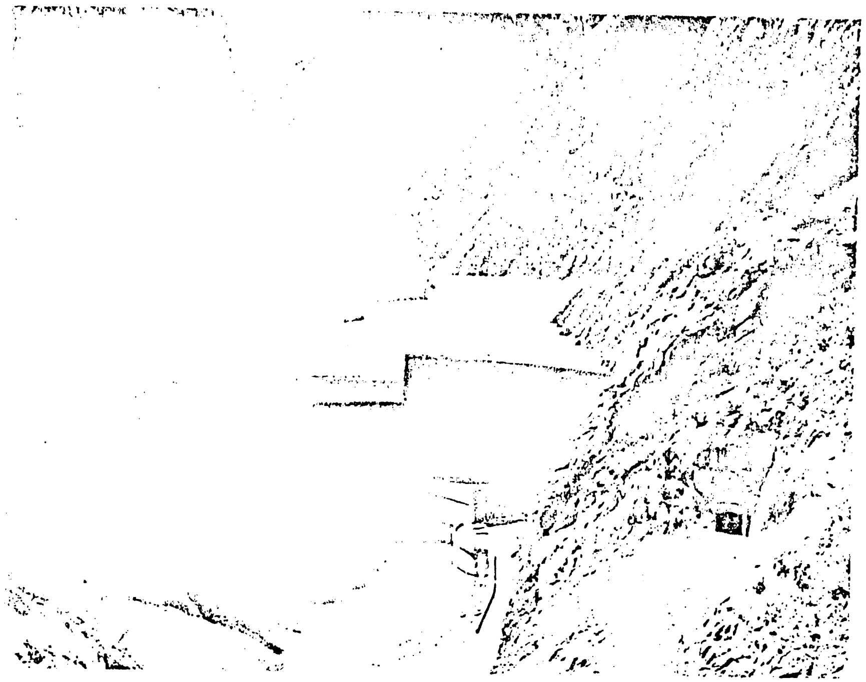
HYDROPOWER ENGINEERING



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HYDROPOWER ENGINEERING

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CONTENTS

PREFACE

Recent escalation of the cost of energy has resulted in a reawakened interest i hydroelectric power as a source of electrical energy. The economic situation no favors smaller hydropower projects, and special equipment is being develope Techniques for making the new low head hydraulic turbines and related equi ment practical and economically viable have introduced new facets to hydropow engineering.

Another influence on the planning and design for hydropower developmen has been the increased public demand to assess the social and environmental impa of the construction and operation of hydropower plants. Techniques for evaluatii those impacts are needed so that an authoritative display and weighing can be o fered to both the decision makers and the public.

This book is an up-to-date textbook on the subject of hydropower enginee ing for use in university courses in hydroelectric technology. It is written primari for persons concerned with planning hydropower projects and for those doing fea bility studies and preliminary design of projects. In addition to the engineerii aspects of hydropower development, this book discusses economic analysis ar environmental considerations. The book should be useful to developers and t people who are required to make decisions as to the acceptability of projects. It w serve as a useful reference to consulting engineers and to engineers and planne serving in governmental agencies.

Throughout the book, detailed examples are given to aid in understanding tl various technical approaches commonly used in making a feasibility determinatic for a hydropower project.

Fundamentals of hydraulics and hydrology are used to present the basic ›ry necessary to understand hydropower engineering. Recent advances in hy- logic analysis for hydropower developed by the author and co-workers are intro- ed. An up-to-date treatment is presented of turbine constants and their use in cting and sizing hydropower facilities. Specific procedures are given for selecting optimum size of units to be used in various situations.

Brief technical information is presented on cavitation problems, design for ›ine settings, analysis for design of water passages for turbine installations, and trical considerations. The intent is to provide enough practical information so engineer can proceed with a feasibility analysis of proposed hydropower devel- ıents. A chapter on speed control and pressure control is presented to familiarize engineer and developer with complex problems that will need more detailed sideration.

A comprehensive treatment is presented on economic analysis for hydro- ’er projects, including the fundamental equations and necessary techniques for :ing a preliminary economic assessment. References to the most recent cost- mating information are included.

One chapter discusses microhydro and minihydro power developments and s useful nomographs for selection of suitable units for very small hydropower elopments. Another chapter treats pumped/storage hydropower developments, uding information on various types of pump/turbines, their selection, and appli- on limitations. Use of pumped/storage units promises to be a challenging engi- ring planning and design realm to meet the demand for peaking power.

The book concludes with a chapter on social, political, and environmental ysis for hydropower developments. Methodologies are presented for making the ssary evaluations to comply with legal requirements. The procedures presented a need for an authoritative technique to help in the social and environmental :cts of planning for potential projects. Pertinent U.S. federal laws are listed, and roaches to state and local problems are outlined.

The appendix provides a listing of manufacturers of hydraulic turbines and resses of principal U.S. federal agencies concerned with hydropower.

The author acknowledges the outstanding cooperation of the many manufac- rs and consulting firms that have provided information for this book, the privi- extended by the University of Idaho to work on the book during a sabbatical e, and the help and inspiration of my students and short-course participants. A ere thanks is expressed to my collaborators, to my wife who provided so much ›, and to Bonnie Milligan for her careful work in typing the manuscript.

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1

INTRODUCTION

Hydropower engineering refers to the technology involved in converting the pressure energy and kinetic energy of water into more easily used electrical energy. The prime mover in the case of hydropower is a water wheel or hydraulic turbine which transforms the energy of the water into mechanical energy. Earlier in the history of energy development and use, water wheels provided power by direct connection or with pulley and gear systems to drive various machines, such as grist mills and textile mills. Since ancient times, water wheels have been used for lifting water from a lower to a higher elevation in irrigation systems. An interesting discussion of early water wheels is contained in Reynolds (1970).

This text treats the technology and developmental considerations that have evolved since the introduction of two well-known types of turbines: the Francis, or reaction-type, turbine and the Pelton, or impulse-type, turbine. These two early types of turbines were developed to a considerable sophistication before 1900 and were important in helping to make the electric generator successful and practical. Later, propeller turbines were developed. Explanation will be given later of the various types of turbines and their origins.

PRESENT AND FUTURE DEVELOPMENT

The existing development of hydropower has resulted from a fairly uniform rate of increase in competition with other modes of electrical energy production. Until the mid-1970s, the pattern of hydro development was to develop bigger and bigger units because smaller hydro plants were not competitive with fossil fuel power plants.

Recently, with rising costs of fossil fuels, the economic feasibility of small-scale hydro has changed. During the period from 1940 to 1970, small units were actually forced out of production because of the high cost of operation and the ready availability of electrical energy from large steam power plants and large high-capacity hydro plants. That situation having changed, small-scale hydropower development is becoming an attractive energy production alternative. The importance of hydropower development in the United States compared to other sources of energy is shown in Table 1.1.

Coal was the most important source of energy in the United States in 1940, but petroleum liquids and gas surpassed coal by 1960. The projection of data into the future envisions that coal will again be the largest source of energy by the year 2000, with marked decreases in the importance of petroleum liquids and gas. Nuclear and "other" sources of energy are expected to increase in importance. Through all these changes, hydropower retained and is expected to retain approximately the same relative importance—about 4% of the total energy supplies in the United States.

Figure 1.1 compares the development of and theoretical potential for hydropower in various regions of the world. Table 1.2 presents information from the U.S. Army Corps of Engineers (1979) on the developed hydropower in each state of the United States and also the potential that exists for further development.

It is interesting to note from these tables and the chart that there is still considerable potential left for developing hydroelectric energy in most parts of the world. The summary by states in Table 1.2 indicates that the Pacific Northwest and

TABLE 1.1 Comparison of Sources of Energy in the United States, Historical and Projected[a]

Date	Petroleum	Gas	Coal	Hydro	Nuclear	Other	Total
			Quadrillion Btu				
1940	7.5	2.7	12.5	0.9	–	1.4	25.0
1950	13.5	6.2	12.9	1.4	–	1.2	35.2
1960	20.1	12.7	10.1	1.7	–	1.0	45.6
1970	29.5	22.0	12.7	2.7	0.2	1.0	68.1
1977	37.4	20.1	13.0	2.7	2.7	1.8	77.7
1985	39.0	17.0	19.0	3.0	6.0	2.7	86.7
2000	28.0	14.0	32.0	3.5	11.0	6.5	95.0
			Percentages				
1940	30.0	11.0	50.0	4.0	–	6.0	
1950	38.0	18.0	37.0	4.0	–	3.0	
1960	44.0	28.0	22.0	4.0	–	2.0	
1970	43.0	32.0	19.0	4.0	–	1.0	
1977	48.0	26.0	17.0	3.0	3.0	2.0	
1985	45.0	20.0	22.0	3.0	7.0	3.0	
2000	29.0	15.0	34.0	4.0	12.0	7.0	

[a]Data for 1985 and 2000 are projections.
SOURCE: Modified from Hayes (1979).

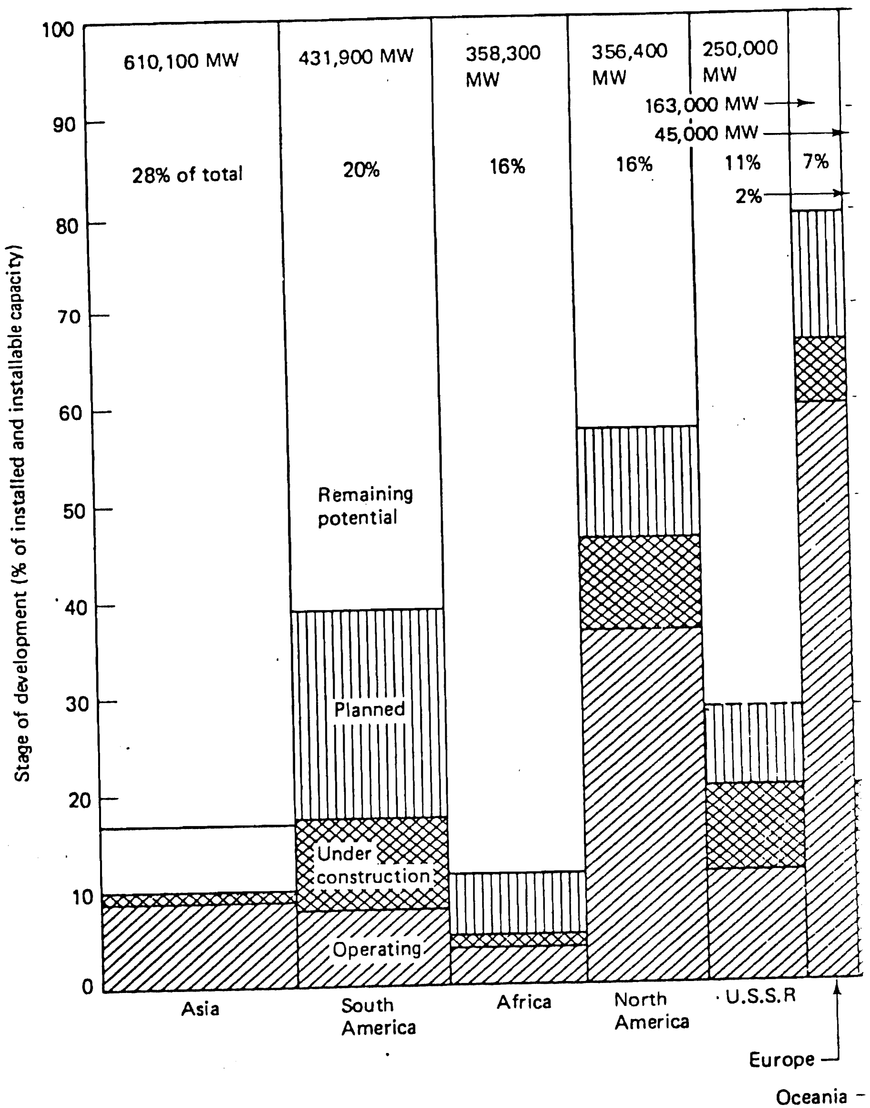

Figure 1.1 World hydropower, developed and potential, by regions. SOURCE: Armstrong (1978).

TABLE 1.2 Developed and Potential Hydropower by State and Region of the United States

	Capacity (MW)		
State and Region	Developed[a]	Incremental[b]	Undeveloped[c]
Pacific Northwest			
Alaska	129	418	166,775
Idaho	2,448	5,172	40,536
Oregon	6,853	14,190	37,453
Washington	17,374	13,482	22,716
Regional Total	26,804	33,262	267,480
Pacific Southwest			
Arizona	1,406	156	13
California	7,636	5,447	13,053
Hawaii	19	31	30
Nevada	677	46	74
Utah	190	148	4,014
Regional Total	9,928	6,028	17,184
Mid-Continent			
Colorado	401	1,593	7,072
Kansas	2	220	480
Montana	2,418	2,332	15,313
Nebraska	136	94	112
New Mexico	24	786	404
North Dakota	430	324	10
Oklahoma	1,029	1,630	1,019
South Dakota	1,500	420	37
Texas	321	372	1,875
Wyoming	227	487	3,546
Regional Total	6,488	7,758	29,868
Lake Central			
Illinois	132	730	259
Indiana	28	96	444
Iowa	135	1,117	257
Kentucky	636	9,271	4,036
Michigan	486	1,133	0
Minnesota	158	989	1,027
Missouri	598	1,368	1,249
Ohio	0	314	90
Wisconsin	429	812	437
Regional Total	2,602	15,830	7,789

TABLE 1.2 (continued)

	Capacity (MW)		
State and Region	Developed[a]	Incremental[b]	Undeveloped[c]
Southeast			
Alabama	2,271	4,121	581
Arkansas	1,080	2,886	6,235
Florida	30	45	30
Georgia	2,050	406	2,060
Louisiana	81	291	2,353
Mississippi	0	133	119
North Carolina	1,937	653	1,553
Puerto Rico	64	92	13
South Carolina	1,532	628	1,175
Tennessee	2,096	3,269	7,261
Virginia	686	497	1,777
Regional Total	11,827	13,021	23,160
Northeast			
Connecticut	103	88	NA
Delaware	0	0	2
Maine	354	369	NA
Maryland	476	532	252
Massachusetts	237	115	NA
New Hampshire	386	261	NA
New Jersey	6	40	647
New York	3,741	12,458	3,127
Pennsylvania	403	1,731	3,245
Rhode Island	2	40	NA
Vermont	197	134	NA
West Virginia	148	2,969	1,184
Regional Total	6,053	18,737	8,457
National Total	63,702	94,636	353,938

[a]Developed = Existing generating capacity
[b]Incremental = Potential for increased capacity at existing plants and dams
[c]Undeveloped = Potential at undeveloped sites
SOURCE: U.S. Army Corps of Engineers (1979).

the New England states have significant potential for development. In 1979 the Federal Energy Regulatory Commission reported that the installed hydroelectric capacity of the United States is 61,000 MW (megawatts), which is estimated to be 36% of the national potential capacity of 176,000 MW.

None of these studies considered the potential for pumped/storage hydropower development. This technique is really an energy-storing system. Water is pumped from a lower reservoir to a higher one using inexpensive "dump" energy produced during periods of low demand by power plants which cannot economically be shut down. The water is then run back down through turbines to produce more valuable power needed during periods of peak demand. The use of pumped/storage for hydropower production is in its infancy in the United States. Barring some now unheard of energy production mode for producing short-term peaking power, pumped/storage appears to have much importance for the future. Chapter 13 presents the theory and engineering aspects of pumped/storage hydropower.

Although the relative percentage of electrical energy produced by hydropower has not increased during the last forty years, the need for additional energy production, the significant local benefits, and the fact that hydropower is a renewable energy source that appreciates with time make hydropower important for future use and development.

The challenge to the engineer is and will be to plan and design economically feasible developments to meet the needs of the future and at the same time protect and preserve the quality of our environment.

TYPES OF DEVELOPMENTS

In studying the subject of hydropower engineering, it is important to understand the different types of development. The following classification system is used in this text:

Run-of-river developments. A dam with a short penstock (supply pipe) directs the water to the turbines, using the natural flow of the river with very little alteration to the terrain stream channel at the site and little impoundment of the water.

Diversion and canal developments. The water is diverted from the natural channel into a canal or a long penstock, thus changing the flow of the water in the stream for a considerable distance.

Storage regulation developments. An extensive impoundment at the power plant or at reservoirs upstream of the power plant permits changing the flow of the river by storing water during high-flow periods to augment the water available during the low-flow periods, thus supplying the demand for energy in a more efficient manner. The word *storage* is used for long-time impounding of water to meet the

seasonal fluctuation in water availability and the fluctuations in energy demand, while the word *pondage* refers to short-time (daily) impounding of water to meet the short-time changes of energy demand.

Pumped/storage developments. Water is pumped from a lower reservoir to a higher reservoir using inexpensive dump power during periods of low energy demand. The water is then run down through the turbines to produce power to meet peak demands.

Tidal power developments. In some estuaries, tidal power can be economically harnessed to develop electric energy. These developments use the water flowing back and forth as a result of tidal action and the fact that there is a significant difference in elevation of the water surface in the estuary from one stage of tide to another.

Single-purpose developments. The water is used only for the purpose of producing electricity.

Multipurpose developments. Hydropower production is just one of many purposes for which the water resources are used. Other uses might include, for example, irrigation, flood control, navigation, municipal, and industrial water supply.

Another way of classifying hydropower development is with respect to the manner in which the hydropower plant is used to meet the demand for electrical power.

Base-load developments. When the energy from a hydropower plant is used to meet all or part of the sustained and essentially constant portion of the electrical load or firm power requirements, it is called a *base-load plant.* Energy available essentially at all times is referred to as *firm power.*

Peak-load developments. Peak demands for electric power occur daily, weekly, and seasonally. Plants in which the electrical production capacity is relatively high and the volume of water discharged through the units can be changed readily are used to meet peak demands. Storage or pondage of the water supply is necessary.

Hydropower plants can be started and stopped more rapidly and economically than fossil fuel and nuclear plants, so the use of hydropower plants to meet peak loads is particularly advantageous. The large hydropower plants of the Pacific Northwest rivers were originally base-load plants but are being used more and more for peaking power as large fossil fuel and nuclear power plants become operative in the region to supply the base load.

Hydropower development has a significant role to play in the future energy oduction of the United States and most countries of the world, because rising sts of alternative energy sources are making small-scale hydropower developments onomically competitive and because hydropower plants are well suited to provide aking power.

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PROBLEMS

1.1. Develop a table or graph showing the relative importance of hydropower compared to electrical energy produced by other means for your area or country.

1.2. Characterize two different types of hydropower developments in your area. Present physical characteristics that demonstrate the type of development.

1.3. Develop a list of sources for finding information on hydropower production, relative importance of hydropower in the energy production realm, and projections for potential development.

TERMINOLOGY AND TYPES OF TURBINES 2

TERMINOLOGY

Important in understanding the fundamental principles and concepts of hydropower engineering is the development of good word definitions that can be extended into visual and mathematical expressions. Such words as *work, energy, power, demand, load, head,* and *discharge* have special meaning in the language of those working in the field of hydropower engineering.

Work is transferred energy and is the product of force times the distance moved.

Energy is the capacity to do work. Water, by its very nature of being a fluid that moves easily by action of gravity, has energy. The work done by water in producing electrical energy is usually measured in kilowatt-hours (kWh). The energy from water can be either potential energy by virtue of position, pressure energy due to the water pressure, or kinetic energy by virtue of the water's moving force or action. Later mathematical expressions and graphical presentations will verify this statement.

Power is the rate of transferring energy or work per unit of time. It is calculated as force times distance divided by time. In hydropower language it is measured in kilowatts (kW) and is also expressed in horsepower (hp) units. Power capacity is often used in referring to the rated capability of the hydro plant to produce energy. Manufacturers of hydraulic turbines are usually required to specify what the rated capacity of their units is in either horsepower or kilowatts.

Two words frequently used in hydroelectric terminology are *demand* and *load.* The terms are often used synonymously, but here they are used with slightly

different meanings. *Demand* refers to the amount of power needed or desired; *load* refers to the rate at which electrical energy is actually delivered to or by a system. In this context, load can include the output from several hydropower plants. It should be noted that load and demand are related to the uses that are being made of the electrical energy. An important job of the engineer is to plan for and match power capacity at hydro plants with energy loads and demands. Implied is the need to have hydropower energy integrated with other modes of energy production.

The power capacity of a hydropower plant is primarily a function of two main variables of the water: (1) water discharge, and (2) the hydraulic head. *Water discharge* is the volume rate of flow with respect to time through the plant. *Full-gate discharge* is the flow condition which prevails when turbine gates or valves are fully open. At maximum rated head and full gate, the maximum discharge will flow through the turbine. *Rated discharge* refers to a gate opening or plant discharge which at rated head produces the rated power output of the turbine.

Hydraulic head is the elevation difference the water falls in passing through the plant. *Gross head* of a hydropower facility is the difference between headwater elevation and tailwater elevation. (*Headwater* is the water in the forebay or impoundment supplying the turbine; *tailwater* is the water issuing from the draft tube exit.) *Net head* is the effective head on the turbine and is equal to the gross head minus the hydraulic losses before entrance to the turbine and outlet losses. Doland (1954) defines *design head* as the effective head for which the turbine is designed for best speed and efficiency. *Rated head* is the lowest head at which the full-gate discharge of the turbine will produce the rated capacity of the generator. It is normally referred to as the *rated net head* in the guarantee of the manufacturer. The term is sometimes used interchangeably with the term *effective head.* Another term used is *critical head. Engineering Monograph No. 20* of the U.S. Department of the Interior (1976) defines critical head as the net head or effective head at which full-gate output of the turbine produces the permissible overload on the generator at unit power factor. This head will produce maximum discharge through the turbine. "Critical head" is used in studies of cavitation and turbine setting, which will be discussed later. Sheldon and Russell (1982) present a composite reference of the various head definitions and terms.

TYPES OF TURBINES

As water passes through a hydropower plant, its energy is converted into electrical energy by a prime mover known as a hydraulic *turbine* or water wheel. The turbine has vanes, blades, or buckets that rotate about an axis by the action of the water. The rotating part of the turbine or water wheel is often referred to as the *runner.* Rotary action of the turbine in turn drives an electrical generator that produces electrical energy or could drive other rotating machinery.

Hydraulic turbines are machines that develop torque from the dynamic and pressure action of water. They can be grouped into two types. One type is an *im-*

ulse turbine, which utilizes the kinetic energy of a high-velocity jet of water to ransform the water energy into mechanical energy. The second type is a *reaction urbine,* which develops power from the combined action of pressure energy and inetic energy of the water. Reaction turbines can be further divided into several ypes, of which the principal two are the Francis and the propeller.

Impulse Turbines

The impulse turbine is frequently called a *Pelton wheel* after one of its early evelopers, Lester Pelton. The potential energy of water flowing from a forebay hrough a penstock is transformed into kinetic energy in a jet or jets of water strikng the single or double bowl-shaped buckets of the impulse runner. The jet of ater strikes the runner tangentially to a circular line of the pitch diameter of the uckets and acts at atmospheric pressure. Figure 2.1 illustrates schematically the rrangement for an impulse-type unit.

The water striking the buckets of the runner is regulated through the use of bulb-shaped needle in a nozzle, as shown in Fig. 2.2. The position of the needle etermines the quantity of water striking the runner. A deflector arrangement in ore sophisticated designs is used to direct the water away from the turbine uckets when there is a load rejection to reduce hydraulic torque on the generator. his deflector is shown schematically in Fig. 2.2.

Figure 2.3 is a picture of an impulse turbine showing a notch in the lip of the ucket. The notch serves to keep the jet from striking the bucket until the jet is sentially tangent with the circular path of the bucket, and thus allows more water act at the outer edge of the bucket before the jet is cut off by the edge of the llowing bucket. Impulse runners have multiple jets and the mounting can be with ther a horizontal or a vertical shaft.

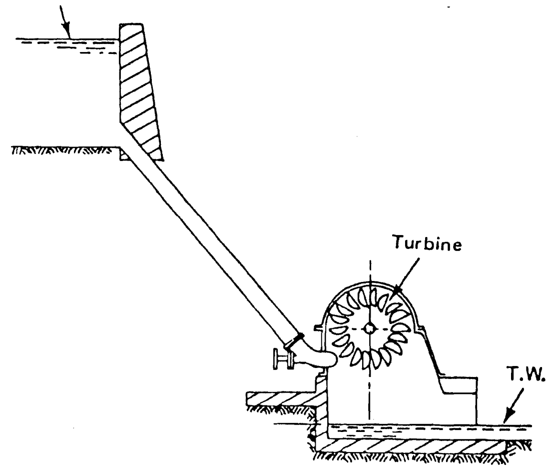

Figure 2.1 Schematic drawing of impulse turbine installation.

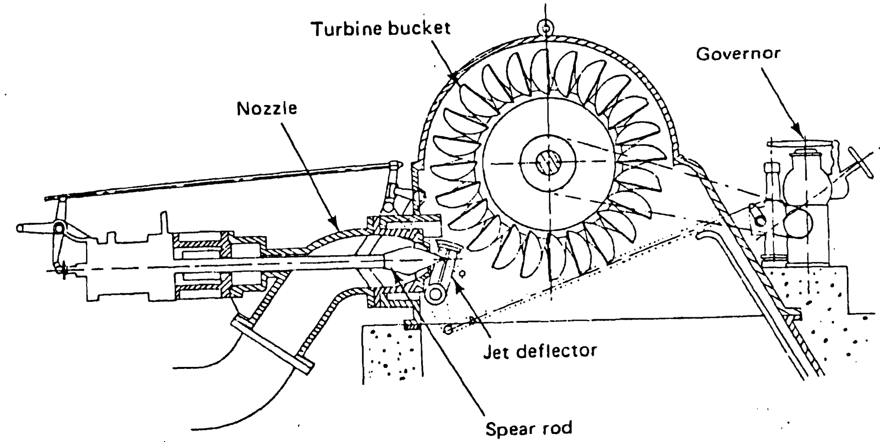

Figure 2.2 Impulse turbine, nozzle and deflector arrangement. SOURCE: Gilbert Gilkes & Gordon, Ltd.

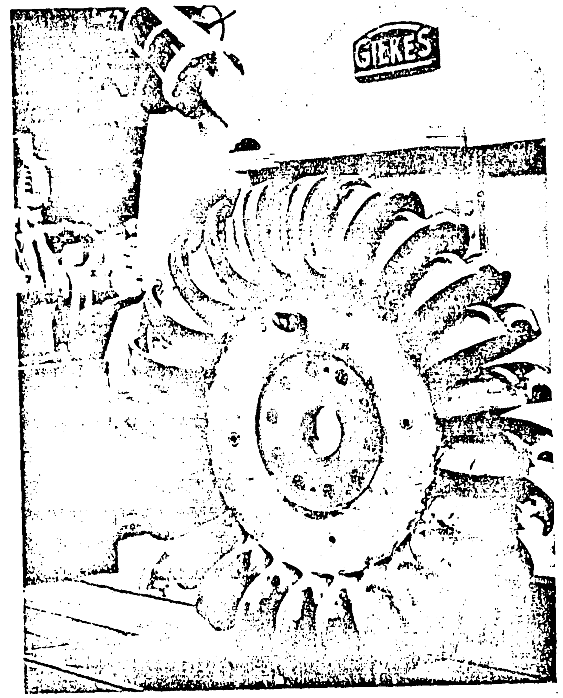

Figure 2.3 Impulse turbine runner showing bucket configuration. SOURCE: Gilbert Gilkes & Gordon, Ltd.

Impulse turbines are usually high head units and used at locations where heads are 1000 ft. or more. They are also used at lower heads for small-capacity units. The ratio of the wheel diameter to the spouting velocity of the water determines the applicability of an impulse turbine.

The impulse or Pelton turbines have advantages for high-head installations, for installations with abrasive matter in the water, and for long-penstock installations where water hammer is critical. The phenomenon of water hammer will be discussed later. Some impulse runners are made with individually bolted buckets and others are solid cast. Double-overhung installations are made with a generator in the center and the runners positioned on the two overhanging ends of a single shaft.

Another design of an impulse turbine is the Turgo Impulse turbine invented by Eric Crewdson of Gilbert Gilkes and Co. Ltd. of England in 1920. The turbine is designed so that the jet of water strikes the buckets at an angle to the face of the runner and the water passes over the buckets in an axial direction before being discharged at the opposite side. The buckets are constrained by a rim on the discharge side of the runner. The advantage claimed for this type of unit is that a larger jet can be applied, resulting in a higher speed with a comparatively smaller machine. A good discussion of this type of unit is presented by Wilson (1974). Figure 2.4 shows the basic difference between a Pelton-type impulse turbine and a Turgo impulse turbine.

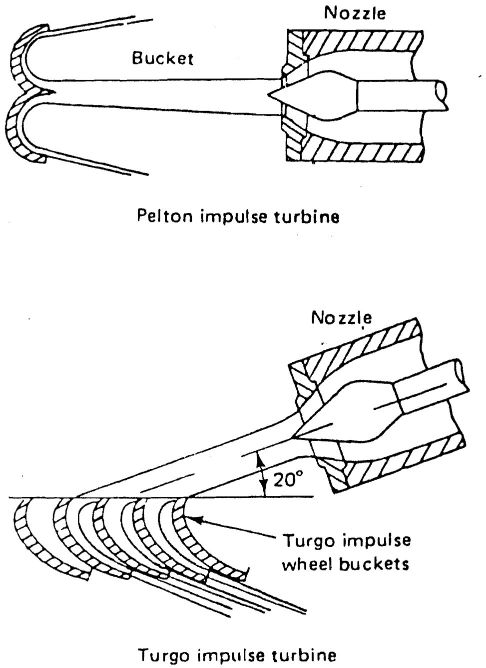

Figure 2.4 Difference between Pelton turbine and Turgo impulse turbine. SOURCE: Gilbert Gilkes & Gordon, Ltd.

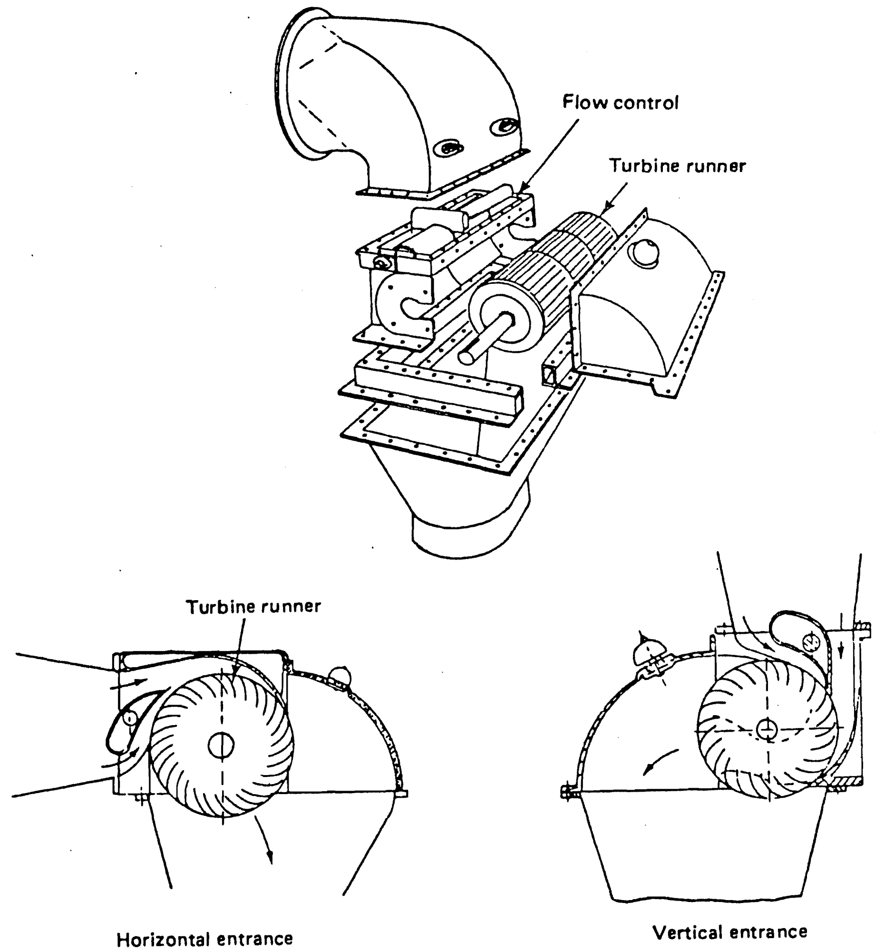

Figure 2.5 Characteristics of a cross-flow turbine. SOURCE: Stapenhorst (1978).

