.125**	**1.956**	**15**

to choose, the process of hydraulic balancing would be achieved in the choice of pipe size. This has only been possible in section 1, and even here there is a small imbalance. You will notice that the branch sizes are 32 mm except for the branch nearest the pump, which is 25 mm for the same flow rate.

6. Assess the balancing resistances

The balancing resistances can be conveniently obtained as follows:

branch 4 pd=(273-168)(8×1.25)=1050 Pa

branch 3 pd=(448-168)(8×1.25)=2800 Pa

branch 2 pd=(560-168×8×1.25)=3920 Pa

branch 1 pd=(756-740)(8×1.25)=160 Pa

These are inserted in the form of regulating valves, which are set accordingly and located in the return pipe of the appropriate branch.

There is a reason for locating commissioning sets in the return rather than the flow. If the balancing resistance is high, the pressure drop created reduces the antiflash margin on the downstream side of the valve, and if it is located in the flow there is a greater chance of cavitation than there is in the return, which is at a lower temperature.

7. Determine the net pump duty

From the total flow rate and the index resistance the net pump duty will be 3.15 kg/s at 48 kPa.

The pump should always be oversized by at least 15% and a regulating valve located on the discharge with an appropriate allowance for its pd in the fully open position. The excess requirement is to allow for variations in pipe routeing to that shown on the drawings, and to have an allowance to assist the commissioning process.

Note: It is normal practice to install a regulating valve on the final branch to the index terminal and include its pd in the fully open position so that *all* circuits can be regulated. This is necessary for commissioning purposes so that if the circuit chosen at the design stage as the index run is not the *longest* circuit in the system, the commissioning engineer has the facility to select the circuit that is. Much time and money is wasted at the commissioning stage because insufficient attention is given to hydraulic balancing and pump selection.

BSRIA have published commissioning codes, which you need to read before attempting the design of a space-heating system.

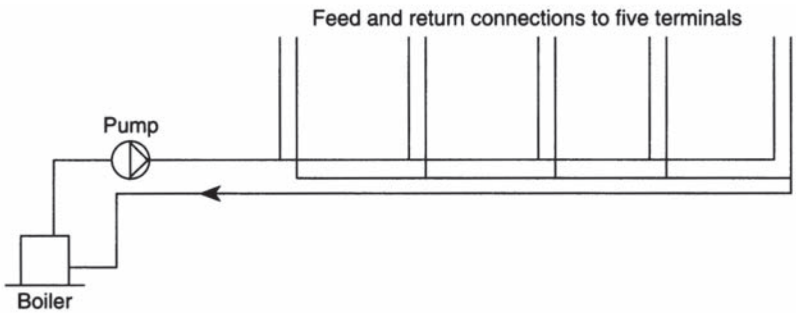


Figure 2.3 Case study 2.1: system in reverse return.

REDUCING THE NEED FOR BALANCING

If the terminals are connected to a pipe circuit of reverse return, each terminal is the same pipe distance from the pump, and the need for balancing is considerably reduced: see Figure 2.3.

USE OF BALANCING PIPES

Although we do not have an infinite number of pipe sizes from which to choose during the sizing procedure outlined above, it is possible to balance the branches by selecting a smaller pipe for part of the branch return than that used for the flow, to create a pd equivalent to that required of the regulating valve. It is not normal practice, but it does solve the mystery when you see, on a site visit, differing flow and return pipe sizes serving a terminal or group of terminals.

The generic equation is simple enough:

$$\text{pd in small pipe+pd in larger pipe=pd required in branch}$$

$$\mathrm{d}p_2 \times L_1 + \mathrm{d}p_1 (L_2 - L_1) = \text{pd required in branch}$$

where L_1 is the unknown length of balance pipe (m); L_2 is the length of flow plus return in branch (m); $\mathrm{d}p_1$ is the pressure drop for pipe selected for branch (Pa/m); and $\mathrm{d}p_2$ is the pressure drop of pipe the next size down (Pa/m).

Consider branch 3 in the case study. The branch size selected was 32 mm at a pd of 168 Pa/m ($\mathrm{d}p_1$). The pd of 25 mm pipe, which is the next size down, is 740 Pa/m ($\mathrm{d}p_2$), and L_2, including the allowance for fittings, is (8×1.25)m.

The pd required in the branch for the purposes of balancing is 448×(8×1.25).

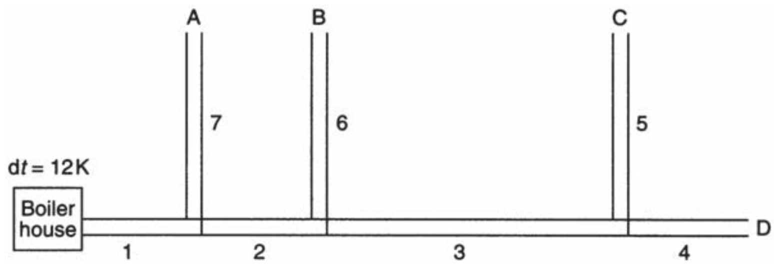


Figure 2.4 Case study 2.2: schematic layout of lphw heating system in plan.

Substituting:

$$740L_1+168\ (8\times1.25-L_1)=448\times(8\times1.25)$$

from which

$$L_1=4.9\ \text{m}$$

Thus of the total equivalent length of the flow and return (8×1.25) m for branch 3, 4.9 m is required in 25 mm pipe and the rest in 32 mm pipe.

Clearly this can only be a somewhat theoretical solution, as installation rarely follows the working drawing exactly. It may also result in infringing upon maximum water velocities. However, consider the potential savings on a project if, say, a long 100 mm diameter branch main can be partially reduced to 65 mm!

In the above case study mass flow rates were given, thus avoiding the problem of pipe emission. The following case study addresses the issue.

Case study 2.2

Consider the diagrammatic layout of an LTHW heating system shown in Figure 2.4. Design data are given in Table 2.6. Allowance for fittings on straight pipe shall be 20%.

Table 2.6 Design data for Case study 2.2

Terminal	A	B	C	D	B/H		
Load (kW)	50	80	30	60	246.7		
pd (kPa)	10	20	10	15	45		
Section	1	2	3	4	5	6	7
Length (flow + return)	30	30	40	20	20	20	20
Pipe emission (kW)	5.6	5.6	6.0	2.5	2.25	2.5	2.25

The solution requires the following.

1. Determine accurately the mass flow rates in each pipe section.
2. Adopt a suitable index pd and size the index circuit.
3. Identify the pressures available at the branches and size the branch pipes.
4. Hydraulically balance the system.
5. Specify the net pump duty.

SOLUTION

You will have noticed that pipe emission has already been estimated and terminal and boiler house pressure drops identified. This has been done so that we can concentrate upon the sizing procedure.

From Table 2.4, the *CIBSE Guide* and section pipe lengths given in the case study, two estimates of pipe emission can be made:

- Assuming 25 mm of pipe insulation having a thermal conductivity of 0.07 W/mK, total pipe emission is 5.86 kW, which is 2.7% of the total net load of 220 kw.
- Assuming that the pipe is uninsulated, total pipe emission is 26.7 kW, and this is 12.1% of the total net load.

If the pipe emission is unusable, and it is taken to be so in this case study, it must be added to the net load.

The solution will assume the worst case, namely that the pipe is uninsulated or so badly lagged that its insulating effect must be ignored. The tabulated data reflect these decisions.

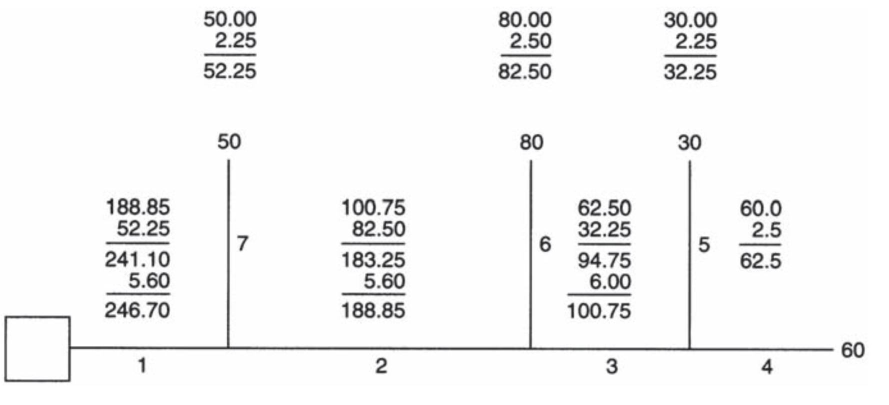


Figure 2.5 Case study 2.2: diagram of system showing heat load in each pipe.

1. Accurate determination of the mass flow rates in each pipe section

The terminal heat loads and estimated pipe emission are annotated on a diagram of the system, as shown in Figure 2.5, for the sake of clarity.

As shown earlier, the formula for mass flow is

$$M = \frac{Q}{C.\mathrm{d}t} \qquad \text{(kg/s)}$$

M_1 therefore carries the total gross heat load, which is given by

$$\frac{246.7}{4.2 \times 12} = 4.9 \text{ kg/s}$$

At the end of section 1 mass flow is the same but the heat load has changed from 246.7 kW to 241.1 kW. The temperature drop must also have changed, and rearranging the formula for mass flow we have at junction 1/2:

$$\mathrm{d}t = \frac{241.1}{4.9 \times 4.2} = 11.72 \text{ K}$$

Thus

$$M_2 = \frac{188.85}{4.2 \times 11.72} = 3.84 \text{ kg/s}$$

and

$$M_7 = M_1 - M_2 = 4.9 - 3.84 = 1.06 \text{ kg/s}$$

At the end of section 2 the heat load has changed from 188.85 kW to 183.25 kW, so again the temperature drop has changed, and at junction 2/3:

$$\mathrm{d}t = \frac{183.25}{3.84 \times 4.2} = 11.36 \text{ K}$$

Thus

$$M_3 = \frac{100.75}{4.2 \times 11.36} = 2.11 \text{ kg/s}$$

and

$$M_6 = M_2 - M_3 = 3.84 - 2.11 = 1.73 \text{ kg/s}$$

Finally at junction 3/4:

$$\mathrm{d}t = \frac{94.75}{2.11 \times 4.2} = 10.7 \text{ K}$$

and

$$M_4 = \frac{62.5}{4.2 \times 10.7} = 1.39 \text{ kg/s}$$

with

$$M_5=M_3-M_4=2.11-1.39=0.72 \text{ kg/s}$$

You win have noticed that as we move further away from the heat source the system temperature drop reduces as a result of pipe emission: Boiler house dt=12 K, junction 1/2 dt=11.72 K, junction 2/3 dt=11.36 K, junction 3/4 dt=10.7 K. However, if the pipework is efficiently insulated the heat loss is minimal (2.7%), and apportioning pipe emission in this way is necessary only for very long runs of pipe, as in a district heating scheme.

As the heat loss is approximately the same in the flow and return, mean water temperature in a system of two-pipe distribution is considered to be constant.

2. Size the index circuit

By analysing the data given about the system, the index circuit will consist of pipe sections 1, 2, 3, 4 *or* pipe sections 1, 2, 3, 5. The pipe section lengths in 4 and 5 are similar but the pd in terminal D is greater than that in terminal C: thus the index circuit will consist of pipe sections 1, 2, 3, 4.

With the aid of the CIBSE pipe-sizing tables for heavy-grade tube the index pipes can now be sized. This and subsequent information is listed in Table 2.7.

3. Size the branches

The pd at branch 5=pd at branch 4+terminal pd=8160+15000 =23160 Pa.

pipe 5=23160-10000=13160 Pa

Table 2.7 Tabulated results for Case study 2.2

Section	*1*	*2*	*3*	*4*	*5*	*6*	*7*
M(kg/s)	4.9	3.84	2.11	1.39	0.72	1.73	1.06
Available pd. (Pa/m)	250	250	250	300	548	576	1263
Diameter *d* (mm)	65	65	50	40	32	40	32
Actual pd (Pa/m)	288	180	222	340	216	520	453
TEL (*L* × 1.2)	36	36	48	24	24	24	24
pd (kPa) (index)	10.37	6.48	10.66	8.16			

$$dp_5 = pd_5/TEL = 13160/24 = 548 \text{ Pa/m}$$

The pd at branch 6=pd at branch 5+dp_3=23160+10660=33820 Pa.

pipe 6=pd at branch 6-terminal pd=33820-20000=13820 Pa.

$$dp_6 = pd_6/TEL = 13820/24 = 576 \text{ Pa/m}$$

The pd at branch 7=pd at branch 6+dp_2=33820+6480=40300 Pa.

pipe 7=pd at branch 7-terminal pd=40300-10000=30300 Pa

$$dp_7 = pd7/TEL = 30300/24 = 1263 \text{ Pa/m}$$

These rates of available pressure drop are included in the tabulated results (Table 2.7) and the branch pipes are sized as shown in the table along with the actual rates of pressure drop.

4. Determine the balancing resistances

Using the calculated available pd's and actual rates of pressure drop obtained from the pipe-sizing tables for the pipes selected (see Table 2.7).

$$\text{Branch } 5 = (\text{available pd-actual pd}) \times TEL$$

$$= (548-216) \times 24 = 7968 \text{ Pa}$$

$$\text{Branch } 6 = (576-520) \times 24 = 1344 \text{ Pa}$$

$$\text{Branch } 7 = (1263-453) \times 24 = 19440 \text{ Pa}$$

5. Specify net pump duty

$$\text{The index pd} = (1, 2, 3, 4) + \text{BH} + \text{index terminal}$$

$$= 35.67 + 45 + 15$$

$$= 95.67 \text{ kPa}$$

The total mass flow handled by the pump will be that in section 1, which is 4.9 kg/s. Thus the net pump duty will be 4.9 kg/s at 96 kPa.

NOTES

1. The selected pd for the index circuit is 250 Pa/m. The branch to the index terminal can have a higher rate of pressure loss, as it does not affect the balancing of the branches near the pump.
2. You will notice that the actual rates of pressure drop have been interpolated from the pipe-sizing tables to gain better accuracy.

3. In estimating the pipe emission in the data, approximate pipe sizes would have been obtained from Table 2.4. These now need to be checked with the pipe sizes selected. Any discrepancy requires correction if a full pipe-sizing analysis is needed.
4. The pump selected should be at least 15% greater in duty than the net value, to allow for changes during the installation process and a margin for the commissioning engineer within which to work.
5. This case study assumes little or no pipe insulation: hence the need to account for pipe emission, which did not form part of the useful heating surface. If pipework is efficiently lagged the heat loss from most installations is minimal. This allows a constant temperature drop to be used throughout the system without seriously affecting design accuracy.

   However, where long pipe runs are prevalent, as in the case of district heating, pipe emission must be accounted for.

   It can also be accounted for when pipe emission is useful and can be said to form part of the heating surface. If this is the case, the pipe emission does not have to be added to the total plant load.
6. The object in calculating the balancing resistances is to ensure that with them in place in the form of regulating valves set to the appropriate pd, each circuit has the same pressure drop as the index run, namely 96 kPa, and the pump will therefore deliver the design flow to each terminal. This principle is identical to those for balancing ducted air systems and steam systems.

We have now considered two case studies related to the pipe-sizing process. Case study 2.3 will look at the pipe-sizing procedure given a fixed pump duty. This situation is not likely to be confronted regularly. It might occur during extensions to an existing installation in which an existing pump is to be commissioned for use.

Case study 2.3

The pump duty that is available at point X is 4.76 kg/s at 114 kPa. The system to which it is to be connected is shown diagrammatically in Figure 2.6. Further design data are listed in Table 2.8.

Allowance for fittings on straight pipe: 15%.

Assume pipework is efficiently lagged and ignore the effects of pipe emission. The design solution requires that the pipework is sized and the system is hydraulically balanced such that the pump having the quoted duty can be installed.

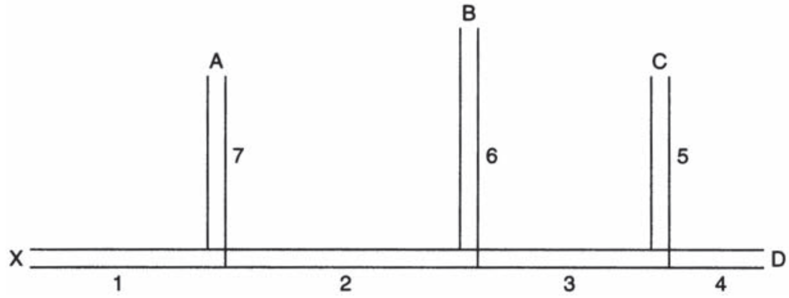


Figure 2.6 Case study 2.3: diagrammatic layout of system.

SOLUTION

1. Determination of system temperature drop d*t*

The total heat load amounts to Σ(A, B, C, D)=200 kW. Adopting the mass flow rate formula to determine the system temperature drop d*t*:

$$dt = \frac{Q}{M.C} = \frac{200}{4.76 \times 4.2} = 10\ \text{K}$$

As pipe emission is to be ignored, this temperature drop will be constant throughout the pipe network.

2. Determination of mass flow rates

The mass flow can now be determined for each terminal and hence the mass flow calculated for the distribution pipes 3, 2 and 1. These calculations are shown in the results listed in Table 2.9.

3. Terminal pressure drops

The pd's at each terminal can be obtained from the manufacturers of the equipment. These are listed in Table 2.9.

4. Identifying the index circuit

There are four circuits in the system: 1, 7; 1, 2, 6; 1, 2, 3, 5; and 1, 2, 3, 4. From an analysis of section pipe lengths and terminal pd's, circuit 1, 2,3, 5 is the longest and also can have the greatest pd.

Table 2.8 Design data for Case study 2.3

Terminal	A	B	C	D			
Load (kW)	40	70	60	30			
Terminal pd (kPa)	20	30	25	15			
Pipe section	1	2	3	4	5	6	7
Length (flow + return)	80	100	70	50	60	80	60

Table 2.9 Tabulated data relating to Case study 2.3

Pipe section	*1*	*2*	*3*	*4*	*5*	*6*	*7*
Load n (kW)	200	160	90	30	60	70	40
M (kg/s)	4.76	3.81	2.14	0.714	1.43	1.67	0.95
Terminal pd (kPa)	–	–	–	15	25	30	20
Available pd (Pa/m)	250	250	250	615	250	420	1000
Diameter *d* (mm)	65	65	50	32	50	50	32
Actual pd (Pa/m)	272	177	228	213	106	140	366
TEL (m)	92	115	80.5	57.5	69	92	69
Section pd (kPa)	25	20.3	18.35	12.25	7.3	12.88	25.25

5. Determination of average available pd for the index circuit

The pump pressure is 114 kPa, so the average rate of pressure drop for the index circuit will be given by

$$\mathrm{d}p/TEL = \frac{\text{available pump pressure} - \text{pd in index terminal}}{310 \times 1.15}$$

$$\text{pd} = \frac{114\,000 - 25\,000}{310 \times 1.15} = 250 \text{ Pa/m}$$

This is added to the tabulated data.

6. Sizing the index circuit

As the pump pressure cannot be exceeded, it is necessary to pay particular attention to the selection of the pipe sizes. You will see from the tabulated data that one section does exceed 250 Pa/m but the others do not. As a check, the total pd in the index pipe sections can be calculated for the pipe sizes chosen. Using the data in the table this comes to 71 kPa.

The maximum pump pressure available is 114-25=89 kPa. If section 5 is reduced to 40 mm pipe the total pd in the index pipe sections comes to 88.6 kPa. Theoretically this would be acceptable; in practice it would not, as it is too close to the maximum and does not allow for variations between design and installation.

7. Sizing the branches

The pressure available at the branches requires calculation in the same way as in the other case studies above. The available pd for the branch pipes is then determined and this is included in the tabulated data, following which the branch pipes are sized and actual rates of pressure drop recorded, as shown by interpolation from the pipe-sizing tables.

8. Determination of the balancing resistances

From the section and terminal pressure drops tabulated the unregulated pressure drop can be determined for each circuit:

circuit 1, 7=25+25.25+20=70 kPa

circuit 1, 2, 6=25+20.3+12.88+30=88 kPa

index circuit 1, 2, 3, 5=25+20.3+18.35+7.3+25=96 kPa

circuit 1, 2, 3, 4=25+20.3+18.35+12.25+15=91 kPa

The pump pressure developed at 4.76 kg/s is 114 kPa, so if a regulating valve is fitted on the pump discharge to absorb 114-100=14 kPa then the index circuit will require a regulation of 100-96=4 kPa, and with the index run now at 100 kPa the regulation required on the other circuits will be

circuit 1, 7=100-70=30 kPa in the return of branch 7

circuit 1, 2, 6=100-88=12 kPa in the return on branch 6

circuit 1, 2, 3, 4=100-91=9 kPa in the return on branch 4

The regulation on the return on the index run will be 4 kPa. The regulation on the pump discharge will be 14 kPa.

Without this regulation the pump is over-sized, with excess pressure and flow rate. With the rationale developed above, the commissioning engineer therefore has room for adjustment at the pump and in each circuit. This offers the best facility, as the installation will not respond exactly as the design data. After hydraulically balancing the four circuits in the pipe network beginning with the shortest circuit, which here consists of sections 1, 7, the final regulation would be done at the valve on the pump discharge.

2.5 Hydraulic resistance in pipe networks

Having dealt with sizing, proportioning pipe emission and hydraulic balancing, the final section of this chapter focuses on hydraulic resistance in pipe fittings. This matter so far has been considered by using a percentage for fittings on straight pipe. Clearly, the approach can only be an approximation, as the percentage will vary according to the density of fittings in a pipe network. The percentage can vary from 10% for long straight pipe runs to 150% in a boiler plant room, where the density of pipe fittings is high.

Chapter 10 addresses design from first principles, and from there you will see that **head loss** dh due to turbulent flow in pipes consists of three kinds:

- that in straight pipes, where $dh=(4fLu^2)/(2gd)$ m of fluid flowing;
- that in fittings where $dh=k\times u^2/2g$ metres of fluid flowing;
- shock losses, where for a sudden enlargement $dh=(u_1^2-u_2^2)^2/2g$ m of fluid flowing, and for a sudden reduction $dh=0.5u_2^2/2g$ m of fluid flowing.

Head loss in pipe fittings is calculated from the product of the kinetic energy expression in the Bernoulli theorem and the velocity head loss factor k for various pipe fittings given in the *CIBSE Guide*. The values of equivalent length l_e given in the pipe-sizing tables against each mass flow are for one velocity head, ie when k=1.0. It is convenient to continue to express head loss and hence pressure loss through fittings in terms of equivalent lengths of straight pipe. Thus total equivalent length *TEL* is used as it has been already, and pressure loss dp in a pipe section containing fittings is

$$dp = \text{pd in Pa/m} \times TEL \qquad \text{(Pa)}$$

where $TEL=L+(k_t\times l_e)$.

DETERMINATION OF l_e

If therefore head loss in fittings is equated to head loss in an equivalent length of straight pipe l_e, then

$$k\times\frac{u^2}{2g}=\frac{4fLu^2}{2gd}$$

from which

$$k=\frac{4fL}{d}$$

and rearranging:

$$L=k\times\frac{d}{4f}$$

Therefore

$$l_e=k\times\frac{d}{4f}$$

and if k=1.0 velocity head:

$$l_e = \frac{d}{4f} \qquad \text{(m)}$$

Alternatively when k=1.0

$$l_e = \frac{dp_{\text{fittings}} \text{ in Pa}}{dp \text{ in Pa/m}} \qquad \text{(m)}$$

$$= \frac{u_2/2g \times \rho \times g}{dp \text{ in Pa/m}} \qquad \text{(m)}$$

Example 2.1

Validate the value of equivalent length l_e when k=1.0 for water flowing at 3.22 kg/s in 50 mm black heavy-grade tube at 75 °C. Data taken from the pipe sizing tables: ρ=975 kg/m^3, dp=500 Pa/m, u=1.6 m/s and l_e=2.5.

Solution

Adopting the second formula:

$$l_e = \frac{(1.6)^2/2g \times 975 \times g}{500}$$

from which

$$l_e=2.496$$

Example 2.2

Check the value of l_e for water flowing at 14 kg/s in 100 mm pipe at 75 °C. Data from the pipe sizing tables: ρ=975 kg/m^3, dp=240 Pa/m, u=1.7 m/s and l_e=5.9.

Solution

D'arcy's formula for turbulent flow in straight pipes:

$$dh = \frac{4fLu^2}{2gd} \qquad \text{(metres of water flowing)}$$

$$dp = \frac{4fLu^2}{2gd} \times \rho \times g \qquad \text{(Pa)}$$

$$\frac{dp}{L} = \frac{4fu^2\rho}{2d} \qquad \text{(Pa/m)}$$

Thus frictional coefficient:

$$f = \frac{2d.\mathrm{d}p/L}{4u^2\rho}$$

Substitute data:

$$f = \frac{2 \times 0.1 \times 240}{4 \times (1.7)^2 \times 975}$$

from which

$$f=0.00426$$

Substitute into

$$\begin{aligned} l_e &= \frac{d}{4f} \\ &= \frac{0.1}{4 \times 0.00426} \\ &= 5.87 \end{aligned}$$

Example 2.3

Part of an LTHW space heating circuit consists of a pipe section having one panel radiator, two 15 mm angle valves and two 15 mm copper bends. Determine, from the data, the pressure loss and the equivalent length of straight pipe for the fittings. Data: u=1.0 m/s, ρ=975 kg/m^3, f=0.006 25, k for radiator=2.5, k for angle radiator valves=5.0 each, k for bends=1.0.

Solution

The total velocity head loss factor k_t=14.5, and from above $\mathrm{d}h=k_t\, u^2/2g$ metres of water flowing. As

$$\mathrm{d}p=\mathrm{d}h\times\rho\times g \qquad \text{(Pa)}$$

then

$$\mathrm{d}p=k_t(u^2/2g)\times\rho\times g \qquad \text{(Pa)}$$

Thus

$$\mathrm{d}p = k_t\left(\frac{1}{2}\rho u^2\right) \qquad \textbf{(Pa)}$$

Pressure loss generated by water flow through the radiator and fittings by substitution:

$$dp = 14.5 \times \frac{1}{2} \times 975 \times (1.0)^2 = 7069 \text{ Pa.}$$

Equivalent length $l_e=d/4f$ when k=1.0. Substitute:

$$l_e = \frac{0.015}{4 \times 0.006\,25} = 0.6$$

Equivalent length of straight pipe

$$=k_t \times l_e = 14.5 \times 0.6 = 8.7 \text{ m}$$

This means that the pressure drop of 7069 Pa through the radiator and fittings in the pipe section is equivalent to that through 8.7 m of 15 mm diameter pipe when water flows at a velocity of 1.0 m/s.

Consider an addendum to this question. Look up the data in the pipe-sizing tables for copper, Table X. On 15 mm pipe at 1.0 m/s, M= 0.14 kg/s, l_e=0.6, which agrees with the calculated value above, and the pressure drop per metre (pd/m)=820 Pa/m. Thus if the length of straight pipe in the section was, say, 6 m, the pressure loss through the pipe *and* fittings would be:

$$\text{pd} = \text{pd/m} \times TEL = \text{pd/m} \times (L + k\text{t} \times l_e) \qquad \text{(Pa)}$$

Substituting:

$$\text{pd} = 820 \times (6 + 8.7) = 12054 \text{ Pa}$$

for that section of pipe, radiator and fittings. This is the standard method by which hydraulic resistance is calculated for each pipe section in the network. The first part of the solution determined from *first principles* the pressure drop through the radiator and fittings alone.

The pd/m of 820 Pa/m from the tables can be checked from first principles from the D'arcy equation:

$$\frac{\text{pd}}{\text{m}} = \frac{4fu^2\rho}{2d} \qquad \text{(Pa/m)}$$

and substituting:

$$\frac{\text{pd}}{\text{m}} = \frac{4 \times 0.006\,25 \times 1.0 \times 975}{2 \times 0.015} = 812.5 \text{ Pa/m}$$

which shows a 1% error.

Example 2.4

Figure 2.7 shows pipe connections to a unit heater. Determine the pressure drop across A and B from the given data from first principles and by the standard method.

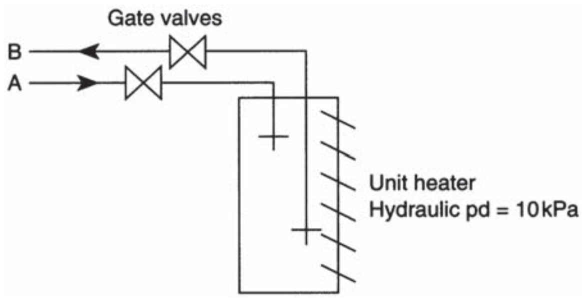


Figure 2.7 Example 2.4: elevation of connections to the unit heater.

Data

Fittings are malleable cast iron and swept and include two gate valves and four bends. Connections are in 40 mm black heavyweight tube carrying water at 75 °C, where M=1.55 kg/s, l_e=1.8, pd=420 Pa/m, u=1.25 m/s, L=5 m and ρ=975 kg/m^3.

Solution from first principles

The numerical data given in the question are obtained from the CIBSE pipe-sizing tables for 40 mm pipe. $l_e=d/4f$, therefore frictional coefficient $f=d/4.l_e$. substituting:

$$f=\frac{0.04}{4\times 1.8}=0.005\,56$$

Hydraulic loss in straight pipe for turbulent flow:

$$\mathrm{d}h=\frac{4fLu^2}{2gd} \qquad \text{(m)}$$

$$\mathrm{d}p=\frac{4fLu^2\rho}{2d} \qquad \text{(Pa)}$$

Substitute:

$$\mathrm{d}p=\frac{4\times 0.005\,56\times 5\times (1.25)^2\times 975}{2\times 0.04}=2117\ \text{Pa}$$

The k values for the fittings are:

2 gate valves at 0.2=0.4

4 malleable iron bends at 0.5=2.0

from which k_t=2.4

For fittings:

$$dp = k_t \frac{1}{2} \rho u^2 \qquad \text{(Pa)}$$

Substitute:

$$dp = 2.4 \times \frac{1}{2} \times 975 \times (1.25)^2 = 1828 \text{ Pa}$$

The pressure drop between A and B, dp=straight pipe+fittings+ terminal

$$\begin{aligned} dp &= 2117+1828+10000 \\ &= 13.945 \text{ kPa} \end{aligned}$$

Solution by the standard method
The pressure loss between A and B, dp=(L+k_t×l_e)×dp/m+terminal (Pa)
Substitute:

$$\begin{aligned} dp &= (5+2.4\times1.8)\times420+10000 \\ &= 13.914 \text{ kPa} \end{aligned}$$

The difference between the two solutions is insignificant, but you can see why the standard method is normally used.

The percentage for fittings on straight pipe can be determined for the pipe section, and will be:

$$\left(\frac{k_t \times l_e}{L}\right) \times 100 = \frac{2.4 \times 1.8}{5} \times 100 = 86.4\%$$

This is high, owing to the density of fittings compared with straight pipe, and is a warning against estimating the percentage unless you have some experience.

Case study 2.4

Figure 2.8 represents the plan view of boiler plant serving an air heater battery with constant temperature variable volume (CTVV) control via a three-port mixing valve providing flow-diverting control. The supply air off the battery is required at constant temperature regardless of the temperature of the air on the upstream side. A thermostat located in the supply air duct therefore controls the three-port valve.

The system (index run) consists of 43.9 m of 32 mm black heavy-grade tube, 13 malleable iron bends, one straight tee, four gate valves and one

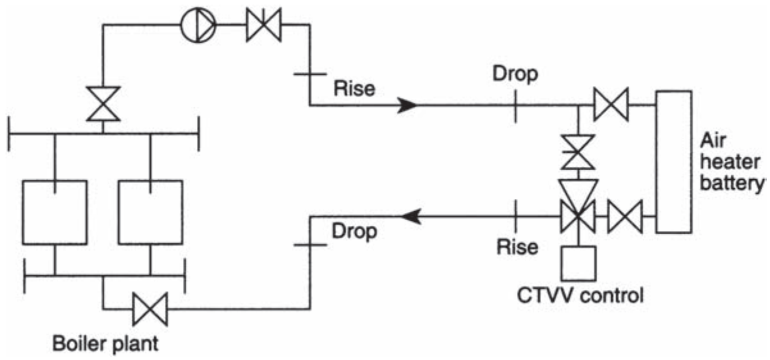


Figure 2.8 Case study 4: diagram of circuit to a heater battery. Refer to text for pipe lengths and fittings. Cold feed, open vent and F&E tank omitted.

regulating valve on the pump discharge. The bypass pipe at the heater battery is 1.5 m long and fitted with a regulating valve. The pipe connections between the bypass and the heater are each 1.2 m in length. The pd across the headers and index boiler is 15 kPa and that across the heater battery is 10 kPa.

From the data, size the three-way valve and determine the net pump duty and the additional pressure drop required across the bypass regulating valve.

DATA

Tube shall be black heavyweight with malleable cast iron swept fittings. Velocity pressure loss factors: gate valve k=0.3, regulating valve k=4.0, bends k=0.5, straight tee k=0.2 and 90 ° tee k=0.5. Mass-flow rate M=1.26 kg/s, and from the pipe-sizing tables pipe diameter d=32 mm, pd=630 Pa/m and l_e=1.5 m. Three-way mixing valve: valve authority N=0.65. Valve manufacturer's data:

Size (mm)	15	20	25	32
K_v	1.9	4.2	9.6	12.4

INTRODUCTION TO THE SOLUTION

You will note that the valve authority N for the three-way mixing valve has been given in the data. This is required to determine the pressure loss dp_v across the valve when it is fully open to the heater. The selection of control valves involves the use and application of two terms: the **flow coefficient,** sometimes called the capacity index, K_v and the **valve authority** N:

$$\mathrm{VFR}=K_v\sqrt{\mathrm{d}p_v} \quad (\mathrm{m^3/h}) \quad \text{from which } K_v \text{ may be determined.}$$

$$N=\frac{\mathrm{d}p_v}{\mathrm{d}p_s+\mathrm{d}p_v} \quad \text{from which } \mathrm{d}p_v \text{ is calculated.}$$

The flow coefficient K_v is needed to select the control valve from the manufacturer's literature. The valve authority N is needed to determine dp_v. For two-port valves its minimum value is 0.5, and for three-way control valves providing flow mixing the minimum value for N is 0.3. For three-way valves providing flow diversion, as in this case study, the minimum value for valve authority N is 0.5.

Generally, the higher the valve authority, the better the control but the higher is the pressure drop dp_v across the valve and hence the higher the pump pressure required. Figure 2.9 identifies the terms dp_v and dp_s for three types of valve control.

SOLUTION

The index circuit is that to the heater, which is the index terminal. It does not include the bypass. The total length of straight pipe is 43.9 m.

The total velocity head loss factor k_t for the fittings on the index run includes:

13 bends@ 0.5=6.5

1 straight tee @ 0.2=0.2

4 gate valves @ 0.3=1.2

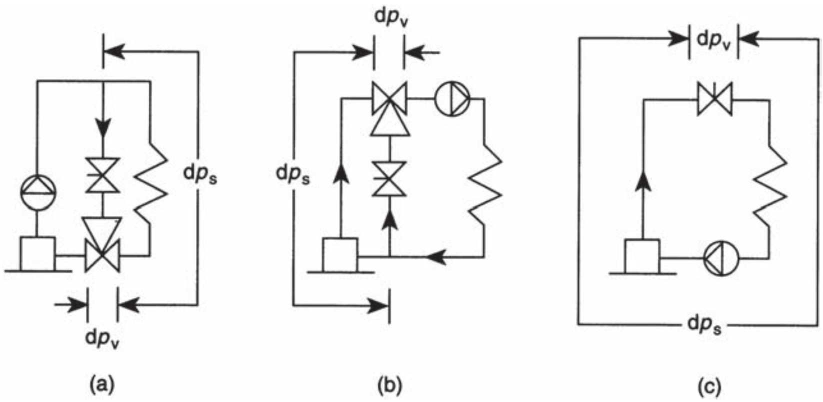


Figure 2.9 (a) Flow-diverting CTVV control; (b) flow-mixing CVVT control; (c) throttling VTVV control.

1 regulating valve @ 4.0=4.0 giving k_t=11.9

$TEL=(L+k_t\times l_e)=(43.9+11.9\times 1.5)=61.75$ m

Pressure drop in the index circuit=$TEL\times dp$ in Pa/m+pressure drop in the index terminal+pressure drop in the boiler plant

$$=61.75\times 630+10000+15000$$

index dp=63.9 kPa

However, this *excludes* the pressure drop dp_v across the three-way control valve, which must be accounted for in the net pump duty as it forms part of the index run.

From the equation for valve authority:

$$N=\frac{dp_v}{dp_v+dp_s}$$

dp_s represents the pressure loss through the heater circuit downstream of the bypass (see Figure 2.9a) and can be calculated from the data. N is known, so the pressure drop across the control valve can be evaluated and added to the index pd. Rearranging the formula for N in terms of dp_v:

$$\frac{1}{N}=1+\frac{dp_s}{dp_v}$$

from which

$$dp_v=\frac{dp_s}{(1/N)-1}$$

dp_s=pd in the pipe connections to the heater downstream of the bypass plus the pd across the heater

$$=(L+k_t\times l_e)\times dp\ \text{Pa/m}+10000$$

$$=(1.2\times 2+2\times 0.3\times 1.5)\times 630+10000$$

$$dp_s=12\ \text{kPa}$$

Substituting N and dp_s into the rearranged formula for N:

$$dp_v=\frac{12}{(1/0.65)-1}=22.3\ \text{kPa}$$

From the formula for K_v:

$$K_v=\frac{\text{VFR}}{\sqrt{dp_v}}=\frac{0.001\ 26\times 3600}{\sqrt{0.223}}$$

$$=9.6$$

From the manufacturer's data, valve size will be 25 mm.
Note: dp_v is in bars and VFR is in m^3/s in the K_v formula.
Thus the *total* index pressure loss will be 63.9+22.3=86.2 kPa.
Note that the pressure drop across the control valve is 26% of the total index pd in this system. It cannot therefore be left unaccounted for.

Net pump duty will therefore be 1.26 kg/s at 86.2 kPa.

The pump selected must have a duty at least 15% in excess of the net value, for the reasons given earlier.

The second part of the solution relates to hydraulically balancing the bypass with the heater circuit downstream of it. When the heater is on full bypass the pressure drop must be similar to that in the index circuit. If the pd across the control valve when it is on full bypass is the same as when it is fully open to the heater, the hydraulic resistance through the bypass pipe will include: straight pipe, a 90° tee and the fully open regulating valve.
Thus

$$\text{pd}=(L+k_t \times l_e)\times dp \qquad (\text{Pa/m})$$

$$k_t=\text{reg valve}+90^\circ \text{ tee}+\text{tee bend}=4.0+0.5+0.5=5.0$$

and

$$\text{pd}=(1.5+5\times1.5)\times630=5.67 \text{ kPa}$$

The pd through the heater circuit is 12 kPa, so the balancing resistance required of the regulating valve is (12-5.67)=6.33 kPa.

Both circuits—that through the heater and that through the bypass—now have similar pressure drops each of 86.2 kPa, and the system will operate hydraulically as specified.

ONE-PIPE RADIATOR SYSTEMS

Figure 2.1c at the beginning of this chapter illustrates one-pipe distribution, which is sometimes used for radiator systems. The ring main connecting the radiators is a common-sized pipe for ease of installation. This has a significant effect upon the ratio of water flow *through* each radiator compared with that through its associated bypass pipe. As the hydraulic pressure drop through the radiator bypass must equal the pressure drop through the radiator it follows that the majority of water in the ring main will flow through each radiator bypass. The ratio M_t/M_r, which is the total mass flow in the ring main to the mass flow through the radiator, is required to determine the temperature drop across the radiator and hence its output in watts.

The method for determining the M_t/M_r ratio is based on trial and error by

Table 2.10 M_t/M_r ratio for sizing one-pipe systems

d	*D*	M_tM_r *ratio where* L_r =					*Approx.*
		1.0 m	*1.3 m*	*1.6 m*	*2.0 m*	*2.3 m*	*value*
15	15	3.46	3.19	3.02	2.88	2.76	3
	20	5.92	5.49	5.18	4.95	4.71	5
	25	9.62	9.01	8.47	8.06	7.41	9
20	20	3.72	3.48	3.31	3.17	3.05	3.5
	25	5.75	5.41	5.13	4.90	4.55	5
	32	9.90	9.43	8.93	8.62	8.26	9
25	25	3.95	3.75	3.57	3.44	3.19	3.5
	32	6.54	6.21	5.95	5.71	5.52	6
	40	8.77	8.40	8.06	7.75	7.52	8

giving values to ratio M_t/M_r, radiator pipe connection sizes *d* and lengths, radiator length L_r, ring main size *D*, and ring main flow rate Mt taken at a pressure drop of between 200 and 300 Pa/m. The hydraulic resistance through the radiator and through its bypass can then be checked, and if they are approximately equal the ratio is validated.

Table 2.10 summarizes some validated M_t/M_r ratios for sizing one-pipe systems for radiators having two angle-type valves and 1 m of flow and return pipe.

Case study 2.5

Consider the one-pipe system shown in Figure 2.10 and determine:

(a) mean water temperature (MWT) in each radiator;
(b) correction factor for each radiator for sizing purposes;
(c) size of the ring main and radiator connections.

DATA

Radiator outputs given for a temperature difference between radiator and room of 60 K; room temperature 20 °C; ignore pipe emission; black medium-weight pipe shall be used.

SOLUTION

The required radiator outputs are obtained from the heat loss calculations. A radiator manufacturer's brochure will specify outputs in watts, usually for a temperature difference between radiator and room of 60 K. For other values of

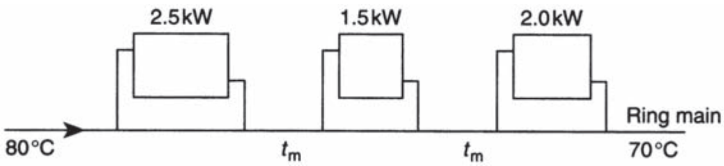


Figure 2.10 Case study 2.5: one-pipe system.

this temperature difference a correction factor must be applied. This is obtained from the following formula:

$$\text{correction factor cf} = \left(\frac{\text{prevailing d}t}{\text{brochure d}t}\right)^n$$

where index n is approximately equal to 1.3 for radiators

The incremental temperature drop across the ring main $\mathrm{d}t_m = \mathrm{d}t(q/Q)$.

The temperature drop across the radiator, $\mathrm{d}t_r$, is obtained from

$$\mathrm{d}t_r = \mathrm{d}t_t\left(\frac{q}{Q}\right)\left(\frac{M_t}{M_r}\right)$$

where $\mathrm{d}t_t$ is the total temperature drop for the circuit, q is the radiator output and Q is the total radiator output for the circuit.

The mass flow in the ring main is

$$M_t = \frac{q}{C\mathrm{d}t_t} + \frac{6}{4.2(80-70)} = 0.143 \text{ kg/s}$$

If 20 mm pipe is used, the pressure drop from the CIBSE pipe-sizing tables is 107 Pa/m, and from Table 2.10 the approximate M_t/M_r ratio is 5 for 15 mm radiator connections.

Thus for the first radiator

$$\mathrm{d}t_r = (80-70)\left(\frac{2.5}{6}\right)(5) = 20.8 \text{ K}$$

This is too high, so the radiator connections need to be increased from 15 mm to 20 mm, for which M_t/M_r=3 and

$$\mathrm{d}t_r = (80-70)\left(\frac{2.5}{6}\right)(3) = 12.5 \text{ K}$$

which is satisfactory.

For the first radiator therefore, the mean water temperature =80-12.5/2=73.75 °C and the mean water to room temperature difference=73.75-20=53.75 K.

Correction factor for the first radiator=$(53.75/60)^{1.3}$=0.867.
For the second radiator

$$dt_m = dt\left(\frac{q}{Q}\right) = 10\left(\frac{2.5}{6}\right) = 4.17\ \text{K}$$

and

$$t_m=80-4.17=75.83\ °C$$

This will be the flow temperature to the second radiator.

The temperature drop across this radiator for 15 mm connections,

$$dt_r = dt_t\left(\frac{q}{Q}\right)\left(\frac{M_t}{M_r}\right) = 10\left(\frac{1.5}{6}\right)(5) = 12.5\ \text{K}$$

and the mean water temperature in the second radiator =75.83-12.5/2=69.58 °C and mean water to room temperature difference for this radiator=69.58-20=49.58 K.
Correction factor for this radiator=$(49.58/60)^{1.3}$=0.78.
For the last radiator

$$dt_m = dt\left(\frac{q}{Q}\right) = 10\left(\frac{1.5}{6}\right) = 2.5\ \text{K}$$

and

$$t_m=75.83-2.5=73.33\ °C$$

This radiator temperature drop using 15 mm pipe connections,

$$\begin{aligned} dt_r &= dt\left(\frac{q}{Q}\right)\left(\frac{M_t}{M_r}\right) = \\ &= 10\left(\frac{2}{6}\right)(5) \\ &= 16.7\ \text{K} \end{aligned}$$

and the mean water temperature in this radiator MWT =73.33-16.7/2=64.98 °C and the mean water to room temperature difference for this radiator=64.98-20=44.98 K.
Correction factor for this radiator=$(44.98/60)^{1.3}$=0.688
The results from the solution are tabulated in Table 2.11

CONCLUSIONS

Note the effect on radiator mean water temperature in the system. It reduces in each successive radiator, requiring the calculation of the correction factor each time. The mean water temperature in a system of two-pipe distribution is

Table 2.11 Results for Case study 2.5

Radiator	*Mean water temperature (°C)*	*Correction factor*	*Radiator Connections (mm)*	*ratio M_t/M_r*	*Radiator dt (K)*
First	73.75	0.867	20	3	12.5
Second	69.58	0.780	15	5	12.5
Third	73.33	0.688	15	5	16.7

Ring main size: 20 mm having a pd of 107 Pa/m.

constant, requiring only one calculation of the correction factor, assuming all rooms are at the same temperature.

Having determined the correction factors, each radiator can now be sized from the manufacturer's brochure.

2.6 Chapter closure

This concludes the work on pipe sizing and hydraulic balancing, proportioning pipe emission, determination of hydraulic resistance in pipe networks and determination of net pump duty.

You now have the skills to undertake the principle design procedures for LTHW heating systems.

Pump and system 3

Nomenclature

BPV	balanced pressure valve
CVVT	constant volume variable temperature
D	impeller diameter (m)
F&E	feed and expansion
g	gravitational acceleration (m^2/s)
h	suction lift (m)
K	constant
M	mass flow rate (kg/s)
N	speed (rev/s)
NP	neutral point
NPSH	net positive suction head
P	pressure (Pa, kPa)
pd	pressure drop (Pa, kPa)
P_w	power (W)
Q	volume flow rate (l/s, m^3/s)
TRV	thermostatic radiator valve
VFR	volume flow rate (l/s, m^3/s)
+ve, -ve	positive, negative pump pressure
ρ	density (kg/m^3)

3.1 Introduction

For pipe networks distributing water and requiring a prime mover, the centrifugal pump is well established. The pressure that the pump must develop is dependent upon its application.

There are essentially two types of pipe distribution in which a pump is used to transport water. **Closed systems** include those applications where water is recirculated around the pipe network for the purposes of transferring heat. **Open systems** are those in which water is circulated and at some point or at a number of points around the pipe network it is discharged to atmosphere for the ultimate purpose of consumption.

3.2 Closed and open systems

CLOSED SYSTEMS

For closed systems such as space heating, the net developed pump pressure must equal the hydraulic resistance in the index run. It does not matter if the system serves a multistorey building, as once water movement is initiated circulation will take place regardless of the static height of the system. However, in such systems the pump must generate sufficient pressure *at no flow* (that is, maximum pressure development) to overcome the system's static height.

OPEN SYSTEMS

In the case of open systems such as boosted water from a ground storage tank the net developed pump pressure must include:

index *pd* plus pressure equivalent to the static lift
plus discharge pressure at the index terminal

OPEN SYSTEMS: SPECIAL CASES

Where a centralized hot water supply system is connected to a high-level cold water storage tank and relies on the secondary pump to overcome hydraulic resistance, the net pump pressure required includes:

index *pd* plus discharge pressure at the index terminal

The static lift for the system is accounted for by the pressure generated by the static height of the water storage tank.

If booster pumps are connected directly to the incoming water main to provide water to a multistorey building, pressure required includes:

index *pd* plus pressure equivalent to the static lift *minus* minimum mains pressure *plus* discharge pressure at the index terminal.

3.3 Pump considerations

SUCTION LIFT

Consider the pump connected to an open system in Figure 3.1. It is clear which part of the system is under negative pump pressure (pump suction) and which is under positive pump pressure. The *change* in pump pressure occurs at the impeller within the pump casing. This is the **neutral point** in the open system.

If the pump is drawing water from some point below the centreline of the impeller, the vertical distance h must not be sufficient to cause cavitation in the suction pipe or on the impeller surface.

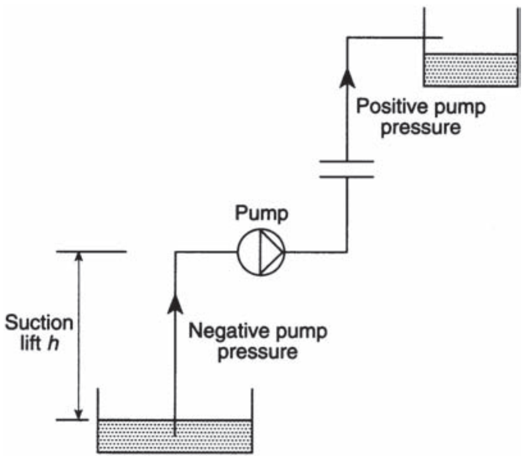


Figure 3.1 Pump pressure effects: open systems.

Theoretical maximum suction lift h for cold water is obtained from

$$P = h \times \rho \times g \qquad \text{(Pa)}$$

from which

$$\text{suction lift } h = P/\rho g \qquad \text{(metres of vertical height)}$$

If all of atmospheric pressure is used:

$$h = 101325/(1000 \times 9.81) = 10.33 \text{ m}$$

This is the maximum theoretical suction lift, as if it were possible to achieve it, the water at the top of the lift would be vapour and at this point we would have also achieved a perfect vacuum. The pump therefore would be attempting to handle vapour, which it is not designed to do. The practical suction lift would be 5 m maximum where the saturation temperature of the water at the top of the lift would be about 82 °C. **Net positive suction head NPSH** is a term used by pump manufacturers:

NPSH = pressure required at the eye of the impeller to *prevent* cavitation in the pump casing.

PUMP PRESSURE DISTRIBUTION AND THE NEUTRAL POINT

In open systems where the water level is below the pump, the pump pressure is negative in the suction pipe and positive in the discharge pipe. The pressure change occurs at the impeller. This is called the **neutral point**.

With closed systems and pumped HWS when there is no draw-off, the pump pressure effect changes from negative to positive or vice versa at the neutral point, which is normally taken at the feed and expansion pipe entry into the system.

PUMP LOCATION

This is important for both closed and open systems. In open systems the suction lift must not be excessive, in case cavitation is promoted in the suction pipe or pump casing. With closed systems, pump location influences the water level in the open vent pipe in the case of vented heating systems. It also influences the magnitude of the antiflash margin around the heating system, whether it is pressurized or vented.

ANTIFLASH MARGIN

The antiflash margin is the temperature difference between the water and its saturation temperature at a point in a heating system. The minimum acceptable value is 10 K, with 15 K being preferred (Figure 3.2).

PUMP SIZING

At the risk of repetition, as this point has been documented elsewhere, it is essential for the purposes of commissioning to ensure that the pump is oversized by at least 15%. A regulating valve, which can be located in the pump discharge, is used to move the operating point of the pump on the system up the pump characteristic (from C to F) to the design flow rate (d). (Figure 3.2).

You will notice that the final pump pressure developed is the sum of the original design system pressure (E) plus the pressure drop *(h)* required across the regulating valve.

MULTIPLE PUMPING

Identical pumps connected in series double the pressure developed. Identical pumps connected in parallel double the flow (Figures 3.3 and 3.4). Dissimilar pumps may only be connected in parallel.

CHOICE OF PUMP

For pumps with steep characteristics, changes in pressure developed produce only small changes in flow rate. This is useful where pipes tend to scale up.

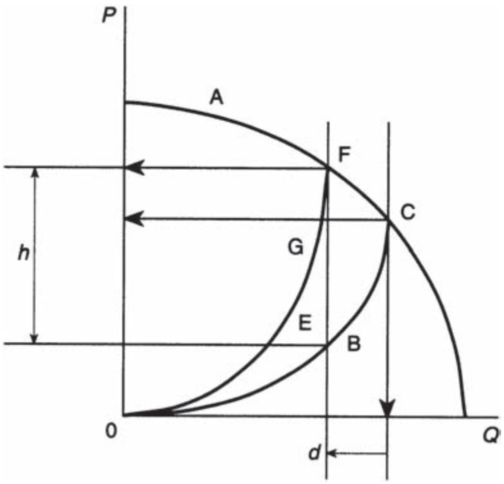


Figure 3.2 The regulating valve produces the required pd *(h)* to achieve design flow *(d)*. A, pump characteristic; B, system characteristic; C, operating point of the pump on the system; *d,* design flow rate; E, original design conditions for the system; F, final operating point of the pump on the system; G, final location of the system characteristic; *h,* required pd across the regulating valve to achieve design flow.

For pumps with flat characteristics, changes in flow produce small changes in pump pressure developed. This might be useful where extensive hydraulic balancing is needed, as changes in flow rate will be small in the operating range of the pump.

For closed systems the pressure developed at zero flow—that is, maximum pump pressure—should be greater than the static height of the system to ensure initiation of flow. Selection should be based upon the intersection of pump and system characteristic at design flow at the point where the pump efficiency curve is at or near its maximum.

PUMP TYPE

There are a variety of pump types: reciprocating, rotary gear, rotary moyno screw, centrifugal single stage, centrifugal multistage, inline and wet rotor. There are also different motor/impeller coupling arrangements: direct coupled, close coupled, belt drive, and wet rotor (rotating magnetic field).

Each of these pump types has a traditional application. Reciprocating pumps, for example, were used exclusively for steam boiler feed because they provide positive displacement. Rotary gear pumps are used to pressurize oil feed lines. Wet rotor pumps tend to be used on domestic and small commercial heating

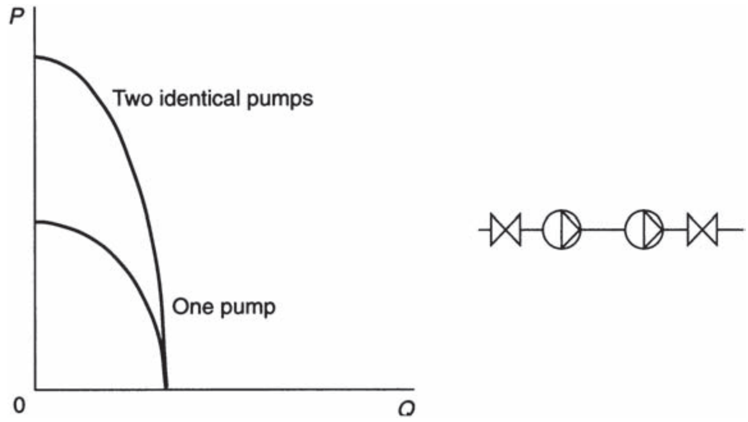


Figure 3.3 Similar pumps connected in series.

systems. Multistage centrifugal pumps are now used exclusively where high pressures are required, as in the case of boosted cold water systems.

The single-stage centrifugal and inline pumps currently have the major market share in the services industry. Recourse should be made to current manufacturers' literature if your knowledge of pumps is limited, before investing further time in this study.

3.4 Pumps on closed systems

(Pumps on open systems are considered in Chapter 9.)

This chapter will focus now on circulating pumps for space-heating systems, both pressurized and vented. As already stated, the net pressure developed by the

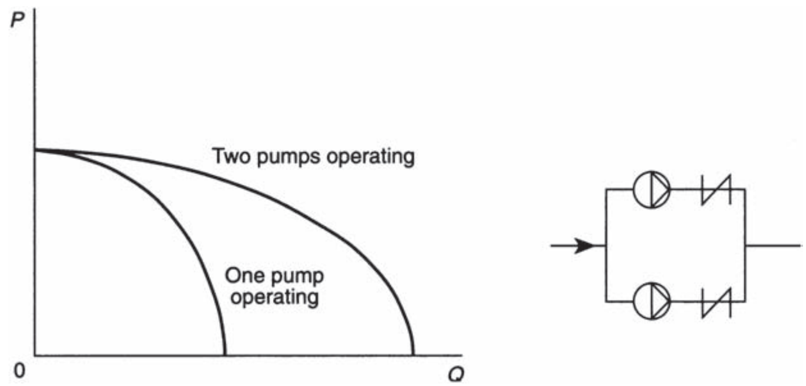


Figure 3.4 Similar pumps connected in parallel.

pump must just equal the hydraulic resistance in the index run, which is that circuit in the system having the greatest pressure drop. Often it is the longest circuit in the system. Pump capacity is the total flow rate in that part of the system that the pump is serving. Pump duty must specify flow rate *and* pressure developed.

THE NEUTRAL POINT

Consider the system shown in Figure 3.5. When the pump runs some water is removed from the feed and expansion tank and is deposited in the open vent. This causes a noticeable rise in water level in the open vent, but the corresponding fall in water level in the tank is negligible, owing to the difference in cross-sectional areas. Thus the pressure change at the cold feed entry to the system is also negligible, whether the pump is operating or not. This point is known as the **neutral point** in the system. It is here that pump pressure passes through zero from positive to negative pressure. From this knowledge it is now possible to identify the positive and negative pump effects around the system.

The systems shown in Figures 3.6 and 3.7 identify different arrangements for pump and cold feed location and the positive and negative pump effects at different points around each system.

Clearly, when the pump operates, the system is under the algebraic sum of static pressure and pump pressure at any point. It is this combined pressure that is measured on a pressure gauge. A matter for investigation occurs when at a point in a system static pressure is low (usually high-level pipework) and *negative pump pressure is high,* as combined pressure or gauge pressure at that point may be

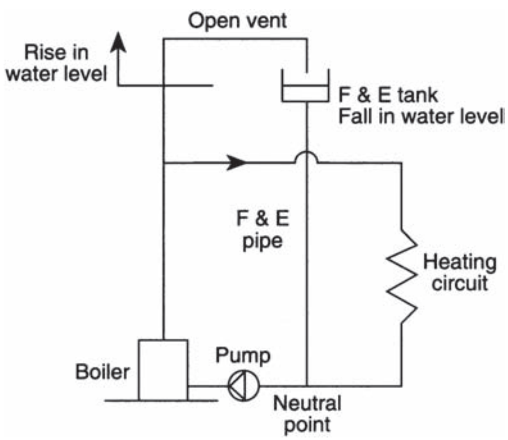
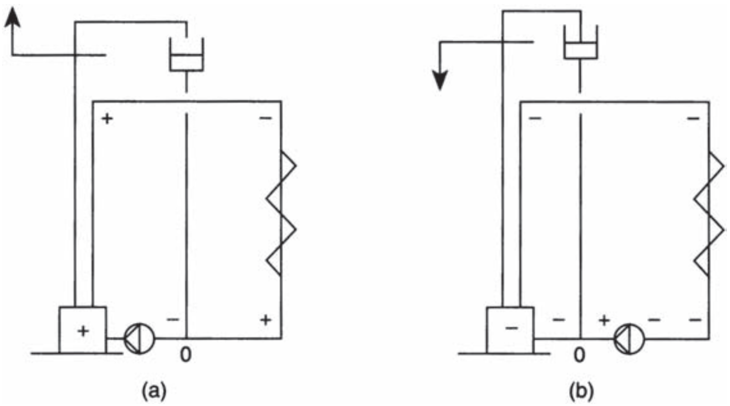


Figure 3.5 Identifying the neutral point in a closed system.



igure 3.6 Positive and negative pump effects, pump in the return: (a) F&E pipe near pump suction; (b) F&E pipe near pump discharge.

subatmospheric. This will have the effect of reducing the antiflash margin, and there is then the potential for the water at that point to cavitate.

Which of the four systems in Figures 3.6 and 3.7 is likely to produce this effect? Which system offers the safest pump location?

Note the location of the open vent connection and cold feed entry. Which is the best engineering solution?

An analysis of the pump effects in each system allows the following conclusions to be drawn.

1. Negative pump pressure in high-level circulating pipes should be avoided, as it reduces the antiflash margin.
2. Negative pump pressure in the boiler, where static head is low (rooftop plant rooms are a case in point), should be avoided for the same reasons. One way of overcoming this is to pressurize the system.

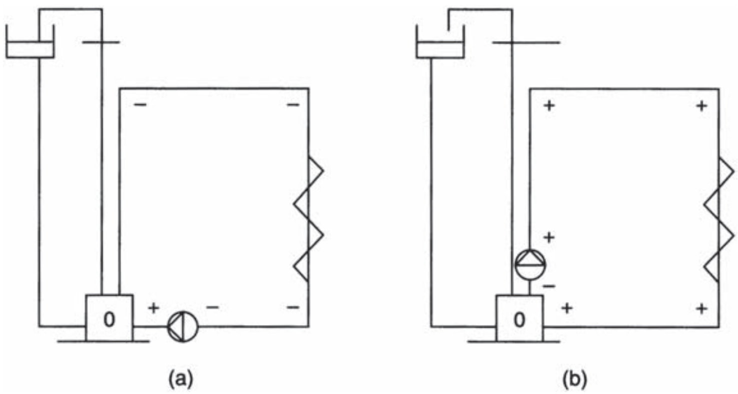


Figure 3.7 Positive and negative pump effects, F&E pipe connected to boiler: (a) pump located in boiler return; (b) pump located in boiler flow.

3. Positive pump pressure at the open vent may cause discharge.
4. Severe negative pump pressure at the open vent may cause air to be drawn into the system at the open vent connection to the system.
5. The open vent should not be extended through the roof to offset the effects of point 3 in case of freezing.
6. Positive pump pressure in the system assists venting. If this is the case, the pump can be operated during system venting.
7. Negative pump pressure in the system does not assist venting—it may allow air in—and therefore it should not be operational during this procedure.
8. The open vent is a safety pipe in case the boiler operating and limit thermostats fail. In this event the boiler water will eventually cavitate, as the burner will run continuously. The open vent therefore must be connected directly to the boiler so that steam can escape in these circumstances.
9. The cold feed and expansion pipe is used for initially filling the system and during heat-up allows expansion water to rise into the F&E tank. Thus during normal operation of the system the ball valve in the tank is submerged. This can cause it to stick in the closed position owing to the upthrust on the ball float, and during the summer make-up water cannot replace water in the system, which evaporates.

   As the expansion takes place at the heat source, which is the boiler, the F&E pipe should be connected to or close to the boiler.

You now have the skills to ensure an engineering design solution with respect to optimum pump/cold feed/open vent location. You should also be able to diagnose faults relating to the potential problems described here, in an existing installation, and offer suitable remedies.

USE OF THE REGULATING VALVE

If the pump is oversized as it should be, the operating point of the pump on the system will give a flow rate greater than design flow. The regulating valve, usually located on the pump discharge, is used to throttle the flow down to the design value on completion of the process of hydraulic balancing. The effect is to move the system characteristic up the pump curve from the operating point where the two characteristics intersect to the new intersection and final operating point. This increases the pump pressure developed by adding the pressure drop required at the regulating valve to achieve design flow (Figure 3.2).

SPEED REGULATION

As many pumps are constant-speed devices, they cannot respond to changes in load. This requires the use of a bypass if part of the load is shut down, as in the case of the use of two-port on/off control valves and thermostatic radiator valves. The bypass pipe will be fitted with a balanced pressure valve (BPV) to ensure that water can flow in the pump circuit at all times when the system is in operation (Figure 3.8).

Bypass pipe A–B prevents high differential pressure in the main distribution being transmitted to the circuit. The BPV, which is spring loaded and normally closed when the TRVs are open, ensures that when all the TRVs are closed the pump can still pump water through the circuit. However as soon as just one TRV starts to close, the other radiators will begin to receive more flow than they require and the circuit becomes unbalanced. This situation is sometimes ignored in design. When it is accounted for, the use of a differential pressure controller located in the flow and controlling a modulating valve in the return ensures that flow through the circuit and the pressure drop across it is correct at all times.

The alternative use of a variable-speed pump, operating on pressure signals, provides for savings in pumping costs during partial circuit load and overcomes some of the difficulties outlined above as long as a bypass is put across the index terminal in the circuit.

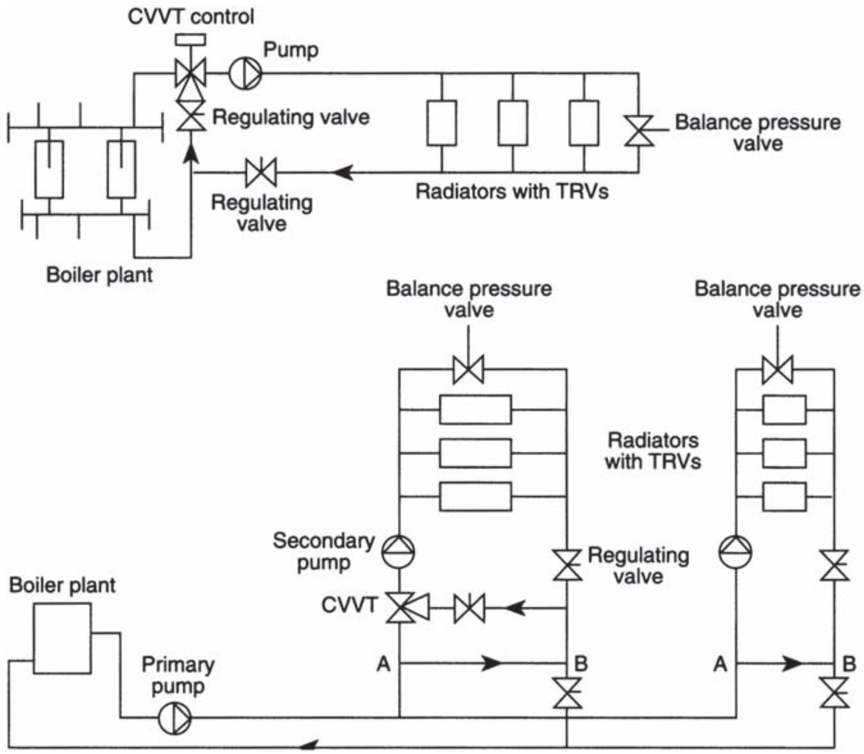


Figure 3.8 Application of balanced pressure valves in circuits subject to interruption of water flow (open vents, cold feeds and f&e tanks omitted).

3.5 Centrifugal pump laws

For centrifugal pumps there is a relationship between Q, N, P, P_w and D such that

$$Q \propto N,\ P \propto N^2, \text{ therefore } P \propto Q^2,$$

$$P_w \propto N^3, \text{ therefore } P_w \propto Q^3$$

and

$$D \propto N$$

One of these laws is useful in obtaining a series of values from the system design flow and index pd. From $P \propto Q^2$: $\frac{P_2}{P_1} = \left(\frac{Q_2}{Q_1}\right)^2$ and $P_2 = \frac{P_1}{Q_1^2} \times Q_2^2$

Thus

$$P_2 = K \times Q_2^2$$

$$P_2 = K \times Q_2^2$$

where $K = \frac{P_1}{Q_1^2}$.

$K = \frac{P_1}{Q_1^2}$.

Using this formula allows other values of system P and Q to be generated for plotting the system characteristic although most solutions can be done without it.

Example 3.1

A heating system has design conditions of 1.5 l/s at 50 kPa. A circulating pump having the characteristic detailed is installed. By plotting the system and pump characteristics determine:

(a) the operating point of the pump on the system;
(b) the pd required across the regulating valve to achieve design flow;
(c) the final operating point of the pump on the system.

Pump characteristic:

P	0	20	40	60	80	100 kPa
Q	3	2.8	2.45	1.95	1.2	0 l/s

Solution

From the derived equation above additional points for plotting the system characteristic can be evaluated:

$$K = \frac{50}{(1.5)^2} = 22.22$$

See Table 3.1.

(a) You should now plot the pump and system characteristics. The operating point of the pump on the system where they intersect is 1.75 l/s at 66 kPa (Figure 3.2).

Table 3.1

Q^2	Q_2^2	K	P_2
0.5	0.225	22.22	5
1.0	1.0	22.22	22.22
1.5	2.25	22.22	50.0
2.0	4.0	22.22	89.0

(b) By projecting a vertical line upwards from the point of design flow (1.5 l/s) on your graph it intersects the pump characteristic at the final operating point of the pump on the system. Refer again to Figure 3.2. The final location of the system characteristic can now be sketched on your graph by hand. Two horizontal lines can now be drawn back to the pressure axis of your graph and the required regulating pd read from the pressure scale. This should come to

$$72.5\text{-}50{=}22.5 \text{ kPa}$$

(c) From your graph, which should now look similar to Figure 3.2, the final operating point of the pump on the system after regulation is 1.5 l/s at 72.5 kPa.

Note: The vertical line from the design flow on your graph and in Figure 3.2 intersects two points: the original system characteristic at a pressure of 50 kPa, and the pump curve at 72.5 kPa. The first intersection is at the design pressure drop.

Example 3.2

A pump having the following characteristic is to be employed in a system whose design conditions are 3.5 l/s at 24 kPa. Determine whether or not two identical pumps operating together are required and state whether a series or parallel arrangement should be adopted. Assess the pressure reduction required to achieve design flow.

Pump characteristic:

P	0	10	20	30	40	50	60 kPa
Q	3.0	2.85	2.6	2.28	1.85	1.3	01/s

Solution

For the series arrangement the pressure developed is doubled. Refer to Figure 3.3:

$2P$	0	20	40	60	80	100	120 kPa

For the parallel arrangement the flow rate is doubled. Refer to Figure 3.4:

$2Q$	6.0	5.7	5.2	4.56	3.7	2.6 0 l/s

By analysing the options *before* plotting the pump characteristics it is clear that:

1. a single pump is not big enough, as the maximum flow of 3.0 l/s is achieved at zero pressure;
2. two pumps simultaneously operating in series do not increase the flow above 3.0 l/s;
3. two pumps operating simultaneously in parallel achieve 3.7 l/s at 40 kPa, which is in excess of the design conditions and is therefore worthy of investigation.

The solution therefore is obtained from the two pumps operating simultaneously in parallel.

After obtaining further values for the system design condition of 3.5 l/s at 24 kPa using $K=P_1/(Q_1)^2=47$, the system and *parallel* pump characteristics are plotted, and from the graph the regulation required to achieve design flow is

$$42\text{-}24=18 \text{ kPa}$$

It is recommended that you now plot the characteristics and check the above solution. What is the final operating point of the pump on the system? You should have 3.5 l/s at 42 kPa.

Example 3.3

Consider the system shown in Figure 3.9(a). Design flow rate is 2.5 l/s and the system and pump characteristics are given in the data below.

Analyse the pump pressure distribution around the system and determine whether or not there will be discharge at the open vent.

If the regulating valve is relocated to the pump discharge will the system operate satisfactorily?

Data

Hydraulic resistances around the system:

Section	A–B	B–C	C–D	D–E	E–F	F–A
Resistance (kPa)	2	2	8	2	2	3

Pump characteristic:

P	50	48	**43.5**	**32.5**	23.5	0 kPa
Q	0	1	**2**	**3**	3.5	4 l/s

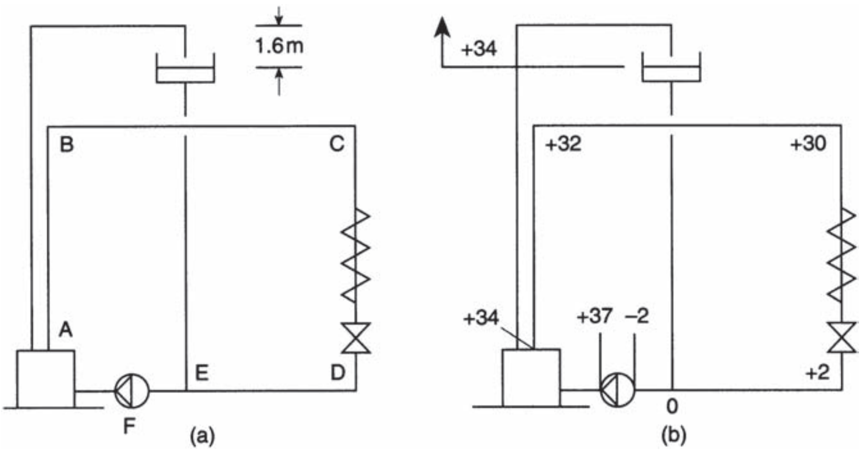


Figure 3.9 Example 3.3.

Solution

The sum of the hydraulic resistances around the system identifies the net pump pressure required, as the single circuit forms the index run. Thus the system design conditions are 2.5 l/s at 19 kPa. Now look at the pump characteristic data and you will see that the system design conditions fall between the values in bold type, so the proposed pump is clearly big enough.

From the system design conditions other values can be calculated as shown in Example 3.1, and the system and pump characteristics can now be plotted and system regulation considered as shown in Figure 3.2.

The operating point of the pump on the system is 3.2 l/s at 30 kPa. By regulating back to the design flow of 2.5 l/s the pump pressure developed is 39 kPa and the regulation required is 20 kPa.

Knowing the pump pressure under operating conditions we can now plot the pump pressure effects around the system as shown in Figure 3.9b.

Pump pressure at the pump outlet will be positive, and the pump pressure effects around the system will therefore be positive up to the neutral point at the cold feed entry, where it must change to a negative pump effect. Remember, at the neutral point pump pressure changes from positive to negative or vice versa. From the data the hydraulic resistance from point E to point F is 2 kPa; this will have a negative value and thus at the pump inlet the pump pressure is -2 kPa. The pressure rise through the pump is 39 kPa: thus the discharge pump pressure will be +37 kPa. By using the other section resistances from the data the remaining pump pressure effects can now be plotted around the system.

Make sure you agree with the analysis in Figure 3.9b.

Now we need to consider what happens at the open vent. If water does *not* flow in the open vent pipe the residual pump pressure effect at point A will be the same at the original water level in the open vent pipe, namely +34 kPa. This is equivalent to a rise in water level of $h=P/\rho g$ metres. Thus

$$h=\frac{34\,000}{1000\times 9.81}=3.46\ \text{m}$$

Clearly water *will* flow in the open vent pipe as it is only 1.6 m above the water level and some of the residual pump pressure here will be absorbed. However, it would be foolhardy to leave the system as it is: the likelihood of flow in the vent pipe is overwhelming, as the height of the pipe above the water level in the F&E tank is only 1.6 m. If water flow takes place, circulation to the system will be reduced and the tank room will fill with vapour.

How would you rectify the situation? There are two or three solutions.

One of them is indicated in the question, for if the regulating valve is relocated to the pump discharge, section F–A now has a hydraulic resistance of 23 kPa and the pump effect at point A and hence at the water level in the open vent is reduced to 14 kPa, from which

$$h=\frac{14\,000}{1000\times 9.81}=1.43\ \text{m}$$

As the height of the vent above the water level in the tank is 1.6 m the system should operate satisfactorily.

Can you suggest other solutions to the problem of pumping over?

Example 3.4

A diagram of a system is shown in elevation in Figure 3.10(a). The design flow rate is 2.5 l/s and the system and pump characteristics are given in the data.

(a) Determine the operating point of the pump on the system.
(b) What is the pressure loss required across the regulating valve to achieve design flow?
(c) Analyse the pump pressure distribution around the system and determine whether or not air will be drawn in through the open vent at point A.

Data

System hydraulic resistances:

Section	A–B	B–C	C–D	D–E	E–F	F–A
Resistance (kPa)	2	2	8	2	2	3

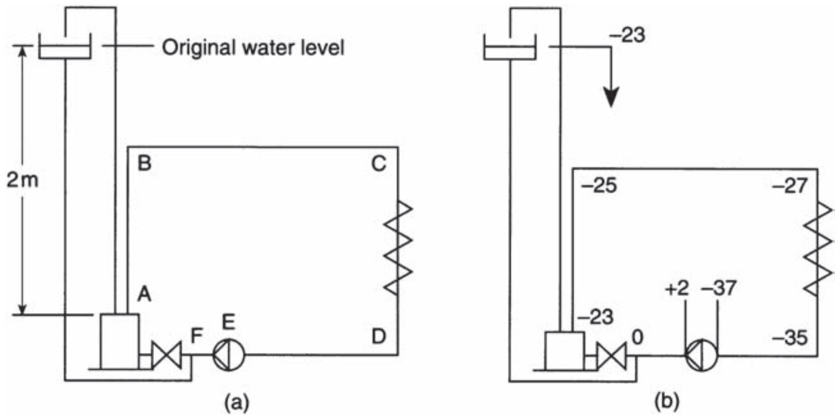


Figure 3.10 Example 3.4.

Pump characteristic:

P	50	48	43.5	32.5	23.5	0 kPa
Q	0	1	2	3	3.5	4 l/s

Solution

You will see that the system and pump characteristics are similar to those used in Example 3.3. The solution to parts (a) and (b) are therefore similar:

(a) The operating point of the pump on the system is 3.2 l/s at 30 kPa.
(b) The regulating pressure drop is 20 kPa to achieve design flow of 2.5 l/s. The pump duty is now 2.5 l/s at 39 kPa.
(c) Here the similarity ends, as the location of the cold feed is now at the pump discharge instead of the pump inlet. This requires another analysis of the pump effects around the system.

As the pump pressure changes from positive to negative at the neutral point F, pump discharge pressure will be +2 kPa, which from the data is the hydraulic resistance in pipe section E–F. For a pressure rise through the pump of 39 kPa the inlet pump pressure must be -37 kPa. The pump pressure effects can now be plotted anticlockwise from pump inlet E around the system or clockwise from the cold feed entry at point F, using the hydraulic resistances for each pipe section from the data.

The results are shown in Figure 3.10(b).

Using a similar argument as in Example 3.3 the negative pump effect at point A is transferred to the original water level in the open vent pipe under no-flow conditions. We are reminded that the reason for this is that under no flow conditions there is no loss in pump pressure. Thus the negative pump effect in the vent is 23 kPa, which is equivalent to

$$h = -\left(\frac{23\,000}{1000 \times 9.81}\right)\text{metres} = -2.34\text{ m}$$

As the F&E tank is only 2 m above point A, air will be drawn into the system at this point, and furthermore the pump will not operate in a stable manner.

Example 3.5

The diagram in Figure 3.11(a) shows a simple LTHW heating system in elevation. From the data determine:

(a) the operating point of the pump on the system;
(b) the presure loss required across the regulating valve to achieve design flow;
(c) the antiflash margin at point E in the system if water temperature at this point is 86 °C.

Data
System characteristic:

Section	A–B	B–C	C–D	D–E	E–F	F–A
Hydraulic resistance (kPa)	5	10	10	20	20	5

System design flow: 4.0 l/s
Pump characteristic:

P	120	114	105	95	83	67	45	0 kPa
Q	0	1	2	3	4	5	6	7 l/s

Steam tables data:

Absolute pressure (kPa)	110	100	80	70	60
Saturation temperature (°C)	102	100	93	90	86

Take atmospheric pressure as 100 kPa.

Solution

By summing up the hydraulic resistances in the index circuit, index pressure drop is 70 kPa at 4.0 l/s. Before proceeding further, look at the pump characteristic in the data, and you will see that the nearest duty is 83 kPa at 4.0 l/s. Clearly this pump should be suitable for the system. It also offers a flow rate that matches the system design flow. It is therefore necessary only to plot the pump and system characteristics to solve part (a) of the question.

(a) Adopting the appropriately adapted pump law ($P_2 = K \times Q_2^2$), a series of

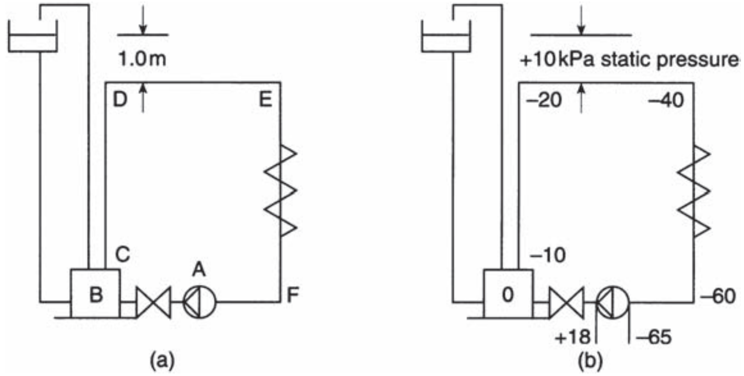


Figure 3.11 Example 3.5.

system pressure and flow readings are obtained and the system characteristic plotted along with the pump characteristic.

You should undertake this piece of work now.

From the graph, which should be similar to Figure 3.2, the operating point of the pump on the system is 4.25 l/s at 80 kPa.

(b) By drawing a vertical line on your graph through the design flow of 4.0 l/s, the required pressure drop across the regulating valve from the graph and from the system design conditions is 83-70=13 kPa.

(c) This solution requires an analysis of the pump pressure distribution around the system, which is shown in Figure 3.11(b).

The initial step commences at the neutral point, where the pump pressure effect passes through zero. Pump discharge pressure must therefore be +18 kPa to overcome the hydraulic resistance of pipe section A–B and that required of the regulating valve. For a pressure rise through the pump of 83 kPa, the inlet pump pressure must be -65 kPa. With this knowledge the pump pressure distribution around the system can now be found with the aid of the system section resistances.

At point E the algebraic sum of pump and static pressure is (+10+-40) kPa=-30 kPa gauge.

Clearly this is a case where subatmospheric pressures exist in this high-level pipe when the pump runs.

As absolute pressure P_{abs}=gauge P_g+atmospheric P_{at}, then absolute pressure will be -30 +100=70 kPa.

From the data in the question relating to the steam tables, the saturation temperature of water at 70 kPa abs is 90 °C. As the water temperature at point E is 86 °C, the antiflash margin is 90-86= 4 K.

This margin is too small; a minimum margin of 10 K is required, as water becomes unstable at 5 °C below its boiling point.

Therefore unless the configuration of the pump and cold feed are changed the maximum water temperature sustainable at point E will be 90-10=80 °C.

Note: Although the examples used here relate to vented systems so that pumping over and air ingress can be identified, the same procedure can be adopted for pressurized systems. In such cases the neutral point is where the cold feed and expansion pipe from the pressurization unit enter the system.

For pressurized systems, potential problems at the open vent do not arise. However, the effects of negative pump pressure around the system should be investigated to ensure that cavitation does not occur when the system is run up to operating temperature with the pump running.

The ideal location for the pump is in the flow as shown in Figure 3.7(b). Do you agree?

3.6 Chapter closure

This completes the work on pump and system. You now have the necessary skills to analyse the effects of different configurations of cold feed and pump locations in closed systems and to select the best position for the pump at the design stage. You are able to offer solutions to some of the problems associated with existing installations, such as continual air in the system, erratic operation of the pump, considerable noise in high-level pipes or from within the boiler (cavitation), discharge from the open vent, and air in high-level pipes at the beginning of the heating season.

Finally, you are now in a position to recommend a suitable pump type for a given application.

4 High-temperature hot water systems

Nomenclature

A	area of surface (m^2)
C	specific heat capacity (kJ/kg K)
CTVV	constant temperature variable volume
dp	pressure difference (Pa)
E	expansion volume (l, m^3)
F&E	feed and expansion
F_1, F_2	temperature interrelationships
h	static head (m)
HTHW	high-temperature hot water
K	constant
LTHW	low-temperature hot water
m	mass (kg)
N	air changes per hour
n	index
P	pressure (Pa, kPa)
P_N	partial pressure of nitrogen
P_T	total pressure
P_{wv}	partial pressure of water vapour
R	constant for nitrogen
T	absolute temperature (K)
t_c	dry resultant, comfort temperature (°C)
t_f	flow water temperature (°C)
t_i	indoor temperature (°C)
t_m	mean surface temperature (°C)
t_o	outdoor temperature (°C)
t_r	return water temperature (°C)
t_s	saturation temperature (°C)
U	thermal transmittance coefficient (W/m^2K)
V	volume (m^3)
γ	index
ρ_1	density at fill temperature (kg/m^3)
ρ_2	density at mean operating temperature (kg/m^3)
Σ	sum of

At the beginning of Chapter 2, Table 2.1 lists operating pressures and temperatures for very-low-temperature, low-temperature, medium-temperature and high-temperature systems, which puts the subject of wet space-heating systems into context.

4.1 Introduction

When water is elevated to a temperature in excess of its boiling point at atmospheric pressure (100 °C at sea level) for the purposes of space heating, the system must be pressurized to provide a minimum antiflash margin of 10 K at the weakest pressure point. This prevents cavitation within the system, keeping the water in its liquid phase. In most cases, high-temperature hot water heating therefore requires a pressurization unit. The specification of plant and fittings must be sufficient for these to operate under the enhanced pressure imposed.

HTHW offers a higher mean water temperature than LTHW, and as the rate of heat flow is dependent upon the magnitude of the temperature difference between the heat exchanger and the room in which it is located, it follows that smaller-diameter pipework can be employed to transport the heating medium for the same heat requirement.

APPLICATIONS OF HTHW

Clearly, the application of HTHW requires that the distribution is over substantial distances, to take advantage of the comparatively small pipe. Large sites, such as those for hospitals, airports and district heating, are worth consideration. The effect of using smaller distribution pipe is extended to lower costs for anchors, guides and brackets as well as pipe insulation and duct size. Weighed against this must be the use of higher-specification plant and fittings, the need for pressurization equipment, water treatment and regular water analysis, as precipitation and corrosion increase with water temperature. Temperature conversion may be required at various points on the site to bring the water temperature down to a lower level locally for radiators.

A final decision rests upon the economy offered by the distribution mains on the one hand and the additional costs on the other.

4.2 Pressurization methods

There are at least four ways in which a system may be pressurized: static head, air, nitrogen or steam. The most common is the use of nitrogen. Occasionally available static head is used for upgrading an existing installation. The use of nitrogen and static head will be considered here. There are a number of manufacturers of pressurization equipment, which is delivered to site in packaged format. You should familiarize yourself with manufacturers' details.

TEMPERATURE CONVERSION

If radiators are used as the space-heating appliances, local conversion from high to low temperature may be required. Ideally, appliances such as natural draught convectors, fan coil units, unit heaters and radiant tube or strip allow direct use of the high-temperature water, thus avoiding the cost of conversion plant.

There are two methods for space heating by which conversion is achieved, and these are shown in Figures 4.1 and 4.2.

Figure 4.1 illustrates the use of the injector valve. The primary circuit is the HTHW distribution and the secondary circuit is the LTHW system. Note that the secondary circuit is subject to the full HTHW pressure. If connection A–B is near the primary pump, d*p* A–B will be high. Without balance pipe 1, flow reversal can occur in pipe 2, as the secondary LTHW pump will develop a lower pressure than the primary pump. Pipe 3 mixes LTHW return with the flow at low load when there is no flow in pipe 2. When the LTHW load increases, water from the return also circulates through pipe 2, with the injector valve providing constant-temperature water to the secondary circuit, which may require further temperature control.

The valves shown are regulating except for the valve on the primary flow balance pipe, which is for isolating purposes.

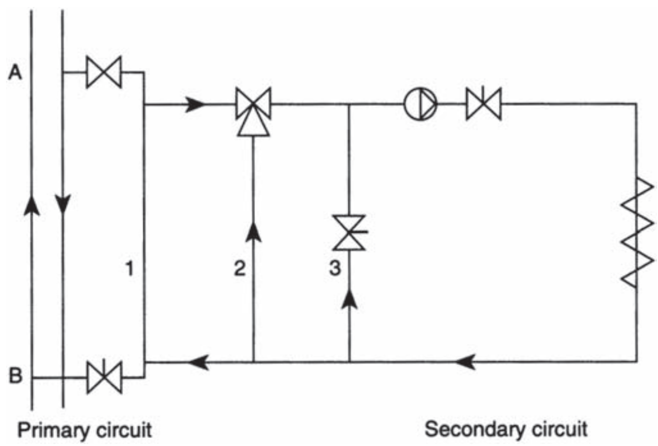


Figure 4.1 Use of the injector valve for conversion from high to low temperature.

Figure 4.2 shows temperature *and* pressure conversion using a non-storage heating calorifier, which effectively takes the place of the boiler. The three-way mixing valve on the HTHW primary circuit controls the LTHW secondary flow temperature at a constant value within the immersion thermostat's differential. A three-way mixing valve is shown on the secondary circuit to provide compensated control via an outdoor detector. You will see that the secondary system is filled and vented via a local F&E tank.

PRESSURIZATION BY STATIC HEAD

Where a higher output is required from an existing LTHW system it may be convenient to utilize the static head imposed by the F&E tank to raise the boiler flow temperature. Figure 4.3 shows the conversion. Note the expansion vessel required adjacent to the boiler. It should be sized to accept all the expansion water from the system, to prevent high-temperature water reaching the F&E tank. The tank accepts the cold water displaced from the expansion vessel on heat-up. The expansion vessel is fitted with sparge pipes to inhibit mixing of the contents during heat-up.

Clearly, the value of static head h needs to be substantial in order to raise boiler flow temperature much above 90 °C, to prevent cavitation in the high-level pipes (the weakest point in the system). One metre of static height h is equivalent to 10 kPa. Five metres is equivalent to half an atmosphere. Recourse to the saturation temperatures in the steam tables shows at 1.5 bar absolute (0.5 bar

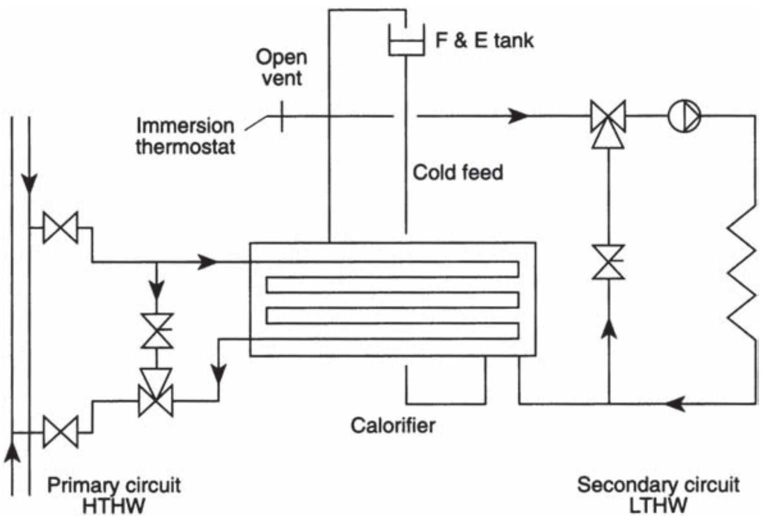


Figure 4.2 Use of the non-storage calorifier for conversion from high to low temperature and pressure.