Another type of impulse unit is the cross-flow turbine (Stapenhorst, 1978), also called the Banki or Michell turbine. This type of unit has been manufactured in Europe for many years. The name "cross-flow" comes from the fact that the water crosses through the runner vanes twice in producing the rotation. The cross-flow principle was developed by Michell, an Austrian engineer, at the turn of the century. Professor Banki, a Hungarian engineer, developed the machine further. Figure 2.5 shows the components of the turbine. Its advantages are that standard unit sizes are available and an even higher rotational speed is obtained than from other impulse turbines. Adjustable inlet valves or gates control flow to separate portions of the runner so that a cross-flow turbine can operate over a wide range of flows.

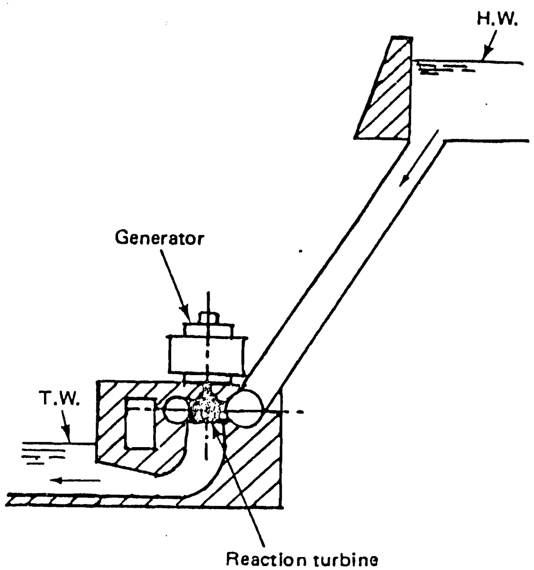

Figure 2.6 Schematic drawing of reaction turbine installation.

Mixed-Flow and Radial-Flow Reaction Turbines

In the operation of reaction turbines, the runner chamber is completely filled with water and a draft tube is used to recover as much of the hydraulic head as possible. Figure 2.6 shows a schematic diagram of the arrangement for this type of unit. Three conditions of flow determine the designs of reaction wheels. If the flow is perpendicular to the axis of rotation, the runner is called a *radial-flow turbine.* An early type of radial-flow machine was the Fourneyron turbine, in which water flow was radially outward. Many early reaction wheels were radially inward-flow runners. If the water flow is partially radial and partially axial, it is called a *mixed-flow turbine.* The most common mixed-flow turbine was developed by James B Francis and bears his name. Francis turbines have a crown and band enclosing the upper and lower portions of the buckets, while a propeller-type runner merely has blades projecting from the hub. Figure 2.7 shows a cross-sectional view through a Francis turbine installation. Figure 2.8 is a picture of the runner of a Francis turbine.

Another type of mixed-flow reaction turbine is the diagonal turbine or *Deriaz turbine.* The runner blades are set at an angle around the rim of a conical hub. There is no band around the blades. Figure 2.9 shows a cross-sectional drawing of a Deriaz turbine. The blades are adjustable and can be feathered about an axis inclined at 45° to the axis of the shaft. The units have been developed for use as reversible pump turbines. They have the advantage of maintaining good efficiency over a wide range of flow, higher-strength attachment of the runner blades to the turbine hub,

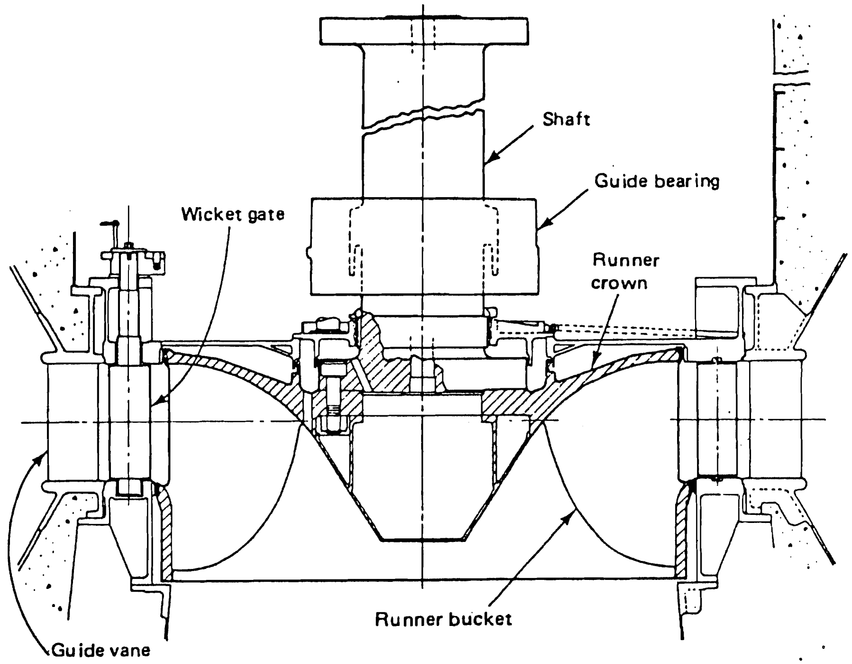

Figure 2.7 Cross-section view of a Francis turbine installation. SOURCE: Allis-Chalmers Corporation.

Figure 2.8 Francis turbine. SOURCE: Vevey.

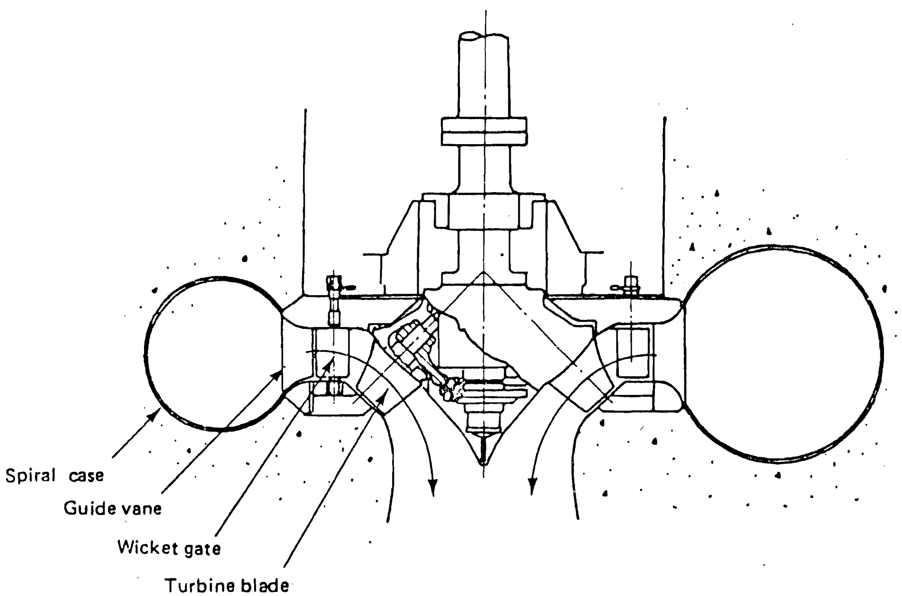

Figure 2.9 Cross-section drawing of Deriaz turbine.

and the arrangement allows higher permissible operating heads than an axial-flow Kaplan or adjustable blade unit, which is discussed in the next section. An excellent reference to the design of Deriaz turbines is Mathews (1970). Other references are Kovalev (1965) and Kvyatkovski (1957). Very few Deriaz turbines have been built.

Axial-Flow Reaction Turbines

Propeller turbines. The direction of flow for most propeller turbines is axial, parallel to the axis of rotation; thus they are classified as *axial-flow turbines.* Early developments utilized propeller units with vertical shafts. More recent developments utilize a horizontal shaft, as discussed and illustrated later. Propeller turbines can have the blades of the runner rigidly attached to the hub; these are called *fixed-blade runners.* The blades of the runners can also be made adjustable so that the turbine can operate over a wide range of flow conditions at better efficiencies. Figure 2.10 shows a cross section through a typical propeller unit. A propeller-type turbine with coordinated adjustable blades and gates is called a *Kaplan turbine* after its inventor, Viktor Kaplan. A propeller turbine with adjustable blades and fixed gates is sometimes referred to as a *semi-Kaplan.* The automatic coordination of the movement of runner blade and adjustment of the gate position provides optimum hydraulic performance and has made such units more efficient for variable-flow and low-head applications. Figure 2.11 is a composite of pictures illustrating the classification and characteristics of reaction-type turbines.

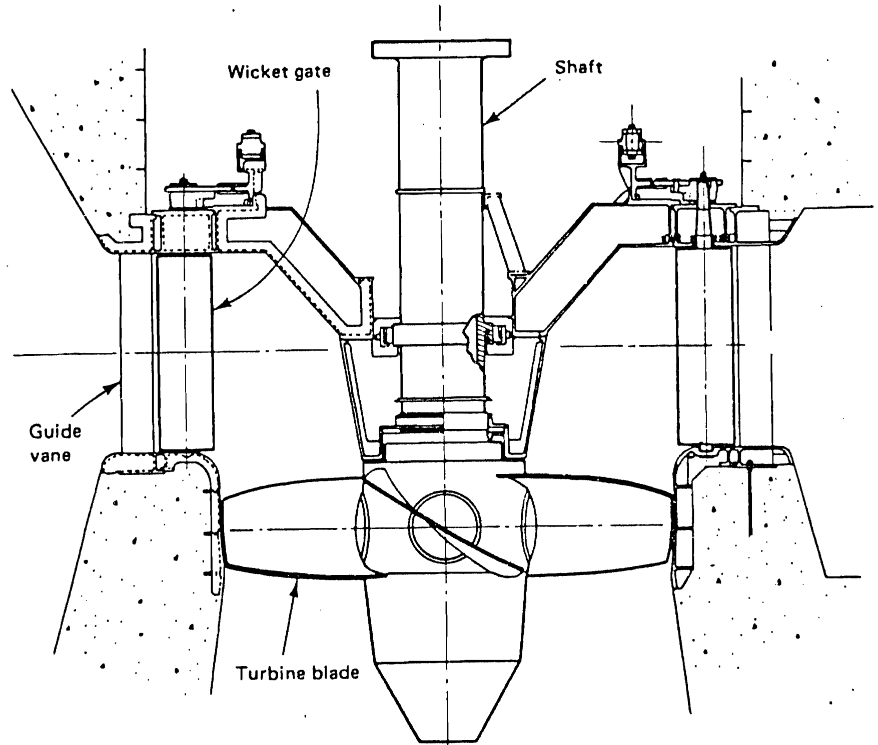

Figure 2.10 Cross-section view of a propeller turbine (vertical Kaplan-type). SOURCE: Allis-Chalmers Corporation.

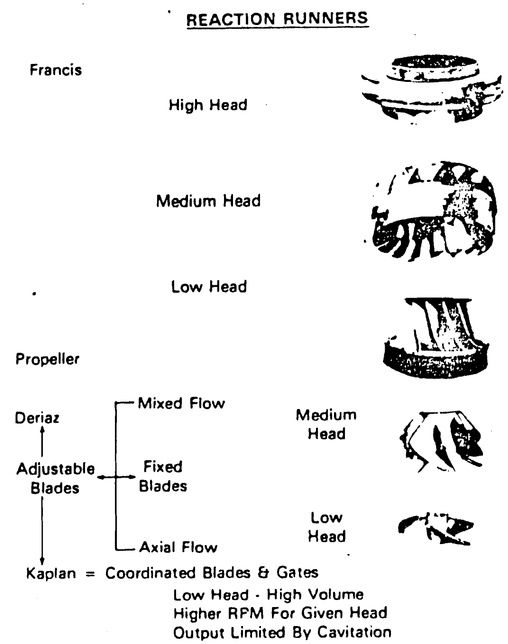

Figure 2.11 Composite illustration and classification for reaction turbines. SOURCE: Allis-Chalmers Corporation.

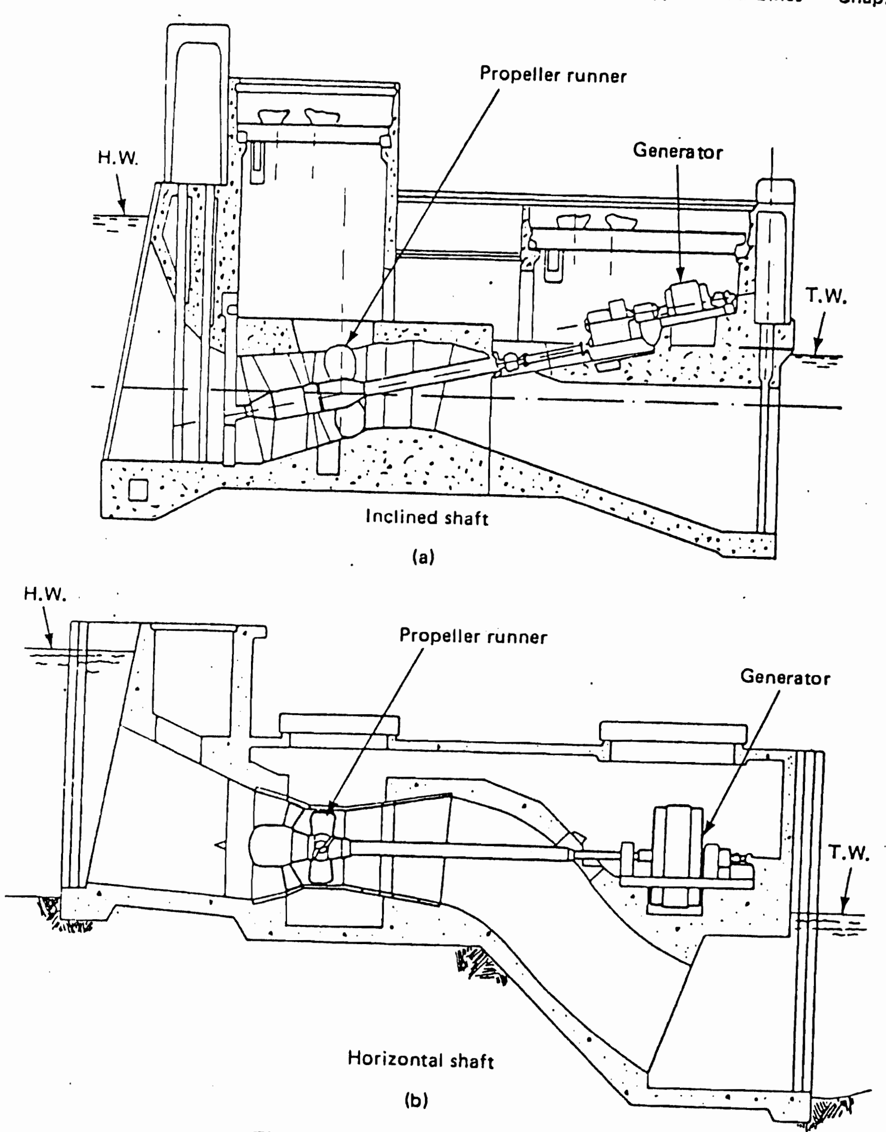

Figure 2.12 Tubular-type turbines.

Recent developments utilizing axial-flow runners have included arranging the runners in specialized configurations sometimes referred to as *tubular-type turbines.* Basically, the units are arranged to minimize the change in direction of flow, to simplify the mounting of the generator, and to provide the best hydraulic characteristics for the water moving through the hydropower plant. TUBE turbine is a registered trademark of the Allis-Chalmers Corporation for a type of unit in which the generator is mounted outside the water passage with direct or gear drive connec-

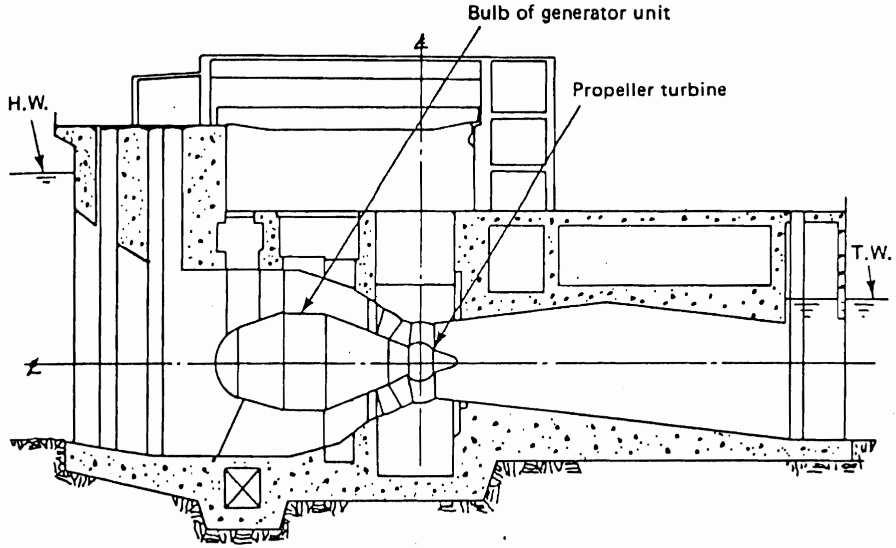

Figure 2.13 Cross-section view of bulb installation.

tion to the generator. Figure 2.12 shows examples of two such installations. These units are now being produced in standardized sizes.

Bulb hydropower units include propeller turbines that drive a generator encapsulated and sealed to operate within the water passage. The generator design is such that all mechanisms are compressed into a diameter that is approximately equal to the propeller diameter. The very compact nature tends to provide some advantages in powerhouse design and in pattern of water flow. It does require special cooling and air circulation within the generator bulb. This type of unit is being manufactured by several companies. Figure 2.13 shows a cross section through a typical bulb installation.

Rim-generator turbines have been developed from an American patent filed by L. F. Harza. Units were built and promoted first by European firms during World War II. The trade name STRAFLO has been used as the proprietary name by the Escher Wyss Company, which builds these rim-generator units. The generator rotor is attached to the periphery of the propeller runner and the stator is mounted within the civil works surrounding the water passage. This arrangement shortens the powerhouse. Only a single crane is required for maintenance, thus reducing space requirements and civil works complexities. The relatively larger diameter rotor provides greater unit inertia than is inherent in bulb generators, an advantage in operating stability. Figure 2.4 shows a schematic drawing of a STRAFLO turbine.

Energy translator. A recent invention of D. J. Schneider reported by Kocivar (1978) is a hydraulic machine known as the Schneider Hydrodynamic Power Generator. It is also referred to as the Schneider Lift Translator. As shown in

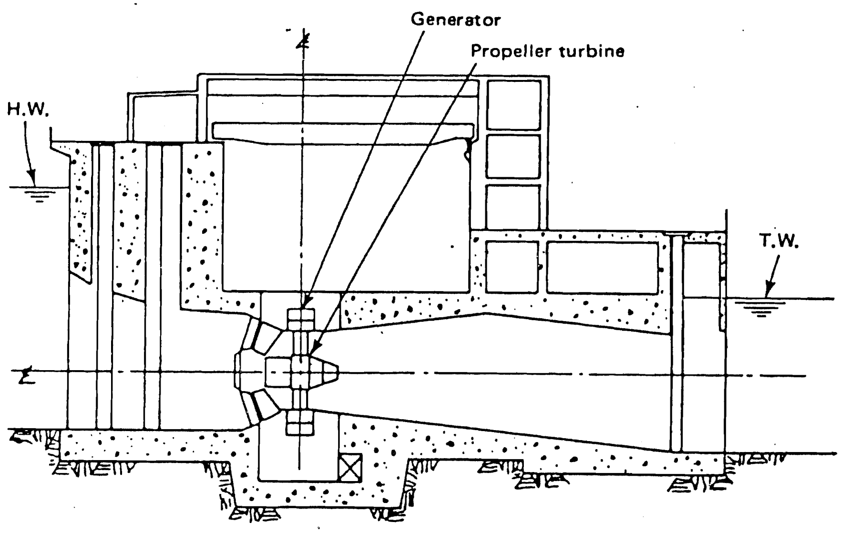

Figure 2.14 Drawing of a rim-generator installation.

Fig. 2.15, it is somewhat like a moving venetian blind being driven by the water passing over the slats in a manner to move the string of slats around the two sprockets. The slats or foils are attached at each end to a moving sprocket chain that moves around the sprockets. The foils are forced downward as the water passes between the foils at the entrance and the water forces the foils upward as it exits on the other side. A generator is connected to the shaft of one of the sprockets. A

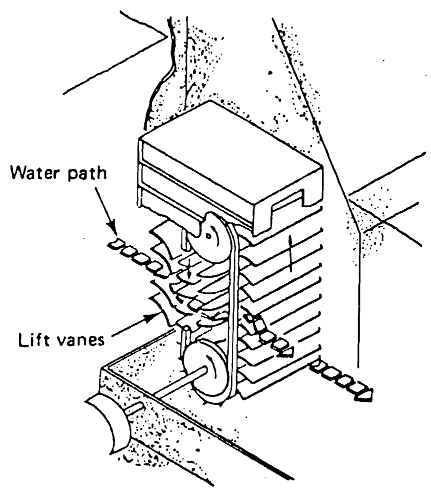

Figure 2.15 Schematic drawing of Schneider hydrodynamic power generator.

report by McLatchy (1980) and a contract report to the Department of Energy (Schneider et al., 1979) cover details of the machine. Two installations are presently being developed on very low head sites on canal systems in California.

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PROBLEMS

2.1. Develop a classification table for types of turbines, showing the characteristics of the various types with respect to range of head, water pressure condition at exit to runner, direction of water flow, and type of energy used.

2.2. Obtain information on an actual installation of each type of turbine, including name of manufacturer, output of unit and/or plant in megawatts, size of runner, net head, rated discharge, and speed.

HYDRAULICS OF HYDROPOWER 3

HYDRAULIC THEORY

In considering hydraulic theory in hydropower engineering, it is important to relate the concept of power to the fundamental variables of head and discharge. As one approach for developing the necessary theory, Fig. 3.1 illustrates certain physical and mathematical concepts.

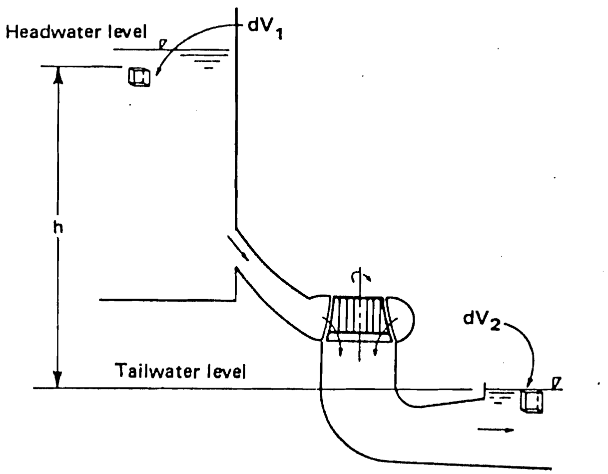

Figure 3.1 Diagram for developing turbine theory.

Energy-Work Approach

If the elemental volume of water, designated dV, moves from position 1 slightly below the headwater level to position 2 at the surface of the tailwater at the exit to the draft tube, the work done is represented by dW in the following equation:

$$\text{Work} = \text{force} \times \text{distance}$$

$$dW = \rho g\, dV\, h \tag{3.1}$$

where dW = work done by elemental mass of water, lb-ft
ρ = density of water, lb-sec^2/ft^4, or slugs/ft^3
g = acceleration of gravity, ft/sec^2
dV = elemental volume, ft^3
h = vertical distance moved by the elemental volume of water, ft.

The h has been purposely designated as slightly below the headwater or forebay level. It is conventional in hydropower computations to treat head as the effective head that is utilized in producing power. Effective head is the difference between energy head at the entrance to the turbine and the energy head at the exit of the draft tube. Hence, in the diagram of Fig. 3.1, the losses of head in the water moving through the penstock to the entrance of the turbine have been accounted for in positioning the elemental cube.

By observing the elements of Fig. 3.1, you should recognize that Eq. (3.1) represents the energy that the water has at position 1 with respect to position 2. Now, consider that if the elemental volume of water moves in some differential unit of time (dt), then the differential discharge (dq) of water can be noted as

$$\text{Discharge} = \text{volume per unit of time}$$

$$dq = \frac{dV}{dt} \tag{3.2}$$

where dq = elemental discharge, ft^3/sec.

The power extracted by the hydropower unit is the rate of doing work and can be represented mathematically as follows:

$$\text{Power} = \frac{\text{work}}{\text{time}} \tag{3.3}$$

$$dP = \frac{dW}{dt}$$

where dP = elemental amount of power, lb ft/sec, or, by substitution from Eq. (3.1),

$$dP = \frac{\rho g\, dV\, h}{dt} \tag{3.4}$$

or, by substitution from Eq. (3.2),

$$dP = \frac{\rho g \, dq \, dt \, h}{dt}$$

which reduces to

$$dP = \rho g \, dq \, h \tag{3.5}$$

Summing the elemental power components of the total discharge passing through the turbine gives the traditional horsepower equation for determining the power capacity of hydropower plants:

$$p_{hp} = \frac{\rho g q h}{550} \quad \text{(theoretical)} \tag{3.6}$$

where p_{hp} = unit power capacity, hp
q = discharge through turbine, ft^3/sec
h = effective head, ft
550 = number of lb-ft/sec in 1 hp.

In the metric or SI system the equation is

$$P_{watts} = \rho' g' QH \tag{3.7}$$

$$P_{kW} = \frac{\rho' g' QH}{1000} \quad \text{(theoretical)}$$

where P_{kW} = unit power capacity, kW
ρ' = mass density of water, kg/m^3
g' = acceleration of gravity, m/sec^2
Q = discharge through turbine, m^3/sec
H = effective head, m
1000 = number of watts in 1 kW.

Note. For equations using the metric or SI system, a prime is added to symbols of density and acceleration, and capital letters are used for power, head, and discharge.

The foregoing equations are for theoretical conditions. The actual output is diminished by the fact that the turbine has losses in transforming the potential and kinetic energy into mechanical energy. Thus an efficiency term (η), usually called *overall efficiency,* must be introduced to give the standard *power equation:*

$$p_{hp} = \frac{\rho g q h \eta}{550} = \frac{q h \eta}{8.81} \tag{3.8}$$

or in the metric system

$$P_{kW} = \frac{\rho' g' QH\eta}{1000} = 9.806 \text{ QH}\eta \tag{3.9}$$

To compare kilowatts and horsepower, remember that

$$P_{kW} = 0.746 p_{hp}$$

Bernoulli Energy Equation Approach

A second approach to basic hydraulic theory of hydropower engineering is the mathematical development in terms of energy grade lines and hydraulic grade lines, using the Bernoulli equation. The Bernoulli equation is related to the energy grade line, hydraulic grade line, and the position grade lines as shown in Fig. 3.2 and by the equation

$$\frac{V_1^2}{2g} + \frac{p_1}{\gamma} + Z_1 = \frac{V_2^2}{2g} + \frac{p_2}{\gamma} + Z_2 = \text{constant} \tag{3.10}$$

where V_1 = water velocity at point 1, ft/sec
p_1 = pressure at point 1, lb/ft^2
$\gamma = \rho g$ = specific weight of water, lb/ft^3
Z_1 = potential head at point 1 referenced to the datum, ft
V_2 = water velocity at point 2, ft/sec
p_2 = pressure at point 2, lb/ft^2
Z_2 = potential head at point 2, ft
h_f = head loss in flow passage between points 1 and 2, ft

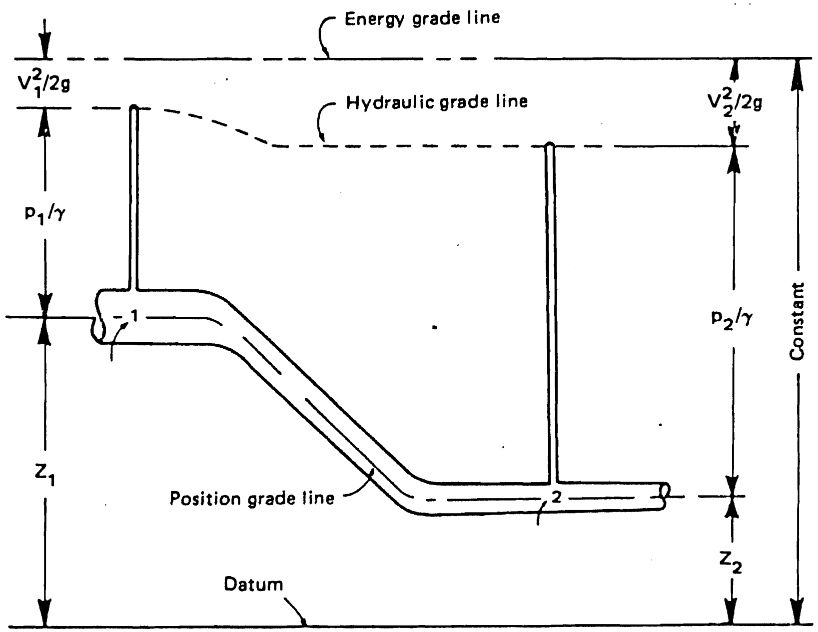

Figure 3.2 Bernoulli diagram relating energy grade lines and hydraulic grade line.

V_3 = water velocity at point 3, ft/sec
p_3 = pressure at point 3, lb/ft^2
Z_3 = potential head at point 3, ft
h = effective head on turbine, ft.

lathematically, then, the Bernoulli equation states that the sum of the component nergies (position energy, pressure energy, and kinetic energy) is constant in a conned moving fluid as the fluid moves along its path. Thus a change in any one of ie component energies at any point along the path of the moving fluid must be ompensated for by an equal change of the water energy components at that point. his mathematical development assumes no friction or head loss in the fluid moving om point 1 to point 2 as shown in Fig. 3.2. In a practical sense, there is friction ss or head loss, h_f, in the case of water flowing from point 1 to point 2. This is ccounted for in the graphical representation shown in Fig. 3.3.