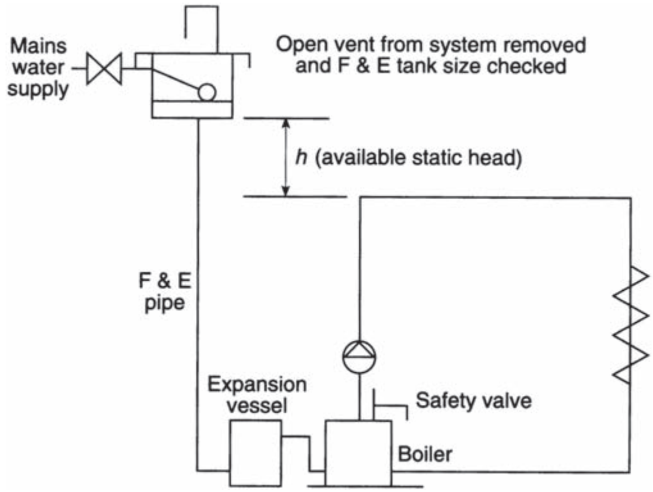


Figure 4.3 Pressurization by static head.

gauge) a value of 112 °C. With an antiflash margin of 12 K the boiler flow temperature can be raised to 100 °C.

Raising the boiler flow temperature can raise the mean water temperature and hence the system output as long as the appliances can operate at 100 °C. We are also assuming here that the boiler has sufficient capacity to service the additional output, otherwise that will need upgrading.

This scenario does not account for the effects of pump pressure around the system, which, when the pump operates, is subject to the algebraic sum of the static and pump pressures at any given point. Chapter 3 shows clearly that negative pump pressure can have an adverse effect upon the combined pressure at a point in the system—particularly along high level pipes. The combined effect of pump and static pressure needs to be used in an analysis of this kind before enhancing the system flow temperature.

Expansion volume *E* for sizing the vessel is obtained from:

$$E = V\left(\frac{\rho_1 - \rho_2}{\rho_2}\right) \tag{4.1}$$

where suffix 1=initial or fill temperature (°C); suffix 2=mean water temperature (°C); and *V*=system contents (litres, l, or m^3).

CONSEQUENCES OF UPGRADING

If limited data are available in a situation where a building's space-heating system is to be upgraded, a heat balance may be drawn with building heat loss and heat emission from the terminals.

For steady-state temperatures, heat loss=heat emission. Thus:

$$(\Sigma(UA)F_1+0.34NVF_2)(t_c-t_o)=KA(t_m-t_c)^n$$

Ignoring the constants:

$$(t_c-t_o)\propto(t_m-t_c)^n$$

Therefore

$$\frac{(t_c - t_o)_2}{(t_c - t_o)_1} = \frac{(t_m - t_c)_2^n}{(t_m - t_c)_1^n} \qquad (4.2)$$

Now consider the following example.

Example 4.1

If an LTHW heating system operating on natural draught convectors is to be upgraded, determine the probable indoor temperature achieved given that the original design conditions were: room t_c=14 °C, outdoor t_o=-1 °C, system flow t_f=80 °C and system return t_r=70 °C, and the proposed new design conditions are: t_o=-1 °C, t_f=100 °C and t_r=90 °C.

Solution

For the original design the difference between mean water and comfort temperature is 75-14=61 K. After upgrading, the temperature difference is greater in value but the new comfort temperature will rise above 14 °C.

Substituting the known data into equation (4.2):

$$\frac{t_c + 1}{14 + 1} = \frac{(95 - t_c)^{1.5}}{(75 - 14)^{1.5}}$$

From which

$$\frac{t_c + 1}{15} = \left(\frac{95 - t_c}{61}\right)^{1.5}$$

Table 4.1 Example 4.1

Comfort temperature (°)	*Heat loss, $(t_c +1)/15$*	*Heat emission, $((95- t_c)/61)^{1.5}$*
18	1.267	1.418
20	1.4	1.363
22	1.53	1.309
24	1.67	1.256

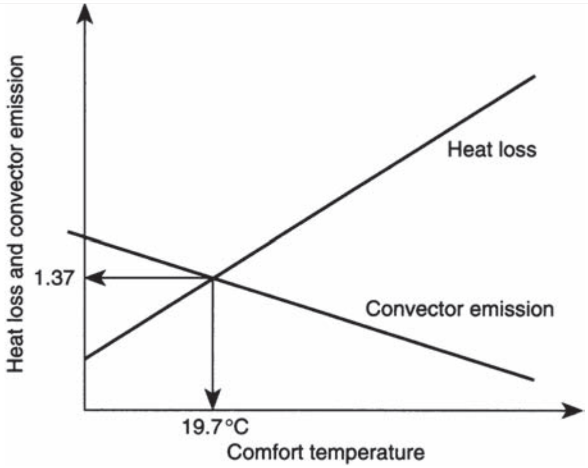


Figure 4.4 The plot for Example 4.1.

By giving values to t_c of say 18, 20, 22 and 24 °C, heat loss and heat emission can be evaluated (Table 4.1) and the results plotted to find probable comfort temperature. The plot is shown in Figure 4.4.

From the plot you will see that the heat loss and convector emission has increased by 37% while comfort temperature has risen from 14 to 19.7 °C.

Summarizing:

Condition	t_m	t_c	$(t_m\text{-}t_c)$
Design	75	14	61
Upgrade	95	19.7	75.3

The effect of the upgrading is an increase in comfort temperature and an increase in system output to offset the corresponding increase in heat loss.

PRESSURIZATION BY GAS

Nitrogen is normally used as the gas cushion because it is inert and therefore does not react with water at its interface in the expansion vessel. Small systems that are pressurized use expansion vessels fitted with a membrane to divide the water and gas cushion. Because of the low static head available for rooftop plant rooms, a pressurization unit would normally be used without necessarily elevating the flow temperature above 85 °C.

Some specifications call for the space-heating system to be pressurized as a matter of course, as this dispenses with the problems associated with the remoteness of the F&E tank and ball valve and the 'out of sight, out of mind' syndrome during maintenance. The pressurization unit, however, is always located close to the boiler plant.

DOMESTIC PRESSURIZATION

Figure 4.5 shows a domestic heating system that is pressurized and connected directly to the mains water service.

Where direct connection to the rising main has not been made, a hose union connection is an alternative. A regular check on the pressure gauge is necessary, however, so that topping up can be undertaken manually.

COMMERCIAL PRESSURIZATION

Pressurization units come as a package, which includes duty feed pump and standby, expansion vessel, feed tank, pressure switches and pressure gauge. It is only a matter of locating the unit adjacent to the boiler plant, and connecting the feed pipe from the unit to the boiler or system return and the mains water service to the ball valve connection on the feed tank.

The expansion vessel and pump need to be sized and the high and low pressures determined for selecting the pressure switches. These matters relate to the system to which the pressurization unit is to be connected, and depend upon the selected design flow and return temperatures, water content and the combined effects of the static pressure and pump pressure around the system.

A typical arrangement is shown in Figure 4.6.

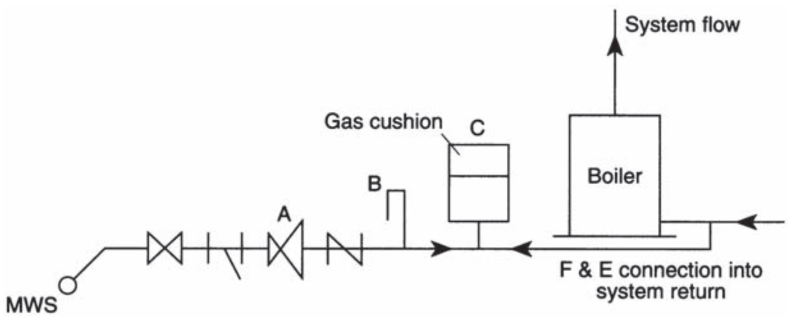


Figure 4.5 Domestic pressurization off the rising main: A=pressure-limiting valve; B=relief valve; C=expansion vessel.

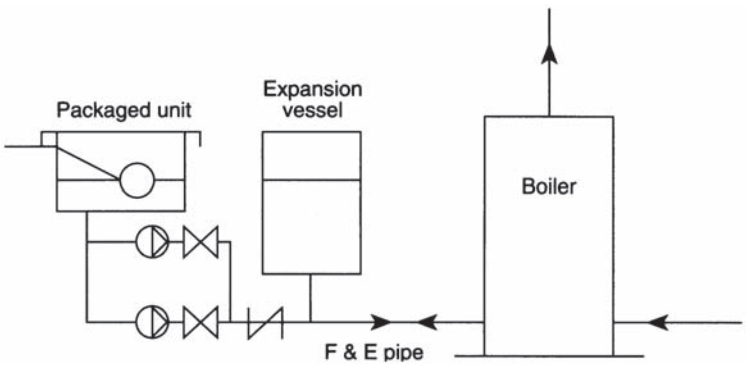


Figure 4.6 Commercial pressurization.

THE NEUTRAL POINT

It is worth reminding you that the neutral point occurs at the entry of the F&E pipe into the system. This is important when considering the combined effects of system, pump and static pressures.

4.3 Pressurization of large systems

Practice in the UK adopts the pressurization unit with the spill tank, whereas in the rest of Europe the expansion vessel is sized to accept all the system expansion water, and therefore the unit is similar to that shown in Figure 4.6. The expansion vessel can become quite large as a result. To prevent this happening, and to avoid the costs of the expansion vessel, which is classed for insurance purposes as a pressure vessel, the spill tank takes most of the system expansion water, thus making the expansion vessel relatively small in size. Figure 4.7 shows this type of unit.

As the system water is heated, the expansion water enters the F&E pipe and passes into the expansion vessel. The high-pressure controller eventually activates the spill valve, and further expansion water is diverted to the spill tank until system operating temperature is reached. Excess pressure is relieved after initial start-up by using the hose union valve and running unwanted water to drain. When the system load fluctuates, the quantity of expansion water varies. On a fall in system heating load, the system water contracts, resulting in a fall-off in pressure. The low-pressure controller is activated, and the feed pump draws water from the spill tank to replenish the system. The pressure difference between the high-pressure and low-pressure controllers must be significant—usually about one atmosphere—to ensure that they operate correctly.

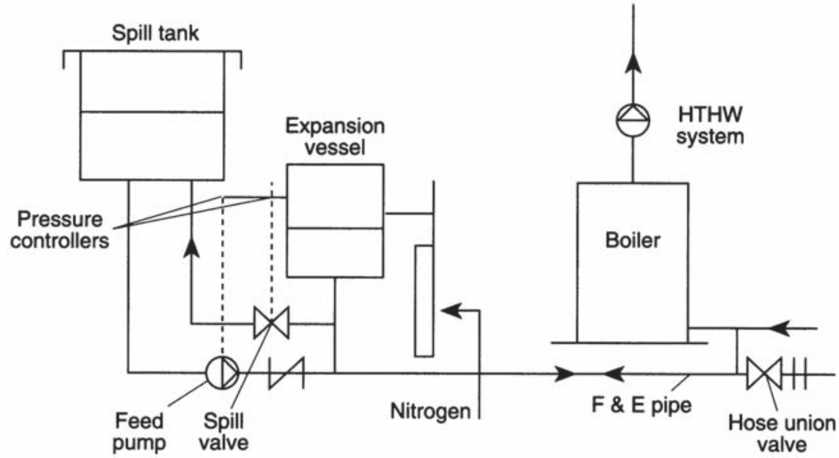


Figure 4.7 Pressurization unit with spill tank.

SIZING THE PRESSURIZATION UNIT

This would normally be done by the manufacturer of the pressurization unit. However, you will have to provide the system specification so that the manufacturer can select the appropriate packaged unit. It is therefore useful to have some knowledge of how selection is made.

System filling and operating cycle

If the filling process takes place quickly with the expansion vessel insulated it will be close to adiabatic, and $PV^{\gamma}=C$.

If the filling process takes place slowly with the vessel uninsulated, it will be close to isothermal and $PV=C$.

Expansion vessels are always left uninsulated, and the time taken to fill is controlled: so the process is polytropic, where $PV^{n}=C$.

The indices $\gamma>n>1$, where index n for the polytropic process is taken as being 1.26.

To size the expansion vessel one of the above laws is adopted. During the filling process there are three stages:

1. system filled—unpressurized;
2. system filled—pressurized and at fill temperature;
3. system pressurized and running at operating temperature.

Consider the three stages in Figure 4.8.

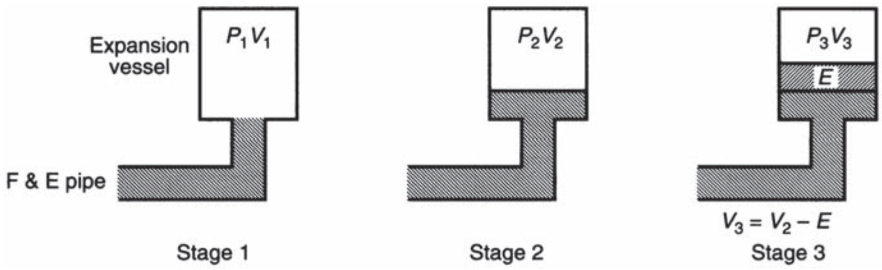


Figure 4.8 The three stages in system pressurization.

SIZING THE EXPANSION VESSEL

Conveniently, V_1 in Figure 4.8 is vessel volume or size as well as the initial gas volume. The other gas volumes are unknown, but the pressure P at each stage is known or can be calculated. A formula needs to be derived, therefore, for determination of vessel size in terms of the stage pressures and the expansion volume E for the system, which can be calculated from equation (4.1).

Using Boyle's law, $PV{=}C$, assuming initially, isothermal conditions.

Now

$$P_2V_2{=}P_3V_3{=}P_3(V_2\text{-}E)$$

Then

$$P_2V_2{=}P_3V_3\text{-}P_3E$$

and

$$P_3E{=}P_3V_2\text{-}P_2V_2$$

So

$$P_3E{=}V_2(P_2\text{-}P_2)$$

from which

$$V_2 = \frac{P_3E}{P_3 - P_2}$$

Now

$$P_1V_1{=}P_2V_2$$

Substituting for V_2:

$$P_1\,V_1 = P_2\left(\frac{P_3E}{P_3 - P_2}\right)$$

from which

$$V_1 = \frac{P_2}{P_1} \times \frac{P_3 E}{P_3 - P_2} \tag{4.3}$$

This is the volume of gas at stage 1 in the pressurization process. It also conveniently is the size of the expansion vessel.

Thus the expansion vessel can be sized from a knowledge of stage pressures and the system expansion volume.

The equation is found in some texts in the following format:

$$V_1 = \frac{P_2}{P_1} \times \frac{E}{(1 - P_2/P_3)} \tag{4.4}$$

Can you reconcile this formula with the derived equation (4.3)?

It is now a simple matter to amend the formula for a polytropic process by introducing index *n:*

$$V_1 = \left(\frac{P_2}{P_1}\right)^{1/n} \times \frac{E}{1 - (P_2/P_3)^{1/n}} \tag{4.5}$$

Can you reconcile this formula with equation (4.4)? Look closely at the derivation of equation (4.3).

EFFECTS OF ALTITUDE

Gravitational acceleration g is taken as 9.81 m/s^2 at sea level. Clearly this is reduced in value with the effects of altitude and therefore should be borne in mind in the design of systems in locations much above sea level.

Example 4.2

Figure 4.9 shows a simple HTHW heating system in elevation. The system data are as follows.

Data

Proposed boiler flow temperature	165 °C
Filling pressure	1.5 bar gauge
System contents	2200 litres
Density on filling	1000 kg/m^3
Density at mean water operating temperature	890 kg/m^3
Antiflash margin	15 K

Section	A–B	B–C	C–D	D–E	E–F	F–A
Hydraulic resistance (kPa)	2	4	5	15	3	4

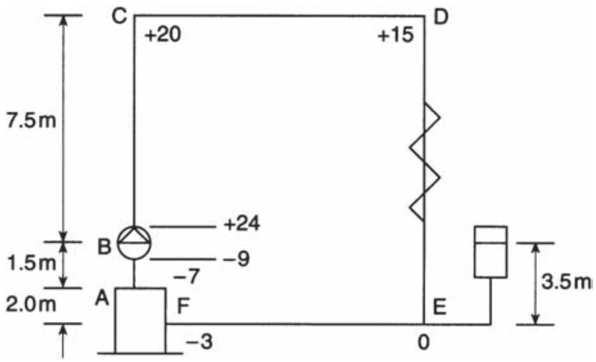


Figure 4.9 Example 4.2: HTHW system pressurized by gas with the pump pressure effects added.

You are asked to check the antiflash margin at the weakest point in the system, determine the minimum operating pressure in the expansion vessel, size the expansion vessel and calculate the mass of nitrogen required for the gas cushion. Assume the pump is operating and then consider the effects of pump failure.

Solution

The hydraulic resistances around the index circuit total 33 kPa, and with the neutral point in the system at E the pump pressure effects can be added to the diagram as shown in Figure 4.9. If you have any difficulty here go back to Chapter 3 and look again at the examples.

The static heights need to be converted to static pressures working from point A:

$$1.5 \text{ m to point B is equivalent to } 1.5\times890\times9.81=13 \text{ kPa}$$

$$9.0 \text{ m to points C–D is equivalent to } 9.0\times9.81=79 \text{ kPa}$$

$$\text{point E is 2.0 m below point A. This is equivalent to } 2.0\times890\times9.81=17\text{kPa}$$

Datum pressure

Point A in Figure 4.9 will be considered the datum, and datum pressure here will need to support with safety the proposed boiler flow temperature of 165 °C. Applying the antiflash margin of 15 K, the supporting pressure will correspond to a saturation temperature of 165+15=180 °C, and from the steam tables this is 10 bar absolute, 9.0 bar gauge.

This *includes* the static pressure imposed by the height of the system above point A. Static, pump and combined pressures can now be tabulated around the system (Table 4.2).

Table 4.2 Example 4.2: pressures round the system (kPa)

Point	*Static*	*Pump*	*Combined*
A	0	−7	−7
B inlet	−13	−9	−22
B outlet	−13	+24	+11
C	−79	+20	−59
D	−79	+15	**−64**
E	+17	0	+17
F	+17	−3	+14

The weakest point in the system

The combined pressures are now considered for pump running conditions. However, these pressures are about the datum at point A of 10 bar absolute, 9 bar gauge. The weakest point in the system is at D, where the combined static and pump pressure is -64 kPa or -0.64 bar. The gauge pressure at this point, taking datum pressure into account, will therefore be: datum minus the combined pressure at point D. That is:

$$9-0.64=8.36 \text{ bar gauge}$$

From the steam tables, the saturation temperature at this pressure is approximately 177 °C. Flow temperature at point D is likely to be close to the boiler flow of 165 °C, so the antiflash margin at this point is (177 -165)=12 K. The minimum acceptable antiflash margin is 10 K, so the proposed boiler flow temperature for this system is acceptable.

Operating pressure in the expansion vessel, P_3

The operating pressure in the expansion vessel can also be determined from the tabulated analysis. At point E the combined pressure is +17 kPa, so gauge pressure here will be datum plus the pressure at point E, namely 9+0.17=9.17 bar gauge. However, the water level in the expansion vessel is 3.5 m above this point. This is equivalent to a loss of 30 kPa static pressure. Thus the pressure at the water level will be:

$$9.17-0.3=8.87 \text{ bar gauge, } 9.87 \text{ bar absolute}$$

This is the **operating pressure,** P_3, in the expansion vessel.

If it is essential to maintain an antiflash margin of 15 K at point D with the pump running, boiler pressure (that is, *datum* pressure) would have to be increased from 9 bar gauge to 9.64 bar gauge, and the resultant pressure at the water level in the expansion vessel, P_3=9.51 bar gauge, 10.51 bar absolute.

Sizing the expansion vessel

Initial pressure P_1 can be taken as 1 atm, 1.0 bar absolute. Filling pressure P_2

is given as 1.5 bar gauge, 2.5 bar absolute. Expansion volume E is obtained from equation (4.1):

$$E = \frac{2200(1000 - 890)}{890} = 272 \text{ litres}$$

Substituting the stage pressures and the system expansion volume for a 15 K antiflash margin at point D, adopting equation (4.3) for isothermal conditions:

$$V_1 = \frac{2.5}{1.0} \times \frac{10.51 \times 272}{10.5 - 2.5} = 893 \text{ litres}$$

For a 12 K antiflash margin at point D:

$$V_1 = \frac{2.5}{1.0} \times \frac{9.87 \times 272}{9.87 - 2.5} = 911 \text{ litres.}$$

Mass of nitrogen
If the vessel accepts all the expansion volume, in this case 2721, the mass of nitrogen m_3 can be determined from the **characteristic gas equation,** which states that

$$PV=mRT$$

Now, as $P_1V_1=P_3V_3=m_3RT_3$:

$$m_3 = \frac{P_1V_1}{RT_3}$$

If water/gas temperature within the vessel is constant at 20 °C, T_3=293 K. P1 is taken as 1 atm, and consists of the sum of the partial pressures of the water vapour and the nitrogen. At 20 °C the partial pressure of the water vapour is negligible. You should check this fact in the steam tables. Thus $P_1=P_N$=1.0 bar absolute=100000 Pa. The gas constant R for nitrogen=297, and taking V_1 as 8931, which is 0.893 m^3, the mass of nitrogen forming the gas cushion will be

$$m = \frac{100\,000 \times 0.893}{297 \times 293} = 1.026 \text{ kg}$$

If you are not familiar with the characteristic gas equation you should note the units of the terms in the formula.

Pump failure
If the pump fails, the system reverts to static head conditions, and the weakest points occur at the lowest static pressure, which is along C-D.

From the tabulated analysis the pressure here will be datum minus 79 kPa.

operating pressure along C-D=9-0.79=8.21 bar gauge, 9.21 bar abs

From the steam tables, saturation temperature at this pressure is about 177 °C. Water temperature along C-D will be approximately the same as the boiler flow temperature, namely 165 °C.

Thus the antiflash margin=(177-165)=12 K, which is within the safety limit.

The polytropic process

This most nearly equates with practical system pressurization. Substituting the stage pressures for a minimum antiflash margin of 15 K throughout the system and expansion volume *E* into the polytropic equation (4.5) for the expansion vessel volume V_1:

$$V_1 = \left(\frac{2.5}{1}\right)^{1/1.26} \times \frac{272}{1-(2.5/10.51)^{1/1.26}} = 827 \text{ litres}$$

The vessel volume is a little less than that for the isothermal process.

Use of the spill tank

This example assumes that all the expansion water from the system enters the expansion vessel. If the pressurization unit includes the feature of a spill tank open to atmosphere, the expansion vessel is much reduced in size and cost.

Thus when, say, 75% of the system expansion water is diverted to the spill tank, the polytropic process yields the following solutions:

spill tank size=827×0.75=620 litres

expansion vessel size=827×0.25=207 litres

The results are summarized in Table 4.3.

Table 4.3 Example 4.2: summary of results

Pump	*Antiflash margin (K)*	*Weakest point in system*	*Pressure at weakest point (bar gauge)*	*Minimum operating pressure in expansion vessel (bar gauge)*	*Size of expansion vessel (l)*	*Mass of nitrogen (kg)*	*Process*
On	12 (15)	D	8.36	8.87 (9.51)	911 (893)	1.026	Isothermal
Off	12	C–D	8.21				Isothermal
On	15				827		Polytropic

Example 4.3

An expansion vessel has a 100 litres nitrogen cushion. If the water is initially at a temperature of 130 °C and the operating pressure P_T in the vessel under these conditions is 10 bar absolute, determine:

(a) the partial pressure of the nitrogen;
(b) the mass of nitrogen forming the gas cushion.

Solution

(a) Dalton's law of partial pressures in this context is $P_T=P_N+P_{WV}$. From the steam tables the partial pressure exerted by the water vapour at a temperature of 130 °C is 2.7 bar absolute. Thus the partial pressure of the nitrogen will be (10-2.7)=7.3 bar absolute.
(b) From Example 4.2, $m=PV/RT$ kg. Selecting the correct units before substitution:

$$m=\frac{730\,000\times 0.1}{297\times(273+130)}=0.61 \text{ kg}$$

Clearly, when the system reverts to ambient temperature there will be a fall-off in pressure within the expansion vessel due to the drop in the partial pressure of the water vapour. Either more nitrogen is required in the vessel or the feed pump will be energized to re-establish the operating pressure in the vessel.

Example 4.4

Figure 4.10 shows in elevation a diagrammatic view of an HTHW system pressurized by gas serving an air heater battery. The required flow temperature at the heater battery is 150 °C with a temperature drop across the battery of 20 K. An antiflash margin of 15 K is required at all points in the system under all operating conditions when the pump is running.

From the data, analyse the performance of the system and offer comment. Propose a minimum acceptable boiler operating pressure.

Data

Take water density as 918 kg/m^3.

Pipe section	A-B	B-C	C-D	D-E	E-A
Hydraulic resistance (kPa)	5	109	5	5	15

Solution and analysis

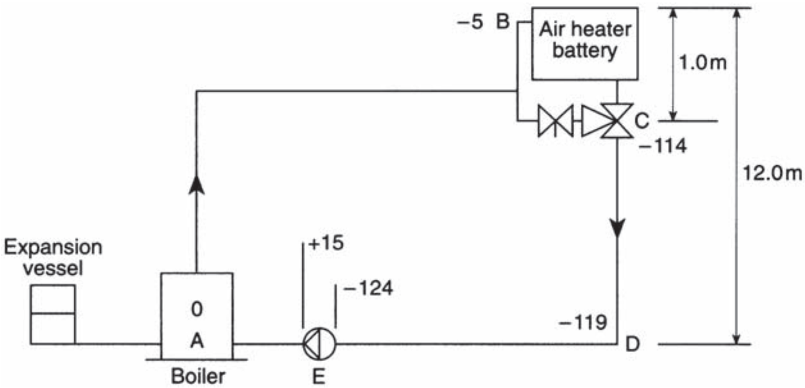


Figure 4.10 Example 4.4: HTHW system pressurized by gas with pump pressure effects added.

Converting the static heads to pressures:

$$\text{for 1 m, } P=1\times918\times9.81=9 \text{ kPa}$$

$$\text{for 12 m, } P=12\times918\times9.81=108 \text{ kPa}$$

The net pump pressure developed will be the sum of the index hydraulic losses=139 kPa.

Ignoring heat loss from the flow main to the battery, the boiler flow temperature will be 150 °C. Using the antiflash margin here of 15 K, the supporting pressure corresponding to a saturation temperature of (150 +15)=165 °C is 7 bar absolute. Taking this initially as datum at point A in the diagram, the static, pump and combined pressure effects can now be tabulated (Table 4.4).

The weakest point in the system is at C, where the pressure=7-2.13 =4.87 bar abs, and from the steam tables saturation temperature t_s= 151 °C.

When the battery load is not required, and this will occur on maximum rise in entering air temperature into the battery, the CTVV control will ensure that

Table 4.4 Example 4.4: pressure effects in the system

Point	*Static*	*Pump*	*Combined*
A	0	0	0
B	–108	–5	–113
C	–99	–114	**–213**
D	0	–119	–119
E in	0	–124	–124
E out	0	+15	+15

the system water bypasses the heater, and its temperature at point C will be the same as it is in the flow main namely 150 °C. Saturation temperature at this point is 151 °C, and serious cavitation will occur. Point D and point E at the pump inlet will also be reduced to approximately 8 K when the system is on full bypass. This is below the minimum acceptable value.

When the heater is operating at design load the system water temperature at point B will be 150 °C, and the antiflash margin at this point is about 8 K. This is below the minimum acceptable antiflash margin.

Check that you agree with the comments at points B, D, and pump inlet E.

Summary and way forward

Clearly, the supporting pressure selected initially is too low and needs to be increased. Taking the weakest point in the system at C and working on the maximum temperature that will occur at this point when the battery is on full bypass, namely 150 °C, and applying the minimum antiflash margin of 15 K means that the supporting pressure here must be equivalent to the saturation temperature of (150+15)=165 °C. This is 7.0 bar absolute. Thus the pressure at the boiler to maintain this pressure at point C when the pump operates must be (7.0+2.13)=9.13 bar absolute, 8.13 bar gauge. Refer to point C in Table 4.4 for confirmation. This is the minimum operating pressure for the boiler.

Check the antiflash margin around the system when the battery is under design load. Do you agree that the minimum operating pressure for the boiler is still 8.13 bar gauge? What effect does the location of the pump in the flow pipe have on the weakest point in the system?

With further data relating to the system it would now be a simple matter to size the expansion vessel.

4.4 Chapter closure

This completes the work on high-temperature hot water systems. You can advise on the suitable applications for HTHW and on its merits and demerits. You now have the skills necessary to check a proposed boiler flow temperature, to ensure an adequate antiflash margin around the system when the pump is operating and on pump failure, to select and size the pressurization unit, to consider the possibility of upgrading an existing system and recommend the necessary alterations. You will also be able to advise on the methods of temperature conversion and identify when it is necessary.

Steam systems 5

Nomenclature

A	surface area (m2).
d	diameter of pipe (m)
dh	difference in enthalpy (kJ/kg)
dp	pressure difference (Pa, kPa)
dt	temperature drop (K)
dZ	Z_1-Z_2
hf	sensible heat in condensate (kJ/kg)
h_{fg}	latent heat of saturated steam (kJ/kg)
h_g	enthalpy of saturated steam (kJ/kg)
h_w	heat in wet steam (kJ/kg)
HP	high pressure
HWS	hot water service
k	velocity pressure loss factor
k_t	total velocity pressure loss factor
K_6	velocity factor
l_e	equivalent length of pipe when k=1.0 (m)
L	length of pipe (m)
LP	low pressure
LTHW	low-temperature hot water
M	mass flow rate (kg/s)
M_f	mass flow of flash steam (kg/s)
P	pressure (Pa, kPa, bar)
pd	pressure difference (Pa, kPa)
PRV	pressure-reducing valve
q	dryness fraction
Q	heat output (kW)
TEL	total equivalent length of straight pipe (m)
u	velocity (m/s)
v	specific volume (m3/kg)
VFR	volume flow rate
V_g	specific volume of the saturated vapour (m3/kg)
Z_1, Z_2	initial and final pressure factors

5.1 Introduction

You are strongly advised to obtain manufacturers' literature relating to the application of steam and condensate equipment and fittings. Boiler manufacturers will provide literature for steam generators. Spirax Sarco will provide literature relating to steam fittings and the handling of condensate and also learning material for interested students.

As a general rule, steam used for space heating is considered only when it is required also for other uses, as in manufacturing processes, for sterilization or as exhaust steam from a turbine. It is rarely generated as a discrete space-heating medium. The applications for steam space heating are therefore limited to factories, chemical plants, hospitals, steam plants for the generation of electricity etc. The reason for this lies mainly in the fact that steam systems, for the purposes of economy, must normally return the condensate for reuse, and herein lies one of the disadvantages. Other disadvantages include plant response to fluctuating loads and the length of the start-up period from cold to operating temperature, and hence its inappropriate use for intermittent operation. There is also a need for water treatment and continuous analysis.

Its main advantage lies in the heat content of the latent heat of evaporation, which is used in the heat transfer process at the heat exchanger, and the high surface temperature, making the exchanger surface smaller than its LTHW counterpart. The rate of mass flow and distribution pipe sizes are different as well.

The mass flow rate of LTHW to serve a terminal whose output is 100 kW is 2.38 kg/s, which will require 50 mm flow and return distribution pipes. The corresponding mass flow of steam at 2.0 bar gauge 0.9 dry is 0.051 kg/s, which will require a 32 mm distribution pipe and a 20 mm condense return. Thus steam as a heat distribution medium is well qualified.

5.2 Steam systems

There are a number of different types of steam system. They include systems working under vacuum and pressurized conditions. Exhaust steam from turbines and process is used after cleaning to remove oil. Steam distribution with condensate returned separately is normal practice. However, there are a few older systems in which the steam and condensate are carried in the same pipe.

OPERATING PRESSURES

Steam is normally generated to pressures above atmospheric level. For space-heating use, low-pressure generation goes up to 3 bar gauge; high-pressure generation starts at around 6 bar gauge. If, as is normally the case, the steam is generated primarily for other uses, much higher pressures may be present, in which case a pressure-reducing valve may be employed on the connection to the space heating system.

To ensure a measure of storage to cope with fluctuations in demand, steam is generated to a higher pressure than required, with pressure-reducing valves used locally at the offtakes to steam-operated plant and heat exchangers.

STEAM PIPE-SIZING PROCEDURES

Steam velocity in distribution mains ranges from 30 m/s for a dryness fraction above 0.8 to 60 m/s for superheated steam. This compares with up to 10 m/s for low-pressure air systems and 1.5-3.0 m/s for LTHW heating systems in black mild steel. As with LTHW and air distribution systems the sizing procedure may be done on velocity or pressure drop. A typical pressure drop for steam distribution (up to 65 mm nominal bore) is 225 Pa/m.

From an inspection of the steam pipe-sizing tables, the pressure loss per metre corresponding to a steam velocity of 30 m/s progressively reduces with increases in pipe size above 65 mm, such that at 250 mm nominal bore it is around 60 Pa/m. It is therefore important to check the steam velocity if pipe sizing is initially undertaken on pressure drop.

STEAM GENERATION

Considering a two-pipe pressurized system, steam is generated from water within the boiler under its own pressure with the crown and isolating valves closed until steam is present and the required operating pressure is reached. The isolating valve is then slowly opened and then the crown valve opened *slowly* a quarter turn at a time to avoid a sudden pressure loss within the generator. Furthermore, on start-up the system is cold, and steam released condenses rapidly. Air and possibly water present in the system must be displaced by the condensing steam, and water and condensate must be allowed to travel at low velocity to avoid water hammer in the steam main, which can be a terrifying and expensive experience.

The warm-up time for a steam system is therefore extended. As steam supplies the system water must be fed into the boiler to maintain its water level. As the boiler is pressurized the boiler feed water must also be pressurized to ensure entry. A boiler feed pump is used for this purpose. It will be a reciprocating pump or, more likely now, a multistage centrifugal pump.

The condensate is returned to the plant room under the residual pressure from the steam supply or via a pumping and receiving unit to the hotwell, which is open to atmosphere, and it is from this tank that the feed pump draws the make-up boiler water. Figure 5.1 shows a two-pipe steam system diagrammatically.

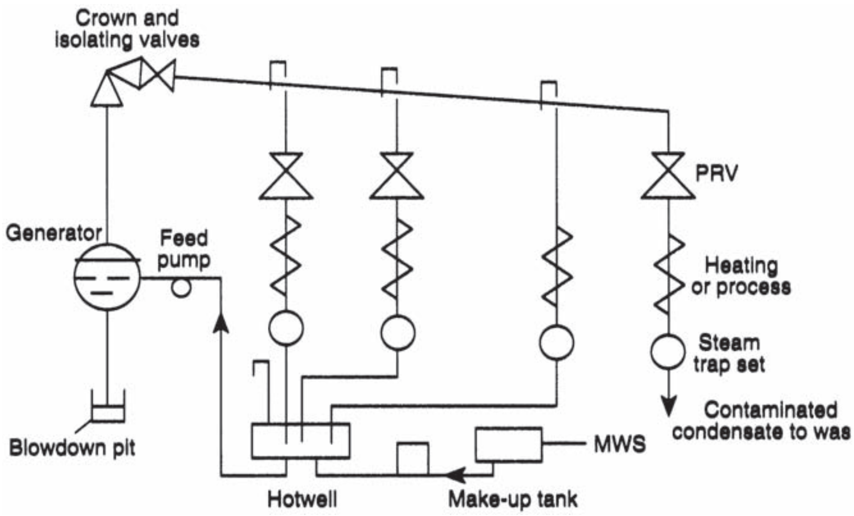


Figure 5.1 Two-pipe pressurized steam system.

The system shows three condensate returns to the hotwell, with one being diverted to waste due to contamination, which can occur in process work.

Make-up water will be required in any event owing to evaporation occurrence at the hotwell. Water treatment will be required for the make-up water, and it is sometimes needed also in the boiler feed line.

Continuous or intermittent blowdown is required to rid the generator of suspended solids deliberately formed by the treatment processes from the impurities in the water.

STEAM PIPE SIZING ON VELOCITY

Typical velocities are given below:

- superheated steam up to 60 m/s;
- saturated steam up to 40 m/s;
- wet steam with a dryness fraction above 0.8 up to 30 m/s;
- steam connections to plant and equipment up to 15 m/s.

A formula for determining steam pipe diameter for a given flow rate and pressure can easily be derived from first principles:

volume flow rate=steam velocity×cross-sectional area of pipe

$$\mathrm{VFR} = u \times A = u \times \frac{\pi d^2}{4}$$

$$\mathrm{VFR} = Mv$$

Thus

$$Mv = u \times \frac{\pi d^2}{4}$$

from which

$$d = \sqrt{\left(\frac{4}{\pi}\right)} \times \sqrt{\left(\frac{Mv}{u}\right)}$$

and

$$d=1.1284\times\sqrt{(Mv/u)} \qquad \text{(m)} \qquad (5.1)$$

As long as the dryness fraction q of the steam is above 0.8, specific volume $v=V_g\times q$, Otherwise specific volume v for the saturated vapour may be read directly from the steam tables as V_g.

Clearly this formula gives *actual* pipe diameter and not standard pipe diameter. The current *CIBSE Guide* adopts a velocity factor K_6 and a table for converting velocity factor values to *standard* pipe diameters.

From : $\mathbf{VFR} = u \times \frac{\pi d^2}{4}$

$$u = \frac{4Mv}{\pi d^2} = Mv \times \left(\frac{4}{\pi d^2}\right)$$

If K_6=(4/pd^2)×1/1000, then:

$$u=K_6\times vM\times 1000 \qquad \text{(m/s)} \qquad (5.2)$$

from which K_6 can be evaluated and standard pipe diameter obtained from the appropriate table in the *Guide*.

Example 5.1

An HWS calorifier stores 600 litres of water at 65 °C from feed water at 10 °C with a regeneration of 1.5 h. If the primary connection is supplied with steam at 3 bar absolute 0.9 dry, determine the mass flow of steam required, the size of the steam connection and the actual steam velocity. Assume a maximum steam velocity of 10 m/s and that condensate leaves the primary heat exchanger at 130 °C.

Ignore the inefficiency of the heat exchange.

Solution

The mass flow of steam M=output in kW/heat given up by the steam in kJ/kg. Thus

$$M = \frac{Q}{\mathrm{d}h} \qquad \text{(kg/s)} \qquad (5.3)$$

$$Q = \frac{600 \times 4.2 \times (65-10)}{1.5 \times 3600} \qquad \text{(kW)}$$

and

$$dh = h_w - h_{f_{130}}$$

and from the steam tables

$$dh=(0.9\times2164+561)-546 \qquad \text{(kJ/kg)}$$

Substituting:

$$M = \frac{25.67}{1963} = 0.0131 \text{ kg/s}$$

mass flow of steam=0.0131 kg/s

Now from equation (5.1):

$$d=1.1284\times\text{v}(Mv/u) \qquad \text{(m)}$$

From the steam tables

$$V_g=0.6057 \text{ m}^3\text{/kg}$$

Thus

$$v=V_g\times q=0.6057\times0.9$$

and

$$v=0.545 \text{ m}^3\text{/kg}$$

Substituting:

$$d = 1.1284 \times \sqrt{\left(\frac{0.0131 \times 0.545}{10}\right)}$$

from which

$$d=0.03013 \text{ m}$$

The nearest standard pipe diameter in mild steel is 32 mm or 0.032 m.

By rearranging equation (5.1) in terms of velocity u and substituting d=0.032 m, the *actual* steam velocity is evaluated to 8.88 m/s. Do you agree?

Summarizing:

- Mass flow of steam is 0.0131 kg/s.
- Standard pipe diameter in steel is 32 mm.
- Actual steam velocity is 8.88 m/s.

Further study

With access to the *CIBSE Guide* you should now find the pipe diameter and actual velocity using the velocity factor K_6 and the appropriate CIBSE conversion table to standard pipe diameters. Clearly the solutions will be similar.

Example 5.2
A steam main conveys 0.063 kg/s of saturated steam at an initial pressure of 2.7 bar absolute. The main is 70 m long and contains six welded bends and one stop valve. Assuming a maximum velocity of 40 m/s determine:

(a) the diameter of the main;
(b) the final steam pressure;
(c) the final steam velocity;
(d) the initial steam velocity.

Data
Velocity pressure loss factors *k:*
welded mild steel bends k=0.3
stop valve k=6.0

Solution

(a) Using equation (5.1), which determines actual pipe diameter, and substituting the data:

$$d = 1.1284\sqrt{\left(\frac{0.063 \times 0.6686}{40}\right)} = 0.0366 \text{ m}$$

The nearest standard diameter for steel pipe is 40 mm.

(b) You will now need access to the steam pipe sizing tables in the *Guide* in order to preceed with the solution.
Given the mass flow as 0.063 kg/s and pipe diameter of 40 mm:

$\frac{Z_1 - Z_2}{L} = 171$ and l_e=1.7 from the pipe-sizing table

The total equivalent length

$$TEL = L + k_t \times l_e \quad \text{(m)}$$

The total velocity pressure loss factor is given by

$$k_t = \text{six welded bends} + \text{one stop valve}$$
$$= 6 \times 0.3 + 1 \times 6.0 = 7.8$$

Thus

$$TEL = (70 + 1.7 \times 7.8) = 83.26 \text{ m}$$

Substituting in dZ/L=171:

$$dZ = 83.26 \times 171 = 14237$$

From the rubric in the pipe-sizing tables, pressure factor $Z=P^{1.929}$ when P is in kPa.

Thus

$$Z_1=270^{1.929}=48989$$

Now

$$Z_1\text{-}Z_2=14237$$

Substituting for Z_1:

$$48989\text{-}Z_2=14237$$

from which

$$Z_2=48989\text{-}14237=34752$$

and

$$P_2={}^{1.929}\sqrt{34752}=226 \text{ kPa}$$

Thus final steam pressure=2.26 bar absolute.

The steam velocities can be obtained from either of the equations derived for actual pipe diameter d or velocity factor K_6.

(c) From equation (5.1):

$$d^2=1.2733\times(Mv/u)$$

From the steam tables, V_g=0.79 m^3/kg at a final pressure of 2.26 bar absolute and pipe diameter d=0.04 m. Mass flow of steam is 0.063 kg/s and therefore by substitution:

$$\text{final steam velocity } u=39 \text{ m/s}$$

(d) From equation (5.2):

$$u=K_6\times M\times v\times 1000$$

The velocity factor table from the *Guide* gives K_6=0.786. Specific volume v at 2.7 bar absolute from the steam tables is 0.6686 m^3/kg for saturated steam, and therefore by substitution:

$$\text{initial steam velocity } u=33 \text{ m/s}$$

Conclusion

You will notice that the final steam velocity is greater than the initial velocity. This is due to the effect of increasing specific volume with decreasing steam pressure. See if you get a similar final velocity by employing equation (5.1).

The critical steam velocity in a section of steam distribution pipe is therefore the final velocity at the end of the pipe section.

The solution takes no account of heat loss from the steam main. It is of interest to know whether the quality of the steam suffers as it is transported over a distance of 70 m.

If the pipe is efficiently lagged with 25 mm of insulation having a. thermal conductivity of 0.055 W/mK, the heat emission from the main is 3146 W for a temperature difference of 107 K. This produces a dryness fraction at the end of the main of 0.997, which is close to saturated conditions at the final pressure of 2.26 bar absolute.

You might like to check this statement, in which case you will need the heat emission from the 40 mm pipe insulated as described, and this is 0.42 W/mK.

Example 5.3

A steam ring main follows the perimeter of a factory at high level with connections coming off the ring as shown in Figure 5.2. Steam is supplied at 2 bar gauge 0.9 dry. Size the ring main on an initial velocity of 30 m/s and the branches on 10 m/s. Determine the steam velocity in the ring between branches 6 and 7 assuming all branches are in use. State any assumptions made.

Solution

When all the connections are in use the total mass flow rate of steam required in the ring main will be the sum of the individual steam flows and M_T=0.173 kg/s. Adopting equation (5.1) and obtaining specific volume V_g from the steam tables at 3 bar absolute:

$$d=1.1284\times\sqrt{(0.173\times0.6057\times0.9/30)}=0.063 \text{ m}$$

Thus the standard diameter for the ring main d=65 mm.