Referring to Fig. 3.3, the Bernoulli equation for a hydropower installation is rst written between point 1 at the surface of the forebay and point 2 at the ntrance to the turbine as

$$\frac{V_1^2}{2g} + \frac{p_1}{\gamma} + Z_1 = \frac{V_2^2}{2g} + \frac{p_2}{\gamma} + Z_2 + h_f \tag{3.11}$$

hen the Bernoulli equation is written between points 2 and 3, the surface of the ater at the exit to the draft tube:

$$\frac{V_2^2}{2g} + \frac{p_2}{\gamma} + Z_2 = \frac{V_3^2}{2g} + \frac{p_3}{\gamma} + Z_3 + h \tag{3.12}$$

ecognizing that for practical purposes, V_1, p_1, and p_3 are equal to zero, then solvg for p_2/γ in Eq. (3.11), the result is

$$\frac{p_2}{\gamma} = Z_1 - \frac{V_2^2}{2g} - Z_2 - h_f \tag{3.13}$$

ext solving for h in Eq. (3.12), the result is

$$h = \frac{V_2^2}{2g} + \frac{p_2}{\gamma} + Z_2 - \frac{V_3^2}{2g} - Z_3 \tag{3.14}$$

ow if the right side of Eq. (3.13) is substituted into Eq. (3.14) for p_2/γ, the sult is

$$h = \frac{V_2^2}{2g} + \left(Z_1 - \frac{V_2^2}{2g} - Z_2 - h_f\right) + \left(Z_3 - \frac{V_3^2}{2g} - Z_3\right) \tag{3.15}$$

mplifying, we have

$$h = Z_1 - Z_3 - h_f - \frac{V_3^2}{2g} \tag{3.16}$$

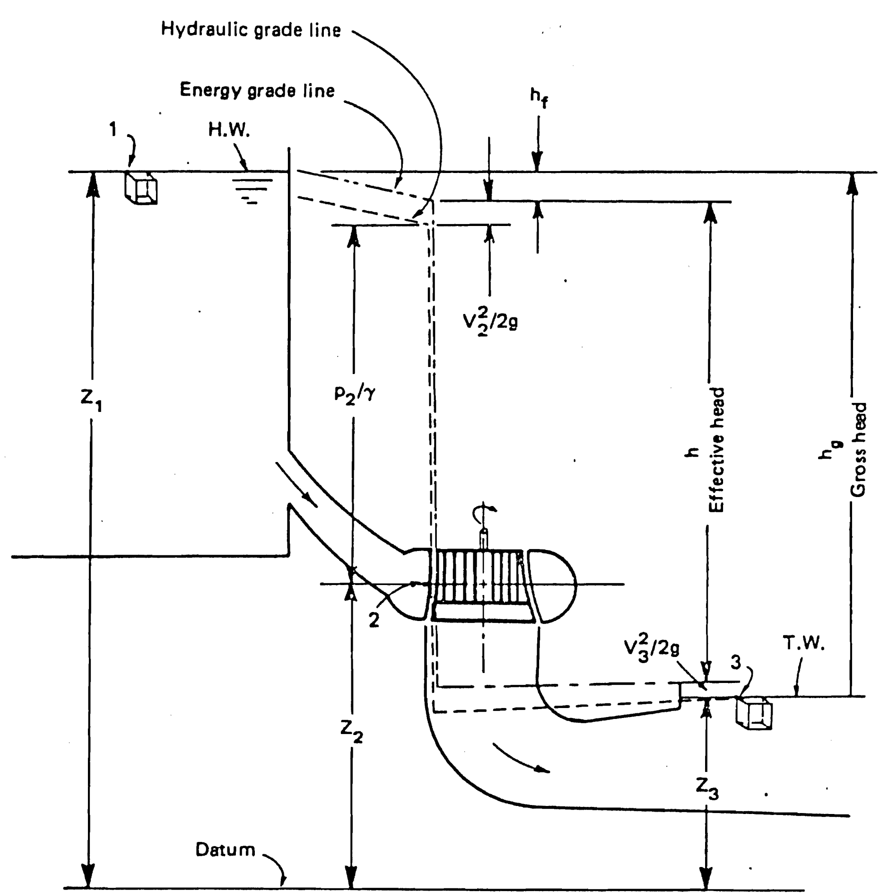

Figure 3.3 Bernoulli diagram for a hydropower installation.

which is the effective head and is so indicated dimensionally in Fig. 3.3.

Because the Bernoulli equation defines terms in units of pound feet per pound of water flowing through the system, it should be recognized that the pounds of water flowing through the turbine per unit of time by definition is $\rho g q$. Now recognizing that energy per unit of time is power, it is simple to calculate by multiplying Eq. (3.16) and $\rho g q$ or γq to obtain the theoretical power delivered by the water to the turbine as

$$P_{\text{lb/ft}} = \gamma q h = \text{theoretical power} \tag{3.17}$$

or Eq. (3.6) results by converting to horsepower as

$$P_{\text{hp}} = \frac{\rho g q h}{550} = \frac{\gamma q h}{550} \tag{3.18}$$

The energy equation in the Bernoulli form, as given in Eqs. (3.10) and (3.11) and illustrated in Fig. 3.3 relating effective head to the energy grade line and the hydraulic grade line, should be referred to often to understand the many concepts of hydropower engineering.

KINETIC THEORY

Further theory related to the speed of the runner and the dynamic action of the water on the buckets and vanes is important for understanding the energy-converting action and is necessary in developing certain turbine constants that are used in the design and selection of runners.

Impulse Runner Force

Figure 3.4 is a diagrammatic sketch of the jet action on an impulse runner. The dynamic force imparted on a moving vane or bucket by a jet of water is given by the equation

$$F = \frac{Wv}{g}(1 - m\cos\theta) \tag{3.19}$$

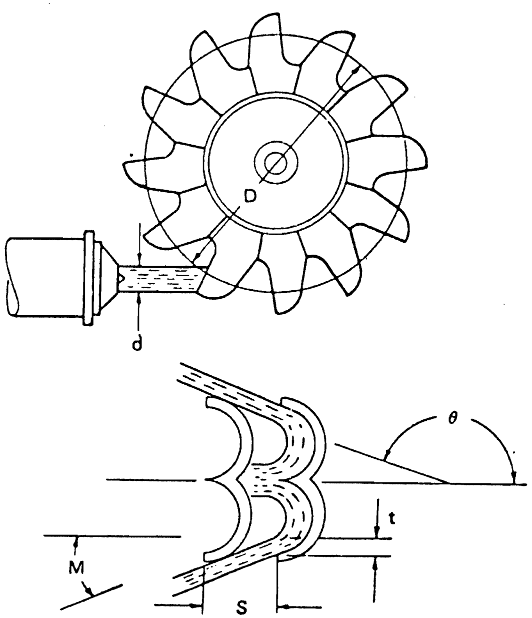

Figure 3.4 Definition sketch for water action on impulse turbine.

where F = dynamic force on the vane, lb
W = weight of water striking vane, lb/sec
g = acceleration of gravity, ft/sec^2
v = relative velocity of the jet of water with respect to the moving vane, ft/sec
m = coefficient accounting for loss of velocity moving across vane
θ = angle of deflection of jet from the original jet direction.

The ideal deflection angle for θ for an impulse runner bucket is 180° but for practical purposes the bucket angle is generally about 165° so that the jet of water does not interfere with the buckets (see Fig. 3.4).

Velocity. The relative velocity v is determined from the relation

$$v = V - u \tag{3.20}$$

where V = absolute velocity of water jet, ft/sec
u = absolute linear velocity of the bucket, ft/sec.

Torque. The torque exerted by the jet of water is the product of the force F and the lever arm r at which the water force is acting. Torque is given by the formula

$$T = \frac{Wvr}{g}(1 - m\cos\theta) \tag{3.21}$$

where T = torque imparted to runner, lb-ft
r = radius of the runner, ft.

Power. The theoretical power imparted is given by the formula

$$p_{hp} = \frac{Fu}{550} = \frac{Wvu}{550g}(1 - m\cos\theta) \tag{3.22}$$

or utilizing the torque equation,

$$p_{hp} = \frac{T\omega}{550} \tag{3.23}$$

where ω = angular velocity of runner, rad/sec.

Since relative velocity, v, from Eq. (3.20) is

$$v = V - u$$

and

$$u = \phi V \tag{3.24}$$

where ϕ = the ratio of wheel speed to the spouting velocity, $\sqrt{2gh}$ then, by substituting into Eq. (3.22), we obtain

$$P_{hp} = \frac{WV^2}{550}(1-\phi)(\phi)(1-m\cos\theta) \qquad (3.25)$$

At the operating point of maximum power and best speed, the relative velocity of water initially striking the bucket is

$$v_1 = V - u \qquad (3.26)$$

and the relative velocity leaving the bucket is

$$v_2 = mv_1 = m(V-u) \qquad (3.27)$$

At the best turbine speed, v_2 has no tangential component of velocity and

$$u = -v_2 \cos\theta = -m(V-u)\cos\theta \qquad (3.28)$$

Therefore,

$$u = \frac{Vm\cos\theta}{m\cos\theta - 1} \qquad (3.29)$$

Substituting into Eq. (3.22)

$$P_{hp} = \frac{W}{550}(V-u)(u)(1-m\cos\theta)$$

$$= \frac{W}{550}\left(V - \frac{mV\cos\theta}{m\cos\theta-1}\right)\frac{mV\cos\theta}{m\cos\theta-1}(1-m\cos\theta)$$

$$= \frac{WV^2}{550}\,\frac{m\cos\theta}{m\cos\theta-1}$$

r

$$= \frac{W}{550}V^2\phi \qquad (3.30)$$

Best linear velocity. The best linear speed of the turbine can be determined using Eq. (3.29) and

$$V = C_d\sqrt{2gh} \qquad (3.31)$$

where C_d = velocity coefficient and then using Eq. (3.29)

$$u = C_d\sqrt{2gh}\,\frac{m\cos\theta}{m\cos\theta-1}$$

For example, if $C_d = 0.98$, $m = 0.96$, and $\phi = 165°$

$$u = 0.98\sqrt{2g}\,\frac{(0.96)\cos 165°}{0.96\cos 165°}\sqrt{h}$$

$$u = 3.78\sqrt{h}$$

Jet diameter. Jet diameter can be determined using the equation

$$q = C_d A\sqrt{2gh} \qquad (3.32)$$

where q = jet discharge in ft^3/sec
A = area of the jet in ft^2
d_j = jet diameter in ft

$$q = C_d\,\frac{\pi d_j^2}{4}\sqrt{2gh}$$

and

$$d_j = \sqrt{\frac{4q}{C_d\,\pi\sqrt{2gh}}} = \frac{0.398\,q^{1/2}}{C_d^{1/2}\,h^{1/4}}$$

for

$C_d = 0.98$ as usual value

then

$$d_j = 0.402\,\frac{q^{1/2}}{h^{1/4}} \qquad (3.33)$$

Bucket spacing. In Fig. 3.2 it can be noted that if wheel diameter is known, the spacing for buckets can be determined by the simple equation

$$s = \frac{\pi d}{n_b} \qquad (3.34)$$

where s = bucket spacing, in.

Runner diameter. Brown and Whippen (1972) indicate that a good rule of thumb for impulse turbines is to make the diameter of the runner in feet equal to the diameter of the jet in inches. However, the ratio of the diameter of the runner to the diameter of the jet in feet will vary from a low of 9 for low-head impulse turbines to a high of 18 for high-head impulse turbines. The ratio is limited by the physical restraints of attaching the buckets to the disk.

Reaction Runner Flow

For a reaction runner, the torque (T) imparted by the water to the runner is given by the equation.

$$T = \frac{W}{g}(r_1V_1\cos\alpha_1 - r_2V_2\cos\alpha_2) \qquad (3.35)$$

where $W = \gamma q$ = quantity of water flowing, lb/sec
r_1 = radius of runner in feet at the periphery where the water first strikes the runner vane, ft
V_1 = absolute velocity of the water at the entrance to the runner, ft/sec
α_1 = angle that the absolute velocity vector V_1 makes with tangent to the runner circumference
r_2 = radius of the runner in feet at point where water leaves the runner, ft
V_2 = absolute velocity of water at the exit to the runner, ft/sec
α_2 = angle that the absolute velocity vector V_2 makes with tangent to runner circumference.

The mathematical relationship is better understood by referring to Fig. 3.5, a vector diagram of the flow acting on the blade of a reaction turbine. In that diagram it is assumed that the flow is two-dimensional radial inward flow. The relative velocity v_1 is made up of a component of the absolute velocity V_1 along with u, the linear velocity of the moving runner. The relationship between the various velocity terms is given by Eqs. (3.36) and (3.37):

$$v_1 = \sqrt{u_1^2 + V_1^2 - 2u_1 V_1 \cos\alpha_1} \tag{3.36}$$

$$\frac{V_1}{v_1} \sin\alpha_1 = \sin(180^\circ - \beta_1) \tag{3.37}$$

where v_1 = relative velocity of water at the entrance to the runner, ft/sec
u_1 = linear velocity of the runner at the periphery, ft/sec
β_1 = angle between a tangent to the runner and the relative velocity of the water entering the runner.

The blade angle of the turbine is normally designed such that the angle between the tangent to the entrance edge and the tangent to the circumference is equal to β_1. The angle is usually greater than 90° and, according to Brown and Whippen (1977), may be as great as 135°.

The required height, B, of the passage at the entrance to the runner or exit from the guide vanes is given by Brown and Whippen (1977) as

$$B = \frac{A_1}{\pi C_1 D_1 \sin\alpha_1} \tag{3.38}$$

where B = height of passage at the entrance edge to the runner, ft
A_1 = required cross-sectional area of water passages at right angles to the direction of flow, ft^2
C = coefficient, usually about 0.95
$D_1 = 2r_1$ = diameter of the circle at the entrance to the runner, ft
α_1 = as previously defined and normally equal to the guide vane angle.

For best performance of the runner, the water should leave the runner in an axial direction and with a very small absolute velocity. In a practical sense it is not possible to obtain completely axial flow at all gate openings. To determine the absolute velocity, it is common practice to consider the absolute velocity of the

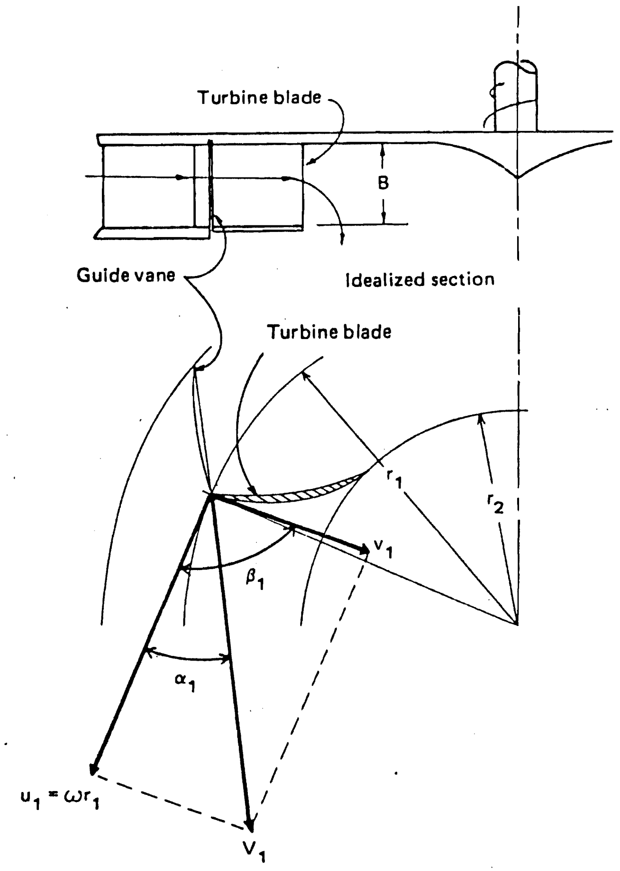

Figure 3.5 Vector diagram of water action on reaction turbine.

water as it exits from the runner equal to the discharge divided by the area of the draft tube at its entrance.

Power. The theoretical power imparted by the water moving through the reaction turbine is given by the formula

$$P = T\omega = \frac{W\omega}{g}(r_1 V_1 \cos\alpha_1 - r_2 V_2 \cos\alpha_2) \tag{3.39}$$

where ω = rotative speed of runner, rad/sec. More detail on particular characteristics of impulse turbines and reaction turbines is given in the next chapter.

The kinetic theory of axial-flow turbines is not treated in this text. A good reference to this is the work of J. W. Daily (1950). Brown and Whippen (1977) indicate that the absolute velocity of water discharged from the runner of a propeller turbine ranges from $4\sqrt{h}$ to $6\sqrt{h}$, where h is the effective head.

The absolute velocity, V_1, at the entrance to the reaction turbine runner is related to effective head by the formula

$$V_1 = c_v\sqrt{2gh} \tag{3.40}$$

where c_v = velocity coefficient. For reaction turbines, Brown and Whippen (1977) indicate that c varies between 0.8 and 0.6.

The angle α_1 between the tangent to the circular path of a point on the periphery of the runner at the entrance and the direction of the water entering the runner and passing the guide vanes (see Fig. 3.5) ranges from 15° to 35°. It is often assumed that the relative velocity of the water leaving the runner blade is in the same direction as the blade angle at the exit. The magnitude of the relative velocity may be computed by dividing the discharge, q, by the total exit area of the runner passage at right angles to the blade edges. In modern turbines that area would be a very complex surface, but for a truly radial flow runner it would be a cylindrical area equal to $2\pi r_2 B$ (see Fig. 3.5).

REFERENCES

Brown, L. P., and W. E. Whippen, *Hydraulic Turbines,* Parts 1 and 2. Scranton, Pa.: International Textbook Company, 1972, 1977.

Daily, J. W., "Hydraulic Machinery." Chapter XIII in *Engineering Hydraulics,* H. Rouse, ed. New York: John Wiley & Sons, Inc., 1950.

PROBLEMS

3.1. Show why the maximum theoretical power from an impulse turbine would occur when the bucket angle is $\theta = 180°$, and explain why net head for such a turbine is taken only to the centerline elevation of the jet striking the turbine buckets.

3.2. An impulse turbine is to be used to develop the energy at a site where water discharge q = 20 ft^3/sec and the effective head is 980 ft. Using theoretical and empirical equations presented in this chapter, find the following:

(a) Required jet diameter for a single jet
(b) Approximate runner diameter
(c) Theoretical best linear speed of runner
(d) Absolute velocity of water at impact with runner
(e) Relative velocity of water striking the runner
(f) Theoretical torque imparted by the water to the runner
(g) Theoretical output of runner, in horsepower.

Assume the deflection angle to be 130°.

3.3. A reaction turbine with an estimated overall efficiency of 0.91 is to utilize 500 ft^3/sec of water, operating at a synchronous speed of 450 rpm under an effective head of 200 ft. If the acute angle between the guide vane and a tangent to the outer periphery of the runner is 30° and the runner diameter at the entrance is 3.83 ft, determine the following:

(a) Absolute velocity at entrance if the coefficient of velocity is assumed to be 0.80
(b) Linear speed of a point on the runner at the entrance to the runner
(c) Relative velocity of water at the entrance edge of the runner bucket
(d) Runner blade angle, β
(e) Approximate height at the entrance edge of the runner
(f) Output of the turbine in horsepower and kilowatts
(g) Torque to be delivered by the turbine shaft

3.4 Would it be possible in Problem 3.3 to find the relative velocity of the water exiting the runner with information given and discussed in the text?

TURBINE CONSTANTS 4

SIMILARITY LAWS

Similarity laws have been developed for characterizing turbine performance of units of different size and type. They provide a means of predicting performance based on the performance of models or the performance of units of design similar to those that have already been built. The fact that the similarity laws can be used is often referred to as the *homologous* nature of turbines. When turbines of different sizes are designed to have corresponding linear dimensions with a common geometric ratio, the turbines are said to be homologous. The power outputs, speeds, and flow characteristics are proportional and they tend to have equal efficiencies. These similarity laws are developed and presented in a series of formulas that define what are called the *turbine constants.* The equations are derived from fundamental physical concepts of motion and hydraulic theory.

Unit Speed

Consider the speed ratio or peripheral speed coefficient as defined in the formula

$$\phi = \frac{u}{\sqrt{2gh}} \qquad (4.1)$$

where ϕ = ratio of linear velocity of the periphery of the turbine runner to the spouting velocity of the water
u = linear velocity of the turbine runner at the reference diameter, ft/sec

$\sqrt{2gh}$ = theoretical spouting velocity of the water operating under a give head, ft/sec
g = acceleration of gravity, ft/sec^2.

Now if the linear speed of the runner is defined in terms of rotating speed an diameter of the runner, the following results:

$$\phi = \frac{\pi dn/[12(60)]}{\sqrt{2gh}} \qquad (4.2$$

where d = reference diameter of the runner, in.
n = runner speed, rpm.

By grouping all the known constant terms, the equation takes the form

$$\frac{\phi(12)(60)\sqrt{2g}}{\pi} = \frac{nd}{\sqrt{h}} \qquad (4.3$$

The speed ratio variable times the constant terms is replaced with a single variabl n_1, known as the *unit speed.* Then

$$n_1 = \frac{nd}{\sqrt{h}} \qquad (4.4$$

Then n_1 is the speed in rpm of a theoretical turbine having a unit diameter an operating under a net head of unity.

Dimensionless Constants

Recently, an international system has been put forth by various manufacturer to make the turbine constants more convenient and to utilize a consistent system o measurement units. Turbine constants under this system use dimensionless ratio and metric, SI, units for the various parameters. For the unit speed the equation i

$$\omega_{ed} = \frac{\omega D}{\sqrt{g'H}} \qquad (4.5$$

where ω_{ed} = unit speed
ω = angular velocity of runner, rad/sec
D = reference diameter of runner, m
g' = acceleration of gravity, m/sec^2
H = head, m.

The advantage claimed for these dimensionless constants is that the units of measur are more easily converted and the terms are more rational to work with in mathe matical expressions.

Note. In Eq. (4.4) the units of n_1 are

$$\frac{(\text{rpm})(\text{in.})}{(\text{ft})^{1/2}} = \frac{(1/T)(L)}{L^{1/2}} = \frac{L^{1/2}}{T}$$

where T = time units
L = length units,
while in Eq. (4.5) the units of ω_{ed} are

$$\frac{(\text{rad/sec})(\text{m})}{[(\text{m/sec}^2)(\text{m})]^{1/2}} = \frac{(1/T)(L)}{[(L/T^2)(L)]^{1/2}} = \frac{(1/T)(L)}{(1/T)(L)}$$

dimensionless

This is a ratio of the peripheral speed of the runner to the theoretical spouting velocity of the water.

Unit Discharge

The unit discharge equation is developed in a similar manner, as follows:

$$q = F(A, h) = CA\sqrt{2gh} \tag{4.6}$$

where q = design discharge flowing through turbine, ft^3/sec
A = circular area opening at the exit from the runner through which water passes, ft^2
C = an orifice-type coefficient relating flow to head and area A.
Writing this in terms of diameter of the runner in inches, the following equation results:

$$q = \frac{C\pi d^2\sqrt{2gh}}{4(144)} \tag{4.7}$$

By grouping all constant terms on one side, the equation takes the form

$$\frac{C\pi\sqrt{2g}}{4(144)} = \frac{q}{d^2\sqrt{h}} \tag{4.8}$$

Then

$$q_1 = \frac{C\pi\sqrt{2g}}{4(144)} = \frac{q}{d^2\sqrt{h}} \tag{4.9}$$

Then q_1 is the discharge of a runner with unit diameter operating under a unit head. The corresponding dimensionless unit discharge specified by international standards is as follows:

$$Q_{ed} = \frac{Q}{D^2\sqrt{g'H}} \tag{4.10}$$

where Q_{ed} = unit discharge
Q = design discharge flowing through turbine, m^3/sec.

Unit Power

The unit power equation is developed from the following:

$$p = \frac{\rho g q h \eta}{550} \tag{4.11}$$

where p = turbine power output, hp
ρ = density of water, slugs/ft^3
η = turbine overall efficiency
550 = number of ft-lb/sec in 1 hp.
By substituting the value of $q = q_1 d^2\sqrt{h}$ from Eq. (4.9) in Eq. (4.11), the following results:

$$p = (q_1 d^2\sqrt{h})\,\frac{\rho g h \eta}{550} \tag{4.12}$$

By grouping the variable q_1 and all the constant terms as before, the equation has the form

$$p_1 = \frac{q_1\rho g\eta}{550} = \frac{p}{d^2h^{3/2}} \tag{4.13}$$

Then p_1 is the power produced by a runner with a unit diameter operating under a unit head. The corresponding dimensionless unit power term is

$$P_{ed} = \frac{P}{\rho' D^2(g'H)^{3/2}} \tag{4.14}$$

where P_{ed} = unit power, dimensionless
P = turbine power output, watts
ρ' = mass density of water, kg/m^3
g' = acceleration of gravity, m/sec^2.

Specific Speed

To develop a more universal constant that embodies all the equations, it is necessary to operate on Eqs. (4.4) and (4.13). The value of d equal to $n_1\sqrt{h}/n$ from Eq. (4.4) is introduced in Eq. (4.13) so that the following equation results:

$$p_1 = \frac{p}{\left[\dfrac{n_1\sqrt{h}}{n}\right]^2 h^{3/2}} \tag{4.15}$$

Grouping the n_1 and p_1 on one side of the equation, the following results:

$$p_1 n_1^2 = \frac{n^2 p}{h^{5/2}} \tag{4.16}$$

Taking the square root of both sides of the equation gives the term n_s, the specific speed, as follows:

$$n_s = \frac{n\sqrt{p}}{h^{5/4}} \tag{4.17}$$

Then n_s is the speed of a unit producing unit output under a head of unity.

In common usage in Europe is a similar form of the specific speed:

$$N_s = \frac{N\sqrt{P}}{H^{5/4}} \tag{4.18}$$

where N_s = specific speed, units of rpm, kW, and m
N = rotational speed of turbine, rpm
P = power output at best efficiency, kW
H = net head, m.

Note. Capital letters are used when metric SI units are used.

The corresponding term from the dimensionless-type constant has been altered to include in the definition the turbine discharge rather than the power output. The development is described below:

First, Eq. (3.9) is modified to express power in watts as $P = \rho' g' QH\eta$. This is then introduced into Eq. (4.14) so that

$$P_{ed} = \frac{\rho' g' QH\eta}{\rho D^2 (g'H)^{3/2}} \tag{4.19}$$

Now the value $\omega_{ed}\sqrt{g'H}/\omega$ is substituted for D in Eq. (4.19), giving the following:

$$P_{ed} = \frac{\rho' Qg' H\eta}{\rho'(\omega_{ed}\sqrt{g'H}/\omega)^2 (g'H)^{3/2}} \tag{4.20}$$

By grouping P_{ed}, ω_{ed}, and η on one side, the equation takes the form

$$\frac{P_{ed}\,\omega_{ed}^2}{\eta} = \frac{Q\omega^2}{(g'H)^{3/2}} \tag{4.21}$$

Taking the square root of both sides of the equation gives the equation for the dimensionless specific speed:

$$\omega_s = \sqrt{\frac{P_{ed}\,\omega_{ed}^2}{\eta}} = \frac{\omega Q^{1/2}}{(g'H)^{3/4}} \tag{4.22}$$

where all units are as defined before in the metric SI system.

This is the form of the specific speed equation that is being advocated for international standardization. In addition to this form, Csanady (1964) reports the specific speed in a similar form as follows:

$$\Omega = \frac{\omega q^{1/2}}{(gh_n)^{3/4}} \tag{4.23}$$

where Ω = specific speed
ω = angular velocity of runner, rad/sec
q = water discharge, ft^3/sec
g = acceleration of gravity, ft/sec^2
h_n = net head, ft.

Other variations of the turbine constants with relation to torque have been developed for ease of analyzing certain characteristics of turbines. Table 4.1 is a summary of the various forms of the turbine constants giving the equation for each and conversions for converting the specific speed from one system to another. A specific speed may be calculated for any point of turbine operation, but the usual specific speed of a turbine is defined as that value calculated at the point of peak efficiency. Not shown in the table is the equation presented by Csanady (1964), Eq. (4.23). Csanady shows that $\Omega = n_s/42$.

However, this is true only if the turbine efficiency, η, is 93.2%, while in actuality, it can be shown that

$$\Omega = \frac{1}{43.5\sqrt{\eta}} \tag{4.24}$$

where η = turbine efficiency. By substituting $\omega = 2\pi n/60$ and $q = \sqrt{8.81p/h\eta}$ from Eq. (3.8) in Eq. (4.23) we have

$$\Omega = \frac{\omega q^{1/2}}{(gh)^{3/4}} = \frac{2\pi n}{60}\left[\sqrt{\frac{8.81p}{h\eta}}\right]\frac{1}{(gh)^{3/4}}$$

Rearranging terms, we have

$$\Omega = \frac{n\sqrt{p}}{h^{5/4}}\,\frac{2\pi}{60(g)^{3/4}}\,\frac{\sqrt{8.81}}{\sqrt{\eta}}$$

but

$$\frac{n\sqrt{p}}{h^{5/4}} = n_s$$

from Eq. (4.17) so that

$$\Omega = \frac{n_s}{43.5\sqrt{\eta}}$$

A similar development shows the relation between ω_s and n_s, since from Eq. (4.22)

$$\omega_s = \frac{\omega Q^{1/2}}{(g'H)^{3/4}}$$

Then by substituting

$$Q = \frac{P}{9.806\,H\eta}$$

TABLE 4.1 Comparison of Turbine Constants in Different Systems of Units and Forms of Equations[a]

	American System (hp, in., ft^3/sec, rpm)		European System (kW, m, m^3/sec, rpm)		Dimensionless System	
Parameter	Designation	Formula	Designation	Formula	Designation	Formula
Speed ratio	ϕ	$\phi = \dfrac{dn}{43.368(h)^{0.5}}$	k_u	$k_u = \dfrac{D_3 N}{60(2g'H)^{0.50}}$		
Unit speed	n_1	$n_1 = \dfrac{dN}{h^{0.5}}$	N_{11}	$N_{11} = \dfrac{DN}{H^{0.5}}$	ω_{ed}	$\omega_{ed} = \dfrac{\omega D}{(g'H)^{0.5}}$
Unit discharge	q_1	$q_1 = \dfrac{q}{d^2 h^{0.5}}$	Q_{11}	$Q_{11} = \dfrac{Q}{D^2 H^{0.5}}$	Q_{ed}	$Q_{ed} = \dfrac{Q}{D^2 (g'H)^{0.5}}$
Discharge coefficient	–	–	–	–	$Q_{\omega d}$	$Q_{\omega d} = \dfrac{Q}{\omega D^3}$
Unit torque	–	–	–	–	T_{ed}	$T_{ed} = \dfrac{T}{\rho' D^3 g H}$
Torque coefficient	–	–	–	–	$T_{\omega d}$	$T_{\omega d} = \dfrac{T}{\rho' \omega^2 D^5}$
Energy coefficient	–	–	–	–	$E_{\omega d}$	$E_{\omega d} = \dfrac{g'H}{(\omega D)^2}$
Unit power	p_1	$p_1 = \dfrac{p}{d^2 h^{1.5}}$	–	$P_{11} = \dfrac{P}{D^2 H^{1.5}}$	P_{ed}	$P_{ed} = \dfrac{P}{\rho' D^2 H^{1.5}}$
Power coefficient	–	–	–	–	$P_{\omega d}$	$P_{\omega d} = \dfrac{P}{\rho' \omega^3 D^5}$
Specific speed	n_s	$n_s = \dfrac{np^{0.5}}{h^{1.25}}$	N_s	$N_s = \dfrac{np^{0.5}}{H^{1.25}}$	ω_s	$\omega_s = \dfrac{\omega Q^{0.5}}{(g'H)^{0.75}}$
Conversion term	n_s	$n_s = 0.262\, N_s$	N_s	$N_s = 166\sqrt{\eta\omega_s}$	ω_s	$\omega_s = \dfrac{N_s}{43.5\sqrt{\eta}}$

[a] h, net head, ft of water; H, net head, m of water; d, runner diameter, in.; D, runner diameter, m; q, discharge, ft^3/sec; Q, discharge, m^3/sec; ω, angular velocity, rad/sec; T, torque kg – m; g', acceleration due to gravity, m/sec^2; ρ', mass of density of water, kg/m^3; η, efficiency.

from Eq. (3.9) and

$$\omega = \frac{2\pi n}{60}$$

in Eq. (4.22) we have

$$\omega_s = \frac{\omega Q^{1/2}}{(g'H)^{3/4}} = \left(\frac{2\pi n}{60}\right)\frac{(P/9.806\, H\eta)^{1/2}}{(g'H)^{3/4}}$$

Inserting $P = 0.746\ p$ and $H = 0.3048\ h$ in order to convert kilowatts to horsepower and meters to feet, and collecting terms we have

$$\omega_s = \frac{2\pi\sqrt{0.746}}{60(9.806)^{5/4}\,(0.3048)^{5/4}}\,\frac{n\sqrt{p}}{h^{5/4}}\,\frac{1}{\sqrt{\eta}}$$

but from Eq. (4.17)

$$\frac{n\sqrt{p}}{h^{5/4}} = n_s$$

Therefore

$$\omega_s = \frac{n_s}{43.5\sqrt{\eta}} = \Omega$$

TURBINE CONSTANTS AND EMPIRICAL EQUATIONS

Hill Curves

Manufacturers working with the similarity laws and using the various turbine constants have developed what are known as *hill curves.* These are three-dimensional graphical representations of the variation of various turbine constants as related to common parameters of head, speed, and power output. A typical representation of a hill curve is shown in Fig. 4.1. This is a plot of several different parameters (turbine constants) on a single performance map. The parameters used in the hill curve of Fig. 4.1 are as follows:

$$Q_{\omega d} = \frac{Q}{\omega D^3} \tag{4.25}$$

$$E_{\omega d} = \frac{g'H}{(\omega D)^2} \tag{4.26}$$

$$P_{\omega d} = \frac{P}{\rho'\omega^3 D^5} \tag{4.27}$$

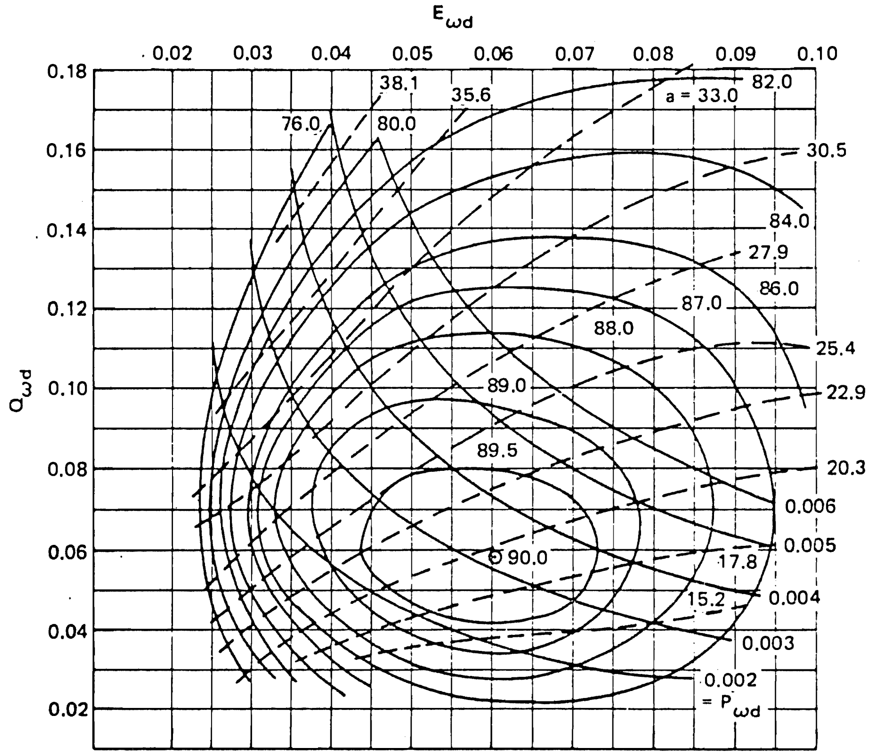

Figure 4.1 Typical turbine hill curve (dimensionless form). SOURCE: Allis-Chalmers Corporation.

where $Q_{\omega d}$ = discharge coefficient, dimensionless
$E_{\omega d}$ = energy coefficient, dimensionless
$P_{\omega d}$ = power coefficient, dimensionless

and the Q, ω, D, g' ρ', H, and P are as previously defined, expressed in metric SI units. The discharge coefficient, $Q_{\omega d}$, is plotted against the energy coefficient, $E_{\omega d}$, and a contour map of lines of equal power coefficient, $P_{\omega d}$, values are generated. Superimposed on this performance map are contour lines for equal turbine efficiency that correspond to the simultaneous values of $Q_{\omega d}$ and $E_{\omega d}$ and also the corresponding values of the gate-opening a. Early versions of hill curves use unit power, p_1, and unit speed ratio, ϕ, as the respective ordinate and abscissa parameter in performance mapping. Figure 4.2 is representative of a hill curve developed using p_1 and ϕ as the variables.

Hill curves are developed from extensive tests of model turbines, and the homologous nature of turbines makes it possible to predict prototype performance from the turbine constants. For instance, it should be noted that the energy coefficient, $E_{\omega d}$, is directly proportional to net head, H, for a given prototype diameter, D, and speed of runner, ω. These hill curves are used in making final design selection and for verifying performance guarantees. Normally, these hill curves are proprietary information of a particular manufacturer.

In actual practice it is found that there is not precise equality between the model and the prototype with regard to efficiency due to differences in boundary layer friction and turbulence effects. To correct for this, an efficiency step-up equation is normally used, of the form

$$\frac{1-\eta_p}{1-\eta_m} = \left(\frac{D_m}{D_p}\right)^a \tag{4.28}$$

where η_m = model turbine efficiency
η_p = prototype turbine efficiency
D_m = model turbine diameter
D_p = prototype turbine diameter
a = step-up coefficient.

Moody (1926) proposed that a have a value of 0.2. Kovalev (1965) lists the

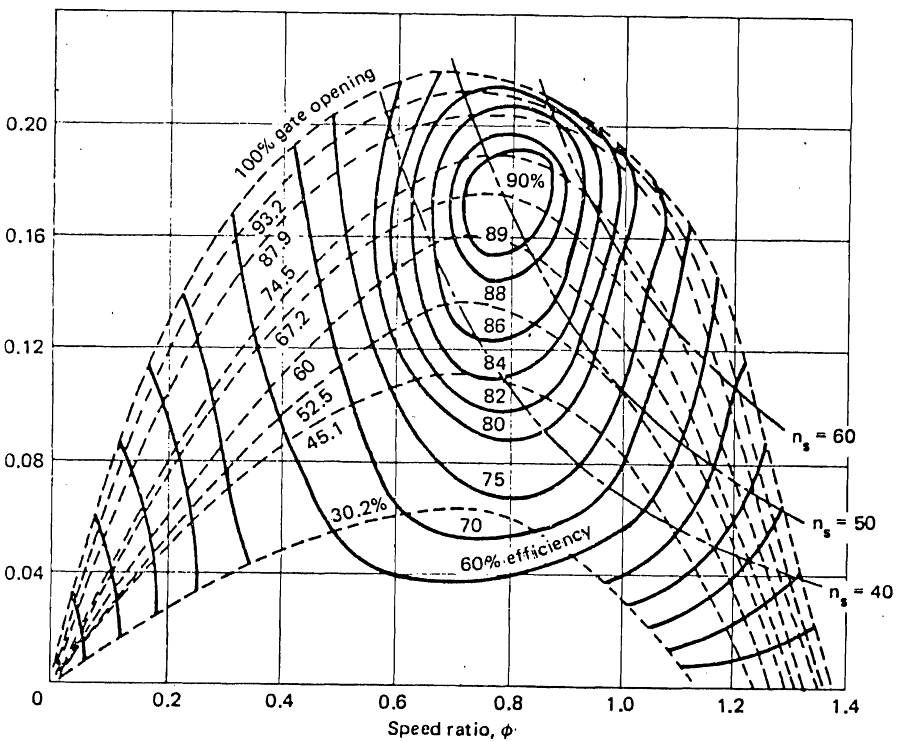

Figure 4.2 Representative turbine hill curve.

following equation for predicting efficiency of prototype turbines from model performance:

$$\eta_p = 1 - (1 - \eta_m)\left(\frac{D_m}{D_p}\right)^{m-1}\left(\frac{H_m}{H_p}\right)^{1-0.5n} \tag{4.29}$$

where the coefficients m and n are related to the coefficients used in calculating head losses in the described system.

More recent work of Hutton (1954) and Hutton and Salami (1969) have related the efficiency step-up to the Reynolds number of the model and prototype for different losses within a turbine system, including the casing, the guide vanes, the turbine, and the draft tube. This alternative form of the step-up equation is

$$\frac{\eta_p}{\eta_m} = 1 - K\left[1 - \left(\frac{R_m}{R_p}\right)^{1/a}\right] \tag{4.30}$$

where K = a coefficient shown to vary from 0.5 to 0.81
R_m = Reynolds number for the model turbine
R_p = Reynolds number for the prototype turbine.

Sheldon (1982) has presented data on the efficiency step-up relations showing actual model-prototype test data from numerous U.S. Army Corps of Engineers installations. Results of that study give specific values for the exponent coefficient in the Moody form of the step-up equation.

Experience Curves

In preliminary design and feasibility studies, it is often necessary for the engineer to get information on power output of a given plant, the synchronous speed, and an estimate of runner diameter to determine preliminary costs and to decide on particular arrangements for hydropower plant units before the final selection is made. Extensive experience curves have been developed for this purpose. Because the specific speed is an integrated universal number it has been used as the means of relating other needed parameters to a common base characteristic. The logical relation to be developed is the relation between specific speed, n_s, and net effective head, h, because a site is normally developed for a particular average head.

An excellent treatise of these experience curves is presented in a series of articles on Francis turbines (deSiervo and deLeva, 1976), on Kaplan turbines (deSiervo and deLeva, 1978), and on Pelton turbines (deSiervo and Lugaresi, 1978). The U.S. Department of the Interior (1976) has also presented information relating specific speed to the design head and other parameters of design. Figure 4.3 is a comprehensive compilation of these various empirical relations that has been developed and plotted so that different forms of the specific speed equation can be used for making preliminary planning and design studies. Example 4.1 will show how this set of curves can be used.

It is useful to determine just what type of turbine is suitable for a particular

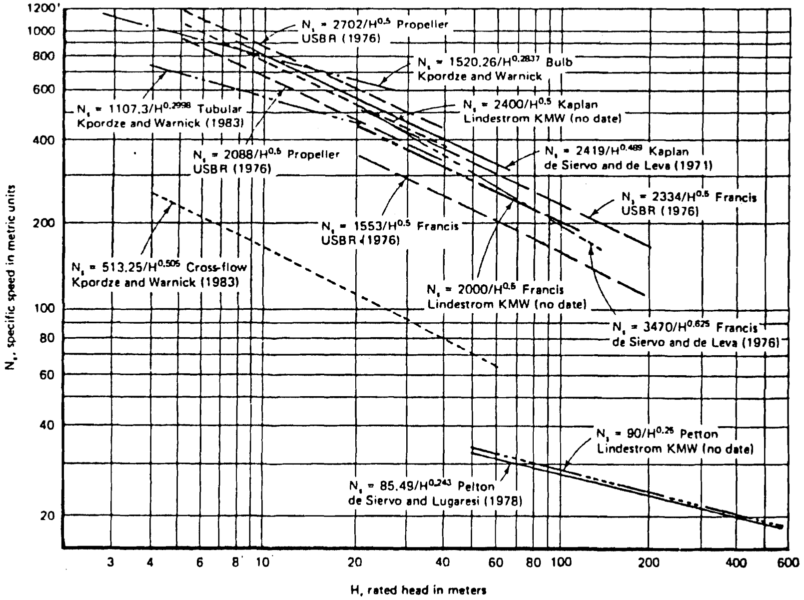

Figure 4.3 From Kpordze, C. S. K. and C. C. Warnick, "Experience Curves for Modern Low-Head Hydroelectric Turbines," Research Technical Completion Report, Contract No. 81-V0255 for Bureau of Reclamation, U.S. Department of the Interior, Idaho Water and Energy Resources Research Institute, University of Idaho, Moscow, Idaho, May 1983.

development, and a simplified approach is valuable. The applications chart in Fig. 4.4 gives a graphical means of defining the range of practical use for various types of turbines.

Speed

To make necessary calculations for determining runner speed, it should be determined in a special way if a synchronous speed must be used to drive the generator. If the turbine is directly connected to the generator, the turbine speed, n, must be a synchronous speed. For turbine speed, n, to be synchronous, the following equation must be fulfilled:

$$n = \frac{120(f)}{N_p} \tag{4.31}$$

where n = rotational speed, rpm

f = electrical current frequency, hertz (Hz) or cycles/sec; this is normally 60 Hz in the United States

N_p = number of generator poles; multiples of four poles are preferred, but generators are available in multiples of two poles.

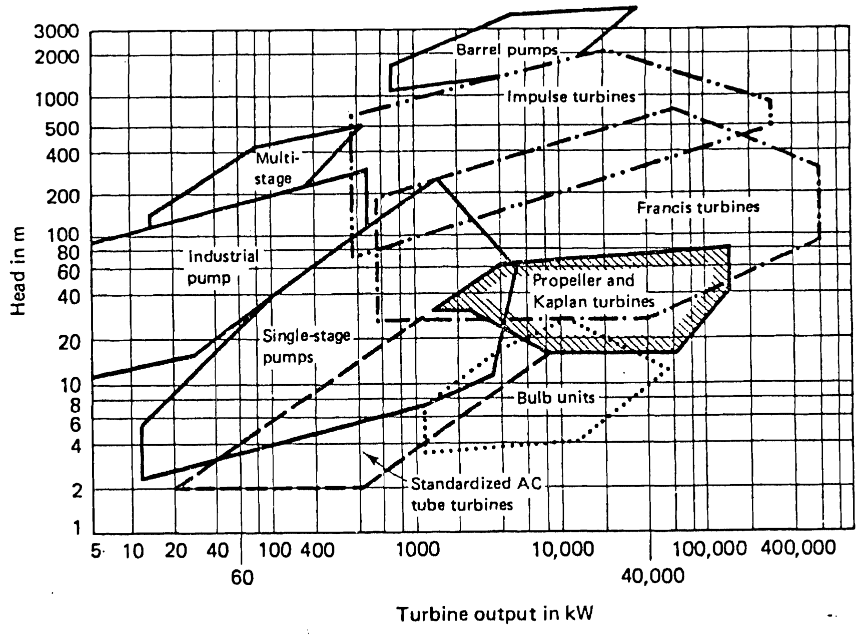

Figure 4.4 Application ranges for conventional hydraulic turbines. SOURCE: Allis-Chalmers Corporation.

If the net head varies by less than 10%, it is customary to choose the next greater synchronous speed to the calculated theoretical speed found from the specific speed equation. If the head varies more than 10%, the next lower synchronous speed should be chosen.

Diameter

To make estimates of turbine diameter, it is again necessary to depend on empirical equations or experience curves that have been developed from statistical studies of many already constructed units. It is customary to relate the variable of diameter to the universal number, the specific speed, n_s, N_s, ω_s, or Ω.

The work of deSiervo and deLeva (1976) shows the following equation for the Francis runner:

$$D_3 = (26.2 + 0.211N_s)\frac{\sqrt{H}}{n} \tag{4.32}$$

where D_3 = discharge or outlet diameter, m. (*Note:* This is different from throat diameter.)

In conventional American system of units and constants the equation becomes

$$d_3 = (569.5 + 17.4n_s)\frac{\sqrt{h}}{n} \tag{4.33}$$

where d_3 = discharge diameter, in.

For Francis runners the U.S. Department of the Interior (1976) gives a similar equation, of the form

$$d_3 = 104.65 n_s^{2/3} \frac{\sqrt{h}}{n} \tag{4.34}$$

For propeller turbines, deSiervo and deLeva (1977) show the following equation for determining design diameter:

$$D_M = (66.76 + 0.136 N_s) \frac{\sqrt{H}}{n} \tag{4.35}$$

where D_M = outer diameter of propeller, m. (This is throat diameter minus clearance.) In the conventional American system of units and constants, the equation becomes

$$d_M = 115.67 n_s^{2/3} \frac{\sqrt{h}}{n} \tag{4.36}$$

where d_M = outer diameter of propeller, in.

For Pelton turbines, deSiervo and Lugaresi (1978) show that the following equations can be used for estimating the turbine diameter:

$$\frac{D_3}{D_2} = 1.028 + 0.137 N_{sj} \tag{4.37}$$

$$\frac{D_j}{D_2} = \frac{N_{sj}}{250.74 - 1.796 N_{sj}} \tag{4.38}$$

$$N_{sj} = \frac{(P/i)^{0.5}}{H^{1.25}} \tag{4.39}$$

where N_{sj} = specific speed for impulse runner per jet
i = number of jets used by impulse turbine
D_2 = wheel pitch diameter, m
D_3 = outer wheel diameter, m
D_j = jet diameter, m
P = turbine rated capacity, kW.

Doland (1954) gives the following equation for determining the size of Pelton wheels:

$$d_2 = 830 \frac{\sqrt{h}}{n} \tag{4.40}$$

where d_1 = diameter of circle passing through the centers of the buckets (pitch diameter), in.

The literature of deSiervo and others gives much greater detail on other design features and provides a basis for providing information that is necessary in determining the actual shape of the wheel and outside controlling dimensions, the dimensions of the draft tube, and other portions of the characteristic dimensions of the civil works that are necessary to enclose the turbines. Chapter 11 will treat elements of civil works powerhouse design.

For illustration purposes, Example 4.1 has been worked out to show the use of the turbine constants, the typical specific speed experience curves, and the empirical equations that relate diameter of the turbine runner to the specific speed. Before proceeding it is necessary to consider a generalized characteristic of turbine efficiency.

Turbine Efficiency

The typical variation of the different types of turbines in their expected operating range is shown in Fig. 4.5. This figure shows typical efficiency curves for different types of turbines over the full range of power output. In this form even though somewhat vague and in some cases tied to a given installation, the information is useful to make estimates in later phases of the problem of selecting units for given sites.

Example 4.1

Given: A proposed hydropower development has a net head of 250 ft and design discharge of water flow of 580 ft^3/sec.
Required: Determine the type of turbine to be used, the normal plant capacity, the operating speed, the suitable specific speed, and the estimated turbine diameter.

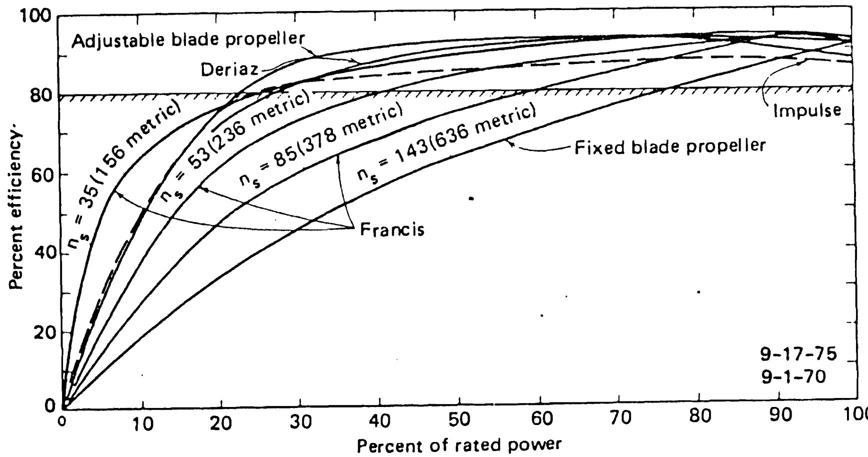

Figure 4.5 Curves of turbine efficiency variation. SOURCE: U.S. Bureau of Reclamation.

Analysis and solution: Refer to Figs. 4.3 and 4.4, which show that the unit should be in the *Francis turbine* range. From Fig. 4.5 assume an efficiency of $\eta = 0.91$. Make a trial selection from Fig. 4.3 of $n_s = 60$ or use the equation from Fig. 4.3, using the upper-limit equation:

$$\text{Trial } n_s = \frac{950}{(h)^{1/2}} = 60.1$$

Calculate the turbine power output from Eq. (4.11):

$$p = \frac{\gamma q h \eta}{550} = \frac{(62.43)(580)(250)(0.91)}{550}$$

$$= 14{,}980 \text{ hp} = 11.2 \text{ MW} \quad \text{ANSWER}$$

Solve for the trial value of n from the specific speed equation (4.17):

$$n = \frac{n_s h^{1.25}}{p^{0.5}} = \frac{(60.1)(250)^{1.25}}{(14{,}977.5)^{0.5}} = 488.2 \text{ rpm}$$

From the equation for synchronous speed, Eq. (4.30), $n = 7200/N_p$:

$$N_p = \frac{7200}{488.2} = 14.75 \qquad \text{choose 14 poles}$$

$$n = \frac{7.200}{14} = 514.3 \quad \text{ANSWER}$$

Calculate the actual specific speed, n_s, from $n_s = np^{0.5}/h^{1.25}$:

$$\text{Actual } n_s = \frac{(514.3)(14{,}980)^{0.5}}{(250)^{1.25}} = 63.32 \quad \text{ANSWER}$$

Now using Eq. (4.33), calculate the runner diameter:

$$d_3 = (569.5 + 17.4n_s)\frac{h^{0.5}}{n}$$

$$= [569.5 + (17.4)(63.32)]\frac{(250)^{0.5}}{514.3}$$

$$= 51.4 \text{ in.} \quad \text{ANSWER}$$

Now check with Eq. (4.34):

$$d_3 = 104.65 n_s^{2/3}\frac{h^{0.5}}{n}$$

$$= (104.65)(63.32)^{2/3}\frac{(250)^{0.5}}{514.3}$$

$$d_3 = 51.1 \text{ in.} \quad \text{ANSWER}$$

This appears to be a good check. Turbine manufacturers will use actual model test

data to make diameter selection after estimating specific speed based on head range, flow, and setting with respect to tailwater elevation.

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PROBLEMS

4.1. Obtain information from a nearby operating hydroplant showing such data as type, rated head, rated power capacity, design discharge, design diameter of runner, and synchronous speed, and calculate a value for specific speed, n_s, for the turbine.

4.2. Develop a turbine constant for the unit diameter.

4.3. Using the hill curve of Fig. 4.1 or 4.2 and assuming that a prototype unit is to produce 10,000 kW of turbine power at best efficiency and operate at a synchronous speed of 112.5 rpm, develop a turbine performance curve for the unit showing how efficiency and power output will vary with varying discharge. Make any necessary assumptions as to the specific speed or head to be used.

4.4. A study of a power development site shows that design discharge should be 108 m^3/sec and the design head should be 18 m. Determine the type of turbine to be used, the rated power output, a desirable runner speed, and an approximate runner diameter.

4.5. A power site has a head of 800 ft and design discharge of 15 ft^3/sec. Determine the type of turbine, the rated power output, a desirable runner speed, and a design diameter of the runner. Show what will happen to the runner diameter if the next higher synchronous speed were used. Where would the specific speed, N_s, be plotted on Fig. 4.3?

HYDROLOGIC ANALYSIS FOR HYDROPOWER 5

PARAMETERS TO BE ANALYZED

Earlier it was pointed out that the principal parameters necessary in making hydropower studies are *water discharge* and *hydraulic head.* The measurement and analyses of these parameters are primarily hydrologic problems. Remembering that hydrology is the study of the occurrence, movement, and distribution of water on, above, and within the earth's surface, it is easy to see that a part of the hydrology problem is to identify the vertical distance between the level of water in the forebay or headwater of the hydroplant and in the tailrace, where the water issues from the draft tube at the outlet to the turbine.

Thus the determination of the potential head for a proposed hydropower plant is a surveying problem that identifies elevations of water surfaces as they are expected to exist during operation of the hydroplant. This implies that conceptualization has been made of where water will be directed from a water source and where the water will be discharged from a power plant. In some reconnaissance studies, good contour maps may be sufficient to determine the value for the hydraulic head. Because the headwater elevation and tailwater elevations of the impoundment can vary with stream flow, it is frequently necessary to develop headwater and tailwater curves that show variation with time, river discharge, or operational features of the hydropower project.

The water discharge is a much more difficult problem to cope with because the flow in streams is normally changing throughout the length of the stream as tributary streams increase the flow and some diversional water uses decrease the flow. Similarly, the flow changes from one time to another due to hydrologic variation caused by the variation in precipitation, evaporation, snowmelt rate, and groundwater recharge that affects the magnitude of stream flow. Details are given later on how to treat the flow variation problem.

TYPES OF HYDROPOWER STUDIES

In hydropower studies the degree of sophistication to which the analyses are made varies with the type of study. Three types of studies are commonly made: (1) reconnaissance studies, (2) feasibility studies, and (3) definite plan or design studies.

Reconnaissance resource studies are made to find potential energy sources and to estimate the energy available in streams, and may not be too site specific. A resource-type study (Gladwell, Heitz, and Warnick, 1978) has been made in the Pacific Northwest region of the United States in which an inventory of the theoretical energy in the streams by reaches of rivers was made. This study used contour maps to determine head available in the streams, and water flow was estimated by using parametric curves of the flow duration in the streams. Some resource studies have been site specific and used mean annual runoff or a characteristic such as the 95%-of-time flow to determine flow available for energy development. More sophisticated resource evaluations were completed under the *National Hydropower Survey* of the U.S. Army Corps of Engineers (1980).

Feasibility studies are made to formulate a specific project or projects to assess the desirability of implementing hydropower development. These normally require duration flow data that give time variability of water discharge sufficiently accurate to make possible capacity sizing of the plants.

Definite plan or design studies are made before proceeding with implementation of final design and initiation of construction. These studies normally require daily or at least monthly flow data and usually require operational studies to determine energy output and economics over critical periods of low flow in the supplying river.

ACQUISITION OF HEAD AND FLOW DATA

Basic data and maps for determining forebay elevation and tailwater elevation can often be obtained from such sources as the Army Map Service, U.S. Geological Survey, State Geological Survey offices, and county government offices.

In the United States, stream discharge or flow data are usually best obtained from the U.S. Geological Survey, which is the basic water-data-gathering agency. The format will vary, but these data normally are available as mean daily flow records at a network of stream gages that usually have rather long-term records. These data usually appear as Water Resources data for [state] water year [given year]. The data are also available as computer printout and in flow duration format.

Other agencies and entities that gather flow data include:

1. U.S. Forest Service
2. Bureau of Reclamation
3. U.S. Soil Conservation Service
4. U.S. Army Corps of Engineers
5. Environmental Protection Administration
6. State water resource departments
7. Irrigation districts
8. Flood control districts
9. Water supply districts

Making correlation studies may require obtaining supporting data from precipitation, temperature, and evaporation records. Such information is normally available from the Weather Service of the U.S. Department of Commerce. In other countries, similar government entities to those named for the United States have data for making studies.

In some cases, it may be necessary to install gages and make field measurements to obtain adequate data on which to proceed with feasibility and design studies. Good references for conducting such data measurement and acquisition programs are those by the World Meteorological Organization (1970) and Buchanan and Somers (1969). The extent to which measurements and more sophisticated calculation of hydrologic data are made will depend on the time available and amount of money allocated for the hydrologic analysis.

Many times the flow data records may be incomplete, or be from locations that are upstream or downstream from the specific hydropower site, and need some adjustment to be useful. The records may be short-time records. Extrapolations and correlations with nearby gaged records may be necessary. Techniques for making such extrapolations are covered in such hydrology texts and references as Linsley, Kohler, and Paulhus (1975) and Viessman, Harbaugh, and Knapp (1972).

In some cases it may be necessary to make estimations of flow and runoff magnitude using precipitation data and estimates of runoff coefficients. Examples of how precipitation data can be used effectively in estimation of flow duration analyses will be presented in Example 5.1.

FLOW DURATION ANALYSIS

Flow Duration Curves

A useful way of treating the time variability of water discharge data in hydropower studies is by utilizing flow duration curves. A *flow duration curve* is a plot of flow versus the percent of time a particular flow can be expected to be exceeded. A flow duration curve merely reorders the flows in order of magnitude instead of the true time ordering of flows in a flow versus time plot. The flow duration curve also allows the characterizing of the flow over long periods of time to be presented in one compact curve. A typical flow duration curve for a gaged stream location in Idaho is shown in Fig. 5.1.

Two methods of computing ordinates for flow duration curves are the rank-ordered technique and the class-interval technique. The *rank-ordered* technique considers a total time series of flows that represent equal increments of time for each measurement value, such as mean daily, weekly, or monthly flows, and ranks the flows according to magnitude. The rank-ordered values are assigned individual order numbers, the largest beginning with order 1. The order numbers are then divided by the total number in the record and multiplied by 100 to obtain the percent of time that the mean flow has been equaled or exceeded during the period of record being considered. The flow value is then plotted versus the respective computed *exceedance percentage.* References to the flow duration values at specific exceedance value are usually made as Q_{50}, Q_{30}, Q_{10}, and so on, indicating the flow value at the percentage point subscripted. Naturally, the longer the record, the more statistically valuable the information that results.

The *class-interval* technique is slightly different in that the time series of flow values are categorized into class intervals. The classes range from the highest flow value in the series to the lowest value in the time series. A tally is made of the number of flows in each, and by summation the number of values greater than a given upper limit of the class can be determined. The number of flows greater than the upper limit of a class interval can be divided by the total number of flow values in the data series to obtain the exceedance percentage. The value of the flow for the particular upper limit of the class interval is then plotted versus the computed exceedance percent. A more thorough description of both of these methods and a listing of computer programs for processing such data are available in a University of Idaho Ph.D. dissertation (Heitz, 1981). Another good reference on duration curves is Searcy (1959). It is possible to obtain digital printouts of the flow duration of all streams measured by the U.S. Department of the Interior from the U.S. Geological Survey by contacting the District Chief of the Water Resources Division, U.S.G.S., in each state. Caution should be exercised in developing flow duration curves that use average values for intervals of time longer than one day. The variations in flow may be masked out if average flows for weekly or monthly intervals are used in flow duration analysis.

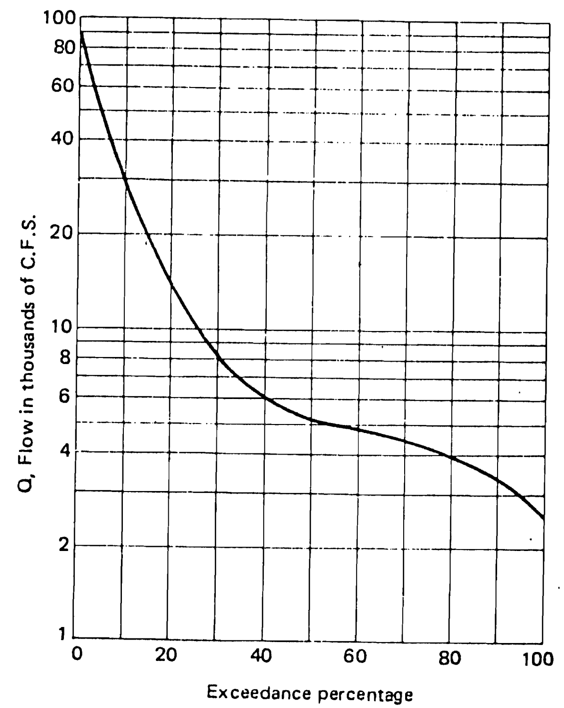

Figure 5.1 Typical flow duration curve.

Extrapolation of Flow Duration Data to Ungaged Sites

All too often the stream flow data that are available from measured gaging stations are not from the location for which a hydropower site analysis is to be made. Methods are required to develop extrapolations of measured flow duration data which will be representative of a given site on a stream. The following method for making synthetic flow duration curves at any point along a stream is based on techniques developed in a hydropower inventory of the Pacific Northwest region of the United States (Gladwell, Heitz, and Warnick, 1978). This type of analysis is particularly useful in regions where stream flow does not vary directly with the area of the contributing drainage. The procedure is to make plots of flow duration curves for all gaged streams within a rather homogeneous drainage basin, as shown in Fig. 5.2. From these flow duration curves are developed a family of parametric duration curves in which flow (Q) is plotted against the average annual runoff ($\bar{R}$), or average annual discharge, $\bar{Q}$, at the respective gages for several exceedance percentages. A separate curve is developed for each exceedance interval used. A correlation analysis is then performed to obtain the best-fitting curve for the data taken from the measured records of stream flow. The result is a parametric flow duration curve such as the one shown in Fig. 5.3.

Another method, developed by Washington State University team members working on the Pacific Northwest regional inventory, utilized the following approach (Heitz, 1978). The values of flow for each flow duration for a given exceedance point are divided by the average annual discharge, $\bar{Q}$, to give a dimensionless flow term. These are then plotted against the particular exceedance interval on logarithmic probability paper as shown in Fig. 5.4 to give a dimensionless flow duration curve. Then a best-fitting curve is developed for a particular area having homogeneous hydrology so that a single curve results that relates a characteristic dimensionless flow term to the exceedance percentage. It is easy to recognize

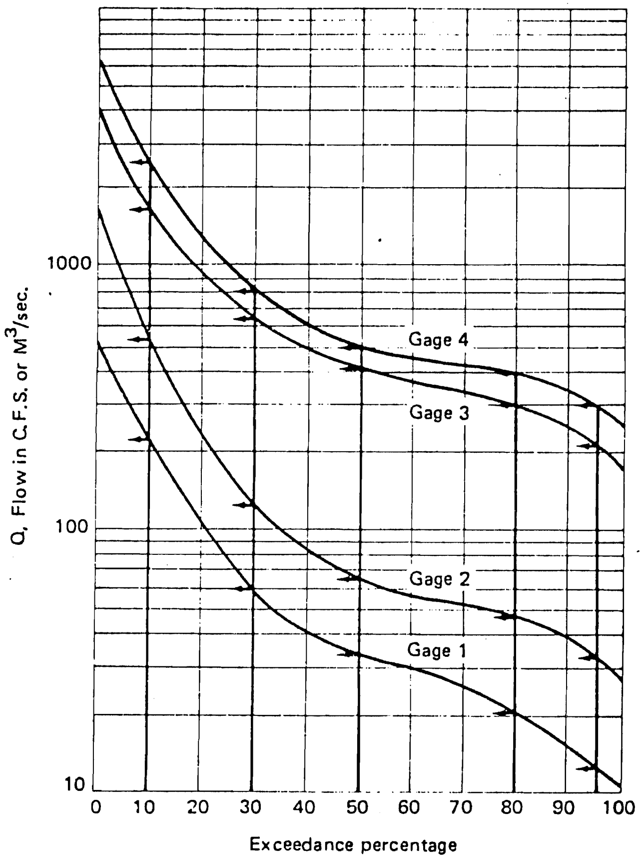

Figure 5.2 Flow-duration curves for gaging stations in a homogeneous drainage basin.

that at the limits of the curve the reliability of the curve is questionable because the number of values are minimal and these outlier values are the unusual occurrences of flash floods or extremely low flows.