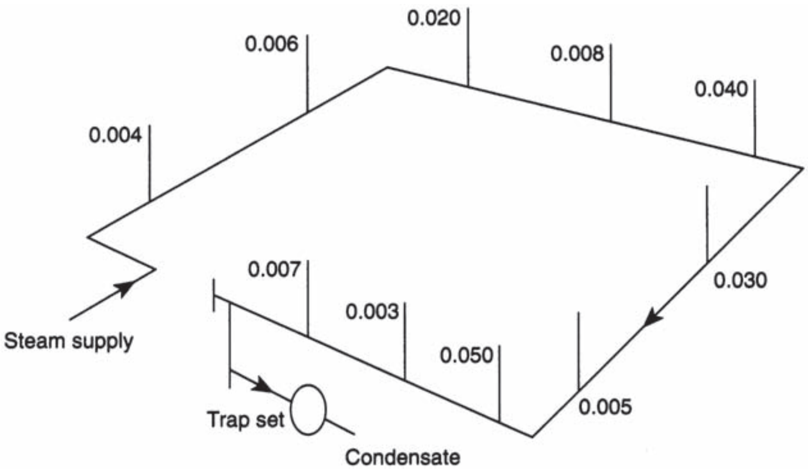


Figure 5.2 Ring main and mass flow rates.

Table 5.1 Example 5.3: standard pipe sizes

Branch no	*Branch*	*Standard diameter (mm)*	*Actual velocity (m/s)*
1	0.004	20	6.94
2	0.006	20	10.41
3	0.020	40	8.68
4	0.008	25	8.88
5	0.040	50	11.11
6	0.030	50	8.33
7	0.005	20	8.68
8	0.050	65	8.21
9	0.003	15	9.25
10	0.007	25	7.77

The diameter for each of the branches may be determined in the same manner if the pressure loss along the ring is ignored. Determine these yourself and see if you agree with the standard pipe sizes listed in Table 5.1. What assumption has been made in determining the branch sizes?

The mass flow required between branches 6 and 7 is 0.065 kg/s. Do you agree?

The ring main size is 65 mm. Rearranging equation (5.1) in terms of velocity:

$$u = \times \frac{Mv}{d^2} = 1.2733 \times 0.065 \times 0.6057 \times \frac{0.9}{(0.065)^2} = 10.7 \text{ m/s}$$

Steam velocity between branches 6 and 7 is 10.7 m/s.

This is well below the initial velocity of 30 m/s because the flow rate has dropped from 0.173 kg/s just before branch 1 to 0.065 kg/s.

The actual steam velocity just before branch 1 is 28.4 m/s. Do you agree?

PIPE SIZING ON PRESSURE DROP

So far we have considered steam pipe sizing on velocity. However, in Example 5.2 the final steam pressure was determined, and the pressure drop along the main can easily be calculated from P_1-P_2: that is, 270-226=44 kPa. This can be expressed as a pressure drop per metre run, thus:

$$\frac{dp}{TEL} = \frac{44\ 000}{83.26} = 528 \text{ Pa/m}$$

Invariably steam is distributed slightly wet, and the recommended rate of pressure loss equivalent to approximately 30 m/s is 225 Pa/m for pipe sizes up to 65 mm. This figure will be used in the example that follows.

Example 5.4

The distribution steam main is shown in Figure 5.3. Steam is supplied at 500 kPa absolute, 0.9 dry. From the data, size the steam pipework and attempt to balance the branch with the index run in the choice of branch pipe size.

Data

Section	1	2	3
Mass flow	0.4	0.2	0.2
TEL	60	50	12

Solution

The index run for the steam distribution is that circuit with the greatest pressure drop and in this case includes sections 1 and 2 if the pressure drops at the terminals are similar. Initial pressure factor Z_1 is calculated from the initial pressure, which is 500 kPa, as 160810. The approximate value of the final pressure factor Z_2 is obtained from 500-(225/1000× (60+50)) and equals 475 kPa. This is equivalent to a pressure factor of 145661.

$$\text{The index } \frac{dZ}{L} = \frac{160\,810 - 145\,661}{60 + 50} = 138$$

Note the difference here from pipe sizing for water distribution: the pressure loss per metre is now pressure factor loss per metre. This takes account of the fact that steam is a compressible vapour whereas water is an incompressible liquid.

The index pipe diameters may now be selected from the pipe-sizing tables and the actual rates of pressure factor loss recorded. The initial and final pressure factor Z is calculated for each pipe section, so that Z_2 for section 1 becomes Z_1 for section 2. The given and calculated data are listed in Table 5.2.

Having determined the actual final pressure in section 2 as 481 kPa (note the approximate value was estimated as 475 kPa), the final pressure in section 3 needs to be the same in order to balance the system. This enables the available *dZ/L* to be calculated as 492, and the table can then be completed.

Figure 5.3 Steam distribution mains to two terminals.

Table 5.2 Example 5.4: given and calculated data

Section	*1*	*2*	*3*
M (kg/s)	0.4	0.2	0.2
TEL (m)	60	50	12
Available dZ/L	138	138	492
Diameter, d (mm)	90	65	50
Actual dZ/L	90	118	443
dZ	5 400	5 900	5 316
Z_1	160 810	155 410	155 410
Z_2	155 410	149 510	150 094
P_1 (kPa)	500	491	491
P_2 (kPa)	491	481	482
v_2 (m^3/kg)	0.3436	0.3507	0.3507
u_2 (m/s)	21	20	34.7

Conclusion

Note that in section 1 the pipe size is above 65 mm but actual *dZ/L* is significantly lower than the available *dZ/L* based on 225 Pa/m.

The process of attempting to balance the system in the selection of pipe sizes is recommended for sizing steam pipe networks, as the use of regulating valves on a steam system is not an option, owing to the eroding effect of steam on valve seatings. The layout of the distribution pipework therefore assumes an important part of the design process for this reason. Ideally, for terminals having similar pressure drops, each needs to be the same distance from the steam supply point. From a practical viewpoint this is rarely achieved, and the balancing process is usually only partially successful. An important part of the sizing procedure is therefore to ensure that the pressure drop in the index run is kept low. This ensures that the first branch does not suffer a high pressure drop and hence an excessive steam velocity.

You will note that the final velocity in section 3 is excessive for wet steam, and this should initiate a review of the pipe layout to see whether branches 2 and 3 can be made more equidistant from the junction.

PRACTICAL PROBLEMS

A more practical approach to steam pipe sizing would be to consider an available steam supply pressure and condition and the needs of the terminal in terms of required steam pressure and condensate condition.

Example 5.5

A steam distribution main is shown in Figure 5.4 serving two terminals, each of which requires a steam pressure of 400 kPa via pressure-reducing

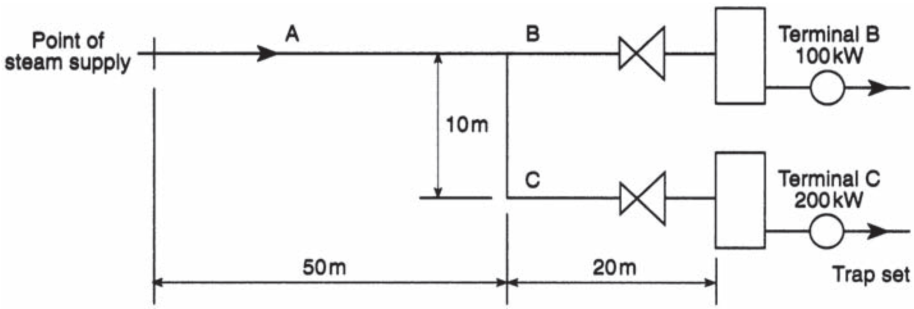


Figure 5.4 Example 5.5: diagrammatic layout of steam distribution.

valves (PRV) with condensate leaving as saturated water at 200 kPa. The initial steam condition is 500 kPa and saturated. Size the pipework assuming the lengths given to be total equivalent lengths and adopting a velocity of 30 m/s as the design parameter.

Determine also the pressure drop across each PRV. Assume also that there is negligible heat loss from the main and that the quoted pressures are absolute.

Solution

Because of the difference in pressure between the steam supply point and the point of use (500-400=100 kPa), there is a measure of storage available in the distribution main in the event of fluctuations in load requirements.

As there is no heat loss to be accounted for along the steam main, heat content is constant along its length at 500 kPa, and h_g=2749 kJ/kg. The heat in the saturated condensate leaving the terminals at 200 kPa, h_f=505 kJ/kg. From these data the mass flows can be calculated:

$$M_B = \frac{100}{2749 - 505} = 0.0446 \text{ kg/s}$$

$$M_C = \frac{200}{2749 - 505} = 0.089 \text{ kg/s}$$

$$M_A = 0.134 \text{ kg/s}$$

From equation (5.1):

$$d = 1.1284 \times \sqrt{\left(\frac{0.134 \times 0.3748}{30}\right)} = 0.046 \text{ m}$$

The nearest standard size for pipe A, d=50 mm

The pressure P_2 at the first branch can now be determined. Knowing the mass flow and pipe size in A, from the pipe-sizing tables: dZ/L=211,

$dZ=211\times50=10550$ and $Z_2=Z_1-dZ$. Thus $Z_2=161000-10550=150450$, from which $P_2=483$ kPa.

Pipe B may now be sized, and

$$d = 1.1284 \times \sqrt{\left(\frac{0.0446 \times 0.388}{30}\right)} = 0.027 \text{ m}$$

The nearest standard diameter for pipe B, d=32 mm.

Pressure at the PRV B can now be determined. Knowing the mass flow and pipe size in B, from the pipe-sizing tables, $dZ/L=200$, $dZ=200\times20=4000$, and $Z_2=Z_1-dZ$. Thus $Z_2=150450-4000=146450$, from which $P_2=476$ kPa.

Therefore the pressure drop across the PRV on pipe B=476-400= 76 kPa.

Determination of the size for pipe C:

$$d = 1.1284 \times \sqrt{\left(\frac{0.089 \times 0.388}{30}\right)} = 0.0383 \text{ m}$$

The nearest standard diameter for pipe C, d=40 mm.

Determining the pressure at PRV C: knowing the mass flow and pipe size, from the pipe-sizing tables, $dZ/L=321$, $dZ=321\times(10+20)=9630$, $Z_2=150450-9630=140820$, from which $P_2=467$ kPa.

Therefore the pressure drop across the PRV on pipe C=467-400= 67 kPa.

Conclusion

It is important to note that the steam pressure available to take the condensate back to the hotwell is 200 kPa absolute, and therefore the effective steam pressure available is 100 kPa gauge.

The actual final velocities in each of the pipe sections can now be evaluated:

$$u_A=26.5 \text{ m/s}, \quad u_B=21.8 \text{ m/s}, \quad u_C=28.5 \text{ m/s}$$

Do you agree?

USE OF THE MODULATING VALVE ON THE STEAM SUPPLY

A difficulty encountered in returning the condensate using the available steam pressure occurs when a modulating valve is employed on the steam main serving the terminal. As the valve modulates, the orifice reduces in size, although the steam velocity tends to increase, thus counteracting the attempt to reduce steam flow.

For satisfactory operation the downstream pressure must therefore not be less than 0.6 of the steam pressure upstream of the valve.

When the valve has modulated to its maximum partially closed position the steam pressure available downstream to drive the condensate back to the hotwell may not be sufficient, in which case the terminal being served becomes waterlogged. In these circumstances a mechanical condense pump operated by the steam on the live side of the modulating valve may be required. Alternatively a mechanically or electrically driven condensate receiver and pumping unit may be the solution. You should make yourself familiar with these items of plant.

Example 5.6

The diagram in Figure 5.5 shows the principal features of a steam supply serving a heat exchanger. Size the steam main on a velocity of 20 m/s and the condensate return on one tenth of the available pressure. Determine the discharge pressure of the condensate at the entry to the hotwell.

Consider the effect that a modulating valve located in the steam supply would have on the return of the condensate under conditions of maximum modulation.

All pressures quoted are absolute.

Solution

With the modulating valve fully open, the mass flow rate M can be determined assuming that the condensate in the steam trap is close to saturation.

Thus

$$M = \frac{80}{546 + 0.9 \times 2174 - 535} = 0.041 \text{ kg/s}$$

$$d = 1.1284 \times \sqrt{\left(\frac{0.041 \times 0.6686 \times 0.9}{20}\right)} = 0.04 \text{ m}$$

Steam supply pipe diameter=40 mm.

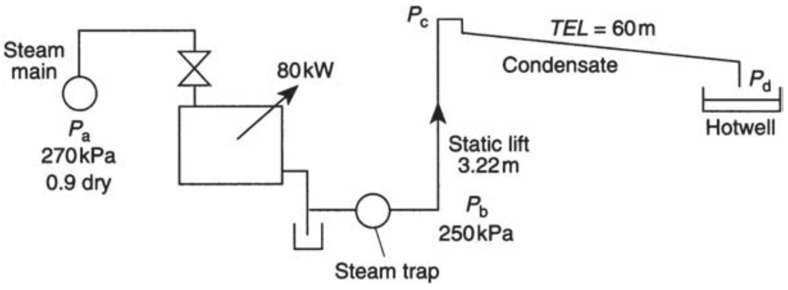


Figure 5.5 Example 5.6: principal features of a steam supply and condensate return to the hotwell in the plant room.

The available pressure for sizing the high-level condensate pipe, P_c, will be P_b—static lift.

From the steam tables the specific volume v_f of water at 127 °C, which is the saturation temperature at 250 kPa, is 0.001066 m^3/kg. The density is therefore 938 kg/m^3.

Thus

$$P_c = 250 - \frac{3.22 \times 938 \times 9.81}{1000} = 220 \text{ kPa absolute}$$

Gauge pressure at P_c is therefore 120 kPa, and if all the pressure is absorbed in sizing the condense return the pressure loss per metre =120000/(60×10)=200 Pa/m. Note that the figure of 1/10 of the available pressure drop given in the data to this example allows for the fact that the condensate return will have to handle condensate, flash steam and air in varying unknown proportions.

Clearly, it is not possible to absorb all of the 120 kPa, as there must be sufficient residual pressure P_d in the condensate outlet at the hotwell for discharge into the tank.

From the CIBSE pipe-sizing tables for X-grade copper, a mass flow of 0.041 kg/s at an available pressure drop of 200 Pa/m requires a 15 mm pipe, where the actual pressure loss per metre is 92.5 Pa/m. This will clearly leave sufficient residual pressure for discharge into the tank. The residual pressure P_d=64.5 kPa. Do you agree?

Condensate return pipe size=15 mm.

If the steam valve modulates down to its minimum downstream pressure of 0.6 of 170 kPa gauge it will have the effect of reducing P_c to 120 kPa gauge×0.6=72 kPa, and the available pressure drop per metre along the condensate return will be reduced to

$$\frac{72\,000}{60 \times 10} = 120 \text{ Pa/m}$$

If 15 mm pipe is used at 92.5 Pa/m, residual pressure at the point of discharge, P_d, into the tank will be 16.5 kPa. Do you agree?

Valve modulation will not affect the return of the condensate.

Conclusion

In this case the modulation of the steam control valve does not affect the return of the condensate, but you can see the potential effect that it could have at a lower steam supply pressure with insufficient pressure left for the return of the condensate to the hotwell. It is in such cases that a mechanical pump referred to earlier would be required.

If the condensate is close to saturation at P_b, its temperature, from the steam tables, will be 127 °C. At P_c the pressure is 220 kPa and saturation temperature is 123 °C. Assuming negligible heat loss from the rising condense pipe, some condensate will therefore flash back into steam at the top of the rise.

When the valve is fully modulating, the steam trap will inhibit the flow of steam through the heat exchanger until its temperature is equivalent to a saturation temperature at 60% of 250 kPa, which is 150 kPa, and therefore, from the steam tables, 111 °C.

STEAM TRAPS

Steam traps are required to inhibit the flow of steam as it passes into the heat exchanger so that it gives up its latent heat. They are also required to prevent live steam entering the condense main.

In most space-heating applications the steam trap must evacuate the condensate as it forms in the heat exchanger. There are a number of different types of steam trap available: float operated, inverted bucket, thermostatic, liquid expansion, bimetallic and thermodynamic. The capacity of the steam trap depends upon the size of its orifice, condensate temperature and the pressure drop across the trap.

As condensate temperature rises so density falls and flow rate is increased. As the pressure drop across the trap increases so does the flow rate. If the condense pipe rises after the trap, back pressure on it is increased, and the pressure drop across the steam trap is reduced, thus reducing the rate of flow.

You should familiarize yourself with the various types of steam trap on the market, and identify which to use for different heating apparatus.

Each heater must be fitted with its own steam trap. Groups of similar heaters with similar traps can be connected to a common condensate return. Dissimilar steam equipment and associated traps must not be connected to a common condensate return. This is due to the way traps operate. Some are intermittent in operation; others are continuous. This and variations in pressure can cause waterlogging in some of the heat exchangers if different pieces of apparatus are interconnected.

This identifies one of the disadvantages of steam systems, in that a number of condensate returns from different apparatuses must be run separately back to the hotwell. An alternative is to employ a receiver and pumping unit into which they are discharged, allowing a single condensate return from the unit to the hotwell.

USE OF THE VACUUM BREAKER

This is considered, for example, on a steam-fed air heater battery, where the duct thermostat is controlling a modulating valve on the steam supply.

As the valve modulates in response to the duct thermostat, the steam in the battery condenses, and this can cause a partial vacuum, which results in condensate not being discharged at the steam trap. The heater battery partially fills with condensate, thus causing a severe temperature gradient in the air stream. The vacuum breaker prevents this occurrence. However, if it draws in very-low-temperature outdoor air to counter the partial vacuum it can cause freezing conditions within the battery. It is therefore important to ensure that the vacuum breaker is located to avoid this possibility.

RELAY POINTS

Particularly on system start-up and to a much smaller extent during system operation, condensate will form in the steam pipework. This must be collected at regular intervals to keep the steam mains largely free of condensate. There are two kinds of relay point, which are shown in Figure 5.6.

Figure 5.6(a) shows relay points for use on a distribution main. Note that the condensate is aided by steam flow and gravity towards the collection point, with the steam main maintaining the same level. Figure 5.6(b) shows a relay point for use in a plant room.

PRESSURE-REDUCING VALVES

Various apparatus using steam requires specific pressures to operate at optimum outputs. It is therefore normal to employ pressure-reducing valves (PRV) on the

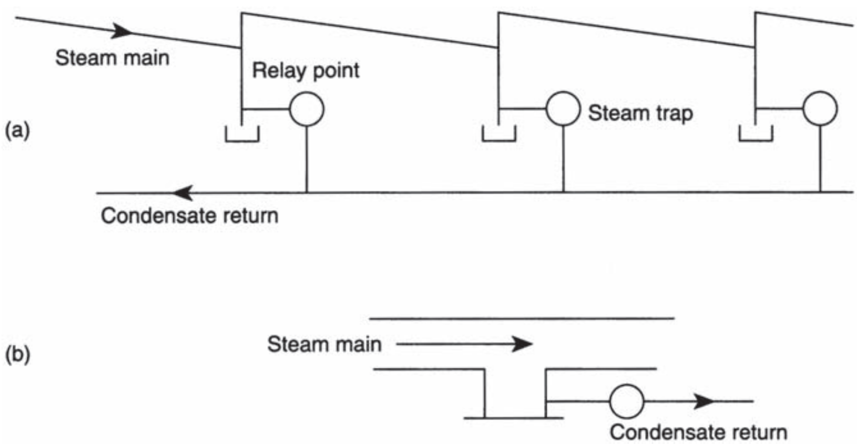


Figure 5.6 Two types of relay point.

live steam supply locally to steam-heated equipment, where it can be reduced to the specified pressure. The data required to size a PRV include the mass flow, expected steam quality, upstream and downstream steam pressures. Steam flow through the PRV follows an adiabatic process, during which the quality of the vapour tends to improve. Consider the following example.

Example 5.7

Steam at 10 bar absolute 0.93 dry is fed through a PRV, which reduces its pressure to 2.0 bar absolute. Determine the condition of the steam on the downstream side of the valve and the temperature drop sustained across the valve.

Solution

The heat content of the steam on the live side of the PRV, h_w=763+ 0.93×2015=2637 kJ/kg. At 2.0 bar the heat content of saturated steam, h_g=2707 kJ/kg. As the process is adiabatic, the heat content of the steam on the low-pressure side is the same as that of the live steam, namely 2637 kJ/kg. Clearly then the steam still has a dryness fraction *q*, and at the low pressure $h_w = h_f + q \times h_{fg}$.

Thus

$$2637=505+q\times 2202$$

from which

$$q=0.97$$

The quality of the steam has improved from 0.93 to 0.97.

From the steam tables the saturation temperatures at 10 bar and 2.0 bar are 180 °C and 120 °C respectively.

The temperature drop across the PRV is 60 K.

SIZING THE CONDENSATE RETURN

On system start-up from cold, the condensate mains will have to handle air and water. If the condense is lifted on the downstream side of the steam trap there is a good possibility that some of the condensate at the top of the rise will flash back into steam during normal operation.

The condensate pipe therefore has to handle a mixture of air, water and flash steam in varying amounts. It is therefore almost impossible to size the condense pipe accurately. It has been found that if it is sized one to three sizes below that of the steam main that it accompanies, it performs its tasks satisfactorily. Table 5.3 is a guide.

Table 5.3 Recommended condensate pipe sizes

Steam d (mm)	20	25	32	40	50	65	80	100	150
Recommended condensate *d*	15	20	20	25	32	32	50	65	80

Example 5.6 identifies an alternative approach where the pressure loss sustained along the condense main is calculated as ten times that for LTHW systems.

Thus dp=10(pd per metre×TEL), from which available pd per metre =dp/(10×TEL).

A further alternative recommends that the condense main is sized on three times the normal hot water discharge.

Both these alternatives tend to yield smaller pipe sizes than those in Table 5.3.

Example 5.8

Figure 5.7 shows a branch taken from a steam main to serve a heat exchanger via a pressure-reducing valve. Pressures quoted are in absolute units.

- Size the branch off the high pressure main on 225 Pa/m.
- Size the low pressure connection on 9 m/s.
- Size the PRV.
- Size the condensate return.

Solution

The mass flow required at the heater can be calculated from the given data and steam tables and M=0.061 kg/s.

Adopting the velocity factor formula, K_6=0.313, and therefore from the velocity factor table in the *CIBSE Guide* d=65 mm.

The size of the LP steam connection is 65 mm.

The listed value for K_6 in the velocity factor table is 0.284, from which the actual velocity of the steam in the heater connecting pipe can be calculated. dZ/L for the HP branch is calculated to be 189, and from the steam pipe-sizing tables d=40 mm for a dZ/L of 162.

The size of the HP branch is 40 mm.

This allows the determination of pressure factor Z_2, from which P_2 at the PRV inlet is determined as 686 kPa.

The data for sizing the PRV are therefore:

inlet pressure=686 kPa

outlet pressure=350 kPa

mass flow=0.061 kg/s

From Table 5.3, the estimated size of the condensate return is 32 mm.

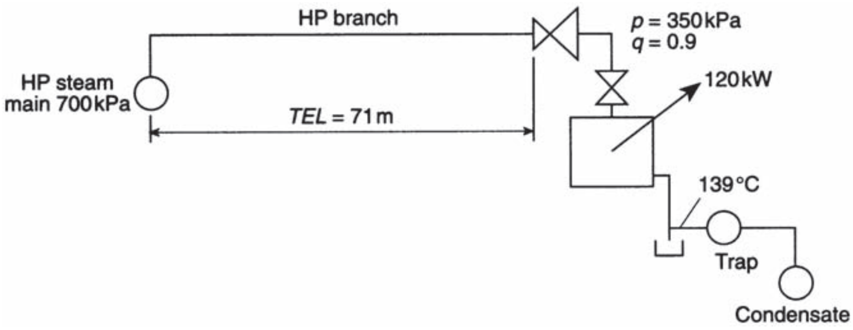


Figure 5.7 Example 5.8: steam connection to heater.

Do you agree with these solutions?

You should now verify these solutions following the procedures in the solutions to earlier examples.

5.3 Steam generation and distribution

There are two ways by which steam may be distributed:

1. Distribution by high pressure from the generator with PRVs located in the branches serving the appliances

   This allows apparatus operating at different pressures to be served from one generating plant. Operating the generator at a pressure in excess of the maximum required offers a measure of storage in the distribution mains, which will be smaller in size compared with low-pressure steam distribution.
2. Distribution by low pressure with the necessary allowance for pressure loss along the index run.

   This may allow the use of lower-specification plant with correspondingly lower capital cost. However, it is likely to cope only with apparatus having a common operating pressure, and there is no inherent margin of steam in the distribution mains to offset a sudden increase in demand. Furthermore, the condensate return may require assistance from pumping and receiving units, as there may be insufficient steam pressure left at the index terminal to drive the condense back to the hotwell.

Case study 5.1

A feasibility study is required for steam distribution on a sloping site to three buildings. Each building requires 1 MW of LTHW heating and 10000 litres/

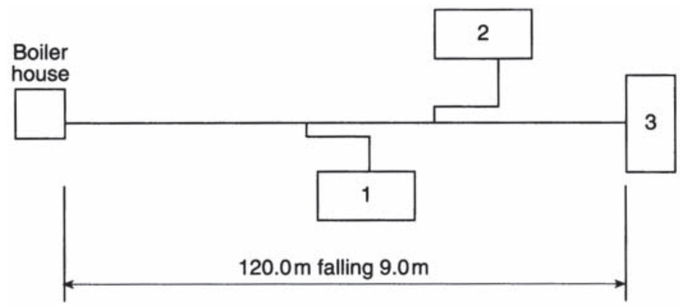


Figure 5.8 Case study on steam distribution, buildings 1, 2, 3. Plan view.

hour of HWS. The minimum pressure in the heating and HWS systems is 1.0 bar gauge.

The study shall include the following outcomes:

(a) generator and distribution pressure;
(b) steam flow rate to each building;
(c) boiler power and equivalent evaporation from and at 100 °C;
(d) recommended method for returning the condensate;
(e) estimates of steam distribution and condensate return sizes;
(f) estimation of the linear expansion in the distribution main and recommendations on methods of accounting for it.

The site is shown in Figure 5.8.

SOLUTION

Clearly there is more than one solution to this case study, if only in the choice of steam pressure for the distribution mains. The distribution steam pressure selected here will be the minimum for satisfactory system operation. It is an easy matter to raise the steam pressure above the minimum to provide for a measure of storage.

Maximum steam pressure at each building

If the temperature controls fail, the secondary water for both the heating and HWS should not boil. Thus the maximum steam temperature should be 10 K below boiling point at the minimum water pressure of 1.0 bar gauge. From the steam tables saturation temperature at 2.0 bar absolute is 120 °C, which less 10 K antiflash margin leaves a maximum steam temperature of 110 °C.

This corresponds to a steam pressure of 1.4 bar absolute at each building.

It is apparent that the steam supply to each building requires a PRV, as 1.4 bar absolute (0.4 bar gauge) is of little use as a distribution steam pressure from the boiler house.

The temperature drop between primary and secondary media should be above 20 K to ensure positive heat exchange. If the LTHW heating mean water temperature is 80 °C, dt=110-80=30 K. Similarly if HWS storage temperature is 65 °C, dt=110-65=45 K. Both temperature drops are satisfactory.

If the condensate is to be returned under its own pressure, steam distribution pressure will depend upon the minimum pressure required to return the condensate to the boiler house from the index terminal. The minimum initial pressure required in the condense return therefore, adopting a pd of 300 Pa/m (remember the pipe is sized on 30 Pa/m if the factor of 10 is used) and making an allowance of 30% for fittings on straight pipe:

$$\text{Initial } P = TEL \times \frac{\text{pd}}{\text{m}} = 120 \times 1.3 \times 300 = 46\,800 \text{ Pa}$$

Taking density as 940 kg/m^3 the initial head

$$h = \frac{46\,800}{940 \times 9.81} = 5 \text{ m}$$

In view of the sloping site:

$$h=5+9=14 \text{ m}$$

This corresponds to a minimum initial pressure in the condensate return of

$$P=14\times940\times9.81=130 \text{ kPa gauge}$$

To ensure satisfactory operation, and allowing for the pressure drop across the steam heat exchanger in the index terminal, minimum final steam pressure available at the PRV of 1.5 bar gauge, 2.5 bar absolute is therefore required on the live side of the PRV. Thus the pd across the index PRV will be from 2.5 to 1.4 bar absolute. Steam from the live side of the PRV will now have sufficient pressure to drive the condensate back to the hotwell via a pumping and receiving unit.

Allowing for a pressure drop of 225 Pa/m along the steam distribution main, minimum boiler pressure can now be established and:

(a) Minimum generator pressure=(225×120×1.3)/1000+250= 285 kPa abs
The demand at each building is: heating 1.0 MW and HWS 10000 l/h. It can be assumed that the quality of the steam on the downstream side of the PRV is 0.95, and from the steam tables at 1.4 bar absolute:

$$h_w = 458 + 0.95 \times 2232 = 2578 \text{ kJ/kg}$$

If condensate leaves at 107 °C:

$$h_f = 449 \text{ kJ/kg}$$

Thus the mass flow for the heating will be

$$M = \frac{1\text{M}W \times 1000}{2578 - 449} = 0.47 \text{ kg/s}$$

The output for the HWS:

$$Q = \left(\frac{10\,000}{3600}\right) \times 4.2 \times (65 - 10) = 642 \text{ kW}$$

Thus the mass flow for the HWS=642/(2578-449)=0.3 kg/s.

Total net mass flow=0.47+0.3=0.77 kg/s.

Allowing for inefficiency of heat transfer of 20%, total mass flow to the index building M=0.92 kg/s.

If steam serving the other two buildings on the downstream side of the PRVs is likewise 1.4 bar absolute:

(b) Steam flow rate to each building is 0.92 kg/s
Total mass flow from the generator will be 3×0.92=2.76 kg/s. If the boiler generates steam at 0.95 dry from feed water at 80 °C, the boiler power required will be:

$M \times dh = 2.76 \times (556 + (0.95 \times 2168) - 335) = 2.76 \times 2281 = 6295$ kW

(c) Boiler output power required is 6300 kW at an operating pressure of 185 kPa gauge
Equivalent evaporation from and at 100 °C is used as the basis for comparing different steam generators for the purposes of selection.

It is equivalent to: ((water evaporated in kg/s)×dh/(fuel input in kg/s))/2257 at atmospheric pressure, where the latent heat of vaporization is 2257 kJ/kg. Assuming boiler efficiency of 75% and a gross calorific value for the fuel of 40 MJ/kg:

$$\text{fuel input} = \frac{6300}{(40\,000 \times 0.75)} = 0.21 \text{ kg/s}$$

$$\text{water evaporated} = 2.76 \text{ kg/s and } dh = 2281 \text{ kJ/kg}$$

(c) Thus equivalent evaporation from and at 100 °C=((2.76× 2281)/0.211/2257=13.28 kg water/kg fuel
The limitation of this comparison is that the fuel must be common to each generator being considered.

The condensate return will need to be via a receiver and mechanical pumping unit operated from the steam supply on the live side of the PRV at a pressure of 150 kPa gauge.

This was established in determining the minimum generator pressure.

Alternatively the pumping unit can be an electrically driven pump in which the net pressure developed would need to be equivalent to a lift of 14 m (130 kPa) and the flow rate equivalent to the condensate flow of 0.92 kg/s.

(d) Methods for returning the condensate

The size of the steam distribution main will be based upon the initial and final pressures, where

$$Z_1 = P_1^{1.929} = 285^{1.929} = 54\ 375$$

$$Z_2 = P_1^{1.929} = 250^{1.929} = 42\ 231$$

$$\frac{\mathrm{d}Z}{L} = \frac{12\ 144}{120 \times 1.3} = 78$$

Initially this *dZ/L* will be used to size each section of the steam main.

From the steam pipe-sizing tables for a mass flow of 2.76 kg/s the pipe size is 200 mm, for which *dZ/L*=61.

As the pipe size is in excess of 65 mm, steam velocity must be checked, and from equation (5.1):

$$u = (1.1284)^2 \times Mv/d^2 = 1.2733 \times \frac{2.76 \times 0.6358 \times 0.95}{0.04} = 53 \text{ m/s}$$

Maximum velocity for wet steam is 30 m/s. The reason for the excessive velocity is the use of 225 Pa/m in determining the minimum boiler pressure. With such large-diameter pipe a figure of 60 Pa/m would be more appropriate.

To be sure of not exceeding the maximum velocity, the distribution mains will be sized on 30 m/s. Thus for a mass flow of 2.76 kg/s, a specific volume V_g of 0.6358, a dryness fraction of 0.95, and adopting equation (5.1), *d*=266 mm.

For a mass flow of 1.84 kg/s, a specific volume of 0.6772 and a dryness fraction of 0.95, *d*=224 mm.

For a mass flow of 0.92 kg/s, a specific volume of 0.7186 and a dryness fraction of 0.95, *d*=163 mm.

The specific volumes were taken at the beginning, mid point and end of the distribution main respectively.

(e) The suggested standard sizes of the steam distribution main are therefore 250 mm, 225 mm and 175 mm

It is left to you to check the steam velocities in each section and decide if you agree. The size of the condense main from the index terminal is based on a mass flow of 0.92 kg/s and a pd of one tenth of the adopted value taken as 300 Pa/m. Thus at a maximum 30 Pa/m and using copper pipe to table X, d=67 mm, for which pd=14 Pa/m.

(e) Three separate condensate returns, each at 67 mm diameter You will find that the size of the condensate return is increased if the alternative estimate is adopted by extending Table 5.3 pro rata. What would you recommend?

There are in fact two options here for returning the condensate. One is to return each of the three condensate pipes back to the hotwell separately. The other is to return the condensate from building 3 into the receiver and pumping unit of building 2, and the common condensate from buildings 3 and 2 to the receiver and pumping unit of building 1, from which the condensate common to the three buildings is returned to the hotwell. This method will save on the length of copper used but at the expense of increases in pipe size to 76 mm from building 2 to 1 and to 108 mm from building 1 back to the boiler house.

Do you agree with these pipe sizes if they are based on a maximum of 30 Pa/m: that is, one tenth of 300 Pa/m adopted in part (a) of this solution?

Figure 5.9 shows the arrangement around the receiver and pumping unit in building 1. A flash steam exhaust pipe fitted with an exhaust head is taken from the receiver to outdoors to stabilize the pressure in the receiver. This fact may help you to decide on the sizes of the condensate returns.

(f) Linear pipe expansion (for detailed consideration refer to Chapter 10).
The total expansion from cold along the steam distribution main is obtained from:

$$\text{expansion coefficient} \times \text{length} \times \text{temperature rise}$$

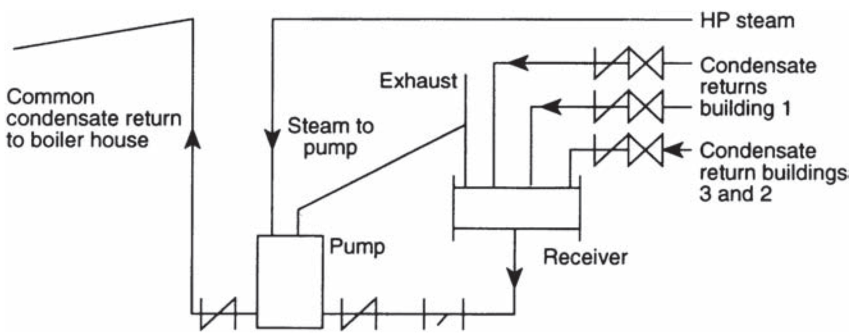


Figure 5.9 Receiver and mechanical pumping unit in building 1.

The coefficient for mild steel is 0.000012 m/mK, and the amount of linear expansion=0.000012×120×(132-10). Therefore the expansion=0.176 m or 176 mm.

This must be accounted for either through changes in the direction of the pipe in the form of expansion loops or double sets, or by employing axial compensators. In either case the pipe will require anchoring either side of the expansion device and at the entry to each building. The pipes will also require adequate pipe guides to control the expansion along the pipe axis and, where necessary, to control the lateral movement of the pipe. Vertical pipe movement must be avoided to ensure against condensate collecting at low points that are not fitted with a steam trap.

The condense return is in copper, which has a coefficient of linear expansion of 0.000018 m/mK. Over the full length the amount of expansion will be:

$$\text{expansion}=0.000018\times 120\times(109-10)=0.214\text{ m}$$

which is 214 mm.

Conclusion of the solution to the case study

Hopefully you have followed the rather tortuous routes to the solutions. You may have noticed that the minimum generator pressure could now be reduced slightly, as the pressure drop along the steam main was taken as 225 Pa/m when subsequently, owing to the large size of the pipe, a pressure of 60 Pa/m would have been appropriate. However, it is not considered necessary to re-work the solutions for such a small drop in boiler pressure.

The final topic for consideration is flash steam recovery and use.

FLASH STEAM RECOVERY

High-pressure condensate coming off apparatus requiring HP steam has substantial heat content, which can be used effectively before returning it to the hotwell. Effective use improves the overall efficiency of the system.

The condensate can be used in a heat exchanger or a flash steam recovery vessel. Figure 5.10 shows a flash steam recovery vessel with its appropriate connections. You will notice that there is a high-pressure make-up line interconnected with the low-pressure flash steam. This ensures that the low-pressure demand is always met. The system is self-regulating in operation. Do you agree?

To obtain maximum useful heat from the HP condensate, the pressure

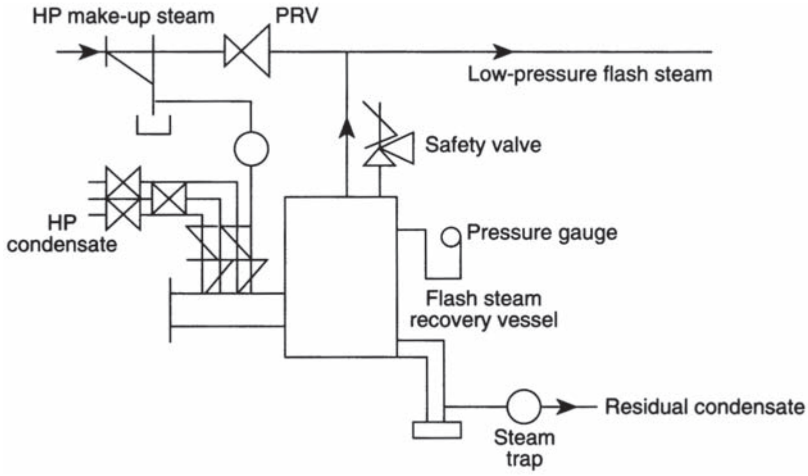


Figure 5.10 The flash steam recovery vessel and connections.

of the flash steam must be as low as possible, which means that it should preferably be used locally to the flash vessel.

Example 5.9

0.22 kg/s of HP condensate at 195 °C is put through a flash steam recovery vessel, where it expands to a pressure of 0.5 bar gauge 0.9 dry.

Determine the mass flow of flash steam derived from the HP condensate, the potential output of this flash steam, the mass flow of residual condensate, and the mass flow of high-pressure steam make-up if the low-pressure demand rises to 100 kW. Assume the HP steam is saturated.

Solution

Mass flow of flash steam:

M_f=mass flow of HP condensate×(difference in h_f/latent heat at LP)

$M_f = M(dh_f/(qh_{fg}$ at LP)) and from the steam tables

$$M_f = 0.22\left(\frac{830 - 467}{0.9 \times 2226}\right) = 0.04 \text{ kg/s}$$

Potential output of flash steam

$$Q_f = M_f \times q \times h_{fg} = 0.04 \times 0.9 \times 2226 = 80 \text{ kW}$$

Residual condensate=0.22-0.04=0.18 kg/s.

$$\text{High pressure make up mass flow} = \frac{\text{increase in demand}}{\text{heat given up}}$$

Increase in demand=100-80=20 kW

Heat given up=heat in HP steam-heat in LP condensate,
and from the steam tables:

=2790-467=2323 kJ/kg

Thus high-pressure make-up=20/2323=0.00861 kg/s.

5.4 Chapter closure

This completes the work on steam heating systems. You should now attempt the steam pipe-sizing solution in Chapter 1. You are now able to undertake initial design studies on the use of steam as a means of space heating for a given site. Conversion from steam to LTHW is done using non-storage (heating) and storage (HWS) calorifiers, and the principle is covered in Chapter 4. You also have the skills now to size systems and plant and to check limiting steam velocities as well as the pressures needed for the return of the condensate to the hotwell in the plant room. If steam is generated to a high pressure, perhaps for uses other than space heating, you are now able to consider the reuse of the high-temperature condensate in the form of flash steam.

6 Plant connections and controls

Nomenclature

BEMS	building energy management system
CPU	central processing unit
CTCV	constant temperature constant volume
CTVV	constant temperature variable volume
CVVT	constant volume variable temperature
DDC	direct digital control
F&E	feed and expansion
HWS	hot water service
LAN	local area network
MWS	mains water service
OEM	original equipment manufacturer
TRV	thermostatic radiator valve
VDU	visual display unit

6.1 Introduction

This chapter focuses upon the way in which systems are put together and automatically controlled. At an early stage in the design process for space heating and water services it is necessary to identify how the building should be zoned, both in terms of time scheduling (those areas whose operating periods may differ) and in terms of temperature control (dependent upon building orientation, exposure and uses to which different parts of the building are going to be put). Figures 6.1 and 6.2 illustrate **zoning diagrams**.

Vertical zoning may be required, for example, to respond to the effects of building orientation. **Horizontal zoning** may be required, for example, to offset the effects of exposure of a multistorey building where the upper storeys are subject to a more severe climate.

Zoning for time scheduling and temperature control marks a major step in the design process, for it implies knowledge on the part of the design team of the client's needs and the architect's vision of the form that the building will take.

Following the completion of the zoning plans, the next stage in the

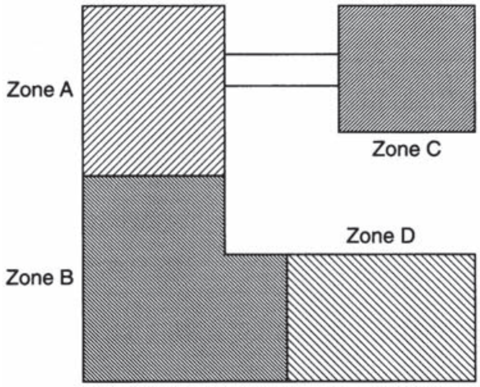


Figure 6.1 Site plan of building showing time-scheduled zones.

design process is to work up a schematic diagram that identifies the services and plant for the project. This chapter focuses on this issue. It will introduce the subject in parts, which can then easily be selected and assembled to form the schematic diagram for the project in hand.

6.2 Identifying services and plant

BOILER PLANT HEADER CONNECTIONS

Single or multiple boilers should always be connected to either a mixing header or separate flow and return headers. This allows heating circuits to be independently connected to the boiler plant, and facilitates maintenance and breakdown of individual circuits and boilers. It is for this reason an indicator of good engineering practice. Figure 6.3 shows plant connected to flow and return headers.

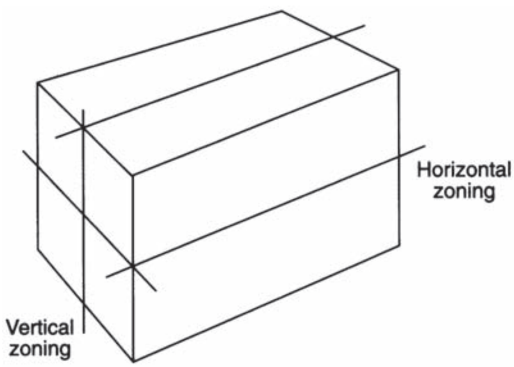


Figure 6.2 Isometric view of building showing zoning for temperature control.

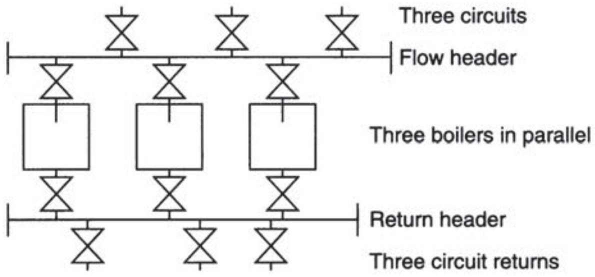


Figure 6.3 Boilers connected in parallel, serving three circuits.

The use of the mixing header has been reinstated since the introduction of modular boilers, and it is normally applied to gas-fired appliances. Figure 6.4 shows a typical layout.

Modular boilers are usually connected as shown in Figure 6.4. Note the use of a primary pump to ensure circulation through the boiler circuit. 'Secondary' pumps are required on each of the three heating circuits. Sequenced boiler control is employed to provide maximum efficiency of plant use.

BOILER PLANT OPEN VENT AND COLD FEED CONNECTIONS

When multiple boilers are used it is not necessary to connect an independent open vent or feed and expansion pipe to each boiler. This clearly saves cost and space, and Figure 6.5 shows typical connections to multiple boiler plant.

Normal position for the manual three-way cock is indicated for boilers 1 and 3. If, for example, boiler 2 needs to be isolated from the common open vent, the three-way cock is positioned as shown, with the boiler port

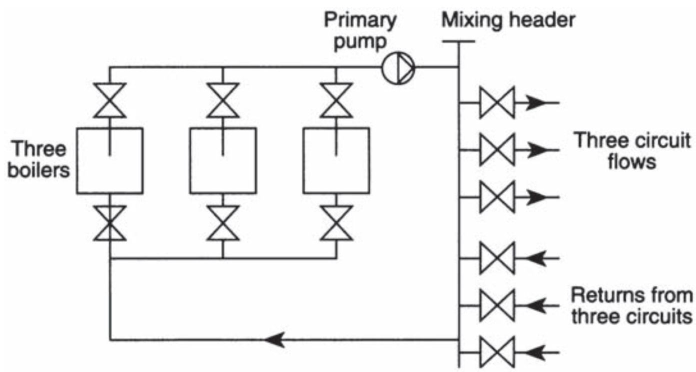


Figure 6.4 Three boilers connected to a mixing header, serving three circuits.

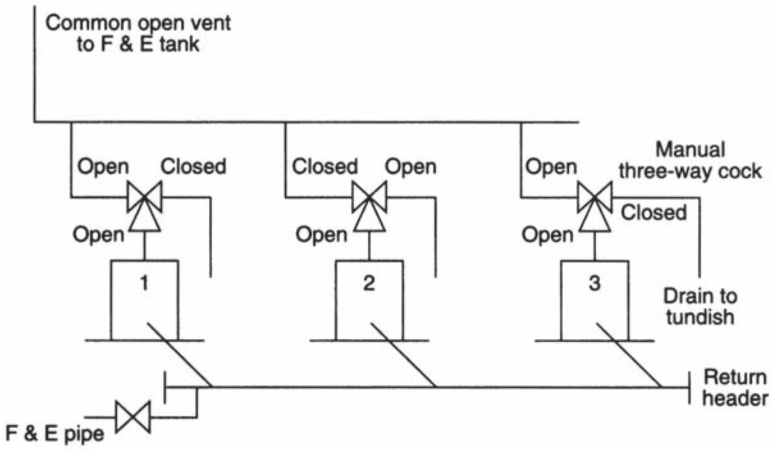


Figure 6.5 Open vent and cold feed connections to multiple boilers. Boiler return isolating valves omitted.

and drain port open and the open vent port closed. This allows release of the static pressure within the boiler, and if draining down is necessary air can enter the drain port, thus facilitating draining down. The remaining boilers are unaffected. The feed and expansion pipe is connected into the return header via a lock shield isolating valve.

BOILER RETURN TEMPERATURE PROTECTION

Heating systems with CVVT control are particularly subject to low return temperatures in mild weather. This can have a detrimental effect on the boiler by inducing condensation in the flue gas in the area of the return connection. Low return temperatures without corresponding modification to boiler flow temperature can also cause differential expansion within some boilers and consequent stress in the boiler metal. The boiler manufacturer will advise.

Figure 6.6 shows means of protection. When the thermostat senses low-temperature water in the boiler return it energizes the pump and cuts it out on a predetermined higher temperature.

Figure 6.7a and b shows alternative arrangements when there is a constant volume flow through the boiler plant and still the requirement for protection. The regulating valve on the bypass pipe will need careful balancing, as it is subject to the full system pump pressure. In both cases the control valves respond to the immersion thermostats when low-temperature water is detected, by opening to ensure flow through the bypass. On rise in temperature the controller closes the valve to bypass.

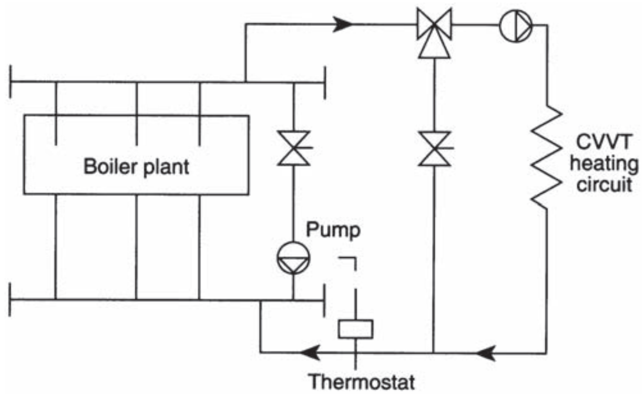


Figure 6.6 Boiler low return temperature protection.

CONDENSING BOILERS

These boilers are specifically designed to operate on low return water temperature so that condensation can form on the flue gas side of a second heat exchanger. In the process, further sensible heat and latent heat of condensation from the flue gas are given up, thus increasing the thermal efficiency of the boiler. The secondary heat exchanger can be integral or separated from the primary heat exchanger in the boiler. If it is separated the boiler is called a **split condensing unit,** in which the primary and secondary heat exchangers can operate at different temperatures and flow rates. This can be put to advantage by allowing the primary water to be used for higher-temperature circuits while still promoting condensation in the secondary heat exchanger.

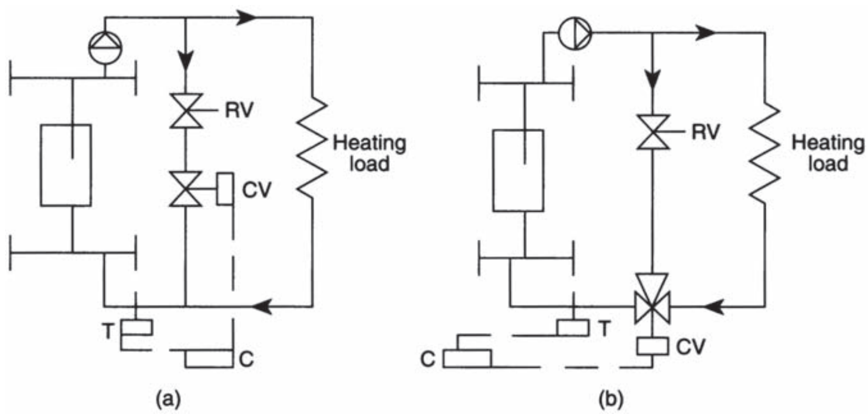


Figure 6.7 Boiler return protection for constant flow through the boiler plant: RV, regulating valve; CV, control valve; T, thermostat.

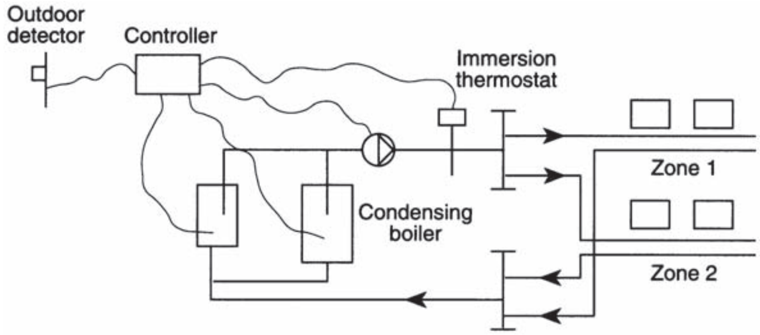


Figure 6.8 Application of the condensing boiler with direct weather compensation giving cvvt control.

Double condensing boilers have three heat exchangers to extract even more heat from the flue gases.

You should familiarize yourself with these boilers because they provide a means of saving primary energy. Figure 6.8 shows a typical application. The condensing boiler is the lead boiler and will be operative throughout the heating season, making use of the latent heat from the flue gases in mild weather when the return temperature from the CVVT zones is low. The lag boiler would be a conventional boiler for use in severe weather with the standby facility of a further conventional boiler if required. Figure 6.8 shows CVVT control, which is achieved directly via the boiler rather than through a three-port control valve and actuator. It is therefore not possible to serve fan convectors or unit heaters from this system as they require constant-temperature constant volume (CTCV) control.

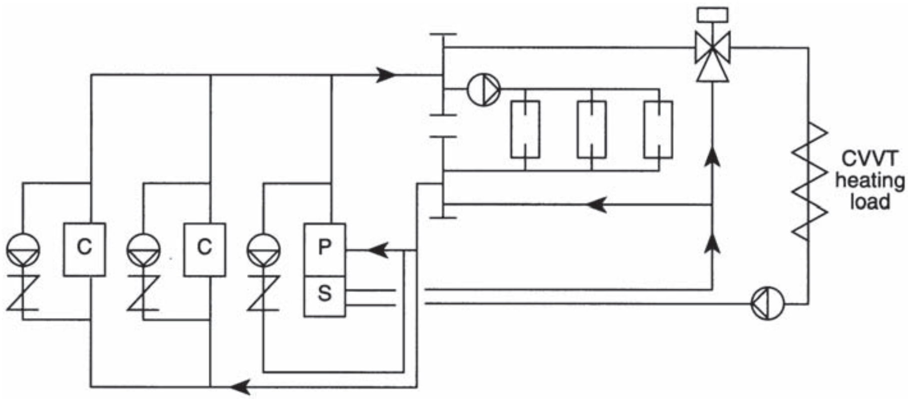


Figure 6.9 Two conventional and one split condensing boiler system: C, conventional boiler; P, primary heat exchanger; S, secondary heat exchanger of double condensing boiler. F&E tank, cold feed, open vent, isolating, regulating valves and control elements omitted.

Figure 6.9 shows multiple boilers: two conventional and one split condensing. There are two heating circuits, one with CVVT control via a three-way mixing valve and the other having CTCV control serving, say, fan convectors. The secondary heat exchanger in the split condensing boiler will increase the unit's thermal efficiency during a mild winter climate because of the effect of weather compensation at the mixing valve. The fan convectors operate at constant temperature, and therefore return water is not low enough to be passed through the secondary heat exchanger. The high thermal efficiencies associated with condensing boilers are dependent upon the return water temperature being low (30–55 °C), and these units are therefore particularly appropriate for underfloor heating, swimming pool heating, and systems employing low-temperature radiators.

TEMPERATURE CONTROLS FOR SPACE HEATING AND HOT WATER SUPPLY

Room temperature control

There are a number of space temperature control philosophies. For appliances such as radiators the use of constant volume variable (circuit water) temperature control (CVVT) employing a three-port mixing valve is well known. In Figure 6.10a, the outdoor detector senses changes in climate, and the controller drives the actuator, thus repositioning the shoe in the valve. At design conditions there is no flow in the bypass, but as outdoor temperature rises the bypass port starts to open and the boiler port begins to close, thus allowing lower return temperature water to mix with the boiler water. The immersion thermostat senses the mixed water temperature and, if necessary, takes corrective action via the controller and valve actuator. A room thermostat or averaging thermostats may also be connected to the controls, although it is difficult to identify a suitable indoor location in practice (Figure 6.10a).

Single heat exchangers required to produce a constant temperature, such as air heater batteries and HWS calorifiers, are fitted with constant-temperature variable-volume (CTVV) control using a three-port mixing valve. Here the boiler water is diverted into the bypass as the air reaches the temperature setting of the duct thermostat located downstream in the supply air duct or in the return air duct (Figure 6.10b).

Note the location of the pump in each of Figures 6.10a and 6.10b. In Figure 6.10a the water flow through the boiler is interrupted when the control is on full bypass.

Constant-temperature variable-volume control is also achieved using two-port valves responding to a room thermostat and by using thermo static radiator

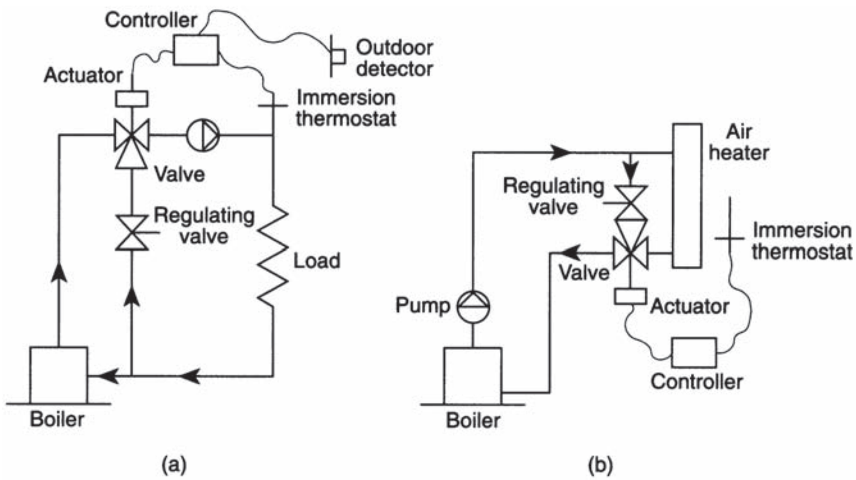


Figure 6.10 (a) CVVT control; (b) CTVV control.

valves. In both cases the water flow varies to maintain a constant room temperature.

It is important to consider the use of balanced pressure valves if the pump is dedicated to the circuit being controlled: see Chapter 3.

Space-heating appliances such as unit heaters and fan convectors are controlled by a room thermostat wired in series with the fan motor. The heating circuit needs only the temperature control imposed by the boiler operating thermostat, thus providing constant-temperature constant-volume control (CTCV). When the fan is off, output is reduced by about 90%, and hot water is instantly available when the fan is reactivated by the room thermostat. (Figure 6.11).

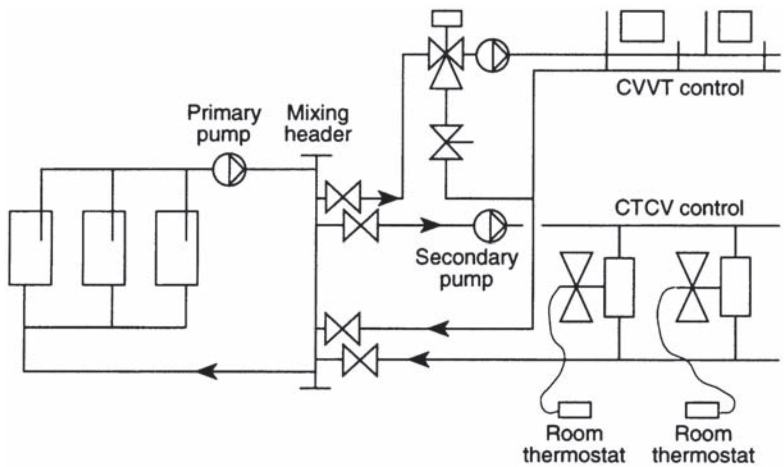


Figure 6.11 A simple schematic showing CVVT control to radiators and CTCV control to unit heaters.

Limitations on CVVT control

This type of control relies on compensating for changes in outdoor climate, and on its own does not give local control of heating appliances. It also suffers from the location of the outdoor detector, which, if located on the north wall of the building, will not compensate for the effects of solar heat gains or for that matter local indoor heat gains. However, CVVT control is to be recommended where appropriate as a means of reducing circuit flow temperatures in mild weather, thus offering energy savings. Local temperature control can be provided by employing, in addition, thermostatic radiator valves or two-port control valves, activated by a room thermostat, to a group of radiators.

It may be appropriate to provide two or more zones of CVVT control when zoning the building. See Figures 6.2 and 6.12.

Temperature control for hot water supply

Figure 6.12 is a schematic showing a combined heating and hot water supply system. The CTVV primary circuit to the HWS calorifier would be controlled by an immersion thermostat located in the secondary water within the cylinder and set to 65 °C. The two CVVT heating zones may be required to offset the effects of building orientation and solar heat gain. If this provides adequate temperature control it may not be necessary to fit TRVs to all the radiators. Remember: without them there is no *local* temperature control. Note that with the pump on the CTVV HWS primary circuit there is constant water flow through the boiler plant and therefore no need for boiler return protection. There is a strong argument for separating the generation of HWS from the space heating plant on the grounds of efficiency. Direct-fired HWS generators connected directly to the rising main are widely used, and for larger rates of simultaneous flow can be connected in banks. See Figure 6.13.

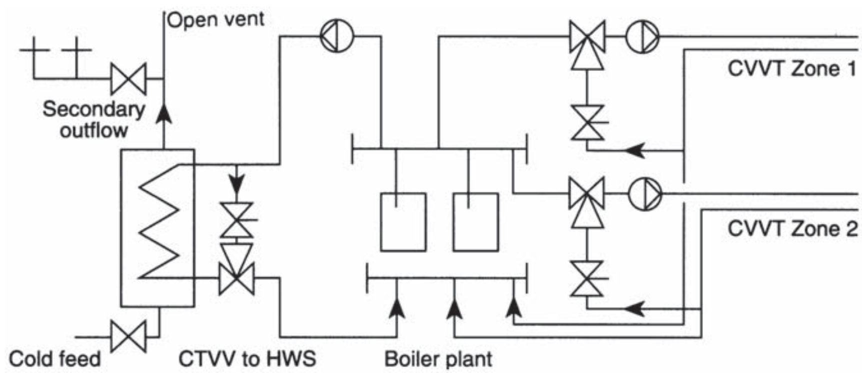


Figure 6.12 illustrating (2) CVVT and (1) CTVV control circuit. Isolating valves, heating F&E pipe, open vent and F&E/CWS tanks omitted.

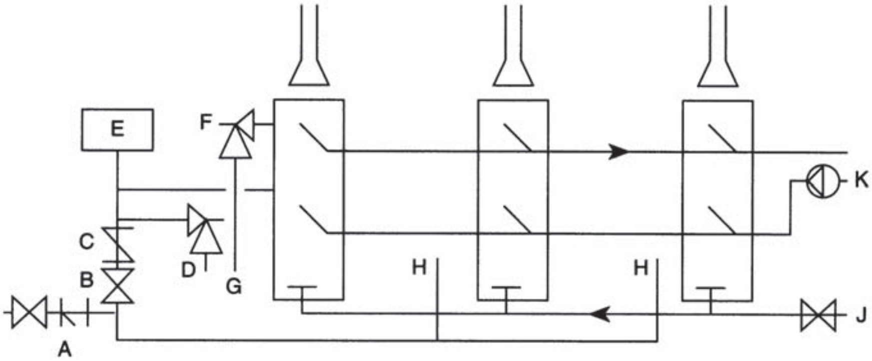


Figure 6.13 Three direct-fired HWS storage heaters showing direct MWS connections: A, MWS with isolating valve and strainer; B, pressure-limiting valve; C, check valve; D, expansion valve; E, expansion vessel; F, temperature/pressure relief valve; G, drains to tundish; H, repeat of items B–G to the two other heaters; J, gas supply; K, HWS flow and return.

LOGIC DIAGRAMS

You are now able to put together a schematic diagram showing space heating and hot water supply systems for a project (diagrams for indirect hot water supply are given in Chapters 8 and 9). It will include all the elements for each temperature control device. For example, CTVV control will include the valve body, valve actuator, immersion thermostat and controller. A logic diagram is a schematic, which has plant duties and main flow rates added: for example, boiler outputs, pump duties, HWS heater outputs and simultaneous flow rate, tank sizes, fan duties, and heater battery duties.

BOILER TEMPERATURE CONTROL

Operating and limit thermostats

The operating thermostat is set to the design flow temperature, and the limit thermostat, which is wired in series, is set 5 K above the temperature of the operating stat. They both control the operation of the boiler fuel burner and are located in the boiler waterways. Control of boilers with modulating burners is more complex but the principle still applies. In the event of failure of both thermostats the burner fires continuously and may cause the water to cavitate within the boiler or high-level pipes where supporting static pressure is at a minimum. This is the purpose of the open vent that is connected to the boiler as a safety device.

Variable-switching thermostat

A sensor takes the place of the operating thermostat and constantly monitors flow temperature, operating with a 5 K switching differential under heavy load. By measuring the angle of decay of the flow temperature the switching differential is widened automatically downwards with decreasing load. This reduces the frequency of boiler operation and brings down the system mean water temperature, thus reducing system output.

Early morning boost

If there is sufficient overload capacity the boiler plant can be set to operate at an enhanced flow temperature on start-up, to heat the building more rapidly before occupation.

Frost protection

If the boiler plant has a shut-down period it may be necessary to ensure that the indoor temperature does not drop below about 12 °C and system water does not freeze. Frost protection overrides the time controls and can energize the pumps to circulate the system water as a first stage and subsequently operate the boiler plant.

Night setback

As an alternative to shutting the boiler plant down, room temperature controls can be set back from say 19 °C to 14 °C during part of the unoccupied period.

Fixed start/stop

Time controls having single or multiple channels with fixed start and stop periods for plant or pump operation are in common use. Plant would be timed to operate over the total heating period, and pumps or three-way valves would be timed to operate for different time-scheduled zones.

Time controls are available for weekly, monthly or yearly time scheduling.

Optimum start/stop

For boiler plant requiring a regular weekly time schedule of, say, five days on and two days off, optimum start/stop control may be appropriate. The self-adaptive system stores and adjusts the start and stop times from the thermal response signature of the building to changes in outdoor climate in respect of indoor design temperature and prevailing indoor temperature. Some of the facilities offered are:

- optimum start/stop of boiler and pumps;
- weather compensation via boiler or three-port mixing valve;
- minimum boiler return temperature control;
- frost protection or night setback;
- day economization;
- multiple zone time scheduling via pumps or valves;
- optimum start boost;
- flue gas temperature monitor and alarm;
- oil level monitor and alarm;
- boiler sequencing;
- lead boiler changeover;
- boiler pump over-run;
- summer turnover for heating pumps.

Sequence control

Multiple boilers can be fired in a sequence dependent upon the prevailing heat load such that on light loads only one boiler is operational, with the other boilers becoming operational sequentially as the load increases.

If the boilers each have a dedicated pump they must operate for a short period after the burner has shut down in response to a fall-off in system load to dissipate the heat in the combustion chamber; otherwise cavitation can occur within the boiler waterways.

Boiler return control

This matter is discussed earlier. Refer to Figures 6.6 and 6.7a and 6.7b

Zoning: time scheduling

This subject is considered in Figure 6.1 and under 'Fixed start/stop.'

CONTROL SYSTEMS

Traditionally many building services systems are controlled using either pneumatics or electric/electronic and mechanical devices such as the five elements in CVVT control: valve body, valve actuator, immersion thermostat, outdoor detector and controller. These may be wired back to a central control panel in which is located the time-scheduling device and frost protection (see Figure 6.14). This type of control still has its place,

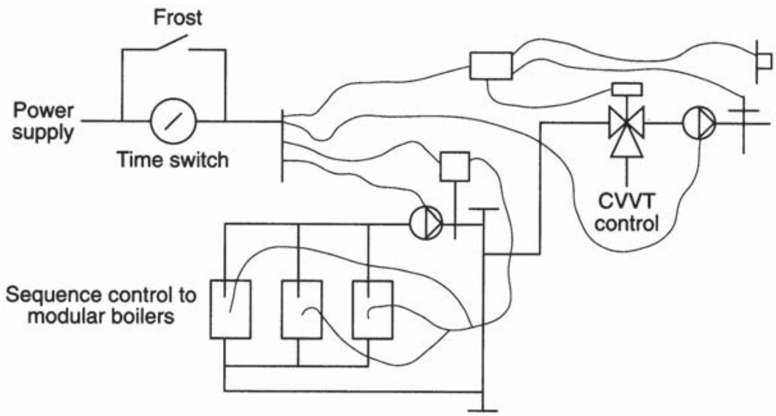


Figure 6.14 Electromechanical system controls.

albeit in an increasingly limited way in commercial and industrial installations.

6.3 Building energy management systems

Direct digital control and supervisory control can be more user-friendly and can give the user more control over the building services systems either locally or remotely via a modem to a **building energy management system** (BEMS). The capital costs and advantages of a BEMS depend upon whether the user has the time and commitment to use this facility and take full advantage of the technology.

BEMS is the subject of another book by the publishers. Here it is therefore only necessary to introduce the concept. The **local area network** (LAN) might include BEMS outstations or original equipment manufacturers' (OEM) outstations, central station and printer. This would be linked to a modem if the final control and monitoring location is remote: say, in another building some miles away. Software is generated and dedicated to operate the controls and relay system conditions such as temperature, relative humidity, pressure, pressure drop, and status such as duty plant operation, standby plant operation. These conditions can be called up on a visual display unit (VDU) or monitor, and will include system logic diagrams. The way a BEMS is connected to a LAN is called the **topology,** of which there are basically three, bus, star and ring. Figure 6.15 illustrates the principles of a BEMS using a **bus** topology.

The keyboard and central processing unit (CPU) complete with visual display unit (VDU), collectively called a **central station,** are connected to the LAN via a modem. The example in Figure 6.15 shows a LAN consisting of four outstations, each with a modem connected on a **bus** topology via a modem to a central station. One outstation is shown providing CVVT control, boiler and

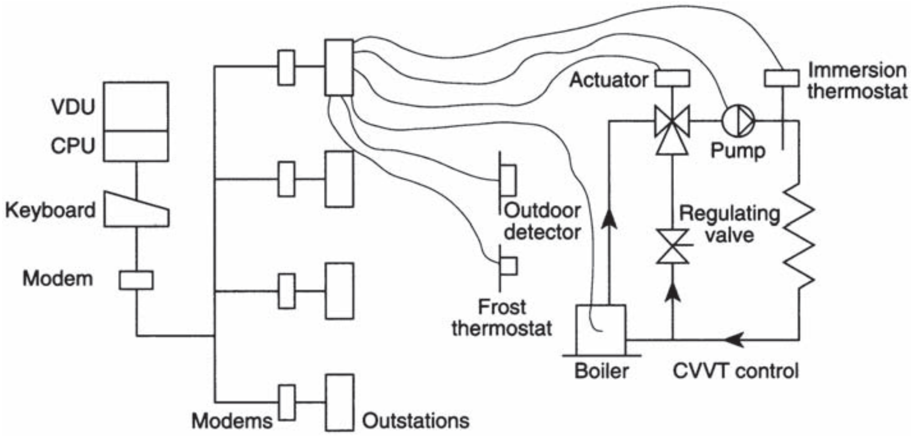


Figure 6.15 Principal features of a BEMS with a bus topology and remote central station.

pump control and frost protection to a circuit of radiators, for illustration purposes.

There are two other topologies in use with BEMS. They are **star and ring,** and are illustrated in Figure 6.16a and 6.16b. With the star network outstations can communicate to each other only through the central station, whereas with the ring network outstations can communicate independently of the central station.

There are two other protocols that are worth mentioning here for the smaller building services systems. The **single packaged local station** shown in Figure 6.17 is suitable for the small commercial/industrial installation. There

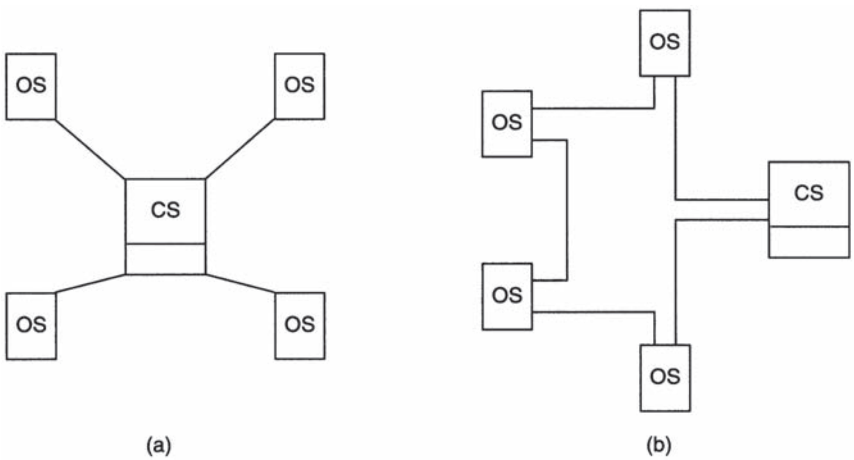


Figure 6.16 (a) Star network; (b) Ring network. CS, central station; OS, outstation.

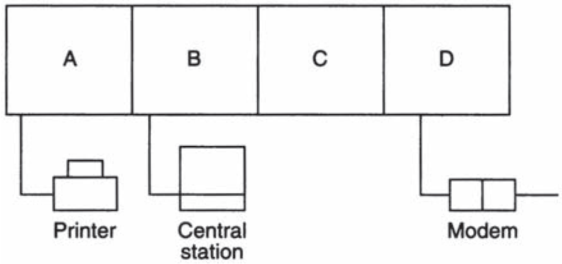


Figure 6.17 Single packaged local station: A, plant and sensor in/out; B, control and monitoring processing; C, user interface and programming; D, data archival remote communication.

is no reason, however, why the single station cannot be linked following expansion and building extension. With the arrangement in Figure 6.18 each local station can communicate following expansion of the enterprise.

OUTSTATIONS

There are two types: the dumb outstation and the intelligent outstation. The **dumb outstation** has the executive control at the central station. This is called **centralized intelligence**. The **intelligent outstation** takes control without input from the supervisor station or central station. This is termed **distributed intelligence**. See Figure 6.19.

OUTSTATION FUNCTIONS

These are split into three levels:

1. high level: remote communication, user interface, optimizer control, cascade control, maintain trend logs, maintain event logs;
2. mid level: proportional plus integral plus differential (PID) control, main data (MD) control, alarm check, alarm communication, program-defined interlocks, calendar/clock control, linearize measurement, convert to engineering units;
3. low level: hard-wired interlocks, elapse timer control, scale measurements, check input limits, debounce inputs, count pulses, plant interface, drive outputs, sensor interface, scan inputs.

INTERLOCK SYSTEMS

Interlocks can be considered as ‘don’t till’ statements, and are of importance in defining the control strategy and in detailing the schematic.

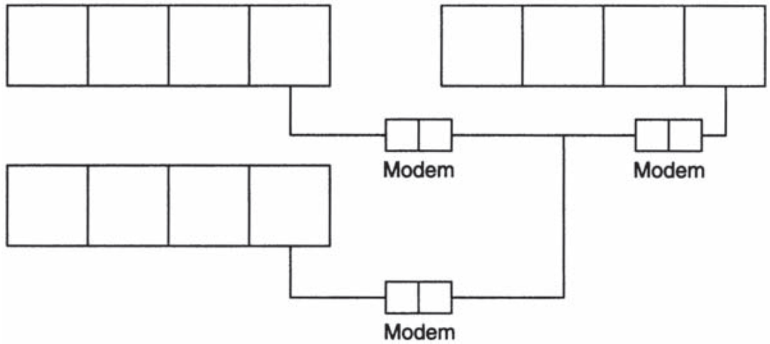


Figure 6.18 Networked local stations.

An example of a system of interlocks at the commencement of plant operation might be:

- *Don't* start primary heating pump *till* the time is right.
- *Don't* start the lead boiler *unless* primary pump is energized.
- *Don't* start secondary pump on zone 1 *till* boiler primary circuit is at 80 °C.
- *Don't* start secondary pump on zone 2 *till* zone 1 is at 80 °C
- *Don't* start unit heater fans on zone 3 *till* the circuit water is at 80 °C.

SUPERVISOR STATION AND CENTRAL STATION FUNCTIONS

These can include the following functions in addition to those listed for the outstation:

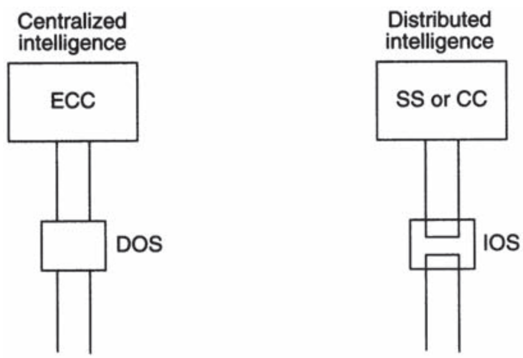


Figure 6.19 The dumb and the intelligent outstation: DOS, dumb outstation; ECC, executive central control; IOS, intelligent outstation; SS or CC, supervisor station or central control.

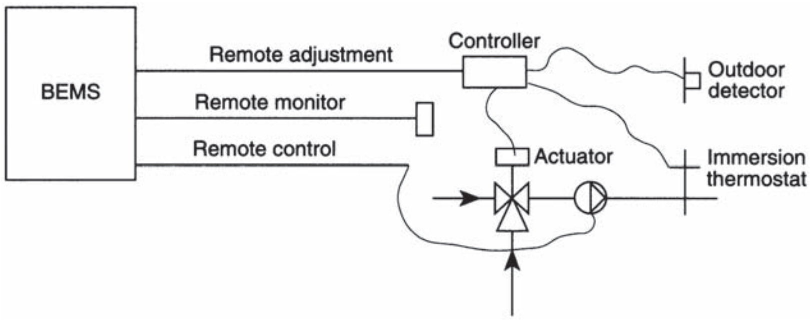


Figure 6.20 A supervisory system providing CVVT control.

- plant supervision;
- maintenance supervision;
- security supervision;
- energy monitoring;
- environmental monitoring;
- system development;
- plant executive control;
- reporting;
- data archival;
- design evaluation.

SUPERVISORY AND DIRECT DIGITAL CONTROL

There is a difference between supervisory control and direct digital control, although the former is often called by the latter name. The supervisory system uses the local controller, whereas direct digital control (DDC) dispenses with it. See Figures 6.20 and 6.21.

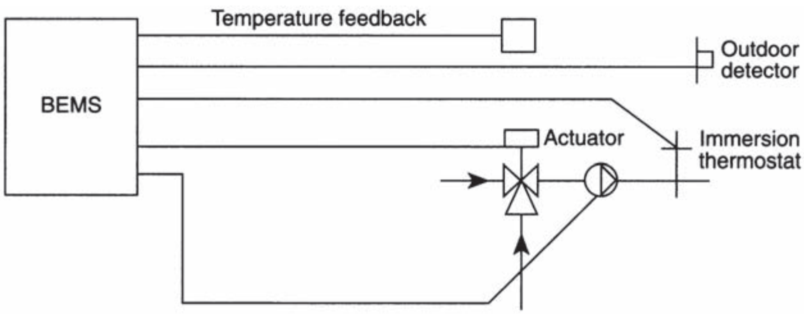


Figure 6.21 Direct digital control system providing CVVT control.

6.4 Control strategies for heating systems

Recommendations have been made relating to controls for heating systems. These include the following.

1. The boiler primary circuit should be pumped at constant volume and be hydraulically independent of the secondary circuits.
2. Domestic hot water should be provided by a separate system.
3. Systems over 30 kW should be controlled by an optimizer.
4. The temperature to each zone should be compensated according to outdoor temperature (clearly this cannot apply to fan convectors or unit heaters).
5. The zones themselves should be selected on the basis of:
   (a) solar heat gain (building orientation);
   (b) building exposure (multistorey buildings—horizontal zoning);
   (c) occupancy times;
   (d) building thermal response;
   (e) types of heating appliance.
6. Space temperature reset of the space heating controls.
7. Frost protection during off periods.

Recommendations have been made for energy-saving features on heating system controllers, and they include:

1. optimizers for loads over 30 kW;
2. time clocks for loads below 30 kW;
3. compensated flow control of the boilers (clearly not possible for fan convectors or unit heaters);
4. protection against frost;
5. sequence control of multiple boilers;
6. control of demand;
7. heating pump run-on facility;
8. summer exercising of pumps;
9. time delays to prevent boilers starting with sudden temporary demand;
10. flue gas monitoring and alarm;
11. seven-day programming;
12. holiday programming;
13. auto summer/winter time change;
14. graphic display of operating times and off times;
15. minimum run times;
16. maximum heating flow temperature.

SYSTEM OPERATION METHOD STATEMENT

An important part of the schematic or logic diagram is a written statement that explains how you intend the systems to operate. The two go hand in hand, and the method statement may indeed help to refine the logic diagram. It will also clarify how, for example, you want the boilers to operate, and may lead to adjustments to the schematic. It is important therefore that both are given time for development in the early stages of project design.

The **method statement** will include:

1. details of zoning by time scheduling and control of temperature;
2. description of plant and circuits, which would include space heating, ventilation, hot water supply, cold water supply;
3. description of interlocks;
4. client interface (how the client can use the systems);
5. specialist interface (where the specialist must be called in to adjust/monitor/refine and maintain).

Case studies

1. Draw a schematic diagram for a system requiring four modular boilers connected to a mixing header and serving two zones of radiators on CVVT control, two air heater batteries located in air-handling units on CTVV control, and one circuit of fan convectors on CTCV control. Secondary hot water supply is required from a bank of two direct gas-fired storage water heaters connected to the MWS. All necessary pumps, valves and controls must be shown, assuming traditional electromechanical control.
2. A multi-storey office block on a north-south axis is for a client who will want it partitioned into offices. It has a central corridor with a staircase each end. Each floor will be let independently.

   Write a system operation method statement for space heating and hot water services in this building.
3. A college campus consists of five buildings, offering the following facilities: building crafts, engineering workshops, catering and restaurant, business studies and college administration, and sports hall.

   There are four plant rooms, with the building craft and engineering workshop buildings having a common plant room and the remaining buildings having independent plant. A BEMS system is specified for the campus.

   Draw a schematic of the plant rooms and example heating, hws and mechanical ventilation systems in each, connected via LANs to a central control station.
4. Describe the protocols for linking local area networks in a building energy management system and identify their strengths and weaknesses.
5. A client wants to update a space heating plant, which is connected to four radiator zones, each controlled by a three-port mixing valve providing CVVT. You decide to use two modular boilers and one condensing boiler, which you hope will give an energy benefit, particularly at the beginning and end of the heating season. Draw a schematic for the boiler plant and

explain how the condensing boiler can take advantage of zone return temperatures below 60 °C.

Now write a plant operation method statement and check that what you intended in the schematic will actually occur.

6.5 Chapter closure

Following the topics introduced in this chapter you are now able to recommend temperature controls for a variety of space heaters. You can prepare schematic diagrams for multi-circuit systems, showing appropriate temperature, boiler and bypass controls and connections to plant. You are in a position to select and propose boiler temperature control options. You are cognizant of the recommended controls on heating systems and the recommended energy-saving features and able to select those that are appropriate. You can write a system operation method statement. Finally, you have a working knowledge of the principles of supervisory and direct digital control offered in building energy management systems.

It is possible to study this chapter without recourse to manufacturers' literature. However, it is strongly recommended that you investigate the market for boilers and controls.

7 The application of probability and demand units in design

Nomenclature

cumulative P_m	probability of satisfying the demand
CWS	cold water service
DU	demand unit
HWS	hot water service
K	constant
m	number of fittings simultaneously discharging
MWS	mains water service
n	total number of fittings in a system
P	probability
P	usage ratio
P_m	probability of occurrence
SD	standard deviation
T	average time between occasions of use
t	average time draw-off is discharging for each occasion of use
X	the mean

7.1 Introduction

Sizing hot water and cold water supply systems, like space-heating systems, is done from a knowledge of flow rates in each pipe section. However, determination of the flow rates for hot and cold water supply is a function of the simultaneous consumption of water. This will vary from one system to the next, depending upon the usage of the draw-off points or fittings on the system, for it would not normally be appropriate to assume that *all* draw-off points in that system will be in use simultaneously.

Simultaneous flow may last only for a matter of a few seconds. It will not necessarily occur from the *same* draw-off points in a system on each occasion of use. There is one application, however, when it is essential to assume that all the draw-off points will be in use simultaneously. This will be mentioned later.

Apart from this and one or two other exceptions, therefore, the concept of **probability** or **usage ratio** P must be introduced into the determination of simultaneous flow before pipe sizing can be considered, otherwise the pipework will be oversized and the system would therefore be unnecessarily costly to install.

7.2 Probability or usage ratio *P*

P is defined as t/T, where t is the average time for which the draw-off point is discharging for each occasion of use, and T is the average time between occasions of use.

The *CIBSE Guide* lists values for usage ratio P for different sanitary appliances: they range from 0.014 to 0.448.

If for a fitting t=60 s and T=300 s then for that fitting the usage ratio P=60/300=0.2.

If, exceptionally, the fitting is used continuously for five occasions, P= (60×5)/300=1.0. That is to say, the fitting is in continuous use. If, exceptionally, a *group* of fittings is being used simultaneously for, say, 60 s, then clearly the total flow when all the fittings are discharging must be used to size the associated pipes even if these fittings are not in use for some time after the event. Clearly the usage ratio is applied only to fittings not in use continuously when $P<1$.

THE BINOMIAL DISTRIBUTION

Probability P follows the binomial distribution, and for hot and cold water supply:

$$P_m = \left(\frac{n!}{m!(n-m)!}\right) \times P^m (1-P)^{n-m}$$

where P_m is the probability of occurrence; cumulative P_m is the probability of satisfying the demand; n is the total number of fittings having the same probability; and m is the number of fittings in use at any one time.
Note:

$$\text{when } m=0,\ P_m=(1-P)^n$$

$$\text{when } m=n,\ P_m=P^n$$

Furthermore, 0!=1 and 1!=1.

Do you agree with these notes relating to the binomial distribution?

Case study 7.1

Ten HWS draw-off points are installed in a building and connected to a common secondary outflow. Each point has a usage ratio P of 0.3. Adopt the binomial distribution and tabulate the probability of occurrence and the probability of satisfying the demand. Draw a probability distribution histogram and calculate the mean and standard deviation. Show that the binomial distribution can be simplified to the formula below for the simultaneous number of points *(m)* discharging for a given common probability when the probability of satisfying the demand is 99.8%:

$$m \approx nP + 1.8[2nP(1-P)]^{0.5}$$

SOLUTION

P_m and cumulative P_m are tabulated in Table 7.1. The number of draw-off points in use simultaneously, assuming for example a 98% probability of satisfying the demand, is, from the tabulated solution, six, at 98.9%. The probability of the event occurring is 3.7%.

You will notice that the highest probability that an event will occur is 26.7% for three draw-off points and not for less or more than three.

The probability distribution histogram is drawn from the values of P_m and the number of draw-off points in use, m (Figure 7.1).

The mean and standard deviation are two measures that describe the frequency distribution of demand. The **mean,** X, is a measure of the central tendency of the distribution, and the **standard deviation,** SD, is a measure of the spread of the distribution:

$$X = \frac{\Sigma fm}{\Sigma f}$$

and

$$SD = \left(\left(\frac{\Sigma fm^2}{\Sigma f}\right) - X^2\right)^{0.5}$$

where

$$P_m = f$$

The values in these formulae, obtained after multiplying $f\,(P_m)$ by 1000 to remove the decimal place, are listed in table 7.2.

Applying the formulae for X and SD:

$$X = \frac{2996}{999} = 2.99$$

and

$$SD = \left(\frac{11\,068}{999} - 8.994\right)^{0.5} = 1.44$$

Table 7.1 Case study 7.1: P_m and cumulative P_m

Number in use, m	*Binomial distribution*	P_m	*Cumulative* P_m
0	$(1-0.3)^{10}$	0.028	0.028
1	$(10!/1!\ 9!)(0.3)^1(1-0.3)^9$	0.121	0.149
2	$(10!/2!\ 8!)(0.3)^2(1-0.3)^8$	0.233	0.382
3	$(10!/3!\ 7!)(0.3)^3(1-0.3)^7$	**0.267**	0.649
4	$(10!/4!\ 6!)(0.3)^4(1-0.3)^6$	0.2	0.849
5	$(10!/5!\ 5!)(0.3)^5(1-0.3)^5$	0.103	0.952
6	$(10!/6!\ 4!)(0.3)^6(1-0.3)^4$	**0.037**	**0.989**
7	$(10!/7!\ 3!)(0.3)^7(1-0.3)^3$	0.009	**0.998**
8	$(10!/8!\ 2!)(0.3)^8(1-0.3)^2$	0.001	0.999
9	$(10!/9!\ 1!)(0.3)^9(1-0.3)^1$	0.000 138	0.999
10	$(0.3)^{10}$	0.000 005 9	0.999

If the number of draw-off points, *m,* varies discretely, i.e. increases by one whole number at a time, and the minimum number is zero and the maximum number is finite, then the distribution is binomial and allows a simplification of the formula such that:

$$X=nP=10\times0.3=3.0$$

and

$$SD=(nP(1\text{-}P))^{0.5}=(10\times0.3\times0.7)0.5=1.45$$

Clearly the error in this context is insignificant, and therefore the binomial formula may be written in its simplified form as: the number of draw-off

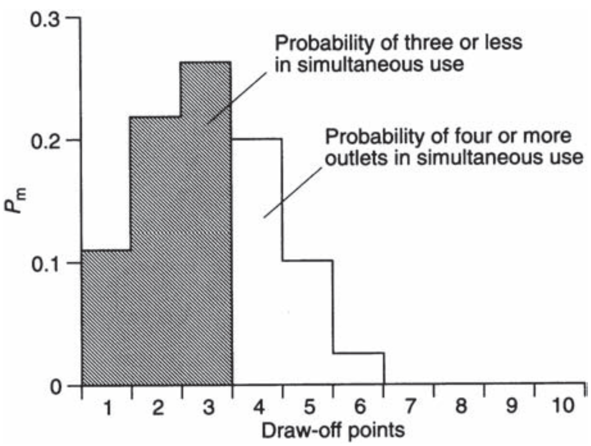


Figure 7.1 Case study 7.1: the histogram of probability distribution of demand.

Table 7.2 Case study 7.1

	f	m	fm	fm^2
	28	0	0	0
	121	1	121	121
	233	2	466	932
	267	3	801	2403
	200	4	800	3200
	103	5	515	2575
	37	6	222	1332
	9	7	63	441
	1	8	8	64
	0	9	0	0
	0	10	0	0
Totals:	999		2996	11 068

points simultaneously discharging, m, is equal to the mean plus a constant K times the standard deviation. This is a mathematical statement that you may know. Substituting for X and SD:

$$m=nP+K(nP(1-P))^{0.5}$$

The constant K is dependent upon the probability of satisfying the demand, P_m, and for a 99.8% probability, K=2.54 taken from Table 7.3. Thus

$$m=nP+2.54(nP(1-P))^{0.5}$$

The simplified formula is more commonly written as:

$$m=nP+1.8(2nP(1-P))^{0.5} \tag{7.1}$$

Do the two formulae for m (the number of fittings simultaneously discharging) agree?