Determination of Average Annual Discharge

To use the parametric flow duration curves effectively, it is necessary to determine the average annual discharge, $\bar{Q}$, at the point or location on the stream for which a hydropower analysis is to be made. A procedure for making that determination follows. First an accurate isohyetal map of normal annual precipitation (NAP) of the river basin involved must be obtained or developed. Isohyetal maps contain lines representing equal precipitation for a geographic region. Care should be taken that the map represents the same period of record as the stream flow data

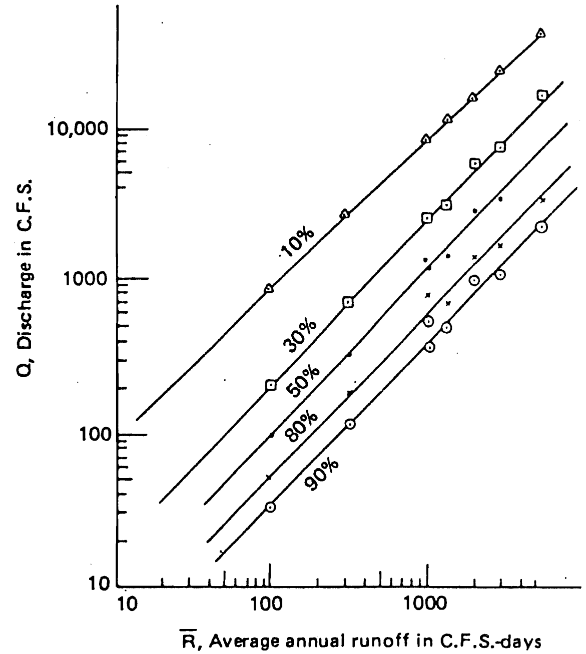

Figure 5.3 Parametric flow-duration curve (Clearwater River, Idaho).

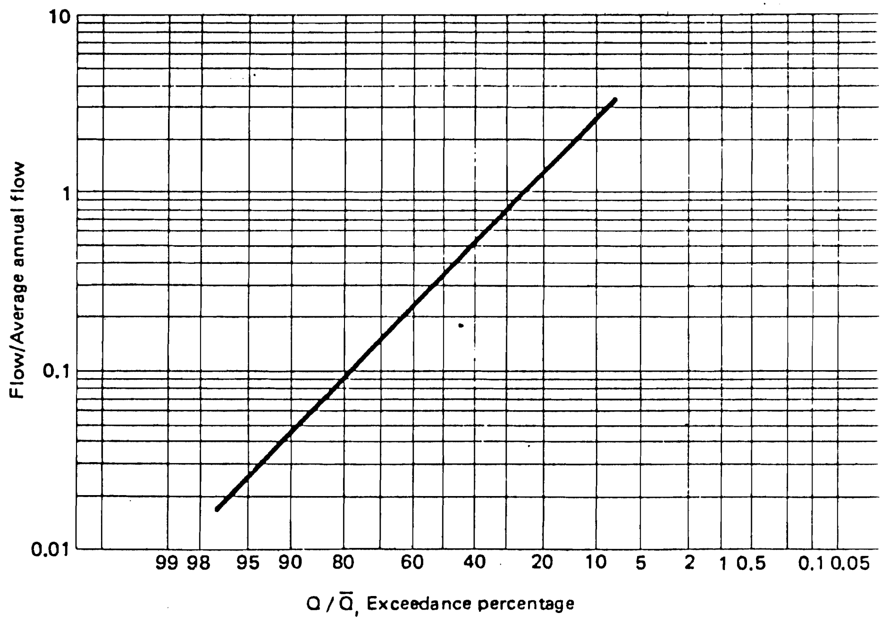

Figure 5.4 Dimensionless flow-duration curve.

for which flow duration data are available and needed. In the United States, normal annual precipitation maps are usually available from the Weather Service of the U.S. Department of Commerce or from a state water resource planning agency, a federal planning agency such as the U.S. Corps of Engineers, the Soil Conservation Service of the U.S. Department of Agriculture, the U.S. Bureau of Reclamation, or a river basin commission. The drainage basin contributing water to the site being investigated is graphically defined on the isohyetal map or a map on which the isohyetal lines have been superimposed. The individual areas between isohyetal lines are planimetered and the areas used to develop a weighted-average precipitation input to a basin on an annual basis. Example 5.1 will present details on how this is done.

Then, utilizing the records of average annual precipitation input to the basins at measured streams nearby or having similar hydrologic characteristics, a runoff coefficient is estimated for the drainage basin being studied. This value can be rather subjective in determination and thus represents a place for making a considerable error. Much care should be exercised in estimating the annual runoff coefficient. The product of this coefficient and the computed normal annual precipitation input to the basin and the basin area can be used to calculate the average annual discharge using the formula

$$\bar{Q} = \frac{K\bar{P}A}{12(31{,}536{,}000)} \tag{5.1}$$

where $\bar{Q}$ = average annual discharge, ft^3/sec
K = annual runoff coefficient as a decimal value
$\bar{P}$ = weighted average annual precipitation, in.
A = drainage area, ft^2
12 = conversion for converting precipitation to ft
31,536,000 = number of seconds in one year.

The annual average runoff term, $\bar{R}$, is sometimes used in computer processing instead of the average annual discharge, $\bar{Q}$. $\bar{R}$ is expressed in cubic feet per second-days [(ft^3/sec)(day)]. This is the summation of the mean daily discharges for all the days of the year. This saves making an extra calculation that involves the constant term of 365 days.

With the average annual discharge estimate it is possible to enter the parametric flow duration curve and determine values of flow for different exceedance percentages for which the parametric flow duration curve has been developed. Example 5.1 has been worked out to illustrate the methodology explained.

Example 5.1

Given: A stream location on the Clearwater River in Idaho has been identified for making a hydropower analysis. The location is at a point where no stream flow record is available. A parametric flow duration curve has been developed for the streams in the river basin being studied and is shown in Fig. 5.3. A normal-annual-precipitation map showing the isohyetal lines is presented in Fig. 5.5. The plani-

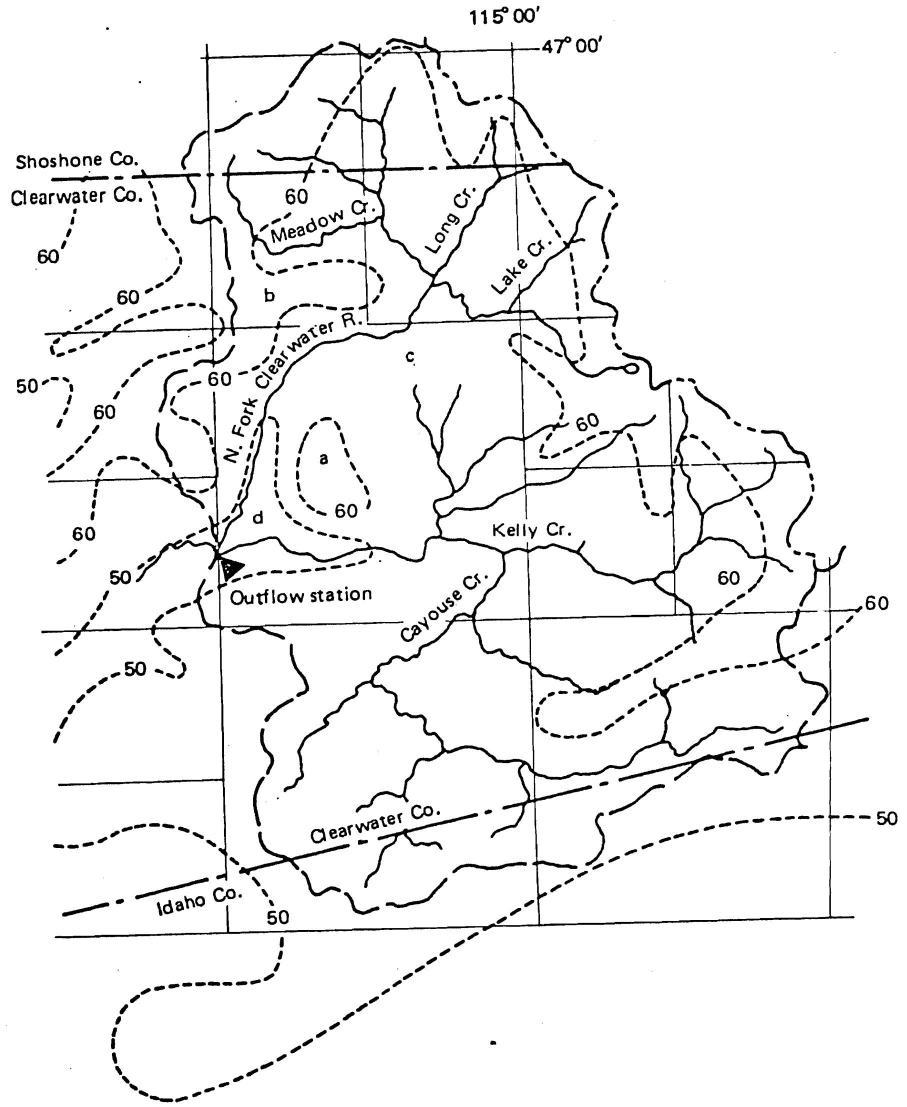

Figure 5.5 Normal-annual-precipitation map for portion of Clearwater River Basin, Idaho.

metering of the respective areas between isohyetal lines for the map areas of Fig. 5.3 is indicated in Table 5.1 and an annual runoff coefficient based on the work of Emmert (1979) has been estimated to be 0.73.

Required: Determine the average annual discharge at the marked location and develop ordinate values for a flow duration curve at the site designated.

TABLE 5.1 Values of Planimetered Areas from the Normal Annual Precipitation Map of Drainage Basins in the Clearwater River System

Area Designation	Average Value of Precipitation Between Isohyetal Lines (in.)	Planimetered Area on Map (in^2)	Percent of Total Area
a	60	0.46	1.24
b	60	8.16	22.01
c	55	27.41	73.92
d	50	1.05	2.83
		37.08	100.00

Analysis and solution: First, determine average annual precipitation input to basin using data from Table 5.1.

$$\bar{P} = \frac{P_a A_a + P_b A_b + P_c A_c + P_d A_d}{A_a + A_b + A_c + A_d} \tag{5.2}$$

where $\bar{P}$ = weighted average precipitation, in.
P_a = precipitation in area *a*, in.
P_b = precipitation in area *b*, in.
P_c = precipitation in area *c*, in.
P_d = precipitation in area *d*, in.
A_a = area planimetered to represent area *a*, in^2
A_b = area planimetered to represent area *b*, in^2
A_c = area planimetered to represent area *c*, in^2
A_d = area planimetered to represent area *d*, in^2

Using Eq. (5.2) and data from Table 5.1, the following computation results:

$$\frac{60(0.46) + 60(8.16) + 55(27.41) + 50(1.05)}{37.08} = 56.02 \text{ in.}$$

Convert this to volume units of runoff per year = *R*. The map used had a scale of 1:250,000, so

$$R = \frac{(56.02)(37.08)(250{,}000)^2}{144 \times 12}(0.73)$$

The 0.73 is the runoff coefficient.

$$R = 5.485 \times 10^{10} \text{ ft}^3/\text{yr}$$

Computing the average flow per year, $\bar{Q}$, it is necessary to divide by the number of seconds per year:

$$\bar{Q} = \frac{5.485 \times 10^{10}}{365(24)(60)(60)} = 1739 \text{ ft}^3/\text{sec}$$

Because the parametric flow duration curve was developed on the basis of average annual runoff expressed in (ft^3/sec)(day) units, it is necessary to convert $\bar{Q}$

to $\bar{R}$(ft^3/sec)(day) which is done by multiplying by 365 (the number of days in a year), so

$$1739 \text{ ft}^3/\text{sec} \times 365 = 643.7(\text{ft}^3/\text{sec})(\text{day})$$

Entering the parametric flow duration curve of Fig. 5.3 or using the regression equation for each specified exceedance percentage, it is possible to arrive at the following values for the ordinates of a specific flow duration curve for flow at the outflow station designated on the map of Fig. 5.5:

$$Q_{95} = 240 \text{ ft}^3/\text{sec} \quad \text{ANSWER}$$
$$Q_{80} = 360 \text{ ft}^3/\text{sec}$$
$$Q_{50} = 690 \text{ ft}^3/\text{sec}$$
$$Q_{30} = 1468 \text{ ft}^3/\text{sec}$$
$$Q_{10} = 5214 \text{ ft}^3/\text{sec}$$

Regulated Flow Considerations

The preceding discussion is based on the assumption that there has been no appreciable regulation of the natural flow of the stream. In many parts of the country there are storage reservoirs that have by their operations altered the flow of the river. It is still possible to use a flow duration analysis if the entire sequence of regulated flow data of a long time period can be obtained or generated by reservoir operation studies. The entire record then must be subjected to either the rank ordered technique or the class-interval technique.

It may be necessary in some cases to combine the flow records of a regulated flow stream with the flow of an ungaged natural stream to make hydropower analysis. Heitz and Emmert (1979) and Emmert (1979) have developed a technique for obtaining the necessary stream flow values and for generating a combined-value flow duration curve.

The basic approach can be explained by referring to the physiographic layout of Fig. 5.6. In this case a measured record for a considerable length of time is assumed to be available at reservoir outlet *A*. The location for which flow data are needed is at point *B*. The flow at *B* is the inflow from an area of considerable extent where there is no stream gage record, plus inflow from the operations of a reservoir at station *A*. A normal annual precipitation map of the entire drainage area is required. Also, records from a nearby stream gage (station *C*) on an unregulated stream that can be considered to represent the sequential variation of runoff from drainage area *M* (crosshatched area) are required. These long-time records must cover the same period for which regulated flow data are available at station *A*.

First an estimate must be made of the average annual runoff from area *M*. This is done by planimetering the isohyetal map of normal annual precipitation as explained in Example 5.1 and getting the normal annual water input into area *M*, as the volume of water per year. Then a coefficient of runoff for the area on an annual basis must be estimated. This can be done by referring to records of nearby gages on streams that have essentially the same hydrologic characteristics. Multiplying the

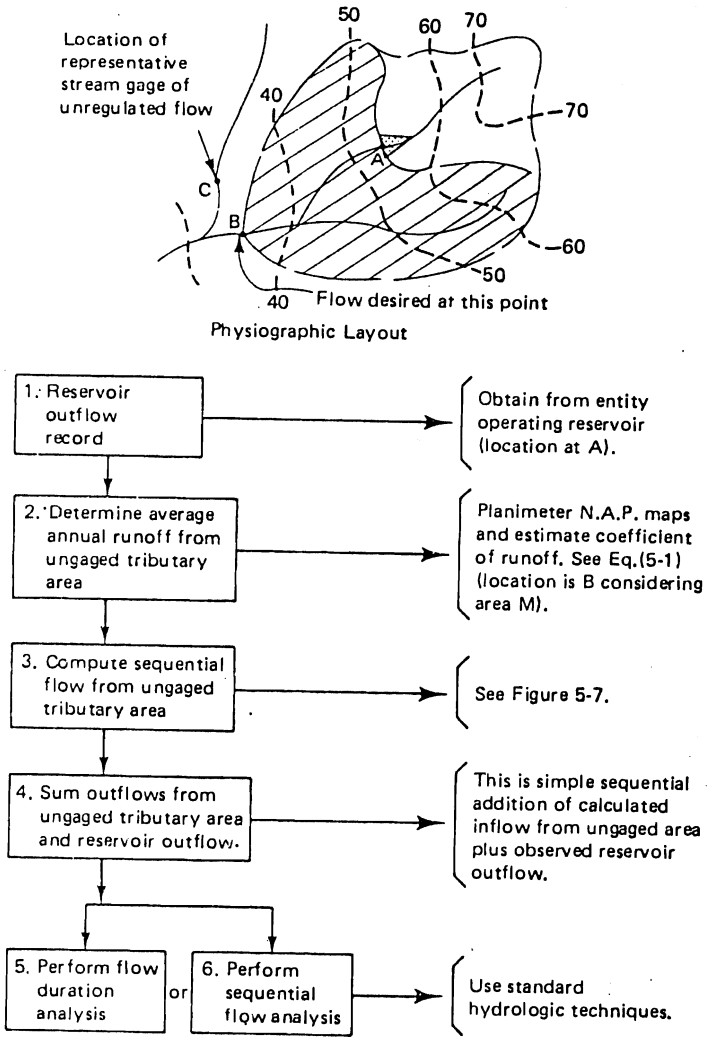

Figure 5.6 Diagram showing method for determining flow duration of regulated flow combined with ungaged inflow.

ormal annual precipitation input value by the runoff coefficient gives the average nnual runoff from the area M. Mathematically, this is indicated as follows:

$$\bar{Q}_M = \bar{P} A_m f_r \tag{5.3}$$

here $\bar{Q}_M$ = average annual runoff volume per year from area M
$\bar{P}$ = average annual precipitation, depth units

A_m = area of the contributing drainage, square-length units
f_r = annual runoff coefficient, a decimal fraction of amount of precipitation that runs off the drainage.

A sequence of flows coming off area M must be computed. The time increments or periods must correspond to the records of discharge available from the reservoir operation. First a flow record at station C must be obtained and studied. The record at C is assumed to have the same time distribution of flow as the runoff coming off area M (refer to Fig. 5.6). An incremental fraction of flow, a_i, for each increment of time in the total desired time period must be obtained for the representative gage C. Figures 5.6 and 5.7 give flow diagrams for a step-by-step process to calculate the sequential inflow from the ungaged area labeled M in Fig. 5.6. Once the sequential flows have been computed it is a simple procedure to add, sequen-

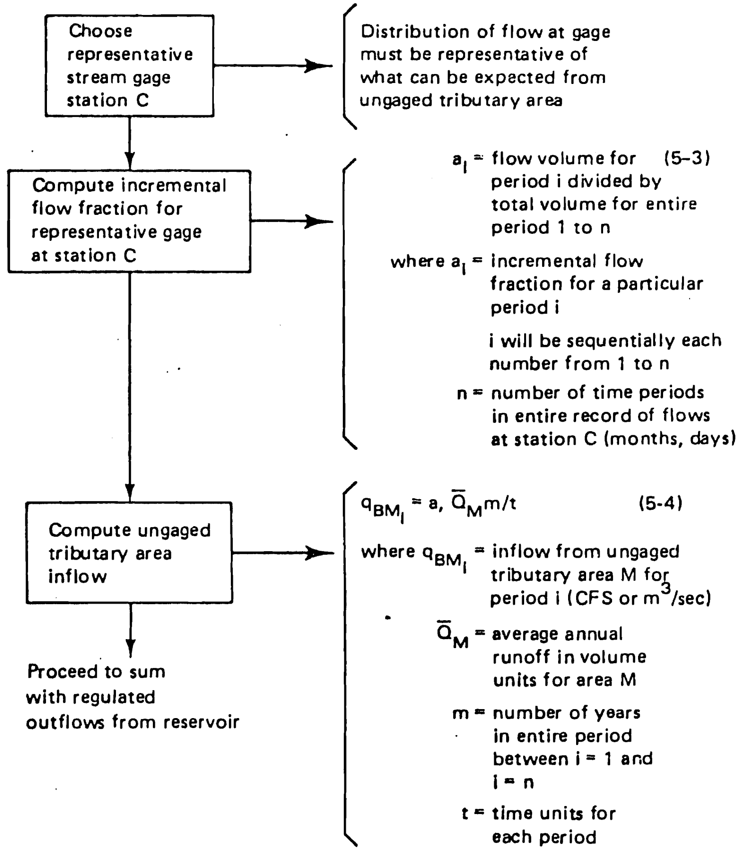

Figure 5.7 Flow diagram for computing sequential flow magnitude from ungaged tributary area.

Outflows from upstream reservoirs

Flows in CFS

Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
50	45	80	100	200	250	250	200	70	65	60	55

Compute average annual runoff from tributary area using NAP maps or by other methods

Average runoff from tributary area = 43,000 CFS-days

Choose representative gage

Flow, CFS

Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
25	20	40	50	100	300	75	60	50	40	30	25

Runoff cfs-days = flow X number of days in month

e.g. Flow (Jan) = 25 cfs X 31 days 775 cfs-days

Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
775	560	1240	1400	3100	9000	2325	1860	1500	1240	900	775

Total runoff for period of record = $\sum_{i=1}^{n} R.O._i$ n = total months in record

775 + 560 + 1240 + 1500 + 3100 + 9000 + 2325 + 1860 + 1500 + 1240 + 900 + 775 = 24,775 cfs-day

Compute flow fraction for representative gage

$$MFF_i = \frac{(R.O.\ \text{for month})_i}{\text{Total runoff for period}} \qquad i = \text{month}$$

$$\text{e.g. MFF (Jan)} = \frac{775\ \text{cfs-day}}{24775\ \text{cfs-day}} = 0.0313$$

Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
0.0313	0.0226	0.0501	0.0565	0.1250	0.3633	0.0938	0.0751	0.0605	0.0501	0.0363	0.0313

Compute tributary inflows

$$\text{Inflow} = \frac{\text{Fract}_i \times \text{avg. annual R.O. from trib. area} \times \text{years in record}}{\text{Days in month}_i}$$

$$\text{e.g. Inflow (Jan)} = \frac{0.0313 \times 43000 \times 1}{31} = 43.4$$

Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
43.3	34.7	69.5	81.0	173.4	520.7	130.1	104.2	86.7	69.5	52.0	43.4

Computer flows at desired point

$\text{Flow}_i = \text{outflow}_i$ from reservoir + tributary inflow

e.g. flow (Jan) = 50 cfs + 43.4 cfs = 93.4 cfs

Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
93.4	79.7	149.5	181.0	373.4	770.7	380.1	304.2	156.7	134.5	112.0	98.4

Figure 5.8 Numeric example of calculations for regulated flow combined with ungaged area inflow.

tially, the flow from the ungaged tributary area to the regulated flows. Figure 5.8 is a numerical example using the technique described above. The increment of time is a month, with the regular number of days in each month being appropriately accounted for in the calculations. Care should always be taken that the correct volume and time units are used in these calculations. With the runoff or flow value for each increment of time i through n generated, it is then possible to generate a flow duration curve for the point B following either the rank-ordered technique or the class-interval technique. A problem using this approach with real data has been included at the end of the chapter to make this more meaningful. More detail on this procedure can be found in Heitz (1981).

Deterministic and Stochastic Flow Methods

With appropriate data on precipitation, antecedent conditions, soil conditions, and terrain characteristics, it is possible to generate flow data for use in hydropower analyses. Numerous deterministic models of varying sophistication are now available to make simulations of hydrologic runoff. Two valuable references are the SSAR model (U.S. Army Corps of Engineers, 1972) and the Stanford Watershed Model (Crawford and Linsley, 1966) and its various improvements, known as the Hydrocomp Simulation Program.

Stochastic models approach the time distribution of flow as statistical and treat the occurrences as probability distributions. With good historical data it is possible to generate a time series of flow data of any length. An excellent reference on this is the work of Haan (1977).

The questions to be asked regarding the use of hydrologic stream flow simulation models are: (1) Do the basic historical and physical data on the river under study justify the use of models? (2) Do the time and cost permit their use?

ENERGY AND POWER ANALYSIS USING A FLOW DURATION APPROACH

In processing regulated and unregulated flow data, it is important to recognize that in the power equation, Eq. (3.8), flow is the primary limiting factor. When a run-of-river type of power study is done and a flow duration analysis is used, the capacity or size of hydropower units determines the maximum amount of water that will go through the unit or units. This is dictated by the nominal runner diameter and the accompanying outlet area and draft tube. A flow duration curve is used to explain discharge capacity, Q_c, as labeled in Fig. 5.9. This Q_c is the discharge at full gate opening of the runner under design head. Even though to the left of that point on the duration curve the stream discharge is greater, it is not possible to pass the higher discharges through the plant. If the reservoir or pondage is full, water must be bypassed by a spillway.

To the right of the runner discharge capacity point, Q_c, it should be noted that all the water that can go through the turbine is the amount flowing in the

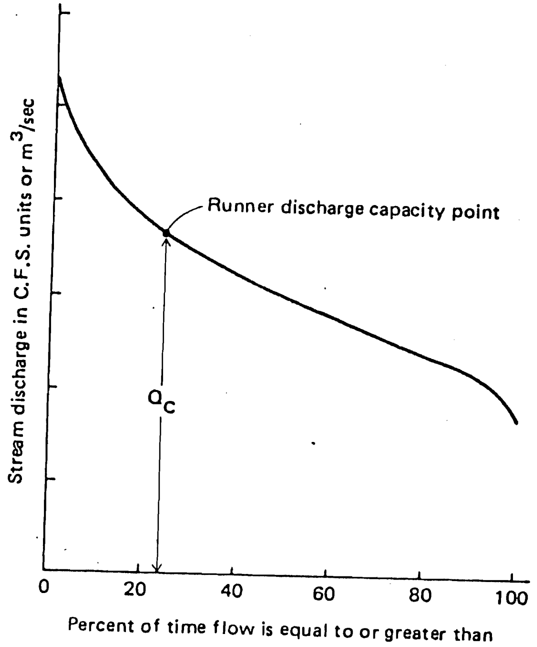

Figure 5.9 Flow-duration curve showing discharge capacity value.

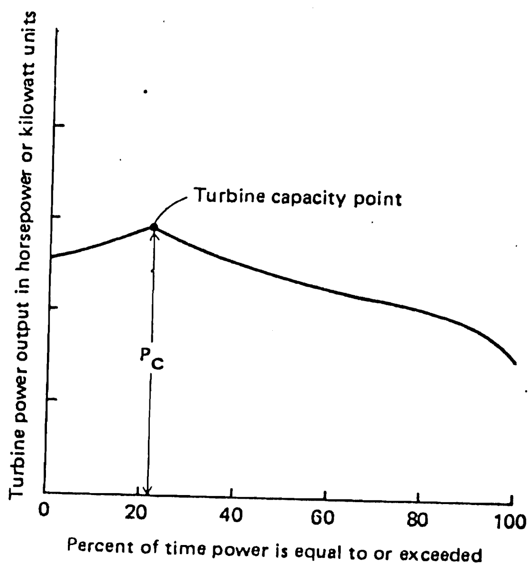

Figure 5.10 Power duration curve.

stream at the particular percent of time point. This shows that full-rated power production will not be produced. With pondage it is possible to alter this for short periods of time, but the total amount of energy output cannot be increased.

If hydraulic head and the expected losses in the penstock are known, it is possible to generate a power duration curve from the flow duration curve. Figure 5.10 gives an example of a power duration curve. The P_c value is the full-gate discharge value of power and comes from multiplying Q_c, discharge capacity value, by the simultaneous value of head, the estimated turbine efficiency, and the appropriate conversion constants [see Eq. (3.9)]. Energy production for a year or a time period is the product of the power ordinate and time and is thus the area under the power duration curve multiplied by an appropriate conversion factor. It is conventional to use the unit of energy measurement as the kilowatt-hour, kWh. An example tabulation of how the calculations might be made is shown in Table 5.2. It should be noted that to the left of the power capacity point the power tends to decrease. This is due to the fact that net head available is decreasing due to a rising tailwater caused by the higher flows that are occurring during that time interval or exceedance period. The calculations in Table 5.2 reflect this situation.

TABLE 5.2 Computational Table for Power Capacity

	Duration (%)					
	0	10	15	20	30	40
River discharge (ft^3/sec)	8000	3400	2700	2150	1550	1150
Head (ft)	80.0	81.6	83.0	83.5	83.5	83.5
Plant discharge (ft^3/sec)	2662	2689	2700	2150	1550	1150
Efficiency	0.88	0.89	0.90	0.90	0.88	0.87
Power (kW)	15,855	16,704	16,920	13,588	9636	7068
	Duration (%)					
	50	60	70	80	90	100
River discharge (ft^3/sec)	850	650	500	420	400	100
Head (ft)	83.5	83.5	83.5	83.5	83.5	83.5
Plant discharge (ft^3/sec)	850	650	500	420	400	100
Efficiency	0.87	0.83	0.75	0.70	0.60	0.50
Power (kW)	5224	3811	2649	2077	1695	353

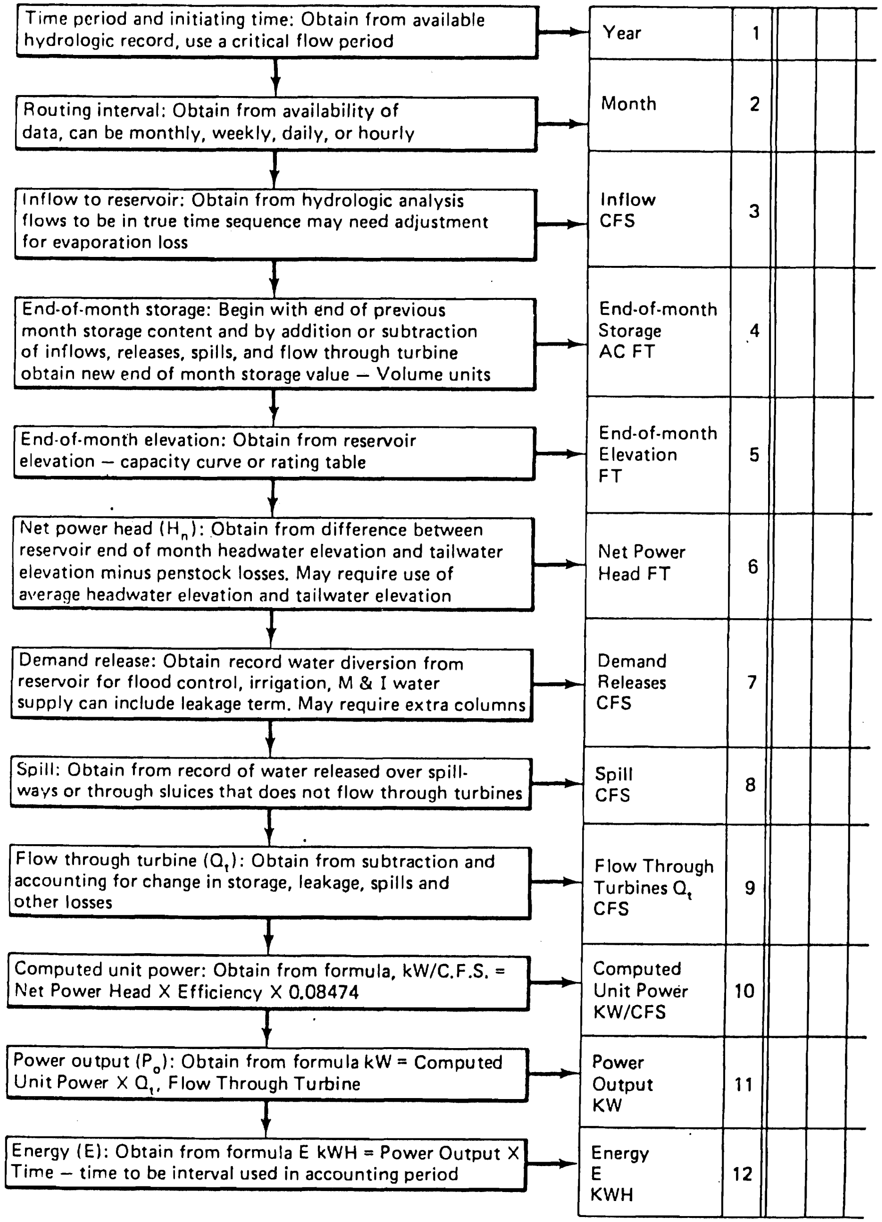

Figure 5.11 Tabular form and flow diagram for sequential flow analysis.