Applying this formula to the solution to the number of points simultaneously discharging:

$$m=10\times0.3+1.8(2\times10\times0.3\times0.7)^{0.5}=7$$

Table 7.3

Values of P_m	Constant K
0.8	0.842
0.9	1.281
0.95	1.645
0.99	2.326
0.998	2.54
0.999	3.09

This solution is appropriate for a 99.8% probability of satisfying the demand. Check with the first set of tabulated data in the solution to this case study. Does it agree? This demonstrates that the simplified formula is valid for use in the determination of the number of draw-off points simultaneously discharging for a common value of probability.

Case study 7.2

Determine the usage ratio P for bathrooms in which 5 min may be taken as the time to fill a bath, after which there is a period of 20 min before the bath might again require filling. Substitute into equation (7.1) and determine the simultaneous demand for 20 baths each having a flow rate of 0.5 l/s.

Determine the simultaneous demand when the total number of baths connected to a common outflow is 20 and the discharge from each is 0.5 l/s for a common usage ratio of P=0.1.

SOLUTION

The first usage ratio P=5/(20+5)=0.2.
Substitute into equation (7.1):

$$m=0.2n+1.8(2n\times0.2\times0.8)^{0.5}$$

This reduces the equation to

$$m=0.2n+(n)^{0.5}$$

Thus for a usage ratio of P=0.2:

$$m=0.2n+(n)^{0.5} \tag{7.2}$$

Substituting n=20:

$$m=0.2\times20+(20)^{0.5}$$

from which

$$m=8$$

and the simultaneous demand

$$=(m/n)\times\text{total flow}=(8/20)\times(20\times0.5)$$

$$=4\ \text{l/s}$$

If all the baths were in use together:

$$\text{flow}=20\times0.5=10\ \text{l/s}$$

This clearly demonstrates the effect of the usage ratio P and its effect on pipe size.

For the second part of the case study the usage ratio P=0.1. Substituting this value for P into equation (7.1):

$$m=20\times0.1+1.8(2\times20\times0.1\times0.9)^{0.5}$$

from which

$$m=5$$

and the simultaneous demand

$$=5\times0.5=2.5\ \text{l/s}$$

or using $m/n\times$total flow:

$$=(5/20)\times(20\times0.5)$$

$$=2.5\ \text{l/s}$$

These results are summarized in Table 7.4.

CONCLUSION

The effect of the usage ratio P on simultaneous flow compared with total flow is clearly apparent.

If the number of baths is doubled the simultaneous flow is not, and for P=0.2 the simultaneous flow for 40 baths is calculated as 7 l/s. For P=0.1 the simultaneous flow is 4.5 l/s. Do you agree?

As the total number of similar fittings (with a common value of P) in a system increases, therefore, the simultaneous flow does not increase at the same rate. Note also that in the solutions for m the numerical value is taken to its nearest whole number.

Case study 7.3

Figure 7.2 shows a centralized HWS system diagrammatically in elevation. Determine the simultaneous flow in each pipe section if the draw-off points have the following common usage ratios: (a) when P=0.2 and (b) when P=0.4.

Each floor has the following fittings: two baths, six basins, one sink and two

Table 7.4 Case study 7.2: summary of results

P	*n*	*m*	*Simultaneous flow*	*Total flow*
0.2	20	8	4	10
0.1	20	5	2.5	10

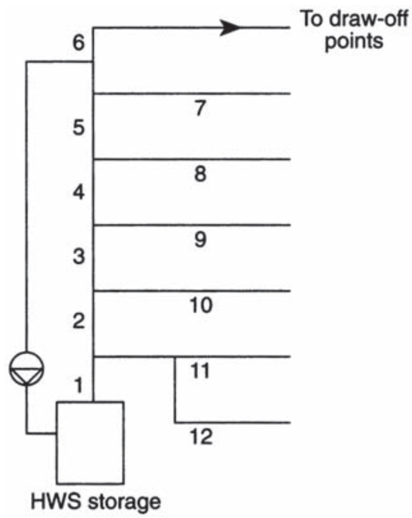


Figure 7.2 Centralized hot water service to seven floors (open vent, cold feed and CWS tank omitted).

showers. The recommended discharge for each fitting is: bath=0.4 l/s, basin=0.15 l/s, sink=0.3 l/s and shower=0.15 l/s.

SOLUTION

The total flow to each floor is calculated to be 2.3 l/s, and for seven floors the total flow will be 16.1 l/s.

(a) Table 7.5a lists the results when P=0.2 and from equation (7.2) $m=0.2n+(n)^{0.5}$.
(b) Table 7.5b lists the results when P=0.4 and from equation (7.1)

CONCLUSION

The simultaneous flows in each pipe section should first of all be compared with the total flow. The simultaneous flows for (a) and (b) should then be compared. These comparisons will give you a feel for the effects of the usage ratio on total flow rates and when the ratio itself is varied.

MULTIPLE PROBABILITIES

The application of the usage ratio *P* is straightforward enough when the same value is adopted for all the draw-off points in a system. However, as indicated at

the beginning of the chapter, the usage ratio varies with the type of fitting in a system of hot or cold water supply, and in such circumstances it is not possible to determine simultaneous flow without initially reducing the fittings on a system to a common value for P.
Consider the next case study.

Case study 7.4

Determine the simultaneous outflow from a hot water service storage vessel serving the following system:

Type of draw-off point	A	B
Number of draw-off points	40	20
Discharge (l/s)	0.2	1
Usage ratio, P	0.2	0.3

SOLUTION

As equation (7.2) is easier to manipulate, it is convenient to convert all the draw-off points in the system to an equivalent usage ratio of P=0.2.
Consider B-type fitting.
Adopting equation (7.1):

$$m=20\times0.3+1.8(2\times20\times0.3\times0.7)0.5$$

$$m=11$$

This is equivalent to n points at P=0.2, and from equation (7.2):

$$m=0.2n+(n)^{0.5}$$

Table 7.5 Case study 7.3: results

Section	1	2	3	4	5	6	7	8	9	10	11	12
(a)												
Total flow	16.1	11.5	9.2	6.9	4.6	2.3	2.3	2.3	2.3	2.3	2.3	2.3
n	77	55	44	33	22	11	11	11	11	11	11	11
m	24	18	15	12	9	6	6	6	6	6	6	6
m/n	0.31	0.33	0.34	0.36	0.41	0.55	0.55	0.55	0.55	0.55	0.55	0.55
Simultaneous flow (l/s)	5.0	3.8	3.13	2.5	1.9	1.27	1.27	1.27	1.27	1.27	1.27	1.27
(b)												
m	42	31	26	20	15	9	9	9	9	9	9	9
m/n	0.55	0.56	0.59	0.61	0.68	0.82	0.82	0.82	0.82	0.82	0.82	0.82
Simultaneous flow (l/s)	8.86	6.44	5.43	4.21	3.13	1.89	1.89	1.89	1.89	1.89	1.89	1.89

Substituting:

$$11=0.2+(n)^{0.5}$$

If $y^2=n$, then

$$11=0.2y^2+y$$

and

$$0=0.2y^2+y-11$$

Adopting the quadratic formula:

$$y=\frac{-b\pm(b^2-4ac)^{0.5}}{2a}$$

Substituting:

$$y=\frac{-1\pm[1+(4\times0.2\times11)]^{0.5}}{2\times0.2}$$

from which the positive solution is y=5.3, and therefore as $n=y^2$, n=28. Thus 28 type B fittings at P=0.2 are equivalent to 20 type B draw-off points at P=0.3.

The *equivalent* system having P=0.2 is therefore:

A	40 points at 0.2 l/s=8 l/s	
B	28 points at 1.0 l/s=28 l/s	
Totals	68	36 l/s

The number of points simultaneously discharging will be, adopting equation (7.2):

$$m=0.2n+(n)^{0.5}$$

Substituting:

$$m=0.2\times68+(68)^{0.5}$$

$$=22$$

Simultaneous demand from the vessel will be:

$$\frac{m}{n}\times\text{total flow}=\left(\frac{22}{68}\right)\times36=11.6\text{ l/s}$$

Case study 7.5

Figure 7.3 shows a cold water down service to a number of fittings from a high-level storage tank. Determine from the data the simultaneous flows in each pipe section.

DATA

Branch	A	B	C	D
n	50	20	30	80
Discharge	0.5	0.15	0.3	0.4 l/s each
P	0.4	0.3	0.2	0.5

SOLUTION

There is again an opportunity to adopt the simplified equation (7.2) if branches A, B and D are reduced to a common probability P of 0.2, which is the usage ratio for branch C.

Converting type A fittings to an equivalent number of fittings having a usage ratio of P=0.2, from equation (7.1):

$$m=(50\times0.4)+1.8(2\times50\times0.4\times0.6)^{0.5}$$

from which:

$$m=29$$

This is equivalent to n points having a probability of P=0.2.

From equation (7.2):

$$29=0.2n+(n)^{0.5}$$

If

$$y^2=n$$

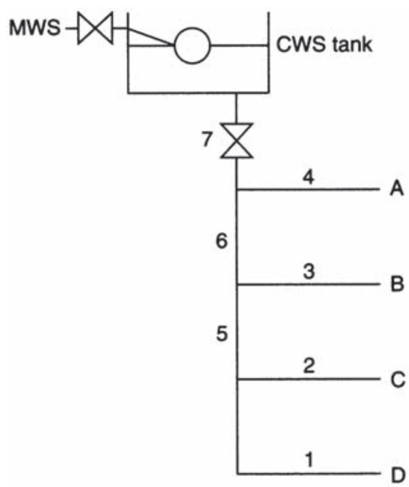


Figure 7.3 Cold water down service to fittings having more than one value of P.

then

$$0=0.2y^2+y-29$$

Adopting the quadratic formula:

$$y=\frac{-1\pm[1+(4\times 0.2\times 29)]^{0.5}}{0.4}$$

from which

$$y=9.8 \text{ and therefore } n=96$$

Thus 50 type A fittings having a probability of 0.4 are equivalent to 96 fittings of the same type having a probability of 0.2.

Converting type B fittings to an equivalent number of fittings having a probability P=0.2, from equation (7.1):

$$m=(20\times 0.3)+1.8(2\times 20\times 0.3\times 0.7)^{0.5}$$

from which

$$m=11$$

This is equivalent to n points having a probability of P=0.2.
From equation (7.2):

$$11=0.2n+(n)^{0.5}$$

If $y^2=n$, then

$$0=0.2y^2+y-11$$

and using the quadratic formula

$$y=5.32 \text{ and therefore } n=28$$

Thus 20 type B fittings having a probability of 0.3 are equivalent to 28 similar fittings having a probability of 0.2.

Converting type D fittings to an equivalent number having a probability P=0.2, from equation (7.1):

$$m=(80\times 0.5)+1.8(2\times 80\times 0.5\times 0.5)^{0.5}$$

from which

$$m=51$$

This is equivalent to n points having a probability of P=0.2.
From equation (7.2):

$$51=0.2n+(n)^{0.5}$$

If $y^2=n$ then

$$0=0.2y^2+y-51$$

and from the quadratic formula

$$y=13.66 \text{ therefore } n=(13.66)^2=187$$

Thus 80 type D fittings are equivalent to 187 similar fittings having a probability of 0.2.

All the fittings in the cold water down service system can now be expressed in terms of an equivalent system of fittings having a common probability. This allows the simultaneous flows in each pipe section to be added back to the common outflow at the tank. It is not possible to add simultaneous flows resulting from different usage ratios because distribution pipes (sections 5, 6 and 7 in this case study) would have to handle simultaneous flows of differing probabilities.

The solutions based upon the common probability for the system of $P=0.2$ are listed in Table 7.6. The determination of m for distribution pipes 5, 6 and 7 highlighted in the tabulation must be done, and a sample calculation is given below for pipe section 5:

187	equivalent fittings at 0.4 l/s=	74.8 l/s
30	equivalent fittings at 0.3 l/s=	9.0 l/s
217	total	83.8 l/s total

Thus for a common probability of 0.2, from equation (7.2):

$$m=(0.2\times 217)+(217)^{0.5}$$

from which

$$m=58$$

Do you agree with the determination of m for distribution pipes 7 and 8?

Case study 7.6

Figure 7.4 shows a system of boosted cold water supply serving four branches, each having a discrete usage ratio. Determine the

Table 7.6 Case study 7.5

Section	*1*	*2*	*3*	*4*	*5*	*6*	*7*	
n	187	30	28	96	217	245	341	
m	51	11	11	29	**58**	**65**	**87**	
Total flow (l/s)	74.8	9	4.2	48	83.8	88	136	
m/n	0.27	0.37	0.39	0.3	0.27	0.26	0.25	
(*m/n*) × total flow	20.4	3.3	1.65	14.5	22.4	23.3	34.7	simultaneous flows (l/s)

simulta neous flow of water in each pipe section for the purposes of pipe sizing.

DATA

Pipe branch	A	B	C	D
n	25	30	20	20
Discharge	0.15	0.3	0.2	0.15 l/s each
P	0.01	0.1 0.	05 0.	01

SOLUTION

It is possible to adopt the simplified equation (7.2) if the system is reduced to the common probability of 0.2. Alternatively, as a usage ratio of 0.01 appears twice in the system, it is equally possible and possibly quicker to reduce the system to P=0.01, in which case equation (7.1) must be exclusively adopted. Thus

$$m=0.01n+1.8(2\times n\times 0.01\times 0.99)^{0.5}$$

Putting this in quadratic form:

$$m=0.01n+1.8(n)^{0.5}\times(0.0198)^{0.5}$$

from which

$$m=0.01n+0.253(n)^{0.5}$$

and if $y^2=n$ then

$$0=0.01y^2+0.253y-m$$

Considering type A fittings:

$$m=(0.01\times 25)+0.253(25)^{0.5}$$

from which

$$m=2$$

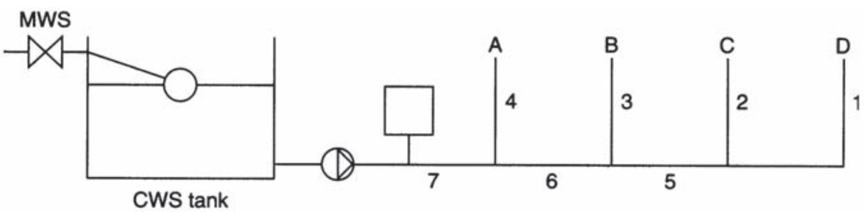


Figure 7.4 Boosted water system with multiple usage ratios.

Considering type D fittings:

$$m=(0.01\times20)+0.253(20)^{0.5}$$

from which

$$m=1$$

Considering type B fittings, equation (7.1) in its original form applies:

$$m=(30\times0.1)+1.8(2\times30\times0.1\times0.9)^{0.5}$$

from which

$$m=7$$

This is equivalent to n fittings having a P=0.01. Thus by substitution:

$$0=0.01y^2+0.253y-7$$

Adopting the quadratic formula, y=16.67 and therefore n=278. Thus 30 B type fittings are equivalent to 278 fittings having a P=0.01 .

Considering type C fittings, equation (7.1) in its original form applies:

$$m=(20\times0.05)+1.8(2\times20\times0.05\times0.95)^{0.5}$$

from which

$$m=3$$

This is equivalent to n fittings having a P=0.01 . Thus by substitution:

$$0=0.01y^2+0.253y-3$$

Adopting the quadratic formula, y=8.79 and therefore n=77. Thus 20 type C fittings are equivalent to 77 fittings having a P=0.01. These solutions are listed in Table 7.7.

As with Case study 7.5, values of m must be determined for distribution pipes 5, 6 and 7. Do you agree with the tabulated solutions?

You are now invited to undertake the solution to this case study by reducing the system to the common probability of 0.2. You can then see whether the simultaneous

Table 7.7 Case study 7.6

Section	*1*	*2*	*3*	*4*	*5*	*6*	*7*	
n	20	77	278	25	97	375	400	
m	1	3	7	2	3	9	9	
Total flow (l/s)	3	15.4	83.4	3.75	18.4	101.8	105.6	
m/n	0.05	0.04	0.025	0.08	0.031	0.024	0.0225	
m/n × total flow	0.15	0.6	2.1	0.3	0.57	2.44	2.38	simultaneous flows (l/s)

Table 7.8 Scales of demand units

Fitting	*Type of application*		
	congested $T=5$ *min*	*public* $T=10$ *min*	*private* $T=20$ *min*
Basin	10	5	3
Bath	47	25	12
Sink	43	22	11
WC	22	10	5
Dishwasher	60	30	20
Clothes washer	40	20	10

flow rates you have calculated for each pipe section agree with the solution above. They may not agree exactly; apart from any errors, can you explain why?

7.3 The system of demand units (DU)

This is not to be confused with discharge units, which are used in the calculation of simultaneous flow for drainage systems.

Clearly, from the work in Case studies 7.4, 7.5 and 7.6, systems having fittings with more than one usage ratio involve tedious calculations, which at best could be transposed to a spreadsheet or database.

The system of demand units simplifies the whole process of determining the simultaneous flow in each pipe section for hot or cold water supply system having more than one usage ratio P.

However, it is based upon a maximum probability of P=0.1, and this imposes limits upon the wider application of demand units. It is important therefore to satisfy oneself that a proposed design for hot or cold water supply can be reconciled with the application of demand units where P is not greater than 0.1, otherwise the approach detailed in Case studies 7.1–7.6 should be adopted.

Table 7.8 lists scales of practical demand units for some typical fittings. The types of application in the table can be misleading, as they refer to the value of the usage ratio, which for *congested* is equivalent to P=0.1. The public and private applications give rise to usage ratios progressively lower than P=0.1.

Having identified the level of use for the various fittings in the system, each can be assigned its practical demand units, and these can be totalled back, section by pipe section, to the common outflow pipe. Table 7.9 is then used to translate the demand units into flow rates.

You will notice that, for example, 200 DU gives a flow of 0.8 l/s; 400 DU gives a flow of 1.3 l/s. In other words there is not a doubling of flow, and as the number

Table 7.9 Conversion of demand units to flow rates

Demand units	*Simultaneous flow (l/s)*																			
	0	50	100	150	200	250	300	350	400	450	500	550	600	650	700	750	800	850	900	950
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

of fittings increases the simultaneous flow rises by smaller increments. This is also apparent if a system is designed by adopting usage ratios.

Case study 7.7

Figure 7.5 shows a centralized HWS installation in elevation. Each branch has the following fittings and demand unit application: one sink (congested), two showers (congested), five basins (public), five basins (private), three baths (congested).

Determine the simultaneous flow in each pipe section.

SOLUTION

From Table 7.8, it is relatively easy to determine the total number of DUs at each branch: one sink at 43, two showers at 10, five basins at 5, five basins at 3 and

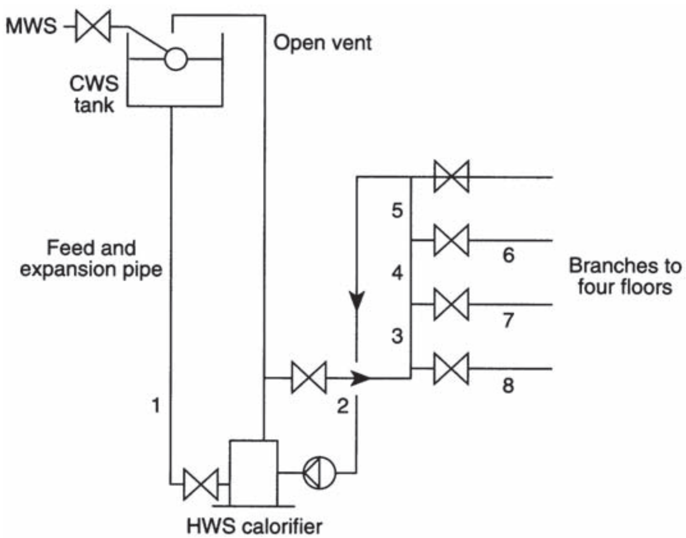


Figure 7.5 Centralized HWS system adopting the scale of demand units.

Table 7.10 Case study 7.7

Section	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*
Demand units	976	976	732	488	244	244	244	244
Simultaneous flow (l/s)	2.6	2.6	2.1	1.5	0.9	0.9	0.9	0.9
Total flow (l/s)	13.2	13.2	9.9	6.6	3.3	3.3	3.3	3.3

three baths at 47, giving a total of 244 DU. It is then appropriate to tabulate the solutions (Table 7.10). The total flow rates are given to show the effect of the application of the scale of demand units, and are obtained from flow rates for fittings given in Case study 7.3.

CONCLUSION

Clearly, the effect of diversity that the application of the usage ratio and the scale of demand units has is considerable on simultaneous flow, on which pipe sizing is based. It is therefore important to apply common sense to this aspect of design. On the one hand, it is likely to be quite uneconomic as well as inappropriate to size the system using total flow rates. On the other hand, application of the scale of demand units reduces significantly the simultaneous flow in each pipe section. The scale of demand units may or may not be appropriate for the project in hand, and experience should be sought and, if necessary, applied if it differs widely from this methodology.

It was identified earlier that there is at least one exception in system design where total flow must be taken for sizing purposes. This occurs where groups of showers are present in, say, a sports centre or sports club or school. In these applications all the showers are likely to be in use simultaneously *and* continuously for more than one group of consumers.

You are also reminded of the range of usage ratios given in the *CIBSE Guide* from 0.014 to 0.448.

7.4 Chapter Closure

You can now apply four methodologies for determining simultaneous flow in pipe networks. You are able to reduce a system having multiple values of probability P to a common usage ratio so that simultaneous flow can be determined in each pipe section. You are able to make decisions or, with the benefit of this underlying knowledge, seek advice from an experienced engineer relating to simultaneous flow in hot and cold water systems for various applications for the purposes of pipe sizing.

8 Hot and cold water supply systems utilizing the static head

Nomenclature

BP	boiler power (kW)
C	specific heat capacity
CWS	cold water service
d	pipe diameter (m)
dt	temperature difference (K)
DU	demand unit
f	frictional coefficient
F&E	feed and expansion
g	gravitational acceleration (m/s^2)
h	static head (m)
L	(circuit) pipe length (m)
m	mass (kg)
M	mass flow rate (kg/s)
P	pressure (Pa)
pd	pressure drop (Pa, kPa)
Q	volume flow rate (m^3/s)
t_f	primary flow temperature (°C)
t_r	primary return temperature (°C)
TEL	total equivalent length (m)
ρ	density (kg/m^3)

8.1 Introduction

Traditionally in the UK water supplies provided within the building originate from a storage tank located at high level, mains water being supplied to the tank from the rising main via a float-operated valve and drinking water being provided off the rising main at locations specified by the water undertaking. This gave the water undertakings some control over the number of connections off their mains water distribution network and hence limited the possibility of back-contamination of the water supply. In current design this is a good starting point, although there are variations around the country and variations are required at any rate for high-rise buildings that exceed the static lift provided by the minimum pressure in the mains water supply at ground level. For most buildings some of the static head provided by the water storage tank located at high level in the building can conveniently be absorbed in sizing the pipework. In centralized hot water supply design, however, the weakest circuit should have available not less than a static head equivalent to 200 Pa/m. Anything below this can lead to uneconomically large pipes, and in such circumstances the use of a pump to overcome the hydraulic resistance in the index run should be considered. This is investigated in Chapter 9.

8.2 Factors in hot water supply design

The following factors need consideration prior to proceeding with system design:

1. the number and type of fittings served (for example, for personal ablutions, for laundering, cooking, dishwashing);
2. number of consumers served;
3. simultaneous flow rates, which may require the application of probability or usage ratio *P* (see Chapter 7);
4. whether fittings are closely grouped or widely distributed (this will define the types of system as centralized hot water generation or point-of-use hot water);
5 nature of the water supply (this will define acceptable materials for the system and the potential need for water treatment);
6. storage temperature (this is now taken as 65 °C minimum to inhibit the growth of *Legionella* spores).

Excessive storage temperatures increase the effects of scale formation and corrosion. The first three factors listed apply in addition to cold water supply. Factors 4, 5 and 6 apply only to hot water supply.

8.3 Design procedures

As implied in the introduction to this chapter, there are two methodologies for centralised systems: sizing on static head, and sizing on a pressure drop of around 300 Pa/m for pumped circulation.

This chapter is confined to the former methodology. The design procedure is as follows:

1. Adopt/select a routine for determining simultaneous demand, and calculate flow rates in all sections of the system (see Chapter 7).
2. Identify the number of circuits in the system, and determine the index run from ratios of height to length *(h/L)* for each circuit in the system.

The index circuit is that circuit whose ratio of *h/L* is numerically the lowest.

3. Rank the remaining circuits from the next lowest value of *h/L* to the highest. They must then be sized in that order after the index run.
4. Determine the available index pressure loss per metre in Pa/m from the available static head after making due allowance for the pressure required at the index fitting, and size the secondary outflow pipes of the index circuit, making sure not to exceed the available static head *h*.
5. Assess the pressure available at the branches.
6. Size the secondary outflow pipes in the remaining circuits in rank order.
7. Size the secondary return and pump.
8. Size the hot water storage calorifier.
9. Determine the required net boiler power.
10. Size the cold water storage tank.

Procedures 7, 8 and 9 do not apply to cold water supply; otherwise, the procedure is similar for both cold water supply and centralized hot water supply.

Case study 8.1

Figure 8.1 shows a centralized system of hot water supply for a factory, which is occupied by 600 employees. The system requires designing given the following information:

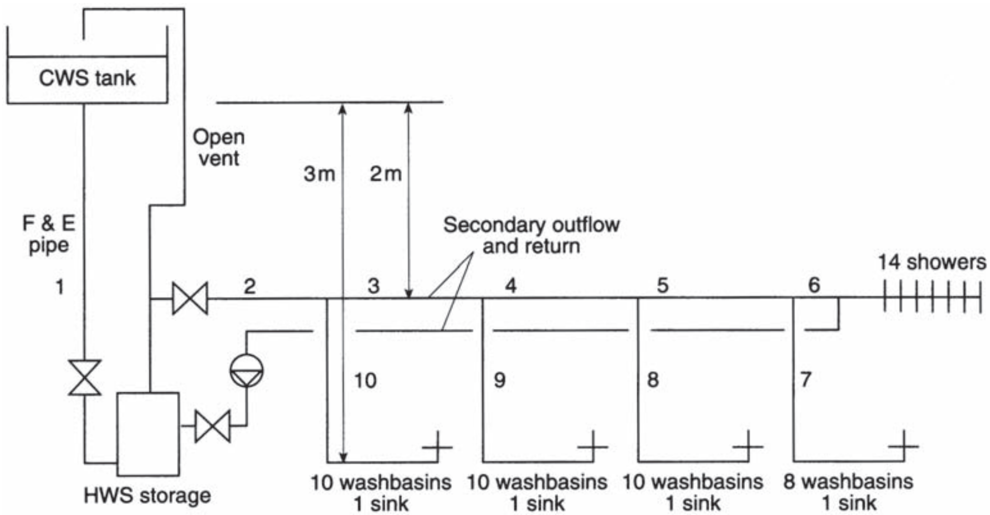


Figure 8.1 Case study 8.1: centralized HWS system for a factory.

Fittings: 38 wash hand basins, 4 sinks and 14 showers.

Pipe section	1	2	3	4	5	6	7	8	9	10
Pipe length (m)	5	6	4	4	4	3	4	4	4	4

SOLUTION

The solution takes account of the factors regarding choice of system listed at the beginning of this chapter and the design procedure.

Without any details of the nature of the water supply, we shall use copper tube Table X and allow 20% on straight lengths for losses through fittings. The question of demand on the draw-off points must now be addressed. It is likely that all the showers will be in use together for each occasion of use. This will affect pipe sizing. It is unlikely that all the wash handbasins will be in use simultaneously, and it is proposed that the scales of demand units are applied to them and to the sinks (see Chapter 7). Section B4 of the *CIBSE Guide* gives data relating to the number of sanitary appliances allocated to staff in most buildings, and this is related to the levels of occupancy and occupation.

For wash basins a total of 22 is allocated for men and 32 for women, taking the occupancy as 600. The factory has a total of 38 wash basins, which accounts for the fact that we are dealing with an industrial building. Clearly a knowledge of the tasks being performed in the factory would be of benefit here.

It is proposed that the wash basins are taken on the congested scale of demand and the sinks, assumed here for cleaners' use, taken on the scale of public demand (see Table 7.8 in Chapter 7). Remember that assumptions must be kept to a minimum, with data from the architect or client obtained in preference.

Determination of simultaneous flow rates

There are 14 showers discharging simultaneously at 0.15 l/s each, giving a total flow of 2.1 l/s.

There are three branches of ten wash basins and one sink, each giving a total of 122 DU, and one branch of eight wash basins and one sink giving 102 DU. This information is best tabulated (Table 8.1).

Number of circuits and the index run

There are five circuits in the system, and the h/L ratios are:

circuit 1, 2, 3, 4, 5, and 6 h/L=2/26=0.077
circuit 1, 2, 3, 4, 5 and 7 h/L=3/27=0.111

Table 8.1 determination of simultaneous flow rates

Section	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*
Shower flow (l/s)	2.1	2.1	2.1	2.1	2.1	2.1	–	–	–	–
Demand units	468	468	346	224	102	–	102	122	122	122
DU flow (l/s)	1.4	1.4	1.2	0.8	0.5	–	0.5	0.5	0.5	0.5
Total simultaneous flow	**3.5**	**3.5**	**3.3**	**2.9**	**2.6**	**2.1**	**0.5**	**0.5**	**0.5**	**0.5**
Available pd (Pa/m)	308	308	308	308	308	308	4564	4815	5121	5255
Actual pd (Pa/m)	188	150	134	306	251	171	1175	1175	1175	1175
Flow diameter (mm)	**67**	**67**	**67**	**54**	**54**	**54**	**22**	**22**	**22**	**22**
Return diameter (mm)	–	**35**	**35**	**28**	**22**	–	–	–	–	–
TEL (m) (+ 20%)	6	7.2	4.8	4.8	4.8	3.6	4.8	4.8	4.8	4.8
pd (Pa)	1128	1080	643	1469	1205	616				

circuit 1, 2, 3, 4 and 8 h/L=3/23=0.130
circuit 1, 2, 3 and 9 h/L=3/19=0.158
circuit 1, 2 and 10 h/L=3/15=0.20

Ranking the circuits

The ratios are listed in rank order of magnitude, with the lowest numerically being the index circuit, namely 1, 2, 3, 4, 5 and 6. This circuit must be sized first, with the remaining circuits being sized in rank order from the next lowest h/L ratio progressively to the circuit with the highest numerical ratio. In fact it is not necessary to follow this ranking procedure here beyond the index circuit, owing to the system layout, but it would be wise nevertheless to keep to the ranking procedure.

Index pressure and index pipe sizing

The available static head to the index circuit from Figure 8.1 is 2 m. Notice that it is taken from the underside of the CWS tank so that the system operates when the tank is almost empty of water. The static pressure is obtained from $P=h$? g=2×1000×9.81. Thus available static pressure is 19620 Pa. The minimum pressure at the index shower head is 10 kPa, so available static pressure for pipe sizing is reduced to 9620 Pa.

Pressure loss per metre, $\mathrm{pd} = \dfrac{P}{TEL} = \dfrac{9620}{26 \times 1.2} = 308\ \mathrm{Pa/m}$

This is the average rate of pressure drop, which can be used to size the index circuit. Note that it is in excess of the minimum quoted in the introduction of 200 Pa/m. Clearly the available static pressure of 19620 Pa must not be exceeded over the index run as there is no more static pressure available to use.

The index run can now be sized using Table X copper tube. This is shown in Table 8.1.

The total pressure absorbed by the index run can now be calculated and compared with the static pressure available:

Pressure absorbed=(pd in pipes 1 to 6)+(pd at index shower)

=6141+10000=16141 Pa

Static pressure available=19620 Pa

It is important not to absorb all the pressure available, as scale build-up will increase the pressure absorbed, and no account has been taken for the pressure drop across the secondary side of the storage calorifier.

Assess the pressure available at the branches

The second circuit in rank order is 1, 2, 3, 4, 5 and 7. The available static head is 3 m and corresponding static pressure is 29430 Pa. Pipe sections 1, 2, 3, 4 and 5 are part of the index run and already sized. The total pressure loss sustained in these pipes is 5525 Pa. The pressure available at branch 5/7 is therefore 29430-5525=23905 Pa.

The third circuit in order of rank is 1, 2, 3, 4 and 8. With a similar static head available the static pressure is again 29430 Pa. The pressure loss sustained in pipes 1, 2, 3 and 4, which are already sized, is 4320 Pa. The pressure available at branch 4/8 is therefore 29430- 4320=25110 Pa.

The fourth circuit in rank order is 1, 2, 3 and 9 having an available static pressure of 29430 Pa. The pressure loss sustained in pipes 1, 2 and 3, which are already sized, is 2851 Pa. The pressure available at branch 3/9 is therefore 26579 Pa.

The fifth circuit in rank order is 1, 2 and 10 having an available static pressure of 29430 Pa. The pressure loss sustained in pipes 1 and 2, which are already sized, is 2208 Pa. The pressure available at branch 2/10 is therefore 27222 Pa.

Size the remaining circuits

You will have noted that the only secondary outflow pipes left to size are branch pipes 7, 8, 9 and 10.

We now have the pressures available at the branches for sizing these pipes. However, the pressure required at the index fitting in each branch should be accounted for before converting these available pressures to pressure losses per metre for pipe-sizing purposes. In each case it will be a sink tap and, like the basin tap, requires 2 kPa discharge pressure.

Thus:

for pipe 7, pd=(23905-2000)/(4×1.2)=4564 Pa
for pipe 8, pd=(25110-2000)/(4×1.2)=4815 Pa
for pipe 9, pd=(26579-2000)/(4×1.2)=5121 Pa
for pipe 10, pd=(27222–2000)/(4×1.2)=5255 Pa.

These available pressures are shown in Table 8.1. Note how much higher in value they are than the available index pressure of 308 Pa/m. This is the result of the increase in static head from 2 m available to the index circuit to 3 m, for the remaining circuits.

The remaining pipe sections can now be sized, and these are tabulated along with the actual rates of pressure loss.

Note that for a pipe size of 22 mm and a simultaneous flow of 0.5 l/s, water velocity from the pipe-sizing tables is in excess of 1.5 m/s, which for copper pipe may generate noise. However, this must be put into context, as unlike a space-heating system any such noise will be intermittent and therefore more acceptable than if it was continuous. Secondly, the system is installed in a factory in which there will be a degree of noise generation in any event.

As a general rule, limits on water velocity in hot and cold water service systems are not imposed owing to their intermittent use, when simultaneous flow is likely to occur only momentarily.

Excessive discharge pressure from hot water taps is potentially dangerous. There is evidence of this in pipe sections 7, 8, 9 and 10, where only a fraction of the available pressure is absorbed by the branch pipes. As a general rule, the maximum pressure at a hot tap outlet should not exceed 50 kPa. In multistorey buildings this limiting pressure is inevitably exceeded, and a pressure-limiting valve should be located behind the fitting.

Size the secondary return and pump

The secondary return is required to ensure hot water at or near the point of use at all times during occupation of the building. This is a regulatory requirement of the water undertaking to reduce the unnecessary consumption of water and to avoid a user waiting for hot water to discharge from the tap.

The secondary pump is required to circulate the water, as it is not usually possible to rely on natural circulation. Note that the pump is only used for this purpose when the system pipework is sized on static head.

It follows therefore that only sufficient water need circulate to offset the heat loss from the circulating pipework. An alternative approach is to dispense with the secondary return and pump and fit electrical tracing tape to the secondary outflow pipe to maintain the supply temperature during periods when water is not being drawn off.

The heat loss from the secondary circulating pipework is a function of its sizes and extent and of the level of thermal insulation applied. It can be estimated using tables of heat loss from insulated pipe given in Section C3 of the *CIBSE Guide*. For this system it is estimated as 3 kW, and taking a temperature drop of 10 K across the circuit the required mass flow to offset the heat losses is 0.071 kg/s.

Adopting a pressure loss of 250 Pa/m, the required size of the secondary return is 15 mm on 240 Pa/m using Table X copper tube. However, the use of 15 mm pipe for this purpose is not recommended owing to the probable effects of scale formation from the constant use of raw water. The minimum pipe size here should be 22 mm. In practice, however, the secondary return is usually taken as one to two sizes below that of the secondary outflow, and this is what is tabulated.

The pressure development required of the pump is based upon the loss sustained in 15 mm pipe. This will generously account for scale build-up between descaling maintenance. The length of the secondary return will be equivalent to sections 2, 3, 4 and 5, namely 18 m, and allowing for fittings the pump pressure required will be

$$P=TEL\times \mathrm{pd}=18\times 1.2\times 240=5184\ \mathrm{Pa}$$

Pressure loss through the secondary outflow, which forms part of the circulating pipework, is negligible as it has been sized on the simultaneous flow rates and not on 0.071 kg/s. Check this out using the pipe-sizing tables: do you agree? Some allowance for the pressure drop across the secondary side of the hot water storage vessel should be made. This would be obtainable from the cylinder manufacturer. Here the required pump pressure for this reason will be increased from 5184 Pa to 7500 Pa.

Pump duty: 0.071 kg/s at 7.5 kPa

On reference to pump manufacturers' literature you will find that it will be the smallest in the range. You will need to specify that it is to be used on secondary hot water, as manufacturing materials are likely to be different from pumps used on closed systems.

Size the hot water storage vessel

There are various ways in which this may be calculated. Table 8.2 is taken from the 1970 *CIBSE Guide* section B4.

For the factory with 600 occupants the size of the storage calorifier will be:

$$600 \times 5 = 3000 \text{ litres.}$$

Net boiler power

From Table 8.2, net boiler power$=600\times0.12=72$ kW.
This should be analysed by determining the length of the regeneration period required. Thus

$$\text{boiler power} = \frac{\text{mass} \times \text{specific heat} \times \text{temperature rise}}{\text{regeneration time}}$$

Substituting:

$$\text{regeneration time} = \frac{3000 \times 4.2 \times (65-10)}{72} = 9625 \text{ s}$$

$$= 2.67 \text{ h}$$

If the factory operates three shifts this would seem satisfactory, for in each 8 h shift approximately 3 h would be required as a heat-up period. For a single shift or two-shift operation the regeneration period could be extended, thus reducing the required net boiler power.

Size the cold water storage tank

Provision of domestic storage to cover 24 h interruption of supply is given in the *CIBSE Guide* and in Table 8.3. The water undertaking will specify for an identified locality the length of the interruption period so that the storage provision can be determined for a building using the data in the table.

There are 600 occupants in the factory, but we do not know whether the factory is on a one-, two- or three-shift system. This may have a bearing upon

Table 8.2 Hot water storage and boiler power

Building type	*Hot water storage at 65 °C (l/person)*	*Net boiler power to 65 °C (kW/person)*
Boarding school	25	0.7
Day school	5	0.1
Dwellings	30	0.7
Factories (no canteen)	5	0.12
Hospitals	30	1.0/2.0
Hotels	35	0.9
Offices (no canteen)	5	0.1
Sports halls	35	0.3

the decision made by the water undertaking relating to the length of the interruption period.

We shall assume here that an 8 h interruption is required, in which case the cold water storage requirement will be: 600×15×8/24= 3000 litres.

CONCLUSION

You will notice that part of the design solution requires the application of logic and common sense: for example, the effect of potential scale build-up on pipe sizes and how much of the available static head can be absorbed in sizing the secondary outflow. It is important to stress the need for these attributes rather than blindly adopting a recommended design procedure. This is particularly true when considering how to arrive at simultaneous flow rates. Adopting a design that has successfully worked on an earlier similar project is not an admission of failure!

The following is an example of the application of common sense.

Case study 8.2

A group of 22 showers, each rated at 0.15 l/s, is attached to a school gymnasium whose timetable has four groups of students using them equally spaced over a 6 h day. The ratio of static head to index length is 0.09, and the cold water storage tank must be sized on an 8 h interruption of mains water service. If the storage per student is 15 litres, estimate without the use of tables the size of: (a) the calorifier, (b) the boiler, (c) the storage tank and (d) the cold feed and secondary outflow.

Table 8.3 Domestic water storage for 24 h interruption

Type of building	*Storage (l)*
Dwellings up to 4 bedrooms	120/bedroom
Dwellings over 4 bedrooms	100/bedroom
Hotels	200/bedroom
Offices with canteens	45/person
Offices without canteens	40/person
Restaurants	7/meal
Schools	
boarding	90/person
day	20/person
Factories	15/person

SOLUTION

(a) If four groups of students use the showers over a 6 h period the calorifier may be sized to supply each group, in which case the regeneration period must be limited to

$$\frac{6}{4} = 1.5 \text{ h}$$

Assuming all the showers could be in use on each occasion:

net size of the calorifier=22×15=330 litres

(b) For a 1.5 h regeneration:

$$\text{net boiler power} = \frac{330 \times 4.2 \times (65 - 10)}{1.5 \times 3600}$$
$$= 14.1 \text{ kW}$$

(c) An 8 h interruption, at worst, would occur at the beginning of the school day, so the storage tank must be sized to cater for a day's use of the showers:

Tank size=4×22×15=1320 litres

(d) The simultaneous flow will be equivalent to all the showers discharging together

=22×0.15=3.3 l/s

The available static pressure for sizing the cold feed and the secondary outflow will be

h/L×?×*g*=0.09×1000×9.81=883 Pa/m

Note: The ratio *h/L* is static height available to index length.

Using Box's formula, which is an adaption of the D'arcy equation for turbulent flow in pipes (see Chapter 10):

$$\frac{\mathrm{d}p}{L} = \frac{Q^2 f \rho g}{3d^5}$$

where the frictional coefficient *f* may be taken as 0.005 and density of water at 65 °C as 980 kg/m^3. Then, rearranging the formula in terms of diameter *d:*

$$d = \left(\frac{Q^2 f \rho g}{3\mathrm{d}p/L}\right)^{1/5}$$

and substituting:

$$d = \left(\frac{(0.0033)^2 \times 0.005 \times 980 \times 9.81}{3 \times 883} \right)^{1/5}$$

from which

$$d = 0.0456 \text{ m} \quad \text{or} \quad 46 \text{ mm}$$

The nearest standard diameter in copper to Table X is 54 mm.

Referring now to CIBSE tables for Table X copper pipe: for a flow rate of 3.3 l/s on 54 mm diameter, pressure drop is 386 Pa/m at 75 °C for the secondary outflow and 483 Pa/m at 10 °C for the cold feed. Do you agree?

Note: There are no pipe sizing tables or corrections for water at 65 °C.

BOILER POWER ASSOCIATED WITH THE GENERATION OF HOT WATER

If the hot water supply system is interconnected with the space heating system the net boiler power BP for the generation of hot water is dependent upon the length of the regeneration period:

$$\text{Net BP} = \frac{m \times C \times \mathrm{d}t}{\text{time}}$$

This calculation is done for both Case studies 8.1 and 8.2. If primary water is used in the heat exchanger within the calorifier, its mean temperature should be at least 10 K above that of the secondary storage temperature of 65 °C for good heat transfer from the primary medium to the secondary medium. It is sensible to have a summer boiler sized to cope with the hot water supply so that the heating boiler plant is not operating during the summer months. Modular boiler plant can be arranged so that one or more of the boiler modules will match the primary hot water demand efficiently. The mass flow of primary water, M, to transfer the heat energy from the boiler to the primary heat exchanger in the calorifier is calculated from:

$$M = \frac{\text{net BP } \textit{plus} \text{ margin}}{4.2 \times (t_f - t_r)} \qquad \text{(kg/s)}$$

where t_f and t_r are the primary flow and return temperatures to the heat exchanger.

GAS-FIRED STORAGE WATER HEATERS

This method of independently heating domestic hot water directly instead of indirectly via a water-to-water or steam-to-water heat exchanger in the calorifier is now frequently specified. Current policy favours the separation of hot water generation and space heating on the grounds of increased thermal efficiency. It also allows the location of the gas-fired storage heaters at or nearer the points of use rather than in the boiler plant room.

Manufacturers offer proven high thermal and combustion efficiencies for their products and supply the heaters for immediate connection to the rising main and gas supply either piped or bottled. Design simultaneous flow is achieved by connecting the appropriate number of heaters in banks to common outflow and return headers. See Figure 6.13 in Chapter 6.