SEQUENTIAL-FLOW METHODS

In some cases it is wise to do a true time analysis of the hydrologic portion of a hydropower study. In cases where flow regulation and storage operation are important, it is necessary to resort to a tabular operations study. This is often referred to as a *sequential-flow* study.

Such a study entails the same problem found in a flow duration study, getting discharge data at the desired location. This means getting records of flow into the reservoir and the use of appropriate hydraulic head, area-capacity curves, and operational decisions on how the water will be released through the turbines or bypassed over a spillway. Losses such as evaporation and leakage may need to be taken into account. Figure 5.11 presents a typical tabular form for such a study with an explanatory flow diagram to show how such a sequential-flow study can be organized.

OTHER HYDROLOGIC CONSIDERATIONS

Hydrologic information is also needed for developing tailwater curves, area-capacity curves for reservoirs, rule curves for operating reservoirs, determination of seasonal losses from reservoirs due to evaporation, and flood analysis for spillways. Brief discussions on each of these topics are presented together with definitive references for making studies of this type related to hydropower studies.

Tailwater Relationships

As releases of water over spillways and any other releases into the stream immediately below a hydropower plant are made, the tailwater elevation below the outlet to the turbines will fluctuate. Therefore, it is important to develop a tailwater elevation versus river discharge curve over the complete range of flow that is to be expected. Figure 5.12 is a typical example of a tailwater rating curve. Preparing such a curve requires an adequate contour map of the channel area and an estimate of velocity in the channel at various stages of flow. Information on normal tailwater, maximum tailwater, and minimum tailwater elevations is necessary to determine design head and to determine the appropriate turbine setting. Estimates of stream channel velocity can be made using slope-area calculations that involve the conventional Manning's open-channel-flow equation.

Area-Capacity Curves

Most hydropower developments involve an impoundment behind a dam. As the water in storage in the impoundment is released the headwater elevation changes and this will influence the design of the plant and the pattern of operation. Therefore, it is necessary to have a storage or pondage volume versus impoundment surface elevation curve or table. At the same time there is need to know water

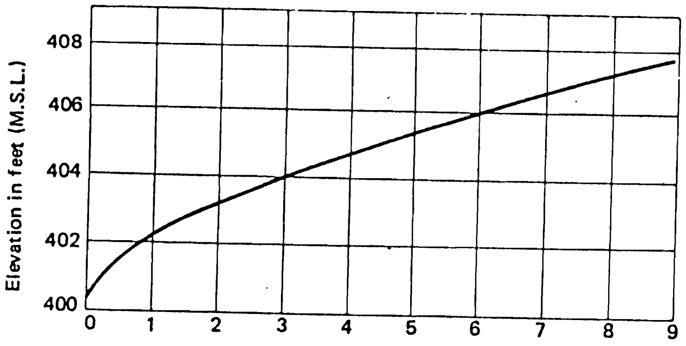

Figure 5.12 Typical tailwater rating curve.

surface area versus impoundment elevation. This information can be obtained by planimetering a contour map of the reservoir area and making necessary water volume calculations and water surface area determinations. The two curves are typically combined into what is termed an area-capacity curve. Figure 5.13 is a typical area-capacity curve for a hydropower development.

Reservoir Rule Curves

When releases from reservoirs are made, the schedule of releases is often dictated by considerations other than just meeting the flow demands for power

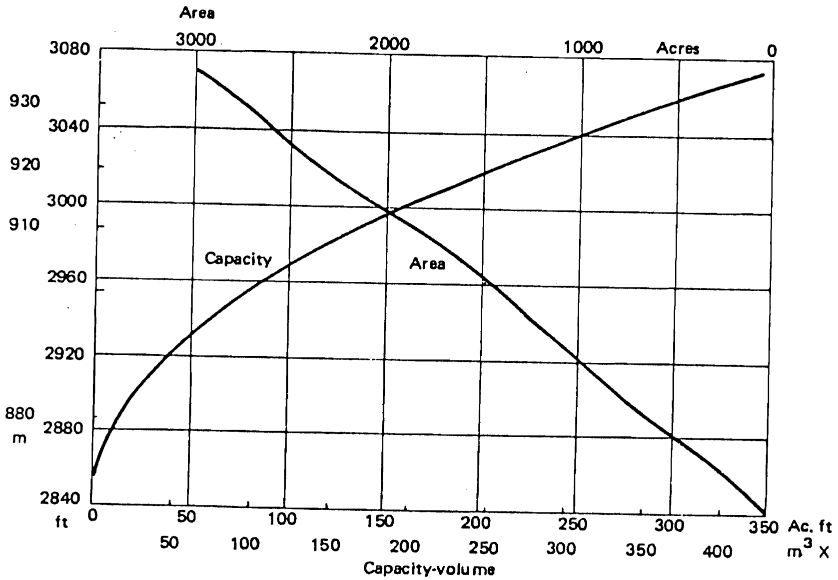

Figure 5.13 Typical area-capacity curves.

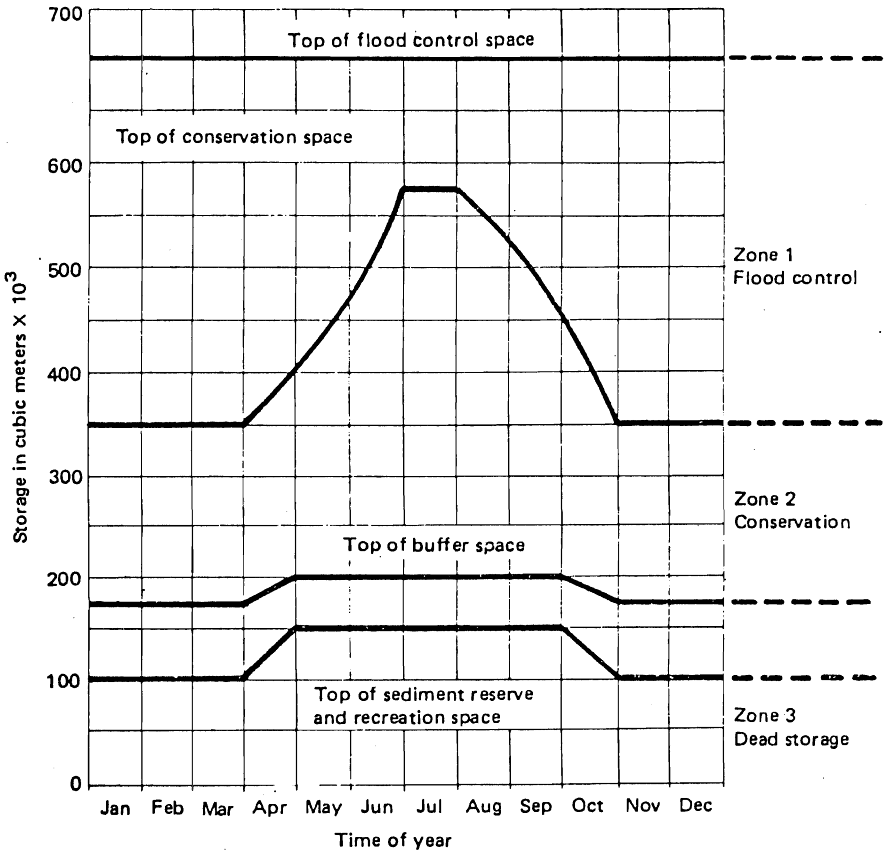

Figure 5.14 Example reservoir operation rule curves. SOURCE: U.S. Army Corps of Engineers.

production. The needs for municipal water supply, for flood control, and for downstream irrigation use dictate certain restraints. The restraints are conventionally taken care of by developing reservoir operation rule curves that can guide operating personnel in making necessary changes in reservoir water releases. Figure 5.14 shows example reservoir operation rule curves.

To be effective, rule curves often require the use of rather careful and extensive reservoir operation studies using historical flow data and estimates of demands for water that are likely to occur in the future. The techniques for study of reservoir management and optimization of reservoir use are many, varied, and presently require a great amount of digital computer time. A report by Chankong and Meredith (1979) gives a modern treatment of the formulation and procedures that can be used. This report contains an exhaustive bibliography on the subject of reservoir operation studies as related to rule making. A good practical treatment has been given by the U.S. Army Corps of Engineers (1977).

Evaporation Loss Evaluation

Where there is an impoundment involved in a hydropower development there is need to assess the effect of evaporation loss from the reservoir surface. This loss in warmer areas can amount to as much as 4 ft. of water evaporated from a reservoir in a summer season. The National Weather Service has developed regional evaporation maps that give lines showing the evaporation rates in various regions of the country. Similarly, there are frequently records of evaporation on at least a monthly basis at nearby reservoirs. In some cases it may be necessary to use empirical equations to obtain information on evaporation. The equation requires various measured data such as radiation, dewpoint readings, air and water temperatures, and wind speed to calculate the evaporation rates. Good treatments of techniques for calculating evaporation from meteorological measurements are those of Veihmeyer (1964) and Viessman, Harbaugh, and Knapp (1972).

Spillway Design Flood Analysis

Many hydropower developments require a dam or a diversion that blocks the normal river flow. This then requires that provisions be made for passing flood flows. Spillway design flood analysis treats a unique type of hydrology that concerns the occurrence of rare events of extreme flooding. Flood frequency analysis has become a well-developed technique, and the U.S. Water Resources Council (1977) has developed a standardized procedure for making flood frequency analyses. It is customary on larger dams and dams where failure might cause a major disaster to design the spillway to pass the probable maximum flood. For small dams, spillways are designed to pass a standard project flood. Detailed procedures and computer programs for these types of analyses are available from the U.S. Army Corps of Engineers (1971). Recently states have implemented federal dam safety regulations which must be met in any developments that involve dams over a minimum height of 20 ft. Normally each state's dam safety regulations should be referred to in making spillway flood determinations. Another good reference is the "Manual of Standards and Criteria for Planning Water Resources Projects" (United Nations, 1964).

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PROBLEMS

5.1. Generate from actual flow data a flow duration curve for use in hydropower computations using a historical record of at least 10 years' length. Compare a flow duration study done with daily flow values with one done with monthly flow values.

5.2. A drainage basin in the Little Salmon River has a possible hydropower site that is not near a stream gage (see Fig. 5.15). Planimetering of the areas between isohyetal lines provides the data given in Table 5.3. The runoff coefficient for the proposed site is estimated to be 0.64. The equations for the parametric flow duration curves similar to Fig. 5.4 that apply to the Little Salmon River basin have the following regression equations for determining the flows at five exceedance percentages:

$$\log Q_{10} = 0.534 + 0.965 \log \bar{Q}$$

$$\log Q_{30} = -0.299 + 1.049 \log \bar{Q}$$

$$\log Q_{50} = -0.574 + 1.084 \log \bar{Q}$$

$$\log Q_{80} = -0.754 + 1.101 \log \bar{Q}$$

$$\log Q_{95} = -0.853 + 1.107 \log \bar{Q}$$

where $\bar{Q}$ is in $[(\mathrm{ft}^3/\mathrm{sec})(\mathrm{day})]$ units and Q is in $\mathrm{ft}^3/\mathrm{sec}$. Develop a duration curve for power studies at the point marked on the map of Fig. 5.15.

5.3. A drainage basin in the Payette River basin has a power site designated at the mouth of Deadwood River (see Fig. 5.16). The Deadwood Reservoir used for irrigation regulates the flow of the upper portions of the drainage. The area of the hydrologic map representative of the drainage basin contributing flow at the mouth has been planimetered and Table 5.4 gives the information needed to compute average annual precipitation input to the basin below the reservoir. A runoff coefficient for the basin on the annual basis is $k = 0.65$. The historic monthly flows of a nearby stream gage on the South Fork of the Payette River are presented in Table 5.5. The gage records are considered to be a good representation of seasonal variation of runoff for the ungaged portion of the Deadwood River drainage. Using techniques discussed in this chapter, develop a flow duration curve for the flow at the mouth of the Deadwood River that would be useful in hydropower studies.

5.4. Obtain a sequential-flow reservoir operation or runoff model and study it for possible application in a hydropower study.

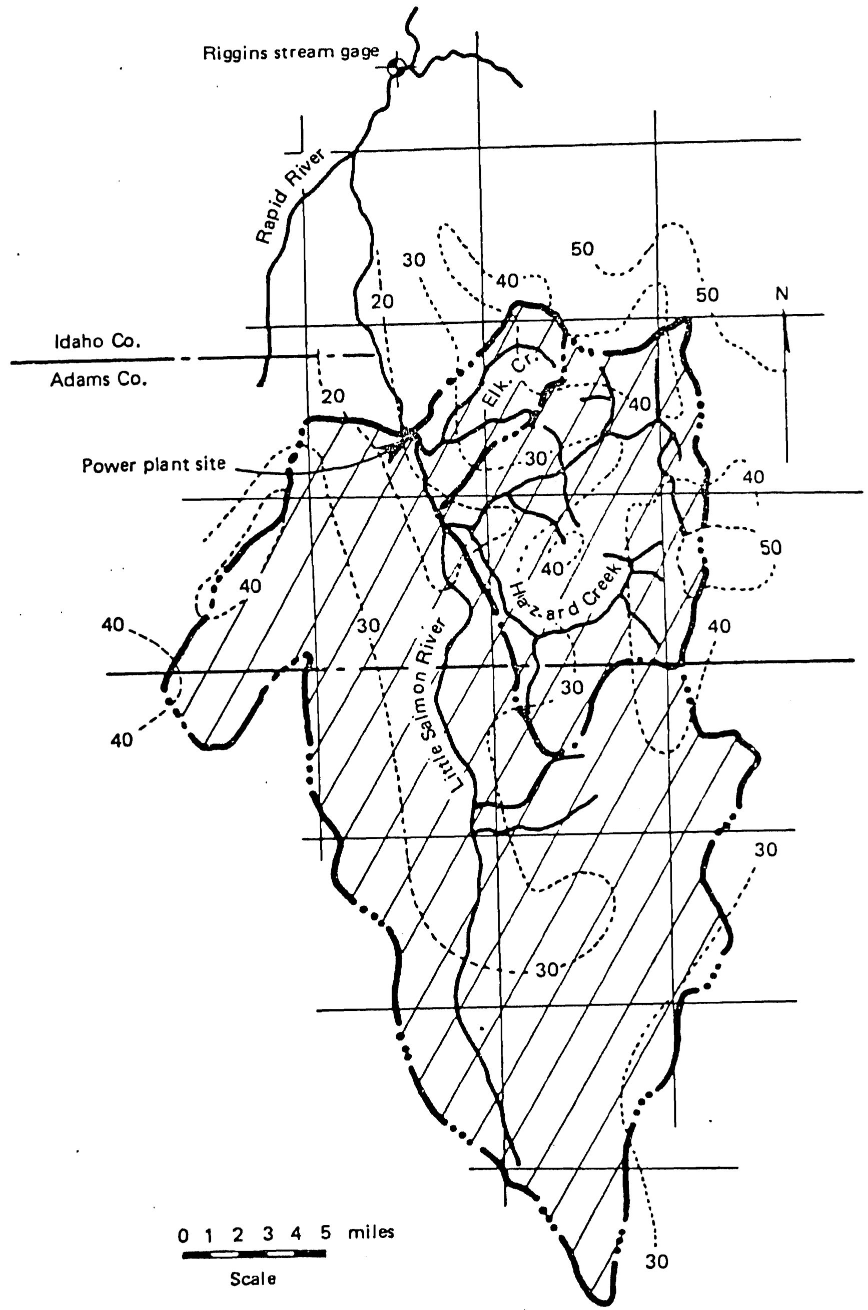


Figure 5.15 Normal-annual-precipitation map of Little Salmon River Basin, Idaho.

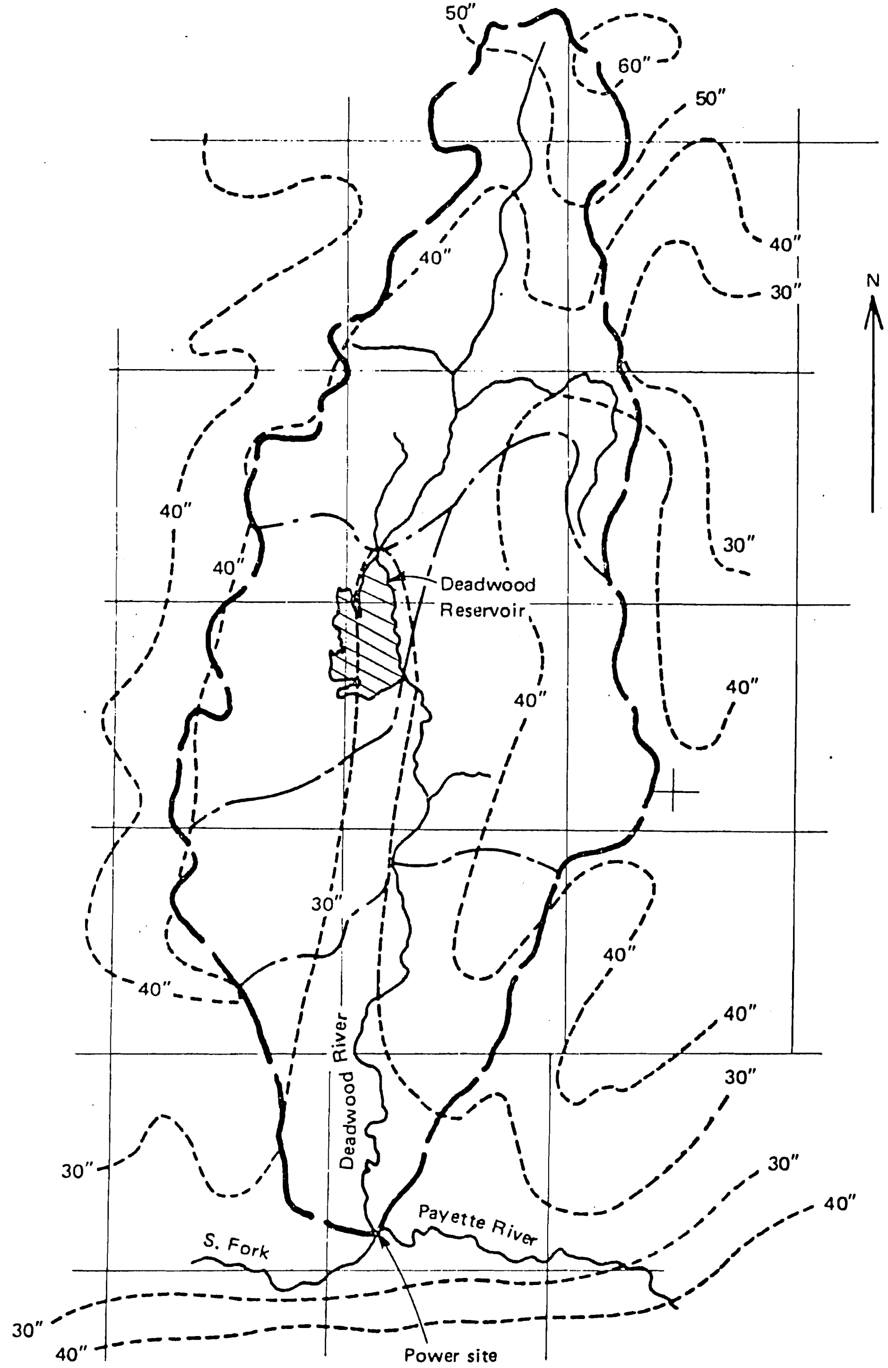


Figure 5.16 Hydrologic map of Deadwood River Basin, Idaho.

TABLE 5.3 Values of Planimetered Areas from the Normal Annual Precipitation Map of Drainage Basins in the Little Salmon River System

Average Value of Precipitation between Isohyetal Lines (in.)	Planimetered Area (in.^{2})
Elk Creek	
25	0.20
35	0.57
40	0.33
Hazard Creek	
20	0.11
25	1.35
35	2.38
40	0.44
45	0.81
50	0.11
Proposed Site (near Fall Creek)	
20	0.65
25	6.40
30	0.41
35	14.14
40	1.02
45	1.16
50	0.10

TABLE 5.4 Values of Planimetered Areas from the Normal Annual Precipitation Map of Drainage Basins in the Payette River System

Average Value of Precipitation between Isohyetal Lines (in.)	Planimetered Area (in^2)
Entire Deadwood River above the Mouth	
30	2.31
35	8.59
40	2.81
45	1.02
55	0.49
60	0.05
Area below Deadwood Reservoir	
30	1.85
35	4.05
40	2.24

TABLE 5.5 Monthly Flows for an Average Year in a Gaged Stream of the Payette River System

Month	Discharge (ft^3/sec)
October	332
November	336
December	298
January	251
February	252
March	349
April	1170
May	2826
June	2271
July	797
August	418
September	332

6 TURBINE SELECTION AND PLANT CAPACITY DETERMINATION

BASIC PROCEDURES

Turbine selection and plant capacity determination require that rather detailed information has been determined on head and possible plant discharge as described earlier. In practice, different selection procedures are used. Engineering firms or agency engineering staff do the selection using experience curves based on data from units that have already been built and installed or tested in laboratories. An excellent description of one governmental agency's approach is *Engineering Monograph No. 20* of the U.S. Department of the Interior (1976). Another approach that is preferred by manufacturers is that they be provided with the basic data on head, water discharge, turbine setting possibilities, and load charactersitics. The selection is then based on hill curves from model performance data that are proprietory in nature. Normally, the manufacturers provide a checklist similar to Table 6.1 which is the basis for making the selection. The manufacturer furnishes preliminary estimates of the cost of the turbines and necessary mechanical equipment and controls, together with basic characteristics and dimensions of the hydropower units.

In a theoretical sense, the energy output, E, can be expressed mathematically as plant output or annual energy in a functional relation as follows:

$$E = F(h, q, \mathrm{TW}, d, n, H_s, P_{\max}) \tag{6.1}$$

where h = net effective head
q = plant discharge
TW = tailwater elevation
d = diameter of runner

TABLE 6.1 Typical Checklist of Selection Information Required by Manufacturers

INFORMATION REQUIRED BY ALLIS-CHALMERS CORPORATION
FOR THE APPLICATION AND SELECTION
OF HYDRAULIC TURBINES

Our complete facilities are at the service of water power users. We provide recommendations, layouts, and proposals for both new developments and the reconstructing of existing plants. However, proper selection and application requires specific, detailed information. To prepare a complete proposal promptly, it is important that the following data, where applicable and available, be sent with each inquiry.

1. Name of firm or corporation, with address.
2. Location and name of plant.
3. Approximate elevation of the plant above sea level.
4. Total quantity of water in cubic feet or cubic meters per second, with comments regarding the variations in daily and seasonal flow as well as storage capacity; flow duration curves; and drainage area.
5. Quality of the water. Does it contain sand, chemicals, or other impurities?
6. Gross head (vertical distance from the headwater level to the tailwater level), with any known variations, preferably correlated with flow.
7. If already determined, net or effective head on which all guarantees are to be based, with any known variations. (If it has not been determined, we are prepared to estimate the net effective head based on the penstock or flume dimensions.)
8. Amount of power desired or required.
9. Discharge or load at which maximum efficiency is desired.
10. Number and size of the units contemplated or required now and for future installation. If new units are to be sized for an existing plant, give space limitations and details of existing foundations and superstructure.
11. Distance from normal tailwater level to powerhouse floor, with variations and relation to flow.
12. Proposed length, diameter, and material of the supply pipe (penstock) if required. If already designed or installed, give complete information.
13. If a surge tank is installed or contemplated on the pipeline, the distance along the penstock from the surge tank to the powerhouse and all available surge tank data.
14. Will the plant operate separately or in parallel with a power system? If in parallel, give the approximate installed capacity of the system and its frequency.
15. Method of intended operation—manual, semiautomatic, fully automatic, or remote control.
16. Supplementary information with drawings or sketches, to assist in proper interpretation of the data.

Since many dimensions may be changed before the plant is actually constructed, the purchaser must be responsible for the net effective head, the elevations of the head and tailwater levels, and the quantity of water, as well as the exactness of all field information on which the final design of the turbines is based.

SOURCE: Allis-Chalmers Corporation.

n = generator speed
H_s = turbine setting elevation above tailwater
P_{max} = maximum output expected or desired at plant.

It is seen that there are numerous parameters that can be varied to achieve the best selection. The usual practice is to base selection on the annual energy output of the plant and the least cost of that energy for the particular scale of hydropower installation. Thus one must recognize that determination of plant capacity requires analyses that vary the different parameters in Eq. (6.1) while applying economic analyses.

For preliminary planning it is sometimes useful to get an order-of-magnitude estimate of the unit (or units) capacity and cost to be expected. A curve (Fig. 6.1) prepared by consulting engineers Tippetts-Abbett-McCarthy-Stratton is useful in making such trial evaluations. If the energy costs exceed 60 mills/kWh (1981 prices), the economics of the development will be marginal. The curves presented are for conditions of water flow and yield of rivers in the northeastern United States.

A similar type of nomograph for sizing standard TUBE units is presented in Fig. 6.2. This gives an idea of the range of unit sizes that can be considered at a particular site. Another useful reference is an engineering manual by the U.S. Army Corps of Engineers (1952).

Figure 4.4 is a useful selection chart for determining the type of unit at the

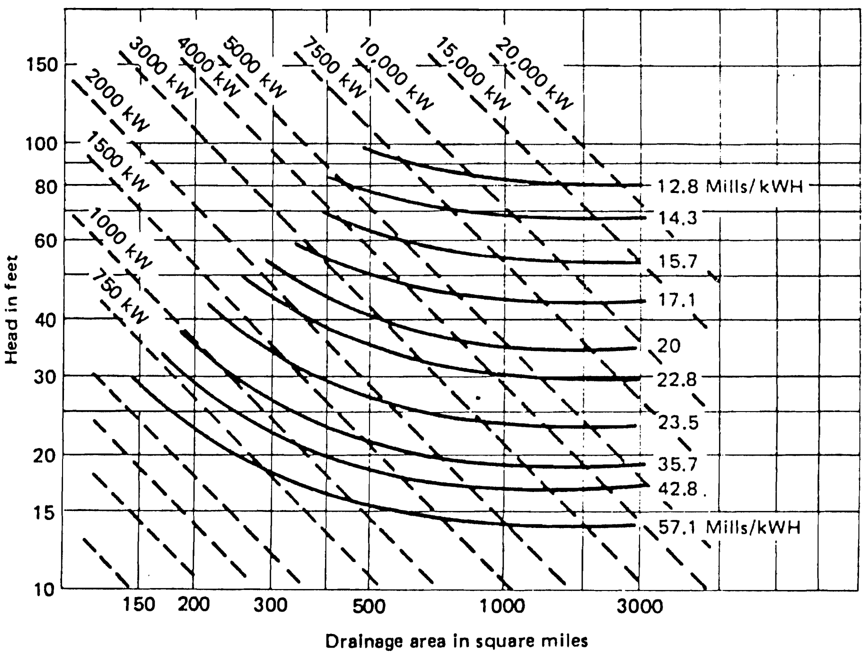
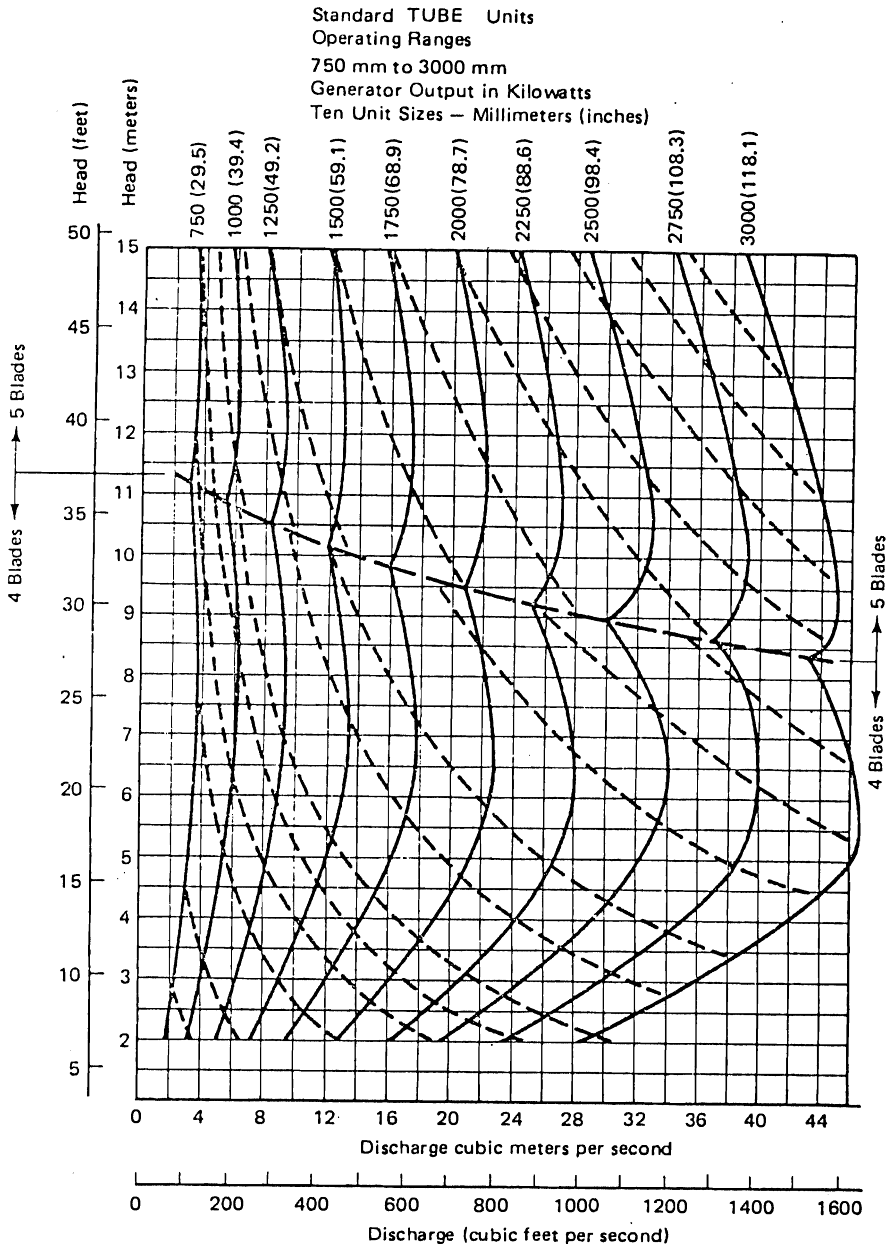


Figure 6.1 Nomograph for determining installed capacity. SOURCE: O'Brien-TAMS.



NOTE: Output based on unit centerline setting of one-half (1/2) runner diameter (dia/2) above tailwater. Centerline of unit at elevation 150-m (500 ft.) above sea level.

Figure 6.2 Nomograph for sizing turbines. SOURCE: Allis-Chalmers Corporation.

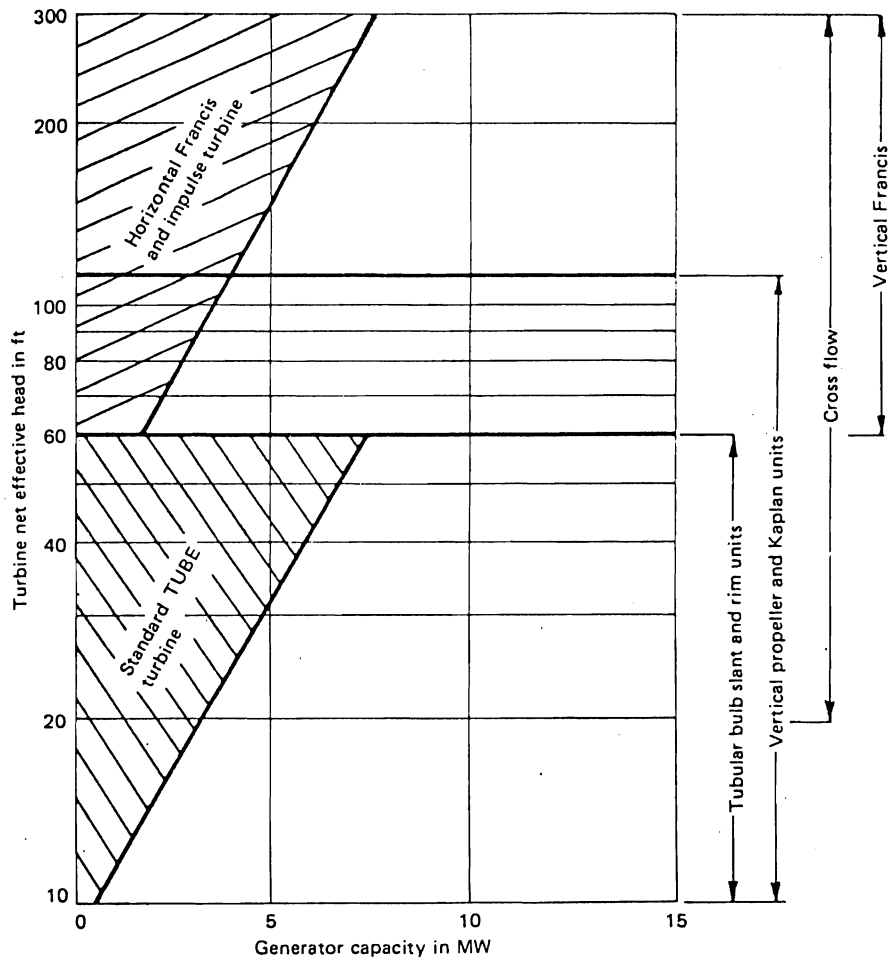


Figure 6.3 Chart for determining type of turbine. SOURCE: Allis-Chalmers Corporation.

stage of preliminary selection. Another type of turbine selection chart, Fig. 6.3, shows the range of types of units that can be used based on the output desired and the net effective head available.

LIMITS OF USE OF TURBINE TYPES

For practical purposes there are some definite limits of use that need to be understood in the selection of turbines for specific situations. Impulse turbines normally have most economical application at heads above 1000 ft, but for small units and cases where surge protection is important, impulse turbines are used with lower heads.

For Francis turbines the units can be operated over a range of flows from approximately 50 to 115% best-efficiency discharge. Below 40%, low efficiency, instability, and rough operation may make extended operation unwise. The upper range of flow may be limited by instability or the generator rating and temperature rise. The approximate limits of head range from 60 to 125% of design head.

Propeller turbines have been developed for heads from 5 to 200 ft but are normally used for heads less than 100 ft. For fixed blade propeller turbines the limits of flow operation should be between 75 and 100% of best-efficiency flow. Kaplan units may be operated between 25 and 125% of the best-efficiency discharge. The head range for satisfactory operation is from 20 to 140% of design head.

An understanding of how the efficiency of turbines varies is useful in the selection of hydropower units. This requires referring back to information in Chapter 4. Figure 4.5 is a generalized graph of how efficiency varies with rated load. These curves can be used in selection analysis and will be referred to again in later explanations and examples.

DETERMINATION OF THE NUMBER OF UNITS

Normally, it is most cost effective to have a minimum number of units at a given installation. However, multiple units may be necessary to make the most efficient use of water where flow variation is great. Factors such as space limitations by geologic characteristics or existing structure may dictate larger or smaller units. The difficulty of transporting large runners sometimes makes it necessary to limit their size. A runner with a maximum overall diameter of 18 ft (5.5 m) is about the largest that can be shipped by rail. Larger units require construction in segments and field fabrication with special care. Field fabrication is costly and practical only for multiple units where the cost of facilities can be spread over many units. Runners may be split in two pieces, completely machined in the factory and bolted together in the field. This is likewise costly, and most users avoid this method because the integrity of the runner cannot be assured. In some cases a reduced project cost may justify the more costly split runners.

Figure 6.4 shows how multiple units can be used effectively to take advantage of low-flow variation. Two-, three-, and four-unit plants of equal unit capacity are preferred and should provide for any variation in flow. At the design stage of analysis and with the availability of standardized units, it may be desirable to consider as alternatives a single full-capacity unit, two or more equal-size units, and two or more unequal-size units to determine the optimum equipment selection.

SELECTION OF THE MOST ECONOMICAL UNIT

Engineering consultants using experience curves for the selection of runner type, speed, and diameter, and utilizing experience curves for costs of turbine installation,

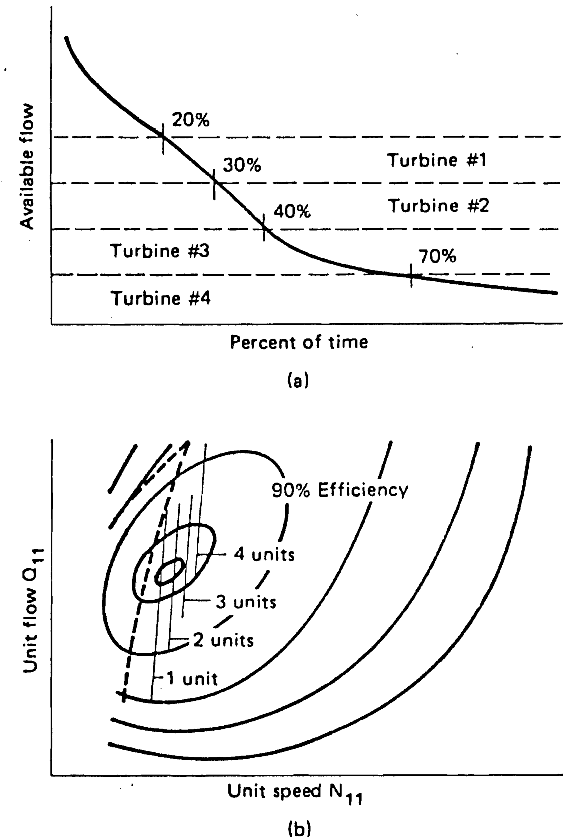


Figure 6.4 Effective use of multiple units. SOURCE: Allis-Chalmers Corporation.

select units in a size or combination of sizes that gives the most economical selection. A step-by-step process for this procedure is presented in a flow diagram of Fig. 6.5. At the time of preliminary analysis and in the early stages of feasibility analysis it may not be necessary to choose the number of units, depending on how detailed the feasibility study is. For the purpose of explaining the technique used for plant capacity determination, an example problem is presented next.

Example 6.1

Given: Information on a proposed run-of-river hydropower development is shown in computational form (Table 6.2). The net head has been computed for the corresponding river flow. To obtain this information a tailwater rating curve was developed and used to calculate net head by taking the difference between headwater and tailwater and then subtracting head losses in the penstock. Note that in this case, at very high flows the head is less than at low flow. This is because normally the tailwater will be higher at the higher flows. A study must also have

1. Obtain river flow data for each percent of time. 0% through 100% in at least 10% increments.
2. Determine headwater elevation at each flow characterized by flow duration curve. On run-of-river plants this is often constant.
3. Determine tailwater elevation at each flow characterized by the flow duration curve.
4. Estimate head loss through hydro system. This will vary with penstock and draft tube.
5. Compute a net head for each of the flows characterized. Note: as river flows increase, tailwater rises and reduces net head.
6. Estimate a plant efficiency, either a constant one or a varying one to be expected as flow is reduced. See Fig. 4-5.
7. Choose a wheel or plant capacity flow. This full-gate flow will be limited by runner diameter and selected penstock size.
8. Compute plant discharge at all flow values for each exceedance percentage. Note: at river flows greater than plant capacity the plant discharge will be less than full capacity. See sample computational table. (Table 6-2).
9. Compute power output at each percent time under investigation.
10. Compute annual energy output for given plant capacity. Repeat this for four to five plant capacities.
11. Estimate the annual plant costs for each of the plant capacities investigated, using estimating curves.
12. With annual energy output calculate plant benefits based on average expected value of power, mills per KWH.
13. Plot a curve or develop table to show where maximum net benefit is obtained.

Figure 6.5 Flow diagram of turbine selection procedures.

been made of headwater fluctuation. Referring to the flow diagram of Fig. 6.5, steps 1 through 5 have been completed at this stage of the problem.

Required: Make a plant capacity selection based on optimum net benefits of annual energy production.

Analysis and solution: In the computational table (Table 6.2) the efficiency has been entered as a constant value for the entire range of flows. This is step 6 of the flow diagram (Fig. 6.5). For a preliminary analysis this use of constant efficiency may be justified, but for more practical application, estimates should be made of the variation in efficiency based on relative output expected due to reduced discharge and reduced head (see Fig. 4.5).

Beginning with the row labeled "Plant discharge" in Table 6.2, a plant capacity discharge must be chosen. As a first alternative, choose a value at an exceedance percentage of 25 to 45% for base-load plants and 15 to 20% for peaking plants. The selection of the plant capacity discharge determines the size of the penstock, runner gate height, and runner discharge diameter. In the example and in the alternative presented, the plant discharge is chosen as 4700 ft^3/sec, the river flow at an exceedance percentage of 20%. This involves step 7 of the flow diagram of Fig. 6.5. The plant discharge is equal to the river discharge for all duration values or exceedance percentages greater than the exceedance percentage of the chosen plant capacity discharge–in this example, 20%. For exceedance percentages of less than 20% or the capacity value point, the plant discharge must be calculated. This is step 8 in the flow diagram of Fig. 6.5. The plant discharge will be less than the full gate capacity discharge due to the fact that the rising tailwater caused by high river discharges will decrease the head available through the turbines and penstocks.

To calculate the plant discharge, the following formula is used:

$$q_l = q_c\sqrt{\frac{h_l}{h_c}} \tag{6.2}$$

where q_l = plant discharge at the percent exceedance, ft^3/sec
q_c = plant discharge at design full gate capacity, ft^3/sec
h_l = net head at the percent exceedance being studied, ft
h_c = net head at the percent exceedance at which flow in the river is at full design gate magnitude, ft.

In the tabular information of Table 6.2, note that at the 10% exceedance the discharge through the plant has been reduced to 4447 ft^3/sec from the full-gate discharge of 4700 ft^3/sec.

With the foregoing information, it is now possible to complete the table. Calculate the power output at each exceedance percentage [Eq. (3.8)]. This is step 9 of the procedure outlined in Fig. 6.5. The energy for each increment of 10% of the time can be determined by considering that specific power output to represent the average output for that percent of time. The total energy produced then is the sum of the 10 increments. This is step 10.

The calculations from steps 7 through 10 need to be repeated for several alternative plant capacity flow values. Table 6.3 is a continuation of the computational procedure. The second row gives the value for various flow capacities for alternative sizes of power plants. In this example the plant capacity was varied from 11.6 MW to 6.2 MW. Using flow capacities for 0, 8, 10, 20, 30, and 40 exceedance

TABLE 6.2 Computational Table for Turbine Capacity Selection (Example 6.1)[a]

	Duration (%)										
	0	10	20	30	40	50	60	70	80	90	100
River discharge ($ft^3/sec \times 10^3$)	10.00	6.35	4.70	3.90	3.40	3.10	2.80	2.65	2.55	2.25	100
Head (ft)	15.50	18.80	21.00	23.00	24.50	26.10	27.50	28.50	29.50	30.50	31.20
Plant discharge [$(ft^3/sec) \times 10^3$]	4.038	4.447	4.70	3.90	3.40	3.10	2.80	2.65	2.55	2.25	1.00
Efficiency (%)	89	89	89	89	89	89	89	89	89	89	89
Power output[b] (kW)	4694	6270	7403	6728	6248	6068	5775	5664	5642	5147	2340
Percent time		10	10	10	10	10	10	10	10	10	10
Energy[c] (MWh)		4802	5989	6189	5683	5394	5187	5010	4952	4725	3279
Total energy	51,213 MWh (turbine output)										

[a]Turbine full-gate discharge 4700 ft^3/sec at 21.0 ft.

[b]$\text{Power} = \dfrac{POE}{11.81} = 0.075PQ.$

[c]$\text{Energy} = \dfrac{\%\ \text{time}}{100} \times \dfrac{P_1 + P_2}{2} \times 8760\ \text{hr/yr} = (P_1 + P_2) \times 438.$

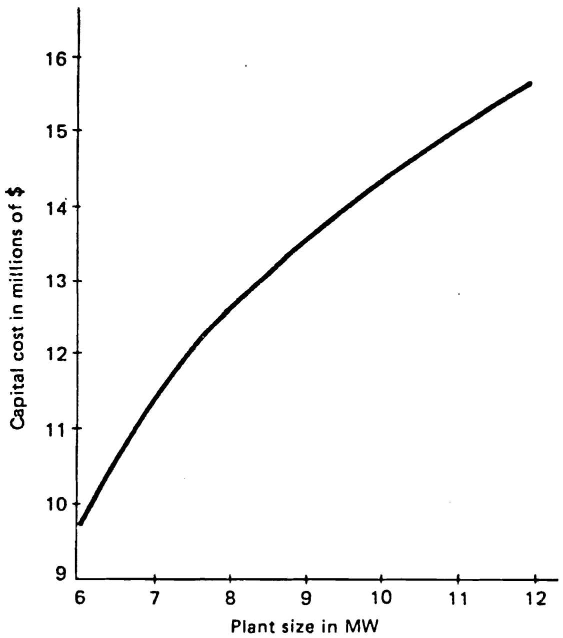


Figure 6.6 Cost estimating curve (Example 6.1).

percentages, the table was completed to determine net annual benefit and thus the most economical size of unit. This required a determination of the project life and the discount rate for money necessary for the capital investment. The capital recovery cost was computed using a 7% discount rate and a plant life of 50 years.

Annual operating costs had to be estimated. This is step 11 in the flow diagram of Fig. 6.5. Figure 6.6 is a cost-estimating curve showing how capital cost varies with plant size. Cost curves for various components of the development must be available for the cost analysis portion of the solution as well as cost curves for estimating annual operating costs. The fourth row of Table 6.3 gives the capital costs for each capacity of plant for six different plant capacities indicated in the third row as determined from Fig. 6.6. The annual operating costs are estimated and entered in the sixth row and when summed with the fifth row, capital recovery cost, the total annual cost is obtained and is given in the seventh row.

Using the kilowatt-hours of energy as calculated for the case of $q_{20} = 4700$ ft^3/sec and $h = 210$, yielding 51,200 kWh, the annual benefit can be calculated by multiplying by the unit sale value of the energy. This is step 12. In this case the unit rate value was taken as 30 mills or \$0.03/kWh (1981 prices). Plotting annual cost and annual benefits against the installed plant capacity will then permit a determination of the optimum plant capacity by showing where the maximum net benefit is or where marginal benefit equals marginal cost. This is shown in Fig. 6.7 and is step 13 of the flow diagram of Fig. 6.5. More detail on procedures, formulas, and approaches to economic analysis of hydropower is presented in Chapter 12.

TABLE 6.3 Computational Table for Economic Capacity Selection (Example 6.1)

	Exceedance % for plant capacity					
	0	8	10	20	30	40
Plant capacity flow, Q_p (ft^3/sec × 10^3)	10	6.90	6.35	4.70	3.90	3.40
Plant capacity, P (MW)	11.625	9.418	8.954	7.403	6.728	6.248
Capital cost of plant (millions of dollars)	15.4	13.95	13.55	12.0	10.95	10.16
Capital recovery cost (millions of dollars/year)	1.116	1.011	0.982	0.870	0.793	0.736
Annual operating cost (millions of dollars/year)	0.717	0.620	0.602	0.595	0.573	0.565
Total annual cost (millions of dollars/year)	1.833	1.631	1.584	1.465	1.366	1.300
Annual energy output (MWh)	56,605	55,096	54,449	51,213	48,307	45,721
Annual benefits, millions of dollars/year ($30.00 × annual energy)	1.698	1.653	1.633	1.536	1.449	1.372
Net benefits (millions of dollars/year)	-0.135	0.021	0.049	0.071	0.083	0.072

This example is based on a flow duration analysis. A similar procedure could be applied to sequential-flow analysis of operation studies by choosing different alternatives of varying the capacity of the units. In that case one must choose enough alternatives for plant capacity to bracket the scale size that represents the most economic unit size.

Consulting firms, manufacturers, and governmental agencies have digital computer programs that will quickly run either a duration-type study or a sequential-flow study to determine the annual energy output. In the case of the U.S. Army Corps of Engineers there is an H.E.C.-3 program (1968), an H.E.C.-5C program (1976), and a HYDUR program (1980). With these computer programs it is possible to make computations similar to those in the example, but which require making several alternative runs at different discharges to find the optimum installed capacity.

As the mathematical expression in Eq. (6.1) indicated, there are variables other than flow and head that can influence the selection of the type and size of turbines. At the final design stage it should be worthwhile to try alternatives such as variation in runner speed and variation in the height of the runner above minimum

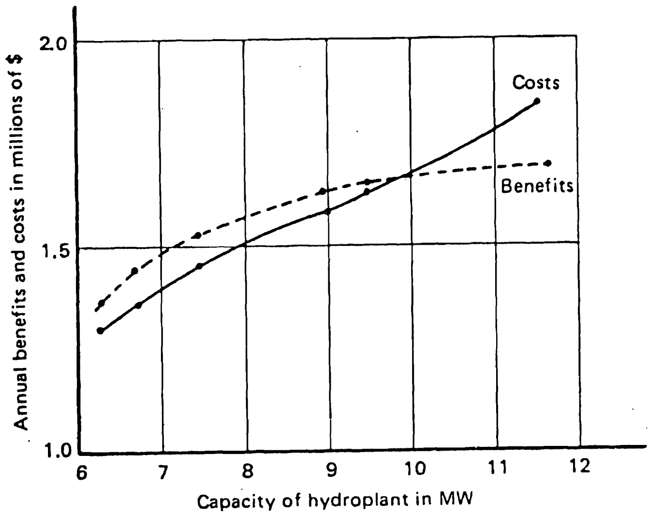


Figure 6.7 Benefits and costs versus plant capacity (Example 6.1).

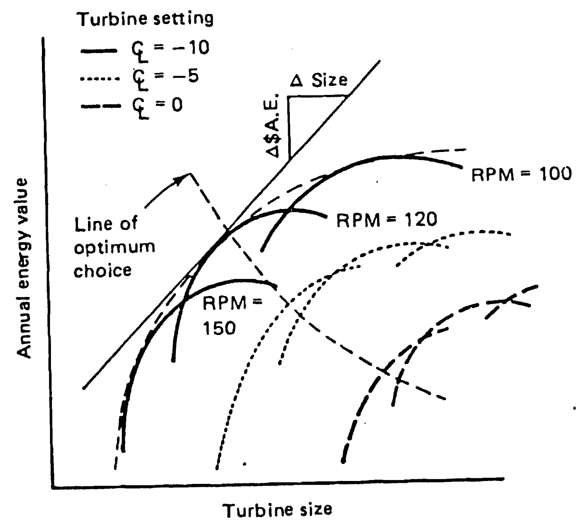


Figure 6.8 Cost variation with turbine parameters of speed and turbine setting. SOURCE: Allis-Chalmers Corporation.

tailwater. Manufacturers making selections make these more detailed analyses with more precise variations in the size, runner speed, and turbine setting evaluation. Figure 6.8 is a qualitative graph showing how parameters of speed and turbine setting can be used in making an optimum selection of a turbine for a given location and sequence of flows.

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PROBLEMS

6.1. A hydropower development has been proposed as a run-of-river plant in which headwater will remain essentially constant. The flow duration data at various exceedance percentages, the expected corresponding net head, and an expected turbine efficiency are given in a computation table, Table 6.4. Figure 6.9 gives a cost curve of annual cost versus plant capacity for the type of plant being considered. Assuming a value of 40 mills/kWh for the energy that could be produced, determine the optimum installed capacity using one unit operating through the entire range of river flows.

6.2. In Problem 6.1, limit the use of the turbines to the restraints that are in practice (see the section on limits of use) and select the size and number of units that you would propose for the installation (refer to Fig. 6.3).

6.3. Obtain a sequential flow power operation study, then vary the plant size and make estimates of power plant costs to determine two points on a curve for determining optimum plant size.

6.4. Develop a solution flow diagram similar to Fig. 6.2 for determining the optimum size of a hydropower plant using a reservoir operation with a sequential-flow study rather than with a flow duration study.

TABLE 6.4 Computation Table for Problem 6.1 (Characteristics for Plant Capacity Determination)

	Exceedance (%)										
	0	10	20	30	40	50	60	70	80	90	100
River discharge (ft^3/sec)	16,000	11,600	9000	7900	7800	7400	7300	7100	6900	6850	6810
Estimated head (ft)	84	90	94	95	96	96.5	97.0	97.2	97.4	97.5	97.5
Plant discharge (ft^3/sec)											
Efficiency (%)	86	88	89.5	91	89.5	89.0	88.5	88	87.5	87.2	87.0
Power output (kW)											
Percent of time											
Energy (kWh)											
Total energy (kWh)											

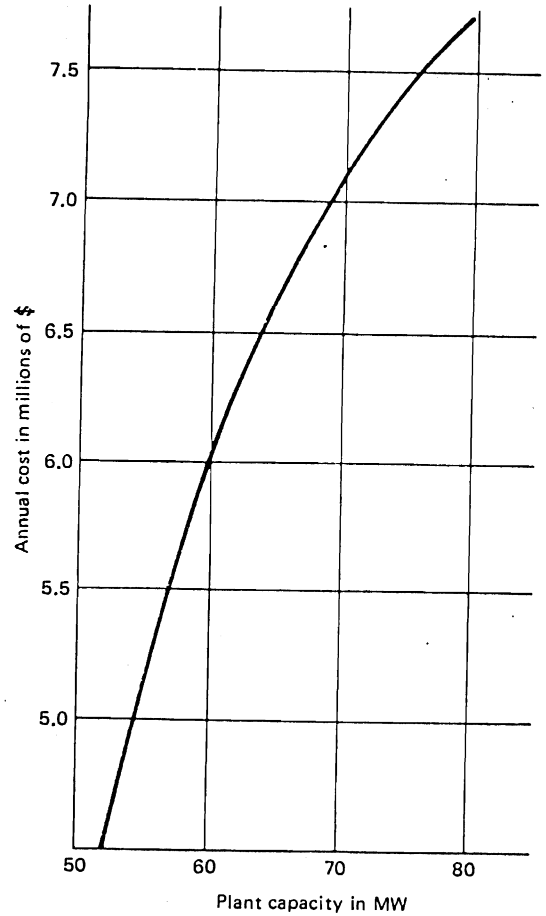


Figure 6.9 Annual cost curve (Problem 6.1).

CAVITATION AND TURBINE SETTING 7

CAVITATION

As water passes through penstocks, through spiral cases, around guide vanes and gates, through the turbines themselves, and out the draft tubes, there are phenomena at work that can cause problems not easily understood or controlled. Cavitation and water hammer are two of these phenomena. The subject of water hammer is discussed in Chapter 10.

Cavitation is defined as the formation of voids within a body of moving liquid (or around a body moving in a liquid) when the local pressure is lower than vapor pressure and the particles of liquid fail to adhere to the boundaries of the passageway (U.S. Department of the Interior, 1975). The failure of the particles to adhere to boundaries occurs when there is insufficient internal pressure within the liquid to overcome the inertia of the moving particles and force them to take sufficiently curved pathes along the boundary. The voids thus formed fill with vapor of the liquid and result in vapor bubbles. Because the inertia of a moving particle of a liquid varies with the square of the velocity, and because the greater the inertia, the greater the pressure required to force the particles to take a curved path, it becomes obvious that cavitation is associated with three conditions: high-velocity flow, low pressures, and abrupt changes in the direction of the flow.

The effect of cavitation is to cause pitting of the boundary surfaces. This pitting is the actual removal of the metal because of the violent collapse of the vapor bubbles formed by cavitation. Preferred practice is to make a distinction between the words "cavitation" and "pitting," identifying cavitation as a cause and pitting as an effect. In the past there was not clear understanding of whether destructive pitting was caused by the chemical action of oxygen being released from water under a high vacuum, or by collapse of bubbles, or by both actions. The lack of knowledge was due to the difficulty of observing the phenomenon. As reported by the U.S. Department of the Interior (1975), Knapp and Hollander (1948) made very high speed photographic studies of the cavitation phenomenon in a laboratory. Their pictures revealed the formation, growth, and collapse of individual bubbles. These bubbles had diameters of about 0.25 in. and had life spans of only 0.003 sec; their velocity of collapse was calculated to be 765 ft/sec. Using water hammer theory, it was calculated that the bubbles' collapse could cause pressures of at least 50,000 lb/in^2. The resulting pressure was concentrated on a microscopically small boundary area of metal. This then resulted in fatigue-like destruction and failure of the metal. The work of Knapp, Daily, and Hammitt (1970) indicates that there is a preponderance of evidence that the effects of cavitation in turbines operating with cold water are due primarily to the mechanical collapse of the vapor-filled bubbles and the impact of the pressure pulse or shock wave that results. Figure 7.1 shows photographs of cavitation pitting on turbines.

Cavitation Parameter

To understand cavitation more fully, it is necessary to define it mathematically. A term that has been developed to characterize cavitation is the *cavitation parameter.* It is a quantitative index that defines the principal dynamic flow and pressure conditions involved in the phenomenon. Traditionally, it is the ratio of pressure head to the velocity head. The formula reported by Knapp, Daily, and Hammitt (1970) for the parameter is

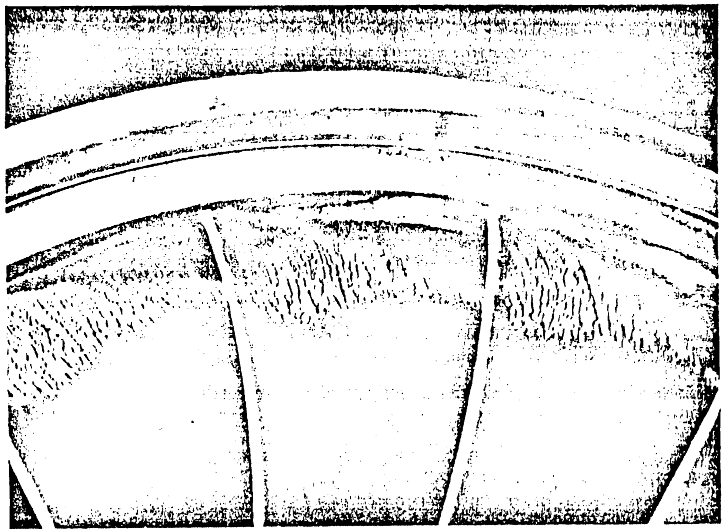
$$K = \frac{(p_0 - p_b)/\gamma}{V_0^2/2g} \tag{7.1}$$

where K = cavitation parameter, dimensionless
p_0 = absolute pressure at some reference point in the flowing water
p_b = absolute pressure in cavity or bubble
γ = specific weight of the water
V_0 = reference velocity in the flowing water
g = acceleration of gravity.

For use in hydraulic turbines, this can be written in the form

$$K_d = \frac{H_d - H_v}{V_d^2/2g} \tag{7.2}$$

where K_d = cavitation parameter for the flow passage at exit of turbine, dimensionless
H_d = absolute pressure head, feet of water at exit of turbine
H_v = vapor pressure head, feet of water
V_d = average velocity, ft/sec at exit of turbine runner
g = acceleration of gravity.



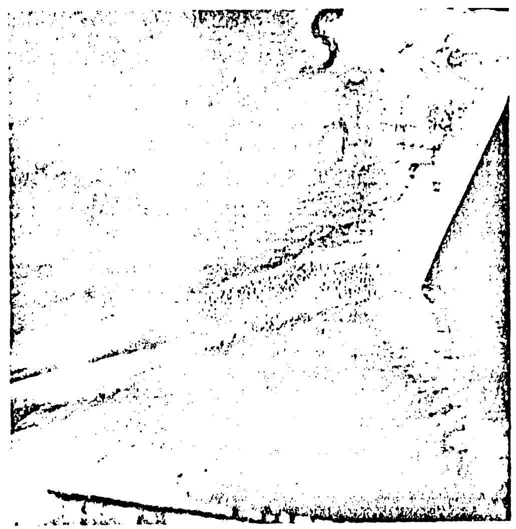


Figure 7.1 Cavitation damage on turbines. SOURCE: Escher Wyss.

In a practical sense, the cavitation parameter is simply the ratio of the pressure available for collapsing the cavity ($H_d - H_v$) to the dynamic pressure available for inducing the formation and growth of the cavity, which is the velocity head at a reference point, usually taken at the exit of the turbine runner. The following can affect the parameter: viscosity, gravity, surface tension, and thermodynamic properties of the water and vapor, as well as contaminants within the water and changes in the boundary conditions that will alter the velocity, V_0. For cold water the thermodynamic effects play a minor role.

Cavitation Coefficient

More used in the turbine industry and engineering profession than "cavitation parameter" is a coefficient or turbine constant known variously as the *plant sigma*, Thoma number, or cavitation coefficient. Mathematical development of the equation of the plant sigma is important to an understanding of cavitation and gives emphasis to the use of the plant sigma in the selection of the turbine setting.

Hydraulic equations utilized in the development include the Bernoulli equation, head loss equation, and orifice equation. First consider the elements involved, shown graphically in Fig. 7.2. By writing a Bernoulli equation and accounting for losses [see Eq. (3.12)] between point 2 at the outlet of the runner (top of draft tube) and point 3 at the outlet to the draft tube, the following equation results:

$$\frac{V_2^2}{2g} + \frac{p_2}{\gamma} + Z_2 = \frac{V_3^2}{2g} + \frac{p_3}{\gamma} + Z_3 + h_{f_{2-3}} \tag{7.3}$$

where
V_2 = water velocity at point 2, ft/sec
p_2 = pressure at point 2, lb/ft^2
γ = specific weight of water, lb/ft^3
Z_2 = elevation of point 2 above reference datum, ft
V_3 = water velocity at point 3, ft/sec
p_3 = pressure at point 3, lb/ft^2
Z_3 = elevation of point 3 above reference datum, ft
$h_{f_{2-3}}$ = head loss between points 2 and 3 in draft tube, feet of water.

From the basic hydraulic definitions of the orifice equation, continuity equation, and Darcy head loss equation, the following equations can be written:

$$V_2 = C\sqrt{2gh} = K_1\sqrt{h} \tag{7.4}$$

$$V_3 = \frac{A_2 V_2}{A_3} = \frac{A_2 K_1 \sqrt{h}}{A_3} = K_2\sqrt{h} \tag{7.5}$$

$$h_f = \frac{fL V_{2-3}^2}{2dg} = K_3\frac{\sqrt{h}}{2g} \tag{7.6}$$

where h = net effective head or normal head in feet. Substituting these values for V_2, V_3, and h_f in Eq. (7.3) and grouping the terms results in the following equation:

$$\left[\frac{K_1^2}{2g} - \frac{K_2^2}{2g} - \frac{K_3}{2g}\right] h = \frac{p_3}{\gamma} - \frac{p_2}{\gamma} + Z_3 - Z_2 \tag{7.7}$$

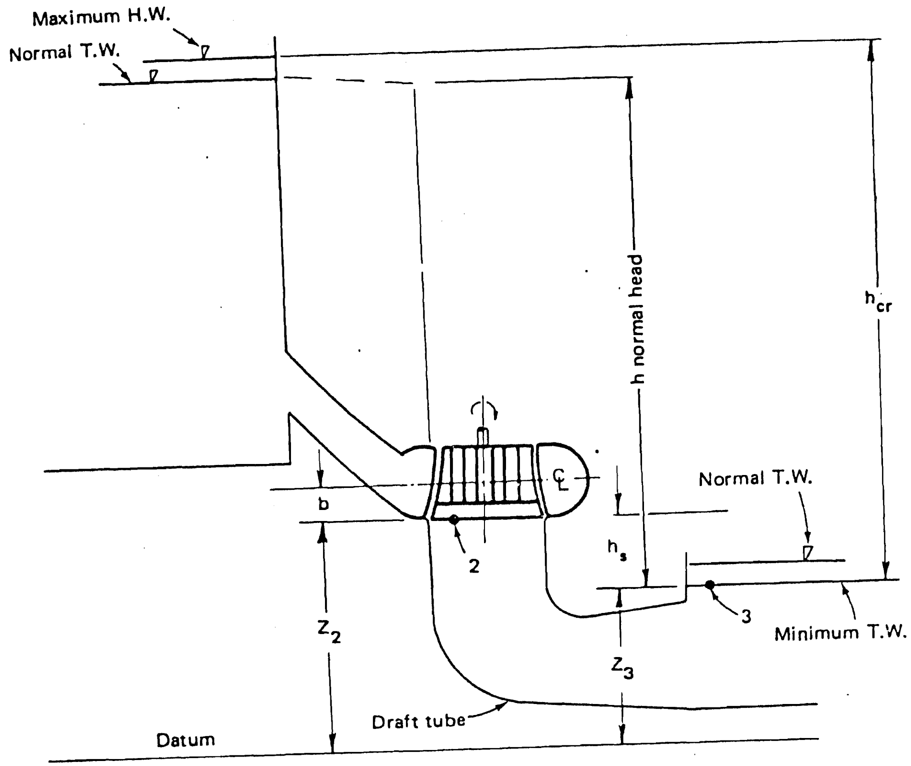


Figure 7.2 Diagram for defining cavitation coefficient parameters.