POINT-OF-USE HOT WATER HEATERS

If fittings are widely distributed it may not be viable to have long runs of secondary outflow and return pipework serving them. In such cases the use of local hot water heaters is justifiable. These may be gas fired or electrically operated, and they come in different forms of which there are four main types:

- **instantaneous:** suitable for hand washing at one draw-off point and connected directly to the rising main;
- **over/under sink:** storage heaters ranging from 5 to 30 litres serving one draw-off point and connected directly to the rising main;

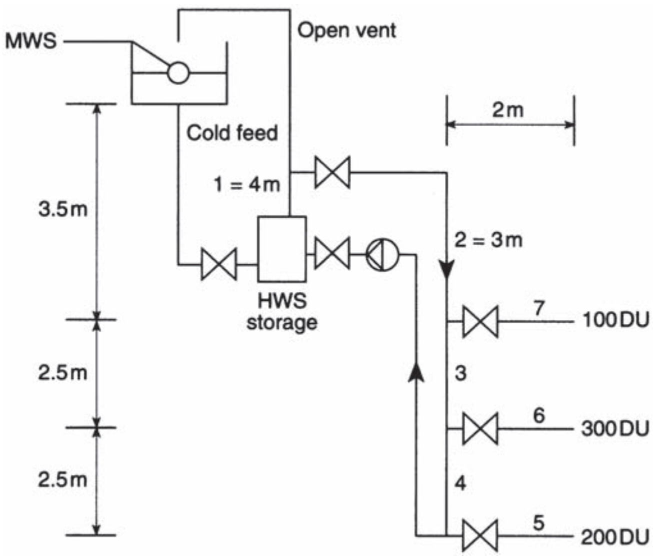


Figure 8.2 Case study 8.3: sizing centralized hot water supply to student accommodation.

- **multipoint:** storage heaters for serving a group of fittings and connected to a high-level storage tank;
- **cistern multipoint:** storage heaters with integral supply tank for serving a group of fittings and connected to the rising main.

Manufacturers are keen to recommend type, size and duty for specified applications.

Case study 8.3

Figure 8.2 shows a centralized HWS system in elevation serving one-bedroom student accommodation for 12 occupants. In accordance with the design procedure set out in this chapter, size the system. The local water undertaking requires an 8 h interruption of supply to be accounted for when sizing the storage tank. Allow 20% on straight pipe for fittings. Copper tube to Table X shall be used. The index draw-off to each student's accommodation is a shower fitting, which requires an operating pressure of 10 kPa.

SOLUTION

The solution is given in Table 8.4. You should now work through your own solution and see if you agree.
Calorifier size=12×30=360 litres
Net boiler power=12×0.7=8.4 kW
This gives a regeneration of 2.75 h. Suggest a regeneration period of 1.5 h, which will require a net boiler power of 15.4 kW.

Net pump duty:

$$M = \frac{\text{estimated 2 kW circulation heat loss}}{4.2 \times 10} = 0.048 \text{ kg/s}$$

P=(124 Pa/m on 15 mm tube at 0.048 kg/s)×8×1.2=1190 Pa

Table 8.4 Case study 8.3: Solution

Section	*1*	*2*	*3*	*4*	*5*	*6*	*7*
Demand units	600	600	500	200	200	300	100
Simultaneous flow (l/s)	1.7	1.7	1.5	0.8	0.8	1.0	0.5
Available pd (Pa/m)	2253	2253	7155	11224	11224	7155	2253
Secondary flow dia (mm)	**35**	**35**	**35**	**22**	**22**	**28**	**22**
Actual pd (Pa/m)	1330	1067	850	Off table	Off table	1200	1175
TEL (+20%) (m)	4.8	3.6	3	3	2.4	2.4	2.4
Pressure drop (Pa)	6384	3841	2550				
Secondary return dia (mm)	–	**22**	**22**	**22**	–	–	–

Say

$$P=3 \text{ kPa}$$

which allows for the pd in the calorifier.
CWS tank size=12×120×8/24=480 litres

8.4 Chapter closure

You are now able to design hot and cold water supply systems utilizing the static head imposed by the cold water storage tank for pipe sizing. You are able to size the cold water storage tank, hot water storage vessel, net boiler power and secondary pump, which offsets the heat loss in the circulating pipework. In conjunction with Chapter 7 you are able to assess the likely simultaneous flow in the secondary circuit for the purposes of pipe sizing and in a multicircuit system put the circuits in rank order for pipe-sizing purposes.

Remember the dictum of applying common sense to the determination of simultaneous flows, calorifier size and regeneration time and hence net boiler power for centralized systems.

You can apply the knowledge you have now gained to sizing point-of-use hot water heaters and associated pipework. You are advised, however, to take account of the heater manufacturer's recommendations.

Hot and cold water supply systems using booster pumps

9

Nomenclature

CWS	cold water service
dp	pressure drop (Pa)
DU	demand units
E	drinking water volume (l, m^3)
F&E	feed and expansion
g	gravitational acceleration=9.81 m/s^2
h	static head (m)
HWS	hot water service
M	mass flow rate (kg/s)
MWS	mains water service
P	pressure (Pa)
pd	pressure drop (Pa, Pa/m)
u	velocity (m/s)
V	volume (m^3)
VFR	volume flow rate (m^3/s)
ρ	density (kg/m^3)
Σ	sum of

9.1 Introduction

In the case of centralized hot water, the economic minimum *available* pressure drop for the index circuit and therefore all the other circuits in the system is about 200 Pa/m. When selecting the pipe size the nearest diameter may yield a pressure drop lower than 200 Pa/m.

If the static head is insufficient to provide this pressure for the index run, a pump should be considered for the secondary circuit to overcome the hydraulic resistance in the index run.

If there is no static head available to a centralized system of hot water supply, as would be the case in the storage of cold water on the same level as the calorifier, a pressurization unit (consisting of a booster pump, pressure vessel and pressure switches) and an expansion vessel located in the cold feed are required to provide the static lift in addition to the secondary hws pump.

In the case of the supply and storage of cold water for sanitary and other uses, when a building exceeds the static lift of the water undertaking's minimum mains supply pressure at ground level, a booster pump is required to get the water to the upper storeys that are beyond the reach of the MWS static lift, and to the high-level storage tank.

Sometimes part of the cold water storage capacity for a building must be held at ground or basement level, owing to the sheer mass of water requiring storage. Clearly, in this case a pressurization unit consisting of a booster pump, pressure vessel and pressure switches is required to transfer the water to all the upper storeys via the high-level tank and to account for the supply of drinking or potable water.

9.2 Pumped hot water supply

Case study 9.1

Figure 9.1 shows a centralized hot water supply system serving a five-storey building.

(a) Show that the system should not be sized on the available static head.

(b) Size the pipework and secondary pump.

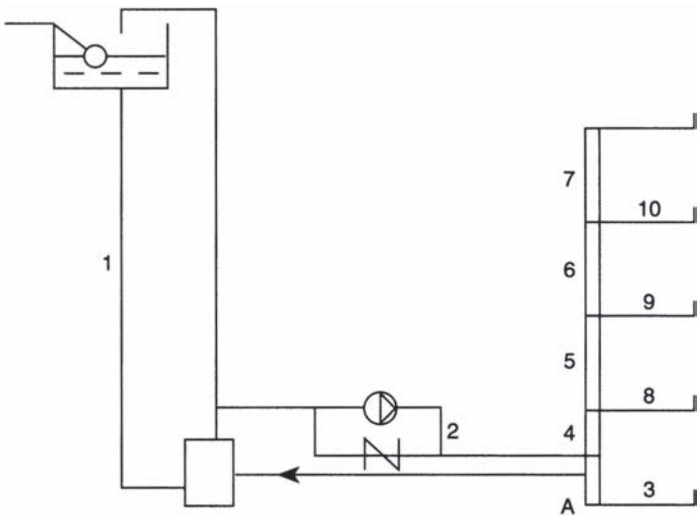


Figure 9.1 Case study 9.1: centralized pumped hot water supply: A, regulating valve on secondary return. All other valves omitted.

Fittings allowance on straight pipe is 25%. The index fittings are basin/sink taps and therefore have a discharge pressure of 2 kPa. Tubing shall be copper to Table X. Each branch carries 100 demand units.

DATA RELATING TO FIGURE 9.1

Vertical height from highest draw-off point to underside of tank is 1 m.

Section	1	2	3	4	5	6	7	8	9	10
Pipe length (m)	15	10	5.5	1.5	3	3	7	4	4	4

SOLUTION

Note how the secondary pump is located in the outflow pipe. If the pump delivery is exceeded by the simultaneous flow it can flow through the pump bypass. As the pump is sized on the simultaneous flow this should happen only infrequently.

(a) Following the procedure and case studies in Chapter 8, the index circuit consists of sections 1, 2, 4, 5, 6 and 7, and the pressure drop available from the static head of 1 m is approximately 152Pa/m. Check this calculation for yourself. This is below the 200 Pa/m threshold, and therefore the system should be pumped.

It could be strongly argued, however, that as there is only one float here attracting a low index pressure it is not worth going to the expense of a pump. In practice there could be many floats at this level, and sizing for a pump would be justified. The solution will proceed on this basis so that you will know how to undertake the design in such an instance.

(b) Adopting a pump pressure of 300 Pa/m the index circuit can be sized. The solution is tabulated in Table 9.1. Note however that section one does not form part of the

Table 9.1 Case study 9.1

Section	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*
Demand units	500	500	100	400	300	200	100	100	100	100
Simultaneous flow (l/s)	1.5	1.5	0.5	1.3	1.0	0.8	0.5	0.5	0.5	0.5
Adopted Pa/m	152	300	2299	300	300	300	300	19 169	13 283	7397
Secondary flow dia (mm)	**54**	**42**	**22**	**42**	**35**	**35**	**28**	**22**	**22**	**22**
Actual pd, (Pa/m)	120	332	1175	257	406	273	340	1175	1175	1175
TEL (m) (flow only)	18.8	12.5	6.88	1.9	3.75	3.75	3.75	5	5	5
pd (flow only) Pa	2250	4150		488	1523	1024	1275			
Secondary return dia (mm)	–	**28**	**22**	**28**	**28**	**22**	**22**	–	–	–

index circuit for the pump and must be sized using some of the available static pressure to the index circuit. The pressure loss is tabulated for the cold feed and expansion pipe and secondary flow only. The secondary return is estimated at one to two sizes below the outflow to maintain hot water at the points of use. The pressure drop in the secondary return is estimated as the same as that in the corresponding outflow pipe sections.

Net pump duty

Pressure loss in the index outflow sections=8460 Pa

Estimated pressure loss in the secondary return=8460 Pa

Total pressure loss in the index pipework=16920 Pa

Pump pressure required=16920+pd at index fitting+pd in hot water storage vessel

=16920+2000+3000 (estimated)

= 21920 Pa

Net secondary pump duty=1.5 l/s at 22 kPa

The pressure drop on the secondary side of the hot water storage vessel consists of two shock losses: a sudden contraction ($0.5u^2/2g$) at the secondary outlet and a sudden enlargement ($u^2/2g$) at the secondary return inlet, assuming water velocity within the storage vessel approaches zero. If water velocity is 1.0 m/s in each of the secondary outflow and return pipes, the pressure loss is approximately 735 Pa. Check this for yourself. Clearly, 3000 Pa is quite sufficient.

Note that the pump pressure does not include pressure to overcome the static lift from the pump outlet to the highest fitting. It is provided in this case study by the static head imposed on the system from the high-level storage tank.

If the hot water storage vessel was supplied from a cold water storage tank on the same level, a pressurization unit would be required to lift the water to the highest fitting (see Figure 9.2).

If the hot water storage vessel is supplied direct off the rising main, minimum mains pressure must be sufficient to overcome the static lift, otherwise a pump is required.

Sizing the F&E pipe, section 1

The cold feed is sized on some of the available static head. The actual rate of pressure loss in section 1 from the table is 120 Pa/m. This is equivalent to

$$120 \times 15 \times 1.25 = 2250 \text{ Pa}$$

and the corresponding static head absorbed will be

$$\frac{2250}{1000 \times 9.81} = 0.23 \text{ m}$$

The static head remaining to lift the water to the index fitting on pipe section 7 will be: available static head at index terminal minus head absorbed by F&E pipe=1.00–0.23=0.77 m, as the pump is sized only to overcome the hydraulic resistance to flow. Some of this residual static head will be needed to size the float attached to section 7.

Sizing the flow, branch 3

Branch 3 flow at junction 4/3 is now considered, taking into account hydraulic balancing with the index run:

Pump pressure available=that absorbed in sections 4, 5, 6 and 7 (flow only)

=(488+1523+1024+1275)

=4310 Pa

Pressure available to size flow branch 3=4310/(1.5×1.25)

=2299 Pa/m

You will see that the attempt to balance hydraulically in the choice of pipe size has been partially successful here, but the valve on branch 3 will need regulation.

Sizing the floats

The float attached to branch 7 and that attached to branch 3, together with floats 8, 9 and 10, are deadlegs and not part of the water circulation. They are subject to the static head from the high-level tank, and are sized on the corresponding static pressure, some of which has been absorbed in sizing the pipe in section 1.

For the float attached to branch 7 the residual static head is 0.77 m. This is equivalent to

$$\frac{0.77 \times 1000 \times 9.81}{5} = 1511 \text{ Pa/m}$$

and 28 mm pipe is chosen at 340 Pa/m to ensure that there is some static head left. These data are not included in Table 9.1.

For the float attached to branch 3 the static head available is:

$$P = (13 - 0.23) \times 1000 \times 9.81 = 125274 \text{ Pa}$$

This is equivalent to

$$\frac{125\ 274}{5} = 25\ 055 \text{ Pa/m}$$

and 22 mm pipe is chosen at 1175 Pa/m. These data are not included in Table 9.1.

Floats 8, 9 and 10 are sized on 22 mm, and it is left to you to check the available static pressures in Pa/m, which are in fact shown in Table 9.1.

When a draw-off point is in use, it is influenced by the algebraic sum of static and pump pressures. The effect in a multistorey building having a centralized hot water supply system results in excessive pressures at the draw-off points on the lower floors. Pressure-limiting valves should be fitted when discharge pressures are likely to exceed 50 kPa at the draw-off point to avoid splash and, in the case of HWS, consequent scalding.

Circuit balancing

This is required to ensure that the circulating pipe section 7 on the index run carries the design flow. Regulating valve A identified in Figure 9.1 will need setting.

BOOSTED HOT WATER SUPPLY

If the cold water supply comes from ground/basement storage there is little or no available static pressure, and the hot water supply system must therefore be pressurized. Consider the centralized system in Figure 9.2.

The system shown is appropriate where the cold water storage tank is located at the same level as the HWS storage vessel at the lowest point in the system or when the minimum MWS pressure is very low. A pressurization unit is located in the cold feed, and a booster pump handles cold water, operating intermittently under the dictates of a pressure switch. The integral pressure vessel has a gas cushion to iron out small pressure variations and so reduce the number of pump operations.

Net pressure developed by the booster pump must include the static lift to the highest fitting. Net booster pump flow rate is equivalent to the simultaneous flow of hot water supply and therefore cold water make-up.

In addition, a further pump is located in the secondary circuit. Its flow rate is the simultaneous flow for the system, and its developed pressure accounts for the sum of the hydraulic loss in the index circuit and the discharge pressure at the index draw-off point.

Note that in addition to the pressure vessel, which forms an integral part

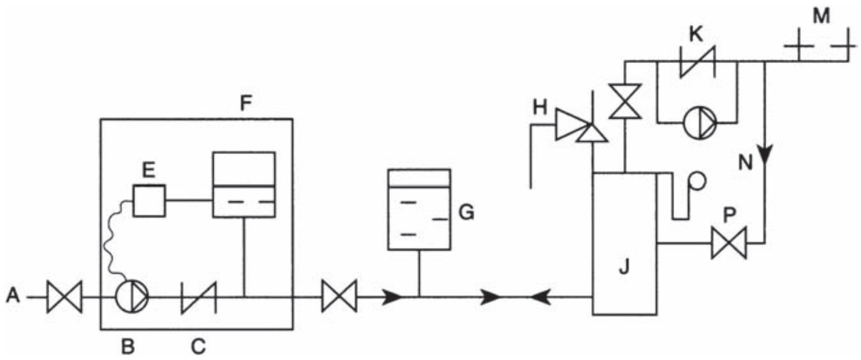


Figure 9.2 Centralized HWS system pressurised at the cold feed: A, low-pressure cold water supply; B, booster pump; C, non-return valve; E, pressure switch; F, pressure vessel; G, expansion vessel; H, pressure relief valve; J, HWS storage vessel; K, non-return valve; M, draw-off points; N, HWS pump; P, pressure gauge. Note: B, C, E and F are the constituent parts of the pressurization unit.

of the pressurization unit, there is also an expansion vessel fitted to the cold feed. This accepts expansion water from the HWS storage vessel on heat-up.

USE OF THE HWS RING MAIN

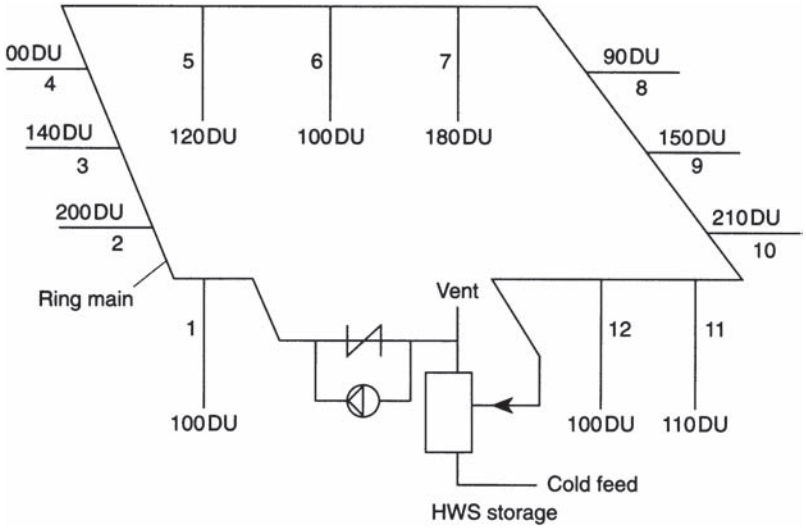
In commercial applications such as laundries, hot water supply is often direct rather than indirect, and use is made of a ring main to supply the draw-off points.

Case study 9.2

Figure 9.3 shows an HWS ring main system. Size the ring main and branches and size the pump. Assume that the static head from the CWS tank is insufficient for sizing the pipework but can provide the static lift.

DATA

Total equivalent length of the ring main is 48 m.
Pressure required at each draw off point is 10 kPa.
Adopt a pressure drop of 300 Pa/m.



igure 9.3 Case study 9.2: HWS ring main system.

SOLUTION

Total demand units ?(DU)=1600, and from Table 7.9 in Chapter 7 simultaneous flow is 3.8 l/s. From the CIBSE pipe-sizing table for copper Table X, the ring main diameter selected is 54 mm on 500 Pa/m. The velocity is approximately 1.7 m/s, but flow of 3.8 l/s is the *total* simultaneous flow for the ring main and immediately reduces as the hot water flows around the ring, thus reducing the pressure loss progressively back to the HWS storage vessel.

Size of ring main is 54 mm

The branches can be sized using copper Table X after converting the demand units to simultaneous flow (Table 9.2).

Pump duty:

pressure developed=pd in index circuit *plus* pd at index terminal

Note: Pressure loss in the index branch is ignored, as it is assumed to be short. The pressure loss in the ring main of 500 Pa/m reduces progressively around the ring as flow reduces. An average pd of 350 Pa/m will be used to assess the pump pressure.

$$\text{Pressure developed} = (350 \times 48 + 10000)\ \text{Pa}$$
$$= 26.8\ \text{kPa}$$

Simultaneous flow=3.8 l/s.

Table 9.2 Case study 9.2

Branch reference	*DU*	*Flow (l/s)*	*Diameter (mm)*	*pd (Pa/m)*
1	100	0.5	28	340
2	200	0.8	35	273
3	140	0.6	35	163
4	100	0.5	28	340
5	120	0.5	28	340
6	100	0.5	28	340
7	180	0.7	35	215
8	90	0.5	28	340
9	150	0.6	35	163
10	210	0.8	35	273
11	110	0.5	28	340
12	100	0.5	28	340

Thus net pump duty is 3.8 l/s at 27 kPa.

Note: The pump flow must be oversized, not least as the flow must be sufficient to ensure some water returns to the storage vessel when the estimated 3.8 l/s is drawn off the ring main.

Case study 9.3

Figure 9.4 shows an HWS ring main serving ten draw-off points. Size the ring main and branches. Size also the pump, calorifier and electric immersion heater. Assume the available static head from the CWS tank is insufficient for pipe sizing but can provide the static lift.

DATA

Total equivalent length of the ring main is 20 m.
Pressure required at the draw-off points is 15 kPa.
Number of people using the system simultaneously is 30.
Hot water storage is 25 l/person.
Regeneration period is 1 h.
Flow rate to each draw-off point is 0.2 l/s.

SOLUTION

Clearly the design solution to hot water supply is made easier if all the parameters are known, as they are here.

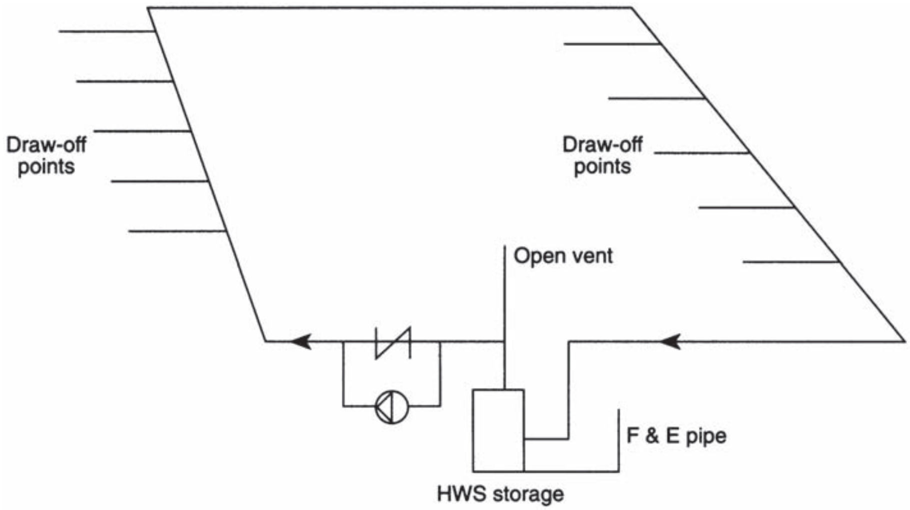


Figure 9.4 Case study 9.3: HWS ring main serving ten draw-off points.

Simultaneous flow in the ring main is 2.0 l/s and using CIBSE Table X copper tube the diameter is 54 mm at a pressure loss of 156 Pa/m and a velocity of 1.0 m/s or 42 mm diameter pipe at a pressure loss of 560 Pa/m and a velocity of approximately 1.7 m/s. If all the fittings are used continuously as well as simultaneously there will be the possibility of persistent noise generation if 42 mm pipe is adopted. This may not be important if background noise is present. This solution will take a ring main size of 42 mm.

Ring main size 42 mm

At 0.2 l/s branch size from Table X will be 22 mm at a pressure loss of 228 Pa/m.

Size of branch pipes 22 mm each

Pump pressure=index pressure loss plus pressure loss at index fitting, index branch ignored. Taking an average pd around the ring main of 400 Pa/m:

$$=400\times20+15000=23000 \text{ Pa.}$$

Flow rate=2.0 l/s

Net pump duty is 2.0 l/s at 23 kPa

The comments referring to pump duty in the solution to Case study 9.2 are the same here.

Calorifier size=30×25=750 litres

$$\text{Net electrical power} = \frac{750 \times 4.2 \times (65 - 10)}{3600} = 48.1\ \text{kW}$$

9.3 Boosted cold water supply

With the advent of high-rise buildings it is essential to ensure that the mains water service is sufficiently pressurized to overcome the static height so as to supply water at a reasonable pressure to the highest draw-off point in the building, which might be the ball valve in a roof-level storage tank. If minimum MWS pressure at ground level is 2.5 bar gauge, this is equivalent to a static lift of 25.5 m. Given floor heights of 3 m, the main can supply seven storeys under its own pressure and still have some surplus pressure available for hydraulic resistance to flow and discharge.

If the building is over seven storeys the MWS must be boosted to the upper floors. The need for water, apart from industrial applications, which may require hot water for process as well, is twofold:

- for a potable water supply suitable for cooking and drinking;
- for a supply of water for washing and sanitary use.

A further requirement may include a water supply for fire hose reels and/or sprinkler system and/or wet fire riser.

There is therefore a need for at least two water supplies within the occupied building. For high-rise buildings that are beyond the static lift of the rising main, there may be a need therefore for the provision of booster pumps serving two or three independent services.

The ground storage of water is becoming the norm for high-rise buildings, and for domestic consumption (drinking, cooking, washing and sanitary use) a common rule of thumb is 2/3 ground storage and 1/3 roof level storage.

Fire hose reel, wet fire riser and sprinkler systems may also be served from ground storage tanks.

In each of these cases booster pumps are required.

For domestic water use in larger buildings beyond the static lift of the MWS there are two systems of water boosting in use: control by water level and control by water pressure.

BOOSTED WATER: CONTROL BY WATER LEVEL

A typical arrangement is shown in Figure 9.5.

There is a tendency to flow reversal, owing to the effects of gravity in the boosted riser when the pump stops. The recoil valve (F) is a spring-loaded non-return valve, which is fitted to insure against this effect.

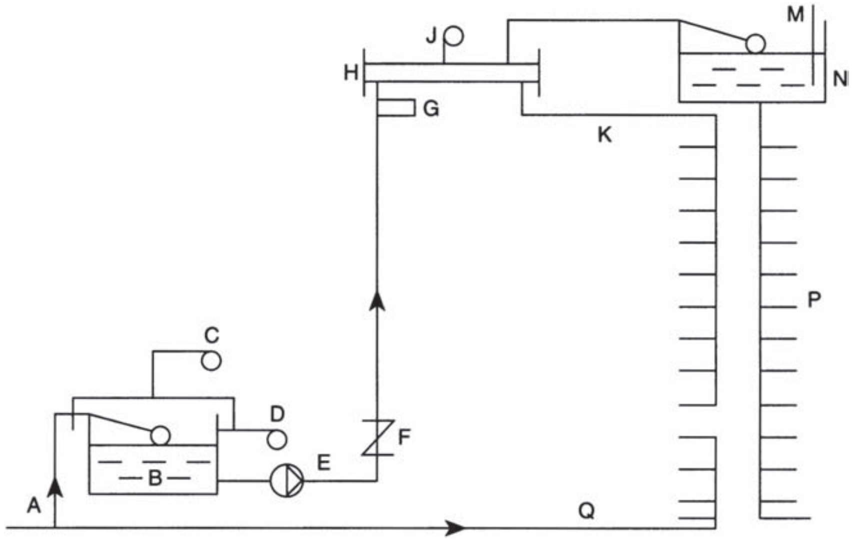


Figure 9.5 Boosted cold water, control by water level, valves omitted: A, mains water service; B, ground storage tank; C, open vent with filter; D, overflow with filter; E, booster pump; F, recoil valve; G, float switch; H, drinking water header; J, air release/intake valve; K, drinking water to upper floors; M, float switch; N, high-level cold water storage tank; P, cold water down-service serving urinals, WCs, basins, sinks etc.; Q, drinking water to lower floors.

Pipes K and P will require water pressure reduction at intervals of five storeys to avoid excessive pressures at the draw-off points. A fixed pressure-reducing valve is used for this purpose.

Referring further to Figure 9.5, a branch from the rising main serves the drinking water points to four lower floors. Another branch from the rising main serves the ground storage tank via a ball valve. This tank in fact carries the drinking water to the upper floors as well as the water for other uses, and therefore must be protected from ingress of foreign matter. The booster pump is activated by either of the float switches: at low water level in the high-level storage tank, which serves all 13 floors with water for general use, or low water level in the drinking water header, which supplies the nine upper floors with potable water.

Case study 9.4

Consider a 15-storey building with four three-bedroom flats per floor, each flat having one drinking water point. Minimum mains water pressure is 2 bar gauge and floor heights are 3 m.

Determine:
(a) cold water storage requirement;
(b) net booster pump duty;
(c) the drinking water storage.

SOLUTION

A typical riser diagram for the system is similar to Figure 9.5 except in relation to the number of storeys in the building.

(a) From section B4 of the *CIBSE Guide* the recommended storage of cold water for a 24 h interruption of supply is 120 litres per bedroom.

$$\text{Gross storage} = 120 \times 3 \times 4 \times 15 = 21600 \text{ litres.}$$

The water undertaking will advise the length of the interruption period, which we shall take here as 12h. Thus gross storage is 21600×12/24=10800 litres.
Applying the rule of thumb:

$$\text{ground storage} = 10\ 800 \times \frac{2}{3} = 7200 \text{ litres}$$

$$\text{high-level storage} = 10\ 800 \times \frac{1}{3} = 3600 \text{ litres}$$

(b) Net pump flow rate consists of the maximum drinking water demand plus the supply for refilling the high-level storage tank.

The lower floors can be supplied from the rising main.

$$\text{Static lift of the rising main} = \frac{P}{\rho g} = \frac{200\ 000}{1000 \times 9.81} = 20.4 \text{ m}$$

This is equivalent to six storeys: thus nine storeys must be pumped.

This represents 9×4=36 drinking water points, and taking 10 demand units per point we have 360 demand units.

From Chapter 7 the simultaneous flow for 360 DU is 1.2 l/s.
If the booster pump must refill the tank in 8 h the water supply will be

$$\frac{3600}{8} \times 3600 = 0.125 \text{ l/s.}$$

Thus the pump flow rate is

$$1.2 + 0.125 = 1.325 \text{ l/s.}$$

Note: If each flat has one bath, two basins, one sink and one shower, demand units total 120, and for the entire building total DU=120× 4×15=7200.

From section B4 of the *CIBSE Guide* this represents a simultaneous flow of 13.8 l/s from the high-level tank. However, simultaneous flow is not continuous flow and will only occur momentarily, so it must not be confused with the calculated fraction (0.125 l/s for 8 h) for replenishing the high-level cold water storage tank by the booster pump. The volume of water in the high-level storage tank is there to iron out the variations of flow into and out of the tank. At the beginning of the day (7 am) the high-level storage tank will be full of water and ready to cope with the momentary simultaneous flow, assuming little is drawn off after 11 pm the previous evening.

Do you agree with the logic here? You can see that the methodology is largely based on common sense.

Pump pressure developed must now be considered and will include: hydraulic resistance in the index circuit *plus* static *lift plus* discharge pressure at the ball valve (index terminal) in the high-level tank.

For a flow rate of 1.325 l/s in 42 mm copper pipe to table X the pressure drop is 337 Pa/m. If the index run is 50 m and taking an allowance for fittings of 15% on straight pipe, the hydraulic resistance is

$$337 \times 50 \times 1.15 = 19.38 \text{ kPa}$$

Static lift is equivalent to:

$$h \times ? \times g = 3 \times 16 \text{ storeys (to tank room)} \times 1000 \times 9.81$$

$$= 471 \text{ kPa}$$

Recommended discharge pressure at the ball valve is 30 kPa.
Pump pressure developed therefore is 19.38+471+30=521 kPa.

Net booster pump duty is 1.325 l/s at 521 kPa

You will have noticed that the discharge pipe from the booster pump has been sized at 42 mm.

Note: If all the water is stored at high level and the booster pump is fitted directly to the rising main, the rate of flow required in 8 h to refill the tank, which now stores 10800 litres of water, will be 10800/8×3600=0.375 l/s. This is added to the drinking water requirement of 1.2 l/s, giving 1.575 l/s.

Pump pressure developed will be 521 kPa *minus* the minimum mains pressure of 2 bar gauge *minus* the original hydraulic resistance at 1.325 l/s *plus* the hydraulic resistance at 1.575 l/s, which in 42 mm copper pipe to table X is 26 kPa. Thus pump pressure =521-200-20+26=327 kPa.

Net booster pump duty is 1.575 l/s at 327 kPa

Do you agree with this calculation? Clearly the pump now only has to generate static pressure equivalent to the static height *above* the minimum mains pressure.

Note: In either case the pressure developed by the booster pump is considerable, and not usually achieved by a single-stage centrifugal pump. Multistage pumps are commonly employed for this service.

(c) The drinking water storage for the upper floors is calculated in two ways. In each case it is relatively small to insure against stagnation in the drinking water header and encourage a high rate of displacement. It is for this reason that the connection to the ball valve comes off the drinking water header.

One method of calculation is to allow two minutes storage: thus $2\times60\times1.2$ l/s=144 litres.

The other method is to allow 4.5 litres/dwelling: thus $4.5\times4\times9$ floors=162 litres.

Size of drinking water header serving upper floors is 162 litres.

BOOSTED WATER: CONTROL BY WATER PRESSURE

This method of providing high-rise buildings with water supplies is more common, as it does not require electrical wiring from ground/basement where the booster pump is situated to the high-level tank room where the float switches are located in the storage tank and drinking water header.

There are a number of specialist pump manufacturers who offer water pressurization plant similar to that shown in Figure 9.2, where all the controls are located in the pressurization unit. See Figure 9.6.

The cold water down service will require pressure reduction at intervals of five storeys to avoid excessive pressures at the draw-off points.

The pressure vessel is partially gas-filled to reduce the number of pump operations. It is sized to hold the calculated quantity of drinking water. As water is drawn off from drinking water to the upper floors the drop in pressure activates the booster pump. The high-level cold water storage tank can be fitted with a delayed action ball valve so that the pump is activated at a predetermined low water level. This also assists in reducing the number of pump operations.

Case study 9.5

The block of flats adopted in Case study 9.4 is considered again for boosted water control by water pressure.

Design procedures include determination of:

(a) cold water storage;

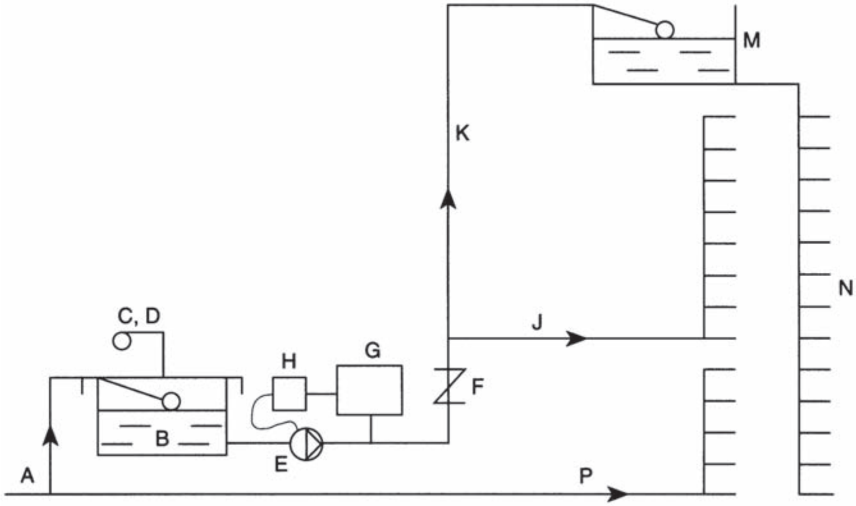


Figure 9.6 Boosted cold water, control by water pressure, valves omitted: A, mains water service; B, ground storage tank; C, D, filtered open vent and overflow; E, booster pump; F, recoil valve; G, pressure vessel; H, pressure switch; J, drinking water header to upper floors; K, boosted water to CWS tank; M, high-level cold water storage tank; N, cold water down-service serving urinals, WCs, basins, sinks etc.; P, mains drinking water to lower floors. E, G and H form the pressurization unit, which is supplied complete.

(b) net pump duty;
(c) drinking water storage;
(d) size of pressure vessel.

SOLUTION

(a) Cold water storage remains the same, at 3600 litres at high level and 7200 litres at ground level.

(b) Pump flow rate remains the same at 1.325 l/s. Pump pressure developed needs further consideration.

At no flow, cut-in pressure for the pump P_2=static lift=471 kPa gauge or 571 kPa absolute. At no flow, cut-out pressure P_3 for the pump=static lift plus 1 atm differential

=471+100=571 kPa gauge or 671 kPa absolute.

Initial pressure P_1 is taken as 1 atm below cut-in pressure and =371 kPa gauge or 471 kPa absolute.

Net pump duty is therefore design flow at pressure P_3 and will be 1.325 l/s at 571 kPa gauge.

This compares with 1.325 l/s at 521 kPa gauge for control by water level.

(c) The reason for identifying the absolute pressures P_1, P_2 and P_3 in part (b) of the solution is that they are needed to size the pressure vessel. The process of pressurization is isothermal, and therefore
Boyle's law can be adopted. From Chapter 4:

$$V_1 = \frac{P_2}{P_1}\left(\frac{P_3 E}{P_3 - P_2}\right)$$

where V_1=volume of the vessel and here E=drinking water storage requirement.

$$\text{The volume of the pressure vessel therefore} = \frac{571}{471}\left(\frac{671 \times 144}{671 - 571}\right)$$

$$= 1171 \text{ litres}$$

Note:

1. The difference in the cut-in and cut-out pressures is 1 atm to ensure sufficient differential to provide clear signals for pump operation.
2. In the determination of pressures P_2 and P_3, when the pump is operating against no flow, it is helpful to remember that there is no hydraulic loss and no pressure required for discharge at the ball valve.
3. The sum of the hydraulic loss and ball valve discharge pressure is less than the 1 atm differential, which is therefore taken to determine pressure P_3.

BOOSTED WATER: CONTINUOUS PUMP OPERATION

The continuously running pump system, which is a useful application for smaller systems, relies for its control upon careful pump selection and a small bleed from pump delivery to suction (Figure 9.7). Potable water and water for washing and sanitary use can be supplied in this way following approval from the water undertaking.

DELAYED-ACTION BALL VALVES

To reduce the number of starts for booster pumps controlled by water/gas pressure, a delayed-action ball valve may be fitted to the mains water supply

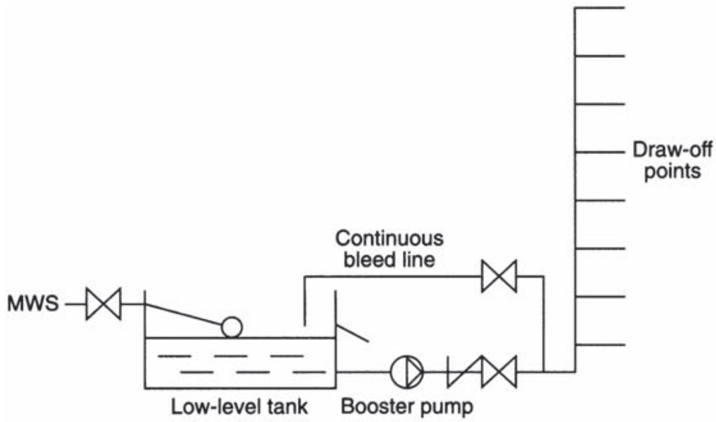


Figure 9.7 Continuously running booster pump. Tank requires sealed cover and filtered open vent and overflow.

serving the high-level storage tank. There are various types on the market, but each ensures that the ball valve remains closed after the tank is full until the water level reaches a predetermined low point in the tank. This can have the effect of countering the need for sufficient storage in the event of an interruption of the water supply occurring at low water level in the tank. Advice should be sought from the water undertaking.

WATER SUPPLIES TO BUILDINGS IN EXCESS OF 20 STOREYS

Cold water storage

As a rule of thumb, cold water storage would be taken via a ground storage tank to intermediate storage tanks at ten floor intervals. (Figure 9.8).

Drinking water storage

The provision of potable water in high-rise buildings must insure against bacterial growth and contamination. This usually requires the use of sealed tanks, vessels or headers. For buildings over 20 storeys, ground storage may be advised, with further storage provided on the upper floors. Alternatively, the booster pump may be connected direct to the mains water supply to serve a storage vessel at roof level (Figure 9.9).

The drinking water storage tank at roof level must be fitted with a sealed cover and filtered overflow and open vent.

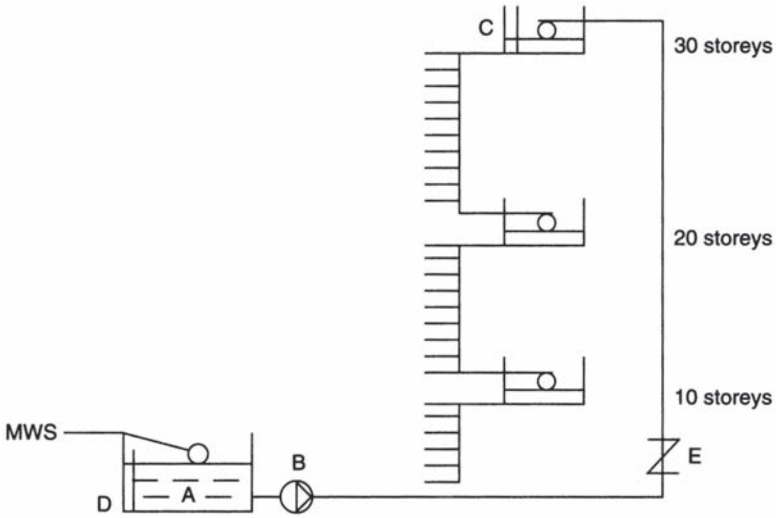


Figure 9.8 Boosted cold water above 20 storeys: A, ground storage tank; B, booster pump; C, float switch pump control; D, low-level float switch pump control; E, recoil valve.

Case study 9.6

A 30-storey office block having a central core of toilet accommodation and basement allocation for water storage is at the feasibility design stage. Estimate the duty and power requirement for the boost pumps handling the domestic water services.

DATA

Occupancy is to be taken as 100 per floor except the ground floor, which consists of the entrance, reception and showcase booths. Floor heights are 3 m.

SOLUTION

Two water services are required here: for washing and sanitary use, and for drinking.

From the *CIBSE Guide* section B4, offices without canteens require 40 l/person domestic storage to cover a 24 h interruption of water supply. Total storage=40×100×29=116000 litres of

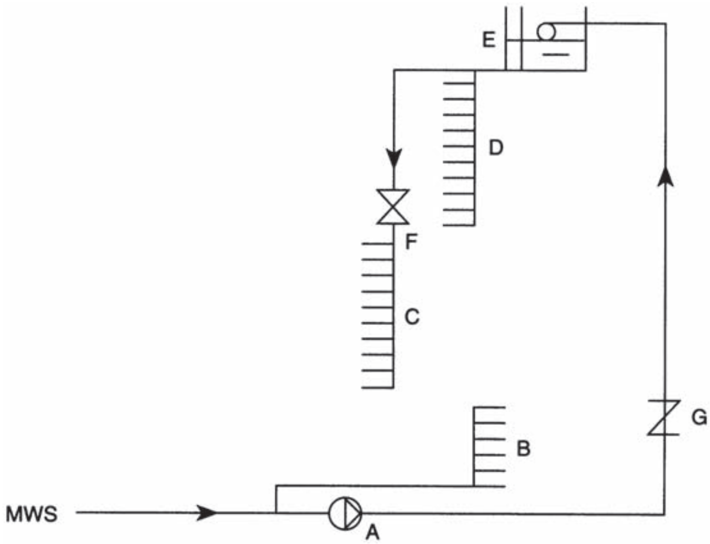


Figure 9.9 Boosted drinking water direct off the MWS, valves omitted: A, booster pump; B, drinking water to lower floors off rising main; C, drinking water to upper floors; D, drinking water to top floors; E, float switch control on pump; F, fixed pressure-reducing valve on drinking water supply to upper floors; G, recoil valve.

water. If 8 h interruption is adequate for offices, the total storage =116000×8/24=38667 litres.

This is equivalent to a mass of 39 t of water! If two thirds is stored at basement level and one third is split between three tanks located at floor 10, floor 20 and the roof (Figure 9.8), the storage arrangements will be:

$$\text{Basement storage} = 38\,667 \times \frac{2}{3} = 25\,800 \text{ litres}$$

$$\text{10th floor, 20th floor and roof storage} = 38\,667 \times \frac{1}{3} \times \frac{1}{3} = 4300 \text{ litres each}$$

If the booster pump must refill these tanks in 4 h the rate of flow handled will be 4300×3/(4×3600)=0.9 l/s.

Pressure developed by the booster pump will be static lift *plus* hydraulic loss *plus* discharge pressure at the index ball valve.

$$\text{Static lift} = 31 \times 3 \times 1000 \times 9.81 = 913 \text{ kPa}$$

Hydraulic loss for 0.9 l/s in copper Table X tube, 42 mm bore is 170 Pa/m and allowing 15% for fittings on straight pipe:

$$\text{hydraulic loss}=31\times3\times1.15\times170=19\text{ kPa}$$

Recommended discharge pressure at the roof tank ball valve=30 kPa

$$\text{Pump pressure}=913+19+30=962\text{ kPa}$$

Estimated net pump duty is 0.9 l/s at 962 kPa.

There are two equations for determining pump power:

$$\text{power}=Mgh=\text{VFR}\times\text{d}p \quad (\text{W}).$$

where M=mass flow rate (kg/s), h=pump head (m), VFR=volume flow rate (m^3/s), and dp=pressure developed by the pump (Pa).

Adopting the second equation:

$$\text{power}=0.0009\times962000=866\text{ W}$$

Taking overall pump efficiency as 50%:

$$\text{Estimated pump power}=\frac{866}{0.5}=1732\text{ W}$$

The drinking water for the offices must now be addressed (see Figure 9.9). Assuming that the rising main can supply the first ten floors, the minimum mains water pressure at ground level will need to be approximately 325 kPa. Do you agree?

The remaining floors must be served by a booster pump.

Normally one drinking water point is allocated to each toilet suite.

One drinking water point will therefore be allocated per floor.

If 10 DU are taken for each drinking water point to the upper floors, total DU=20×10=200, and from Table 7.9 in Chapter 7 this is 0.8 l/s. This is equivalent to 0.8/0.15=5 drinking water points discharging simultaneously out of a total of 20, which we shall take as adequate. Do you agree?

If the booster pump is located in the mains water supply, the pressure developed will be: static lift *minus* lift provided by minimum mains water pressure *plus* hydraulic loss *plus* discharge pressure at the index ball valve:

$$\text{Static lift}=913\text{ kPa}$$

$$\text{static lift provided by mains water pressure}=325\text{ kPa}$$

Hydraulic loss for 0.8 l/s in copper tube Table X and 35 mm bore is 350 Pa/m, and allowing 15% for fittings on straight pipe:

$$\text{hydraulic loss}=31\times3\times1.15\times350=38\text{ kPa}$$

recommended discharge pressure at the roof tank ball valve=30 kPa

$$\text{Pump pressure}=913-325+38+30=656\text{ kPa}$$

Estimated net pump duty is 0.8 l/s at 656 kPa

Estimated pump power is 1050 W

Note the influence that the time to refill the tank has on the duty and power requirement of the booster pump.

Size of DW storage tank=0.8×2×60=96 litres for 2 min storage

9.4 Chapter closure

You now have the skills to design a centralized HWS system requiring secondary pumping where the static head from the high-level storage tank is sufficient to provide static lift but insufficient to overcome the hydraulic losses in the secondary circuits. You are able to employ a pressurization unit for a system of centralized HWS where the cold water supply serving the HWS storage vessel has insufficient or no static head available.

You have been introduced to the special requirements for cold water supplies to high-rise buildings where the mains water service pressure is insufficient to provide the required static lift. The foregoing case studies provide you with the skills to identify the requirements and to design cold water services and drinking water services to multistorey buildings.

It is important to remember to seek advice from the local water undertaking in respect of storage and pumping arrangements before proceeding with the design of water services within buildings.

Loose ends 10

Nomenclature

A	cross-sectional area (m^2)
d	diameter (m)
dh	head loss (m)
dp	pressure drop (Pa)
E	expansion volume (l, m^3)
E	modulus of elasticity (Pa)
F	force (N)
f	frictional coefficient
f	stress (Pa)
g	gravitational acceleration (m^2/s)
I	moment of inertia (cm^4)
k	velocity head loss factor
k_s	absolute roughness (mm)
L	length (m)
LTHW	low-temperature hot water
M	mass flow rate (kg/s)
MWS	mains water service
P	pressure (Pa)
pd	pressure drop
Q	volume flow rate (m^3/s)
q_c	heat transfer by free convection (W/m^2)
q_{cd}	heat transfer by conduction (W/m^2)
q_e	heat transfer by evaporation (W/m^2)
q_r	heat transfer by radiation (W/m^2)
Re	Reynolds number
u	mean velocity (m/s)
V	volume of water (m^3)
VFR	volume flow rate
y	$=d/2$
z	deflection due to expansion (m)
Z_1, Z_2	section or point in a system in which water is flowing
μ	viscosity (kg m/s)
ρ	density (kg/m^3)

10.1 Introduction

With the explosion in the use of software as an aid to design there is a temptation to denigrate the underpinning knowledge that forms the foundation of all vocational disciplines. There are good reasons for relying on dedicated software programs, which have been subject to careful validation for systems design, because there are increasingly many other calls upon one's time, not least on issues of quality control and safety management during the progress of a project: paperwork, in other words! The foregoing chapters in this book unashamedly attempt to instruct you in design by the manual methodology, using tabulated data from the *CIBSE Guide* where appropriate to speed the process, for it is important at least to be able to undertake random checks when using software programs.

Furthermore, software programs are not available for every situation requiring a design process, and it is here if nowhere else that one needs the underpinning knowledge to respond confidently with a design proposal.

In fact you will have noted that many of the design routines identified in the foregoing chapters are not yet to be found in software programs.

This chapter is dedicated to matters that have not been investigated earlier in the text and to those topics that have not yet found a place. This must not be taken to mean that what is set out in this chapter is of lower significance or that it will contain all the other topics that could be included in this book.

The chapter also introduces more underpinning knowledge in fluid flow.

10.2 Water supplies

Case study 10.1

Figure 10.1 is taken from a sketch of water storage needed 45 m above a well. The required supply to the storage tanks is 3 l/s in 50 mm steel pipe reducing to 40 mm connections to the tanks. You are asked to calculate the duty and power requirement of a submersible pump.

SOLUTION

From earlier work on pumps we know that the pressure developed must include: static lift *plus* hydraulic losses in pipe and fittings *plus* discharge pressure from the index ball valve.

Loss in straight pipe may be determined from the D'arcy equation for turbulent flow:

$$dh = \frac{4fLu^2}{2gd}$$

Frictional coefficient f may initially be taken as 0.005.

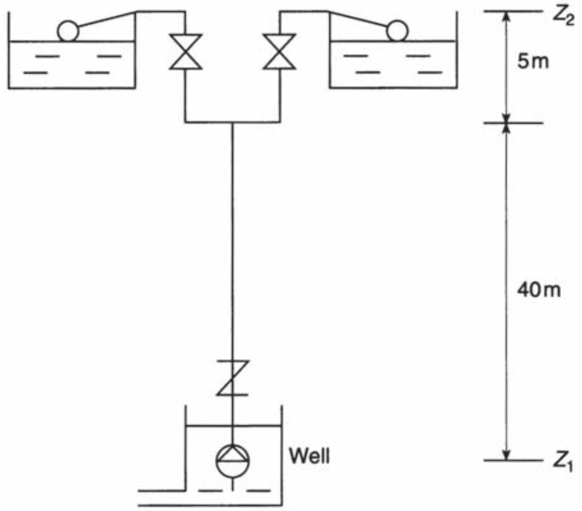


Figure 10.1 Case study 10.1 : high-level storage of water.

Velocity u needs to be determined for 40 and 50 mm pipes from VFR=u×πd^2/4.
For 50 mm pipe velocity is determined as u=1.53 m/s and for 40 mm pipe u=1.19 m/s. Do you agree?

Loss in pipe fittings is obtained from

$$dh = k \times \frac{u^2}{2g}$$

where k is the velocity head loss factor for the fitting. Values of A: are obtained from the *CIBSE Guide* (Greek letter ζ (zeta) is used in the *CIBSE Guide* table).

Here we shall take k=0.4 for bends, k=10 for the recoil valve, k=5.0 for the stop valves and k=0.5 for the tee. The horizontal length of 40 mm index branch is 11 m. The losses are tabulated in Table 10.1.

Table 10.1 Case study 10.1: tabulated hydraulic losses

Item	*Equation*	*Head loss (m)*
50 mm straight pipe	$(4 \times 0.005 \times 40 \times (1.53)^2)/(2 \times g \times 0.05)$	1.91
40 mm straight pipe	$(4 \times 0.005 \times 16 \times (1.19)^2)/(2 \times g \times 0.04)$	0.58
50 mm fittings	$10.5 \times (1.53)^2/2g$	1.25
40 mm fittings	$5.8 \times (1.19)^2/2g$	0.42
Total		**4.16**
Static lift	$Z_1 - Z_2$	45.0
Ball valve	$P/\rho g$	3.06
Total		**52.22**

Net pump duty is 3 l/s at 53 m head or 3 l/s at 520 kPa

Power requirement for the pump=
VFR×*dp*=0.003×520000=1560 W

Taking an overall pump efficiency of 50% the pump power requirement is 3120 W.

A more fundamental approach can be adopted by employing the Bernoulli theorem for frictional flow, for which the total energy in the water at the pump impeller (section Z_1 in Figure 10.1) is equal to the total energy in the water at the index ball valve (section Z_2) plus hydraulic losses. Total energy is made up of: potential energy, which is height Z_2 above some datum; pressure energy $P/\rho g$ metres, which includes static and pump pressure where appropriate; and kinetic energy $u^2/2g$ metres, which is energy due to fluid velocity. Thus

$$Z_1 + \frac{P_1}{\rho g} + \frac{u_1^2}{2g} = Z_2 + \frac{P_2}{\rho g} + \frac{u_2^2}{2g} + \text{losses}$$

Datum is conveniently taken at Z_1. P_1 is taken at the eye of the impeller and is subject to static pressure only, owing to the head of water in the well, and will be disregarded here. Pump pressure at the eye of its impeller is zero for a submersed pump. u_1 and u_2 are water velocities at the impeller and at the point of discharge on the index circuit respectively. As these velocities are similar they will cancel in the formula. The Bernoulli formula thus reduces to

$$Z_1 = Z_2 + \frac{P_2}{\rho g} + \text{losses}$$

The head that the pump must develop therefore

$$= Z_2 - Z_1 + \frac{P_2}{\rho g} + \text{losses}$$

$$= 45 + 3.06 + 4.16$$

$$= 52.22 \text{ m}$$

This represents the static lift *plus* the pressure required at the index ball valve *plus* the losses in straight pipe and fittings accounted for and tabulated above.

You might like to refer back to case studies in Chapter 9 relating to boosted water to compare the difference in approach.

LAMINAR AND TURBULENT FLOW

In the solution to Case study 10.1, reference was made to **turbulent flow**. This is defined as flow in which fluid particles at any cross-section of the pipe or duct are travelling in a random manner; the sum of the random movement, however, is in one direction. Turbulence is due to the frictional resistance offered at the boundary (pipe wall) and to the fluid viscosity μ, which is a measure of the internal friction of the fluid, and is the stress that occurs as one layer of fluid slides over adjacent layers. Fluid viscosity is temperature dependent, and can be likened to comparing the differing effects of pouring cold water and cold engine oil from separate cans.

Laminar flow can be defined as flow in which the fluid particles at any cross-section of the pipe or duct are travelling in an orderly manner in the same direction.

Osborne Reynolds in the 19th century identified a methodology for determining when a fluid was in laminar or turbulent flow by applying what became known as the Reynolds number *(Re)* equation, which is

$$Re = \frac{\rho u d}{\mu}$$

This is one of the many dimensionless numbers used in engineering.

When *Re* is less than 2000, flow is said to be laminar; when *Re* is more than 3500, flow is said to be turbulent. Between the values of 2000 and 3500 flow is said to be in transition and therefore unstable. Have a look at the Moody diagram in section 4 of the *CIBSE Guide*. You can see that the frictional coefficient f for turbulent flow can be determined from the diagram given a knowledge of the Reynolds number *Re* and the relative roughness k_s/d of the pipe. Values of absolute roughness k_s in mm in pipes are listed in Table 10.2.

In Case study 10.1, where the viscosity of cold water is 0.001 501 kgm/s, for the 50 mm pipe

$$Re = \frac{1000 \times 1.53 \times 0.05}{0.001\ 501} = 50\ 966$$

Clearly, flow is fully turbulent and the D'arcy equation can be used. In fact, for most services systems using air, water or steam, flow is invariably in the turbulent region. The relative roughness k_s/d of 50 mm galvanized pipe is 0.15/50=0.003. Given *Re* and k_s/d, from the Moody diagram frictional coefficient f=0.0071. This is substantially different from the value of f taken in the solution to Case study 10.1. However, it does not alter the net pump pressure significantly in this case. Do you agree?

Table 10.2 Absolute roughness k_s in pipes

Material	*k_s (mm)*
Copper pipe	0.0015
Plastic pipe	0.003
Black steel pipe (new)	0.046
Black steel pipe (rusted)	2.5
Galvanized steel pipe	0.15
Cast iron pipe	0.2

Case study 10.2

Cold water flows at 30 kg/s in a 100 mm diameter straight black steel pipe 68 m long. Determine:

(a) the pressure loss due to friction and the specific pressure loss in Pa/m;
(b) the gradient to which the pipe must be laid to maintain a constant water pressure.

SOLUTION

(a) Volume flow rate VFR=u×A=M/ρ (m^3/s), from which mean water velocity $u=M/\rho(4/\pi d^2)$=3.82 m/s.

$$Re = \frac{1000 \times 3.82 \times 0.1}{0.001\ 501} = 254\ 497$$

Clearly, flow is in the fully turbulent region. Relative roughness for black steel pipe k_s/d=0.046/100=0.00046.
From the CIBSE Moody diagram, frictional coefficient f=0.0046.
Substituting in the D'arcy equation:

$$\mathrm{d}h = \frac{4fLu^2}{2gd} = 9.3 \text{ m of water}$$

and therefore

pressure loss=9.3×1000×9.81

=92 kPa over a straight length of 68 m

The specific pressure loss=pd/L=92000/68=1350 Pa/m

Now with access to the CIBSE pipe-sizing tables compare the specific pressure loss for 100 mm galvanized pipe and a flow rate of 30 kg/s. You can also check the water velocity in the tables.

(b) The gradient required to maintain a constant pressure in the pipe is called the **hydraulic gradient,** and can be obtained from h/L. Thus

$$\frac{9.3}{68} = 0.137 \text{ m/m} = 1.37 \text{ m in } 10 \text{ m or } 1 \text{ m in } 7.3 \text{ m}$$

The hydraulic gradient is 1 m in 7.3 m and the increase in static pressure along the length of the pipe (9.3 m) set to this gradient will compensate for the hydraulic loss of 92 kPa, thus maintaining a constant water pressure in the pipe.

Case study 10.3

Two tanks, one vertically above the other as shown in Figure 10.2, are connected together by a 50 mm black steel pipe. Determine the flow rate of cold water from the upper tank to the lower tank. How long will it take to fill the lower tank, given that its dimensions are 4×3×2 m to water level?

SOLUTION

Information required to complete the solution will include velocity head loss factors: *k* for bends=0.3, *k* for stop valves=5.0, hydraulic loss at exit from upper tank=$0.5u^2/2g$ (sudden contraction) and hydraulic loss at entry to lower tank=$u^2/2g$ (sudden enlargement).

Applying the Bernoulli theorem for frictional flow, taking section 1 at the water level in the upper tank and section 2 at the water level in the lower tank:

$$Z_1 + \frac{P_1}{\rho g} + \frac{u_1^2}{2g} = Z_2 + \frac{P_2}{\rho g} + \frac{u_2^2}{2g} + \text{losses}$$

P_1 and P_2 are atmospheric pressure and therefore cancel; u_1 and u_2 are water velocities in the tanks, both of which approach zero and therefore also cancel. Thus rearranging the equation:

$$Z_1 - Z_2 = \text{losses}$$

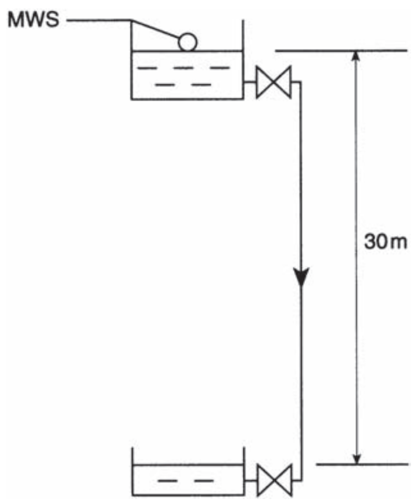


Figure 10.2 Gravity feed to lower tank.

$$Z_1 - Z_2 = \frac{0.5u^2}{2g} + \Sigma k \times \frac{u^2}{2g} + \frac{4fLu^2}{2gd} + \frac{u^2}{2g}$$

The unknown quantity is water velocity u in the down pipe. Taking out the common factors:

$$Z_1 - Z_2 = \left(\frac{u^2}{2g}\right)\left(0.5 + \Sigma k + \frac{4fL}{d} + 1\right)$$

Frictional coefficient f cannot be calculated here, as the Reynolds number requires mean water velocity, which is unknown. Turbulent flow in the down pipe is assumed and frictional coefficient f is taken as 0.005. Substituting values:

$$30 = \left(\frac{u^2}{2g}\right)\left[0.5 + 10.6 + \left(4 \times 0.005 \times \frac{30}{0.05}\right) + 1\right]$$

from which

$$u = 4.94 \text{ m/s}$$

As the pipe is 50 mm diameter the flow rate through it will be

$$= u \times A \text{ m3/s}$$

$$= 4.94 \times \pi(0.05)^2/4 = 9.7 \text{ l/s}$$

Flow rate in down pipe is 9.7 l/s.

Now find access to the CIBSE pipe-sizing tables and check the velocity for 9.7 l/s on 50 mm pipe.

Reynolds number can now be determined:

$$Re = \frac{1000 \times 4.94 \times 0.05}{0.001\,501} = 164\,557$$

Clearly, flow is fully turbulent. Relative roughness:

$$k_s d = 0.046/50$$

$$= 0.0009$$

From the CIBSE Moody diagram, frictional coefficient f is 0.0052, which is close to the estimated value of 0.005. If it was not, a second solution would be done at this point.

$$\text{Time to fill the tank} = \frac{\text{volume}}{\text{flow rate}} = \frac{4 \times 3 \times 2}{0.0097} = 2474 \text{ s} = 42 \text{ min}$$

Case study 10.4

A pipe 50 m long carries cold water at a velocity of 3 m/s. A solenoid valve in the end of the pipe is designed to close in 0.5 s. If the pipe bore is 65 mm, determine the pressure the valve seat must withstand when the valve closes. Take the diameter of the valve orifice as 45 mm.

SOLUTION

Now force = mass × acceleration.

$$\text{The mass of water in the pipe} = \left(\frac{\pi d^2}{4}\right)L\rho \qquad \text{(kg)}$$

$$\frac{\pi \times (0.065)^2}{4} \times 50 \times 1000 = 166 \text{ kg}$$

$$\text{acceleration} = \text{velocity/time} = \frac{3}{0.5} = 6 \text{ m/s}^2$$

$$\text{Force therefore} = 166 \times 6 = 996 \text{ N}$$

$$\text{Pressure on the valve seat, } P = \frac{\text{force}}{\text{area}} = \frac{996}{\pi(0.045)^2/4} = 626 \text{ kPa}$$

The pressure that the valve seat must withstand is 626 kPa.

This is equivalent to six atmospheres! The valve movement must clearly be designed to withstand this pressure.

BOX'S FORMULA

This formula is derived from the D'arcy equation for turbulent flow by implanting an expression for velocity from volume flow rate $Q=u\times A=u\times\pi d^2/4$, from which $u=4Q/\pi d^2$. Thus, substituting:

$$\mathrm{d}h = \left(\frac{4fL}{2gd}\right)\left(\frac{4Q}{\pi d^2}\right)^2$$

Collecting the constants:

$$\mathrm{d}h = \left(\frac{64}{2g\pi^2}\right)\left(\frac{fLQ^2}{d^5}\right)$$

This reduces to

$$\mathrm{d}h = \left(\frac{1}{3}\right)\left(\frac{fLQ^2}{d^5}\right)$$

This formula is used in Chapter 8.

10.3 Linear pipe expansion

A factor that must be accounted for in pipework design associated with the transport of heated water or steam is the effect of pipe expansion. Considerable damage can be sustained by the pipe fittings and equipment to which it is attached and, indeed, on occasion to the structure to which the pipe is fixed if consideration is not given to this matter. A mild steel pipe 30 m long in an LTHW system will expand 27 mm along its axis from fill to operating temperature. This must be recognized and allowance made in the pipe run. The pipe will require anchoring and supporting/guiding to control the expansion and to ensure that fittings and equipment are not subject to excessive stress. There are two ways in which linear pipe expansion may be controlled:

- use of natural changes in direction;
- use of expansion devices

NATURAL CHANGES IN DIRECTION

Figure 10.3 shows three natural changes in direction which allow the absorption of linear pipe expansion. With this approach the bending properties of the pipe are employed to control the effects of pipe expansion. This is the recommended way of dealing with linear pipe expansion.

Anchor points are fixed points in the pipe system and usually consist of

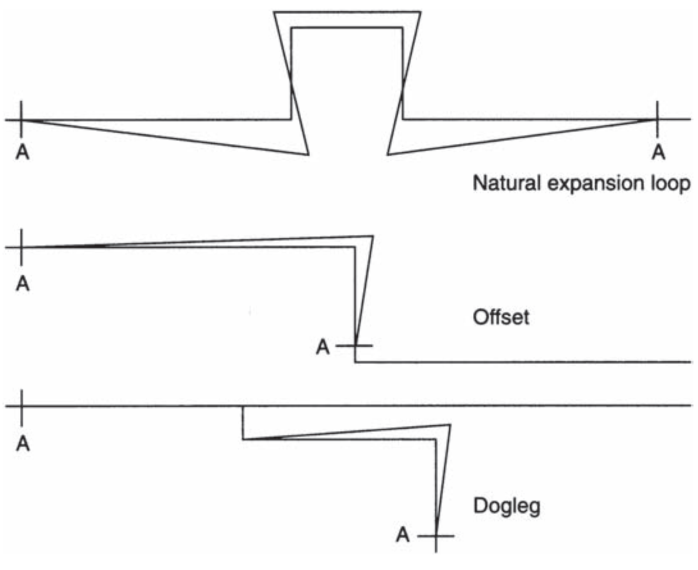


Figure 10.3 Accomodating linear pipe expansion using natural changes in direction: A, anchor point.

welding the pipe to a bearer which is bolted or ragged into a structural element.

You will see from Figure 10.3 that if the pipe *absorbing* the expansion, the offset, for example, is long enough, it too will expand axially, and some of the pipe supports or guides that are required between anchor points must allow lateral pipe movement. This has been exaggerated in Figure 10.3. Vertical pipe movement should be avoided to insure against changes in gradient and allow venting and drainage and avoid the possibility of air locks.

Fifty per cent cold draw is specified to be applied to the loop/offset/dogleg before welding or flanging it into the pipe run. Whether this is carried out in practice depends on the level of site supervision.

EXPANSION DEVICES

Figure 10.4 shows the application of the **axial expansion bellows,** which is the most common expansion device employed in building services when it is not possible to account for pipe expansion by natural changes in direction.

Linear expansion is absorbed by the axial flexibility of the bellows, which are designed to be subjected to compression but not extension. It is therefore important to ensure that the bellows can withstand system operating pressure, which will tend to extend the bellows. The bellows may have to be removed or tied during pressure tests on the pipework.

The pipe guides must be designed to allow axial movement of the pipe only to ensure correct alignment with the axial compensator.

The guide spacing is a function of anchor load. The load on the anchors is the sum of the force resulting from axial pipe movement and coefficient of friction of the guides to pipe movement through them.

Most manufacturers of axial compensators prestress their products before they leave the factory. They are therefore supplied at installation length and must not be subject to extension (cold draw).

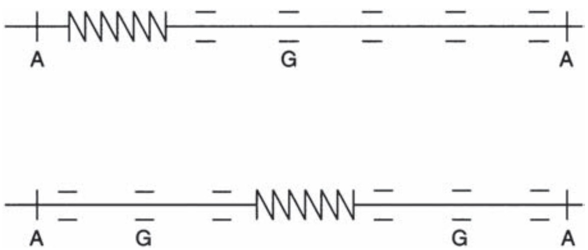


Figure 10.4 Two locations for axial compensators: A, anchor; G, guide.

DETERMINATION OF CRITICAL LENGTHS OF PIPE AND ANCHOR LOADS USING THE OFFSET

From Figure 10.3 it is clear, for example, that there is a minimum length L in the offset for a given amount of linear pipe expansion between the anchor points A-A. This needs to be calculated.

It is also important to have some idea of the force F in newtons or the load in kg that the anchors will sustain during system operation and to a lesser extent the thrust on the bearers of the pipe supports/guides. This information will be required by the structural engineer to design a suitable load bearing structure.

Consider the offset in Figure 10.5. You will note that the axial expansion of offset length L is not considered here. Notation: z= deflection (m), L=length of offset (m), F=force on anchors (N), d= nominal bore of pipe (m), E=Modulus of elasticity (Pa), f=stress (Pa), I=moment of inertia $(\text{cm})^4$, y=distance (m) of the outside of the pipe from the neutral axis=$d/2$.

Figure 10.6 shows the bending moment diagram for the offset length L. When the pipe expands on temperature rise, the offset must bend to take up the movement, as both anchors are fixed points. The deflection is z and at the midpoint of the offset it is **z**/2. Thus

$$\frac{z}{2}=\frac{F(L/2)^3}{3EI}$$

from which the net force sustained by the anchor is given by

$$F=\frac{12EIz}{L^3} \qquad \text{(N)}$$

To this net force must be added the resistance offered by the guides/supports as the pipe moves due to expansion. A coefficient of friction of 0.5 is typical of sliding guides and a coefficient of 0.3 for roller guides.

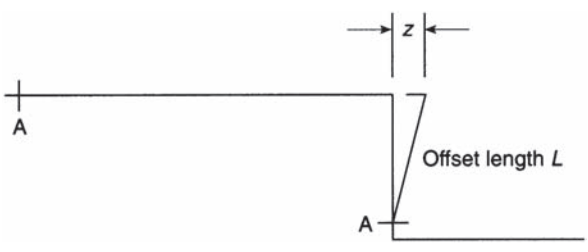


Figure 10.5 Deflection z in offset L caused by pipe expansion between anchors A-A.

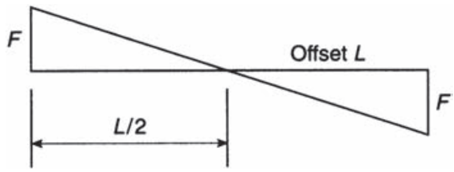


Figure 10.6 Bending moment diagram for offset pipe.

The bending moment in the offset pipe $=F\times L/2$ and $F\times L/2=fI/y$, from which

$$F = 2fI/yL \qquad (N)$$

Putting the force equations together:

$$\frac{2fI}{yL} = \frac{12EIz}{L^3}$$

from which

$$L^2 = 6Ezy/f$$

As $y = d/2$:

$$L^2 = \frac{6Ezd}{2f}$$

For steel pipes, modulus of elasticity is given by

$$E = 200 \times 10^9 \qquad (Pa)$$

and stress by

$$f = 60 \times 10^6 \qquad (Pa)$$

Substituting:

$$L^2 = \frac{6 \times 200 \times 10^9 \times zd}{2 \times 60 \times 10^6}$$

From which the minimum length of offset is given by

$$L = 100(zd)^{0.5} \qquad (m)$$

Note that the offset will be stressed to its maximum if the minimum length is adopted. It may not be appropriate to stress the offset pipe to its maximum and therefore it would be useful to determine the stress f in the offset pipe when it is greater than its minimum length, and from the theory above:

$$\frac{2fI}{yL} = \frac{12EIz}{L^3}$$

from which working stress in the offset pipe is given by

$$f=6Ezd/2L^2 \qquad \text{(Pa)}$$

From the alternative equation for force *F*, the net force sustained by the anchor can also be expressed as:

$$F=\frac{4fI}{dL} \qquad \text{(N)}$$

Do you agree?

Case study 10.5

A 150 mm nominal bore steel pipe extends by 25 mm owing to linear expansion. An offset is to be employed to accommodate the axial movement between the anchor points, which are located as shown in Figure 10.5. Determine the minimum length of the offset and the force on the anchor if the coefficient of friction of the pipe through the supports is 0.3. This is equivalent to the use of rollers.

If the length of the offset is increased by 25% find the force on the anchor and the stress in the offset pipe.

SOLUTION

Minimum length of offset is given by

$$L=100(zd)^{0.5}$$
$$=100(0.025\times0.15)^{0.5}$$
$$=6.12\text{m}$$

Net force on the anchor is given by

$$F=\frac{12EIz}{L^3}=\frac{12\times200\times10^9\times862\times0.025}{(6.12)^3\times10^8}$$
$$=2252 \text{ N}$$

For a coefficient of friction of 0.3 an equivalent of 30% of the force will be resisted by the pipe support bearers. The resistance from the pipe supports=2252×0.3=675 N.
Thus the force on the anchor is given by

$$2252+675=2927 \text{ N}$$

Note that moment of inertia I is tabulated in cm^4 and must be converted to m^4: hence the term 10^8 in the equation. The moments of inertia for pipes of different materials are given in *CIBSE Guide* section B. Some values are given in Table 10.3 for heavy-grade steel pipe

If the length of the offset is increased by 25%:

$$L=6.12\times1.25=7.65\text{m}$$

$$\text{Force } F=\frac{12\times200\times10^9\times862\times0.025}{(7.65)^3\times10^8}$$

from which F=1153 N and force transmitted to the anchor =1153+1153×0.3=1500 N.

The stress in the offset pipe is given by

$$f=6Ezd/2L^2=\frac{6\times200\times10^9\times0.025\times0.15}{2(7.65)^2}$$

from which

$$f=38.4\times10^6\text{ Pa}$$

Note that increasing the minimum length of the offset pipe by 25% has had the effect of reducing the stress in the offset from f=60×10^6 Pa. Summarizing the solution:

Offset pipe length	Anchor force	Stress in offset pipe
6.12m minimum	2927 N	60×10^6 Pa
7.65 m (increase of 25%)	1500 N	38.4×10^6 Pa

Note: A factor of safety must be applied to the calculated anchor force.

DETERMINATION OF MINIMUM LENGTHS OF FLEXIBLE PIPE FOR ABSORBING AXIAL PIPE MOVEMENT IN EXPANSION LOOPS AND DOUBLE SETS

See Figure 10.7. For site-made expansion loops, the determination of length L is the same as for the offset. The value of L is then halved.

Table 10.3 Moments of inertia for heavy-grade steel pipe

Pipe diameter (mm)	*moment of inertia, I* cm^4
50	30.8
65	64.5
80	114
100	272
150	862
200	1880
250	4745

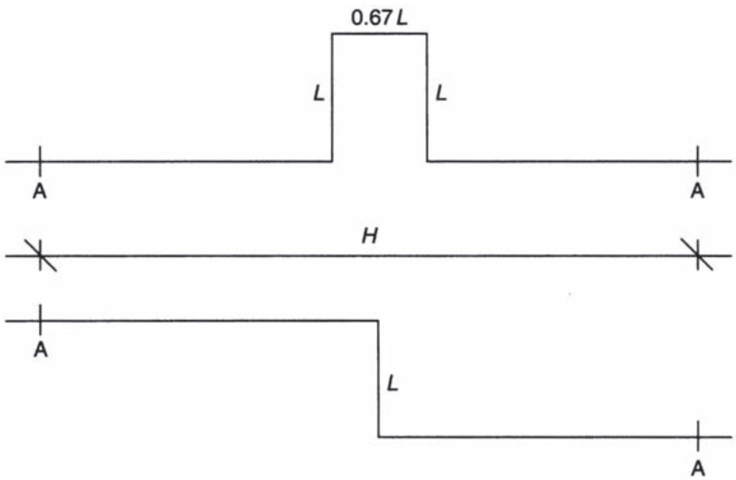


Figure 10.7 Natural expansion loops and double sets: guides and supports omitted.

For site-made double pipe sets, the determination of length *L* also depends upon the distance between anchor points *H* and hence the ratio *H/L*. *L* is first of all determined using the formulae for offsets and then multiplied by a factor that depends upon this ratio:

Ratio	3/1	4/1	5/1	6/1
Multiplying factor	0.7	0.5	0.4	0.3

Case study 10.6

A 65 mm steel pipe measures 40 m between anchors and is subject to a temperature rise of 130 K. Axial expansion can be accommodated by employing a site-made double set. Determine the length of the offset, *L*.

SOLUTION

The coefficient of linear expansion of steel is 0.000012 m/mK. The amount of expansion=0.000012×40×130=0.0624 m. As $L=100(zd)^{0.5}$, by substitution:

$$L=100(0.0624\times0.065)^{0.5}=6.4 \text{ m}$$

The ratio $H/L=40/6.4=6.25$. If a multiplying factor of 0.3 is adopted the offset will be:

$$0.3 \times 6.4 = 1.92 \text{m}$$

Minimum length L for the 65 mm pipe having a double set will be 2 m.

Note:

1. This is based upon a stress $f = 60 \times 10^6$ Pa.
2. If the axial expansion was accommodated by employing a site made expansion loop, the sides of the loop would be $L = 6.4 \times 0.5 = 3.2$ m, and the top of the loop would be $0.67 \times 3.2 = 2.2$ m.

THE DETERMINATION OF ANCHOR LOADS WHEN USING AXIAL COMPENSATORS

When it is not possible to accomodate linear pipe expansion by natural changes in direction, the use of axial compensators is usually considered.

It may be necessary here to undertake two calculations: determination of the load sustained by the anchor under normal operating conditions, and determination of the load sustained under test conditions. The factors accounted for under working temperature and pressure conditions are as follows.

Force F_1 due to internal pressure on the bellows is given by

$$F_1 = \text{working pressure} \times \text{effective bellows area}$$

Force F_2 to compress the compensator is given by

$$F_2 = \text{force per mm} \times \text{mm of expansion}$$

Force F_3 to overcome the friction of the pipe through the pipe guides for sliding guides is taken as 30 N/m of pipeline for each 25 mm of pipe diameter. For hangers or roller guides it is taken as 15 N/m for each 25 mm of pipe diameter.

Force F_4 is specific to test pressure conditions if the compensator can withstand system test pressure. If it cannot, it must be removed, and a distance piece fitted in its place during testing.

$$F_4 = \text{test pressure} \times \text{effective bellows area}$$

Case study 10.7

30 m of 100 mm nominal bore steel pipe expands axially over a temperature rise of 10 to 85 °C. The expansion is to be absorbed by an axial compensator with suitable sliding guides located as necessary along the main. The working pressure is 5 bar gauge. From the data determine the total load on the anchor:

(a) under working conditions;

(b) under test conditions, given that these are twice working pressure.

DATA

Coefficient of linear expansion for steel is 0.000012 m/mK.
Force to compress bellows is 250 N/mm.
Effective bellows area is 118 cm^2.
Total movement of bellows is 50 mm.
Working pressure for bellows 16 bar gauge.

The last four items of data are obtained from N.Minikin & Sons literature for 100 mm Emflex flanged axial expansion joints for use on steel pipe.

SOLUTION

Linear expansion is given by

$$30\times0.000012\times(85-10)=0.027\ \text{m}=27\ \text{mm}$$

This is well within the total movement of 50 mm so one compensator will suffice, fitted as shown in Figure 10.4.

(a) $F_1 = 500 \times 118 \times 10^{-4} = 5.9\ \text{kN}$

$$F_2 = \frac{250}{1000} \times 27 = 6.75\ \text{kN}$$

$$F_3 = 30 \times 30 \times \frac{100}{25} \times \frac{1}{1000} = 3.6\ \text{kN}$$

Total force is given by

F=16.25kN

This is equivalent to 16250/9.81=1656 kg.

Load on the anchor during operating conditions is 1656 kg

(b) Under test pressure

F_4=10 bar×100×118×10^{-4}=11.8 kN

Load on the anchor during pressure tests is 1203 kg
The maximum load on the anchor occurs during operating conditions. The compensator can therefore be left in place during pressure tests.

10.4 Electrothermal storage

he concept omf cooling or heating a medium during unoccupied periods, and when cheaper electrical energy for example might be available, is not an uncommon practice. The stored heating or cooling energy is then distributed around the building as required during the periods of occupancy.

Electrode boilers are particularly suited to generating hot water from offpeak electricity, and those privatized utilities that generate electricity are usually keen to take advantage of improving their base load.

Electric current is passed between electrodes directly through the water in the boiler. The electrical resistance in the water thus generates heat, resulting in a rise in water temperature. Control on temperature is obtained by automatically raising and lowering an insulating shield placed around one of the electrodes.

Figure 10.7 shows the plant arrangement. The thermal storage vessel is fitted with sparge pipes on each of the loading and unloading flow and return connections. These run the full length of the vessel to promote temperature stratification during the loading period at offpeak and the unloading period during occupation of the building. At the beginning of the loading period most of the water in the vessel will be at a common low temperature, and at the end of the loading period the stored water will have reached a common final storage temperature. There should not be any convection movement of the stored water in the vessel at any time.

The heating circuits serving the building take off as much or as little of the stored high-temperature water as required.

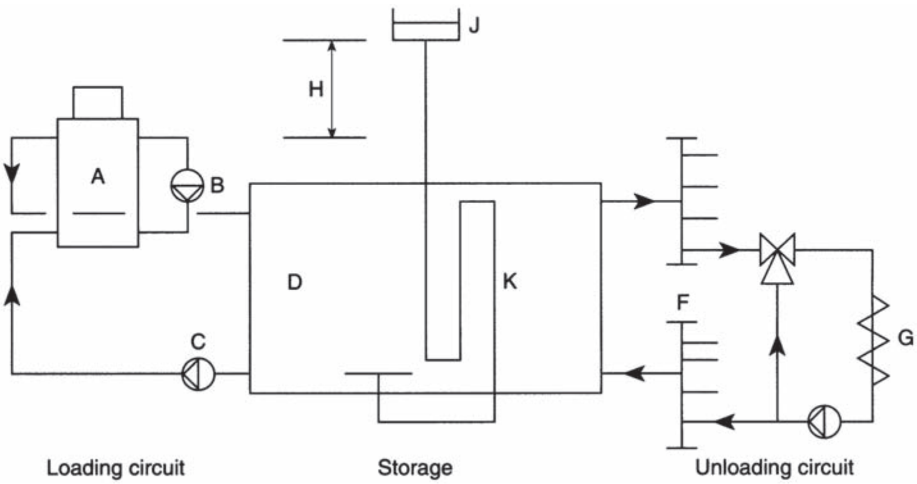


Figure 10.8 Electrothermal storage plant and connections: A, electrode boiler; B, boiler circulating pump; C, primary pump on loading circuit; D, thermal storage vessel; F, flow and return headers on the secondary unloading circuit; G, typical CVVT control to, say, a zone of radiators; H, static head on system; J, feed and expansion tank, MWS omitted; K, antisiphon feed and expansion pipe.

Case study 10.8

A system of electrothermal storage is proposed for a building. From the data below determine the generator output, storage and electrical power requirements and the expansion volume of water stored in the F&E tank at the end of the generating period.

DATA

Plant located in the basement of a 10-storey building with 2.8 m floor heights.
Gross heat requirement is 110 kW
Idling losses are 12 kW
Occupancy is between the hours of 09.00 and 17.00 hours.
Storage temperature rise is 30 K.
Antiflash margin is 15 K.
Electricity supply is from a 3.3 kV three-phase supply.
Offpeak period is from 23.00 to 06.00 hours.

SOLUTION

The output of an electrode boiler based upon the gross heat requirement will be

$$\frac{(110 \times 8) + (12 \times 16)}{7} = \frac{1072}{7} = 153 \text{ kW}$$

The storage capacity of the vessel will be

$$\text{mass of water} = \frac{[1072 - (12 \times 7)] \times 3600}{4.2 \times 30} = 28\ 229 \text{ kg}$$

This needs converting to a volume of water at the storage temperature at the end of the charge period in order to find the volume of the vessel.

Available static pressure, taking a point at high level in the basement of the building where it is assumed the plant will be located, will be $10 \times 2.8 = 28$m. This is equivalent to a static pressure of

$$P = 28 \times 1000 \times 9.81 = 274680 \text{ Pa gauge}$$

Absolute pressure will be

$$274680 + 101325 = 376005 \text{ Pa} = 3.76 \text{ bar absolute}$$

From the steam tables, saturation temperature is given by

$$t_s = 142°C$$
$$\text{Antiflash margin} = 15\ K$$
$$\text{Storage temperature} = 127\ °C$$

This is the temperature that the water in the vessel must reach at the end of the charge period. From the steam tables the specific volume of water at 127 °C, v_f=0.001065 m3/kg. Thus the volume of the vessel will be

$$V = 28229 \times 0.001065 = 30\ m^3$$

As you can see, this is a substantial volume, for which provision must be made at an early stage in the project.

$$\text{Electrical power requirement} = \frac{153 \times 1000}{3300 \times (3)^{0.5}} = 27\ A$$

$$\text{Expansion volume } E = \frac{V(\rho_1 - \rho_2)}{\rho^2} = \frac{30(962 - 939)}{939} = 0.735\ m^3$$

The water densities (obtained from steam tables) are taken at the storage temperature at the end of the charge period, which is 127 °C, and the storage temperature at the beginning of the charge period, which is 30 K lower at design conditions.

SUMMARY

Generator output is 153 kW; storage temperature is 127 °C; storage volume is 30 m^3; electrical power requirement is 27 A, which would be a dedicated supply taken from the live side of the electricity undertaking’s transformer; and the expansion volume of water being transferred from the storage vessel to the feed and expansion tank during the charge period is 0.735 m^3. The antisiphon pipe must hold at least this quantity of water to avoid hot water at 127 °C reaching the F&E tank.

10.5 Heating an indoor swimming pool

If an indoor swimming pool is to be heated, the dry-bulb temperature of the pool hall should be in excess of the pool temperature to inhibit excessive condensation occurrence on the inner surfaces of the pool hall. When the pool is up to temperature and uncovered there will be a latent heat loss from the pool and corresponding latent heat gain to the pool hall. This requires the hall to be mechanically ventilated to insure against any surface condensation. The latent heat should be reclaimed in the air handling plant by raising the temperature of the incoming fresh air, in the interests of energy conservation.

Case study 10.9

From the given data relating to an indoor swimming pool determine:

(a) net heat loss from the pool surface;
(b) heat loss from the tank;
(c) boiler output without pool cover;
(d) boiler output with pool cover in place;
(e) latent heat gain to the pool hall during occupancy;
(f) daily design energy requirement (DDE) with and without pool cover in place;
(g) heat-up time for pool water.

DATA

Pool size 15×7×1.7 m mean depth
Water temperature, t_w 25 °C
Hall air temperature, t_a 28 °Cdb
Relative humidity 70%
Air velocity over pool surface, u 0.5 m/s
Emissivity of water, e 0.96
Mean U value for pool tank 3.0 W/m^{2}K
Ground temperature 10 °C
Mean radiant temperature in pool hall, t_r 26 °C
Absolute temperature, T K

SOLUTION

The total heat loss from an open water surface is given by

$$\Sigma q = q_e + q_c + q_r + q_{cd} \qquad (\mathrm{W/m^2})$$

From the *CIBSE Guide*:

$$q_e = (91.5 + 77.6u)(P_s - P_{wv})$$

$$q_c = 3.18(u)^{0.8}(t_w - t_a)$$

$$q_r = 5.67 \times 10^{-8} \times e(T_w^4 - T_r^4)$$

$$q_{cd} = U(t_w - t_a)$$

where T_w and T_r are the absolute values of water and mean radiant temperature, t_w and t_r.
From hygrometric data in the *CIBSE Guide* the partial pressure of the water vapour P_{wv} and the saturation vapour pressure P_s at 28 °Cdb and 70% RH are 2.675 and 3.779 kPa respectively.

Substituting the data into the equations we have:

$$q_e=+144\ \text{W/m}^2$$

$$q_c=-5.48\ \text{W/m}^2,$$

$$q_r=-5.79\ \text{W/m}^2$$

$$q_{cd}=+45\ \text{W/m}^2$$

Do you agree with these solutions?

(a) The net heat loss from the pool surface
=(144-5.48-5.79)×15×7=13.94 kW.

(b) The heat loss from the tank=45[(44×1.7)+(15×7)]=8.1 kW.
Net heat loss from pool surface and pool tank=13.94+8.1=22 kW.

(c) Boiler output=22 kW without the pool cover.
If a pool cover is used for 12 h each day, qe and qc are cancelled during that period, leaving a net heat gain to the pool of 5.79×(15×7)=608 W. So the net heat loss from the pool and tank

=8100-608=7492 W for 12 h.

(d) Boiler output=7.5 kW with the pool cover in place.

(e) The latent heat gain to the pool hall during use will be

144×(15×7)=15kW.

This must be taken out of the pool hall and reclaimed by the airhandling plant.

(f) DDE=(12×22)+(12×7.5)=354 kWh with the pool cover in place

=(24×22)=528 kWh without the pool cover

The mass of pool water

=15×7×1.7×1000=178500 kg

If a boiler having an output of 22 kW is installed, the net boiler power available for heating the water if the pool cover is used during the heat up period

$$= 22 - 7.5 = 14.5 \text{ kW}$$

Since

$$\text{Boiler power} = \frac{\text{mass} \times \text{specific heat} \times dt}{\text{time}}$$

for a 1 ° rise,

$$\text{time} = (178\ 500 \times 4.2 \times 1)/14.5 = 51\ 703 \text{ s} = 14.4 \text{ h}$$

If initial pool water temperature is 10 °C, a 15 ° rise to 25 °C will therefore take 216 h.

(g) The estimated time to heat the pool to 25 °C will be 9 days. Do you agree?
Note: In practice, if the tank is sunk in the surrounding ground, the mass of earth warms up in time to that of the pool water and then acts as a thermal stabilizer, so eventually boiler output in solution (d) is reduced.

10.6 Chapter closure

You have now continued to develop skills in the design of water systems from first principles, which you can apply to similar situations covered here. This should give you confidence to investigate applications that differ from those encountered in the text. You can now analyse the pressure effects of rapid valve closure in water distribution. The introduction to accounting for linear pipe expansion in systems subject to temperature rise provides you with the skills for determining critical pipe lengths and anchor loads to ensure satisfactory system operation.

You are able to determine loads and capacities of plant associated with electrothermal storage and pool water heating.

Sources of further information

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- Section A4: Air infiltration and natural ventilation (1986)
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