The term with all the constants is replaced with the cavitation coefficient (plant sigma), σ,

$$\sigma = \frac{K_1^2}{2g} - \frac{K_2^2}{2g} - \frac{K_3}{2g}$$

the term $Z_3 - Z_2$ is replaced with h_s

$$h_s = Z_3 - Z_2$$

and

$$\frac{p_3}{\gamma} = \frac{p_{\text{atm}}}{\gamma} = h_a$$

and

$$\frac{p_2}{\gamma} = \frac{p_{\text{vapor pressure}}}{\gamma} = h_v \quad \text{as a limit}$$

The resulting convenient equation is

$$\sigma = \frac{h_a - h_v - h_s}{h} \tag{7.8}$$

where σ = cavitation coefficient (plant sigma)

h_s = difference in elevation between minimum tailwater and the cavitation reference point at the outflow from the turbine, ft

h_a = atmospheric pressure head, ft

h_v = vapor pressure head at temperature of water issuing from the turbine, ft.

In the metric system

$$\sigma = \frac{H_a - H_v - H_s}{H} \tag{7.9}$$

where all the heads are expressed in meters of water head.

In practice, the term h, net effective head, is replaced with a term called *critical head*, h_{cr} (see Fig. 7.2). It should be noted that as the headwater or forebay water level rises and the tailwater lowers, there is an operating head greater than the normal design head that will be functioning at times. This condition must be allowed for in the design of the turbine setting. A later discussion on turbine setting will require use of the plant sigma equation [Eq. (7.9)] and the use of critical head.

The occurrence of cavitation and its inception is usually associated with increased noise, vibration, and a loss in performance of the turbine. Often it is difficult to identify where the cavitation starts because the vapor bubbles may grow rapidly and be swept downstream to the place where collapse occurs and damage through pitting is noted. It is usual practice to study and interpret cavitation by modeling and relate the plant sigma to loss of efficiency and power or to such turbine coefficients as unit power and unit discharge. Figure 7.3 shows representative cavitation coefficient curves for a reaction turbine giving the variation in the efficiency and the unit power with variation in the values of the cavitation coefficient.

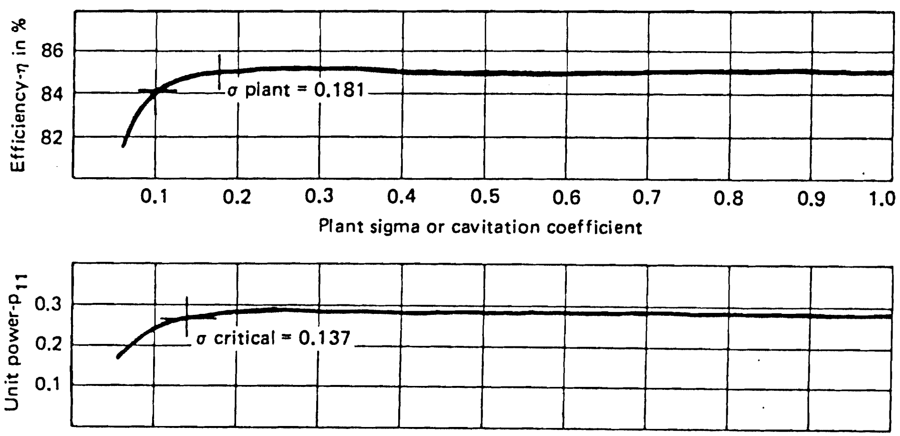


Figure 7.3 Representative cavitation coefficient curves. SOURCE: U.S. Army Corps of Engineers.

In model testing and the activities of the turbine manufacturers it is customary to develop plots of cavitation coefficient, σ, versus turbine efficiency, η, or power output, or unit discharge. From these plots criteria are developed that are used to determine turbine setting elevations and to indicate admissible values of the cavitation coefficient. Figure 7.4 illustrates the characteristic shapes of the different kinds of curves that are common in turbine testing. The curves show how an admissible cavitation coefficient is determined by designating an acceptable change in sigma, $\Delta\sigma$, that must be added to the critical sigma value or in this case the σ_1, which is the cavitation coefficient where the efficiency has decreased by 1%. An excellent article treating details for determining the limits of the cavitation coefficient (plant sigma) is the work of Eichler and Jaeger (1979), from which some of the information in Fig. 7.4 was taken. Other recent information on cavitation is contained in two articles published in 1981 by Arndt.

Normally, the variation in plant sigma, σ, for model tests gives a greater change in turbine efficiency than that which will occur in the prototype units, so the cavitation indicated in the model test will usually be more severe than will actually occur in homologous prototype units.

Control of Cavitation

There are several ways of controlling cavitation. The most satisfactory control is through design and setting the turbine runner so that the pressure and velocity at

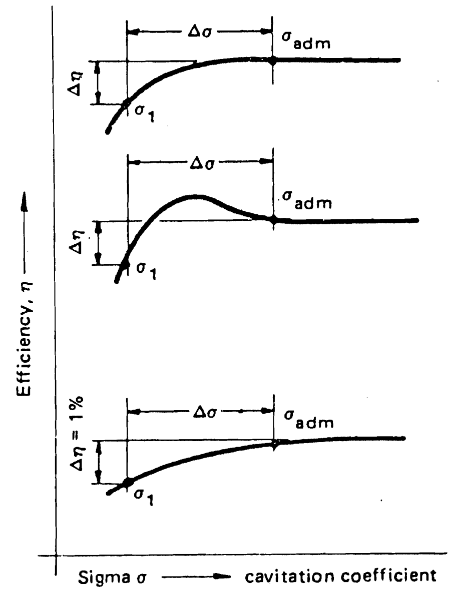


Figure 7.4 Different shapes of cavitation coefficient curves.

critical areas in the system do not permit excessive cavitation to occur. The setting is dictated by economics and may require placing the turbine well below the tailwater, which can be very expensive due to excavation and concrete costs. In some cases it has been economically justifiable to permit some cavitation and use very resistant metal and/or repair of the damage to compensate for the pitting. Table 7.1 gives the weight loss of different metals for the same cavitation conditions as reported by the U.S. Army Corps of Engineers.

Frequently, a layer of cavitation-resistant material is overlaid by welding on the base metal. The locations and size of the areas requiring such overlay are determined *after* model tests have been made and by experience. Nearly all turbines do experience some cavitation damage, so a control measure is to build up the damaged surfaces with stainless steel welding rod or welded-on plates.

Sometimes special anticavitation fins have been added to Kaplan and propeller turbine blades to minimize blade tip cavitation. According to Csanady (1964), Rasmussen in 1956 reported that the addition of air bubbles produces an elastic fluid that apparently cushions the action of the pressure pulses. Csanady indicates that amounts of only 1 to 2 parts per thousand of air produce a significant reduction in damage. According to Csanady, Plesset (1960) found that a thin hydrogen film on a metal surface could stop pitting.

Indirectly through design, the selection of the speed of the turbine can be used to control cavitation. Increasing the speed of the runner results in smaller

TABLE 7.1 Weight Loss in Materials Used in Hydraulic Machines

Alloy	Weight Loss after 2 hr (mg)
Rolled stellite[a]	0.6
Welded aluminum bronze[b]	3.2
Cast aluminum bronze[c]	5.8
Welded stainless steel (two layers, 17% Cr, 7% Ni)	6.0
Hot-rolled stainless steel (26% Cr, 13% Ni)	8.0
Tempered, rolled stainless steel (12% Cr)	9.0
Cast stainless steel (18% Cr, 8% Ni)	13.0
Cast stainless steel (12% Cr)	20.0
Cast manganese bronze	80.0
Welded mild steel	97.0
Plate steel	98.0
Cast steel	105.0
Aluminum	124.0
Brass	156.0
Cast iron	224.0

[a]This material is not suitable for ordinary use, in spite of its high resistance, because of its high cost and difficulty in machining.

[b]Ampco-Trode 200: 83% Cu, 10.3% Al, 5.8% Fe.

[c]Ampco 20: 83.1% Cu, 12.4% Al, 4.1% Fe.

SOURCE: U.S. Army Corps of Engineers (personal communication, 1980).

diameters and thus increases water velocities through the runner, which may, in turn, increase the likelihood of cavitation. To compensate for this, it will normally be necessary to set the turbine lower with respect to the tailwater. Thus, determining the turbine setting is a very important and complex problem relating to several variables of design and operation.

Another way to control cavitation is in the design and shape of the water passage so that the shapes and surfaces offer the minimum opportunity for abrupt changes in flow lines of the water. This involves visual model test analysis and practical experience.

SELECTION OF THE TURBINE SETTING

Determining the turbine setting is based primarily on defining the plant sigma and choosing the vertical distance the critical part of the runner is from the minimum full-load tailwater level. This plant sigma σ must be referred to a specific point on the runner and is assigned an elevation position designation of KS in some literature. For vertical-axis Francis turbines it is customary to choose as the reference elevation, KS, the bottom of the runner as the water exits the bucket vanes. For vertical-axis propeller turbines, the reference point is the centerline of the blades. For horizontal-axis turbines, a point near the upper tip of the runner blade is used because the pitting damage is time related and the most critical position is only exposed instantaneously during each revolution.

Final selection of the turbine setting is done by the turbine manufacturer in a manner similar to turbine size and shape selection, using results from model tests. However, it is often useful to make a preliminary determination of the turbine setting elevation on the basis of the homologous nature of turbines. This must be done on the basis of past performance and relations between models and prototype turbines, so experience curves have been developed for preliminary setting determination. As with turbine capacity selection, the specific speed, n_s, is used as the parameter to characterize the values of the plant sigma, σ, and to develop experience curves that are used in making the setting elevation determination.

U.S. Department of the Interior Procedure

The U.S. Department of the Interior (1976) has published a useful experience curve that can be used in the preliminary selection of the turbine setting elevation. This curve is presented in Fig. 7.5. A curve similar to this is presented in Knapp, Daily, and Hammitt (1970). This curve relates the acceptable plant sigma, σ, to the specific speed, n_s, and thus is using the homologous nature of turbines to indicate that turbines having geometrically similar design and operating under similar hydraulic conditions will have the same value of critical runner sigma, σ_1. The experience curve of σ versus n_s in Fig. 7.5 has been plotted to place the turbine 1 ft (0.3 m) lower than the elevation at which cavitation damage and loss of performance have

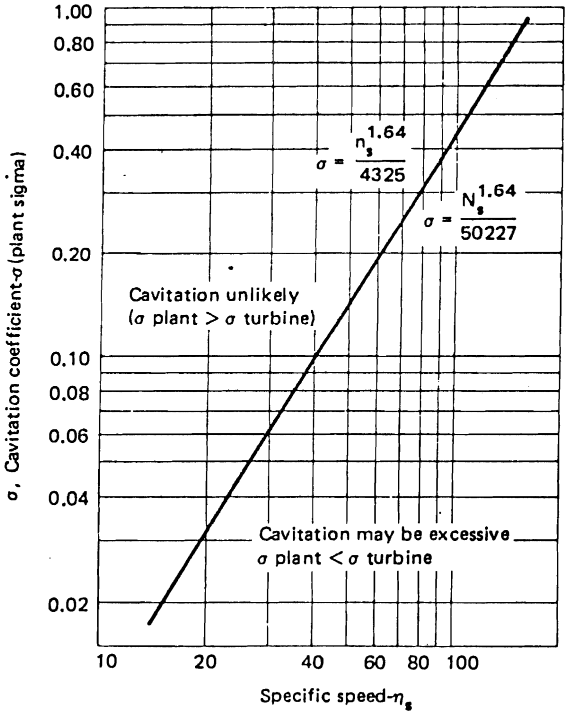


Figure 7.5 Experience curve for recommending cavitation coefficient limit. SOURCE: U.S. Bureau of Reclamation.

approached unacceptable values. This then provides a limited margin of safety to allow for variation in atmospheric pressure and minor variation in the turbine runner characteristics.

With Eq. (7.8) it is possible to proceed with determination of the turbine setting elevation. However, another experience curve is necessary to relate the turbine setting elevation, h_s, to the centerline of the turbine distributor. This experience curve is reproduced as Fig. 7.6 together with a definition sketch from a U.S. Department of the Interior publication (1975). Using the definition of h_s as shown in Fig. 7.6:

$$h_s = h_b - \sigma h_{cr} \tag{7.10}$$

and

$$Z = h_s + b \tag{7.11}$$

and

$$h_b = h_a - h_v \tag{7.12}$$

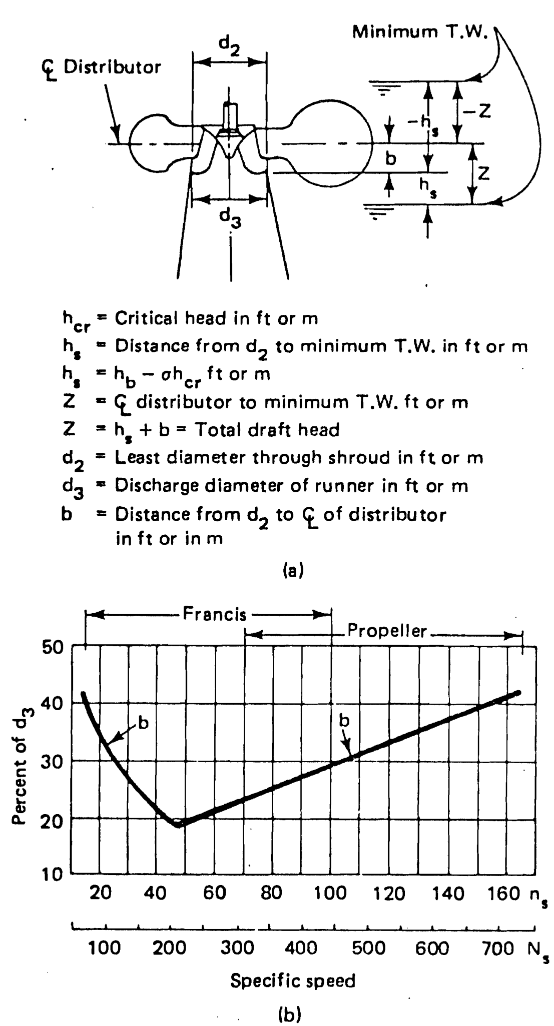


Figure 7.6 Experience curve for recommending total draft head. SOURCE: U.S. Bureau of Reclamation.

where Z = total draft head, ft
h_b = barometric pressure head, feet of water
h_a = atmospheric pressure head, feet of water
h_v = vapor pressure head, feet of water.

Note that Table 7.2 gives data on the relation of atmospheric pressure head, h_a, to the elevation of the tailwater elevation in pounds per square inch and feet above mean sea level (MSL) and the relation of vapor pressure head, h_v, to the water

TABLE 7.2 Atmospheric Pressure and Vapor Pressure Variation[a]

Atmospheric Pressure

Altitude (ft)	h_a (lb/in^2)	h_a (ft H_2O)	Altitude (m)	h_a (mm Hg)	h_a (m H_2O)
0	14.70	33.96	0	760.00	10.351
500	14.43	33.35	500	715.99	9.751
1,000	14.17	32.75	1,000	674.07	9.180
1,500	13.92	32.16	1,500	634.16	8.637
2,000	13.66	31.57	2,000	596.18	8.120
2,500	13.42	31.00	2,500	560.07	7.628
3,000	13.17	30.43	3,000	525.75	7.160
3,500	12.93	29.88	3,500	493.15	6.716
4,000	12.69	29.33	4,000	462.21	6.295
4,500	12.46	28.79			
5,000	12.23	28.25			
5,500	12.00	27.73			
6,000	11.78	27.21			
6,500	11.56	26.70			
7,000	11.34	26.20			
7,500	11.12	25.71			
8,000	10.91	25.22			
8,500	10.71	24.74			
9,000	10.50	24.27			
9,500	10.30	23.81			
10,000	10.10	23.35			

Water Vapor Pressure

Temp (°F)	h_v (ft)	Temp (°C)	h_v (m)
40	0.28	5	0.089
50	0.41	10	0.125
60	0.59	15	0.174
70	0.84	20	0.239
80	0.17	25	0.324

[a] h_a, atmospheric pressure for altitude, ft (m); h_v, vapor pressure of water (use highest expected temperature), ft (m); $h_b = h_a - h_v$, atmospheric pressure minus vapor pressure, ft (m).

temperature. An example problem is presented next to illustrate the use of the experience curves as an approach to determining a turbine setting.

Example 7.1

Given: A hydro development is to operate at a maximum discharge of 1075 ft^3/sec. Normal tailwater elevation is 3163.5 ft, head loss in the penstock is estimated to be 0.5 ft, normal headwater elevation is 3247 ft, and minimum tailwater elevation is 3158.5 ft. Maximum water temperature is 70°F.
Required: Find a suitable centerline elevation for the turbine runner.

Analysis and solution: First determine plant capacity, speed, and runner size. p_{hp}, design capacity, can be determined from Eq. (3.6):

$$p_{hp} = \frac{q\gamma h\epsilon}{550}\text{, the } h \text{ is design head } h_d$$

$$h_d = 3247 - 3163.5 - 0.5 = 83 \text{ ft}$$

Assuming an efficiency, $\epsilon = 0.90$, we obtain

$$p_{hp} = \frac{1075(62.4)(83)(0.90)}{550} = 9111 \text{ hp}$$

Using Eq. (4.17), we have

$$n_s = \frac{nP^{0.5}}{h^{1.25}}$$

Next, we solve for a preliminary value of speed n':

$$n' = \frac{n_s h^{1.25}}{P^{0.5}}$$

Taking a value for n_s from Fig. 4.3 when $h_d = 83$ ft, $n_s = 100$, and

$$n' = \frac{100(83)^{1.25}}{(9111)^{0.5}} = 262.5 \text{ rpm}$$

Using the synchronous speed equation, Eq. (4.30),

$$n = \frac{7200}{N_p}$$

the number of poles would be

$$N_p = \frac{7200}{262.5} = 27.4$$

Use the nearest even number of poles, 28, to obtain a synchronous speed.

$$n = \frac{7200}{28} = 257.1 \text{ rpm}$$

Now find the actual n_s.

$$n_{s\,\text{actual}} = \frac{nP_{\text{hp}}^{1/2}}{h^{5/4}}$$

$$= \frac{(257.1)(9111)^{1/2}}{(83)^{5/4}} = 97.96$$

For a propeller turbine

$$d_3 = 115.67 n_s^{2/3} \frac{\sqrt{h}}{n} \quad \text{from Eq. (4.35)}$$

$$= \frac{(115.67)(97.96)^{2/3}\sqrt{83}}{257.1}$$

$$= 87.12 \text{ inches} = 7.26 \text{ feet}$$

Now, using Fig. 7.5, solve for a limiting plant sigma, σ:

$$\sigma = \frac{n_s^{1.64}}{4325} = \frac{(97.96)^{1.64}}{4325} = 0.426$$

Interpolating h_a from Table 7.2 is

$$h_a = 30.43 - \frac{158.5}{500}(30.43 - 29.88)$$

$$= 30.26 \text{ ft}$$

Also from Table 7.2,

$$h_v = 0.84$$

Using Eq. (7.12), we then have

$$h_b = h_a - h_v = 30.26 - 0.84 = 29.42 \text{ ft}$$

In this case the normal elevation of the headwater is taken for maximum headwater elevation to calculate h_{cr}.

$$h_{cr} = 3247 - 3158.5 - 0.5 = 88 \text{ ft}$$

so that from Eq. (7.10),

$$h_s = h_b - \sigma h_{cr} = 29.42 - (0.426)(88) = -8.07$$

From Fig. 7.6, $b = Kd_3$ and K for this case is

$$n_s = 97.96 \qquad K = 0.284$$

so that $b = (0.284)(7.26) = 2.06$ ft and

$$\text{Elevation of centerline of runner} = 3158.5 - 8.07 + 2.06$$

$$= \mathit{3152.6\ ft} \quad \text{ANSWER}$$

U.S. Army Corps of Engineers Procedure

The Corps of Engineers method recognizes that there must be a limiting or starting value for defining the cavitation coefficient, σ. This value is called *critical plant sigma* and requires that model testing has been done to generate efficiency versus sigma curves. The Corps of Engineers defines critical runner sigma as the point at which there occurs a 1% decrease in power or efficiency, whichever occurs first. This definition follows the pattern defined in Fig. 7.4.

Some turbine units do not have the typical cavitation curves of Fig. 7.3 and the shapes may vary as shown in Fig. 7.4. For cavitation curves of these different forms, the Corps of Engineers has arbitrarily defined critical runner sigma, σ_{cr}, indicated as σ_1, in Fig. 7.4, and has taken the higher value that is determined in such a graphical evaluation. Because of the uncertainty involved, such as knowledge of the prototype temperature, it is customary to work with a safety margin. The Corps of Engineers defines safety margin, SM, as being equal to the minimum difference (in feet of water) between critical sigma and plant sigma, expressed in equation form as

$$SM = h(\sigma_p - \sigma_{cr}) \tag{7.13}$$

Physically, the plant sigma is fixed by the turbine setting and the operating level of the tailwater with respect to the headwater level. The Corps of Engineers also specifies the reference point or elevation, KS, for determining plant sigma. For vertical-axis Kaplan turbines and propeller turbines, the reference point is the centerline of the blades. For vertical-axis Francis turbines, the point is usually the bottom of the runner vanes or at the top of the draft tube. For horizontal-axis machines, the upper tip of the runner blade may be used.

Most purchasers of turbines require a guarantee of performance with respect to cavitation. Recognizing that it is almost impossible to have turbines operate free of cavitation, the guarantee is usually expressed in a maximum rate of metal removal during the guarantee period, usually one year. The maximum metal removal rate allowed for mild steel by the U.S. Corps of Engineers contracts is $0.2D^2/8000$ lb per operating hour each year, where D equals the runner diameter in feet. A lower rate of $0.02D^2/8000$ lb is usually specified for stainless steel runners. This includes the runner and other parts of the turbine installation, such as the discharge ring. To illustrate the U.S. Corps of Engineers method, an example problem is presented next.

Example 7.2

Given: A proposed hydro development is to have four horizontal-shafted 5-MW turbines that will be operating under a rated net head of 16 ft. The turbine runners have been preliminarily sized at a diameter of $d_3 = 217$ in., with a speed, n, of 54.5 rpm. The atmospheric pressure head, h_a, is 33.7 ft of water; the vapor pressure head, h_v, is 0.6 ft of water; and the minimum tailwater elevation is 416 ft MSL. Figure 7.7 shows the results of a model test for a unit of the proper specific speed for the given site conditions.

Required: Using the model test curve of Fig. 7.7, find the following:

(a) Required turbine centerline elevation allowing for a safety margin of 10 ft
(b) Safety margin if the turbine is operated under a head of 13 ft
(c) Turbine horsepower output at the rated head

In developing the model curves the following definition of speed coefficient, ϕ, and unit power, p_1, were used:

$$\phi = \frac{nd_p}{1839\sqrt{h}} \qquad \text{[see Eq. (4.3)]} \tag{7.14}$$

$$p_1 = p\left(\frac{d_p}{D_m}\right)^2 \frac{1}{h^{1.5}} \quad \text{[from Eq. (4.13)]} \tag{7.15}$$

$$p_1 = p\left(\frac{d_p}{12}\right)^2 \frac{1}{h^{1.5}}$$

where d_p = prototype runner diameter, in.
D_m = model runner diameter, in. = 12 in.

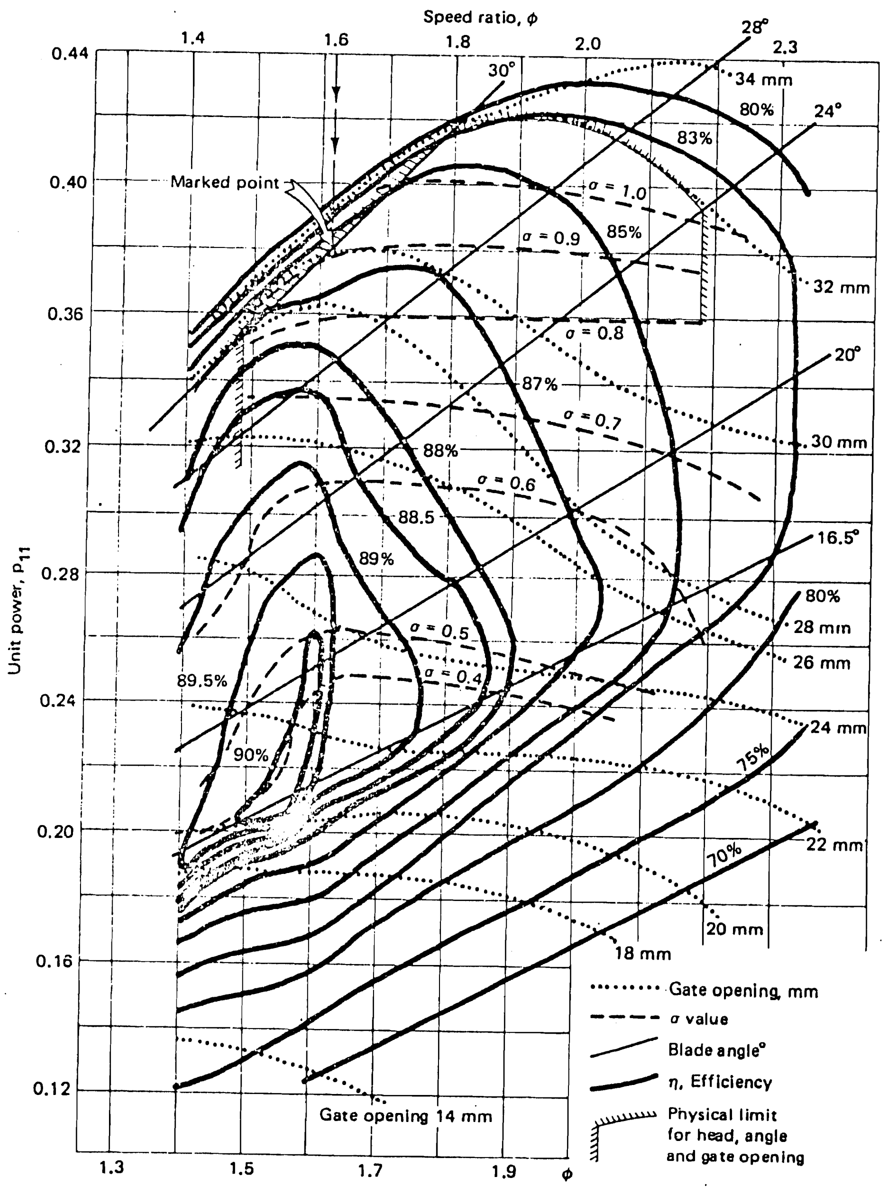


Figure 7.7 Model test curve with values of cavitation coefficient indicated. SOURCE: U.S. Army Corps of Engineers.

Analysis and solution:

(a) Calculating a value for ϕ from Eq. (7.14), we have

$$\phi = \frac{nd_p}{1839\sqrt{h}} = \frac{(54.5)(217)}{1839\sqrt{16}} = 1.61$$

From the model test Fig. 7.7 with $\phi = 1.61$,

$$\sigma_{cr} = 0.91 \qquad \text{as shown by the marked lines}$$

The 0.91 is just inside the operating range indicated by the boundary marking the physical limit of the operating region of the unit as tested. This is where the power output indicated by the unit power, p_1, is a maximum for that value of speed coefficient, ϕ.

Using the U.S. Corps of Engineers definition of safety margin, SM, we have

$$\text{SM} = h(\sigma_p - \sigma_{cr})$$

Then

$$\sigma_p = \frac{\text{SM}}{h} + \sigma_{cr}$$

$$= \frac{10}{16} + 0.91 = 1.535$$

Recognizing from Eq. (7.8) that

$$\sigma_p = \frac{h_a - h_v - h_s}{h} = 1.535$$

then

$$h_s = -16(1.535) + 33.7 - 0.6 = 8.54 \text{ ft}$$

$$\text{Bottom of runner elevation} = \text{tailwater elevation} + 8.54$$

$$= 416 + 8.54 = 424.5 \text{ ft MSL}$$

$$\text{Runner centerline elevation} = 424.5 - \frac{d_p}{2(12)} = 424 - \frac{217}{2(12)}$$

$$= \mathit{415.5\ ft} \quad \text{ANSWER}$$

(b) For the operating head $h = 13$, from Eq. (7.14),

$$\phi = \frac{nd_p}{1839\sqrt{h}} = \frac{(54.5)(217)}{1839\sqrt{13}} = 1.78$$

$\sigma_{cr} = 1.18$ from the model test curve, Fig. 7.7. Since σ_p will not change, then

$$\text{SM} = (\sigma_p - \sigma_{cr})h = (1.535 - 1.18)13$$

$$= \mathit{5.65\ ft} \quad \text{ANSWER}$$

This is less than the specified value of 10 ft, so the setting may need to be modified and be governed by the minimum operating head, h_{min}.

(c) At rated head, h_d, of 16 ft, p_1 = 0.38 from the model curve Fig. 7.7. Using Eq. (7.15) yields

$$p = p_1\left(\frac{d_p}{12}\right)^2 h^{1.5}$$

$$= 0.38\left(\frac{217}{12}\right)^2 (16)^{1.5} = 7953\ hp \quad \text{ANSWER}$$

Assuming a generator efficiency of 0.97 at maximum output, the kilowatt power output should be

$$P_{kW} = (0.97)(0.746)(7953) = 5750 \text{ kW}$$

Based on the rated output of 5 MW, this indicates

$$\frac{5.75}{5}(100) = 115\%$$

and that the turbines would be operating at 115% of rated capacity, which is quite common in U.S. Army Corps practice.

One caution should be mentioned: this method assumes that model test data are available. The U.S. Corps of Engineers has, as a result of normal contract requirements, accumulated a representative group of such curves, but normally model test curves are not readily available. A method of estimating the value of critical sigma should be obtained and thus experience curves like Fig. 7.5 are useful.

Manufacturers' Procedures

Each manufacturer has sets of model test curves for its own designs so that an approach to determining turbine setting can be similar to that of the U.S. Corps of Engineers method.

One company, KaMeWa of Sweden (Lindestrom, n.d.), gives a useful approach for preliminary planning which is more a rule-of-thumb approach: that the submergence of the unit centerline should be

$$Z = -KD \tag{7.16}$$

where Z = total suction head, m

D = runner diameter, m

K increasing from K = 0.5 for a design net head H_d of 10 m to K = 1.0 for a H_d = 30 m net head. This is for vertical-axis machines and would normally apply for Kaplan and fixed-blade turbines.

Experience curves compiled by KaMeWa from worldwide installations and KaMeWa units manufactured since 1970 are shown in Fig. 7.8. Superimposed on the experience curve of KaMeWa is the curve of the U.S. Department of the Interior

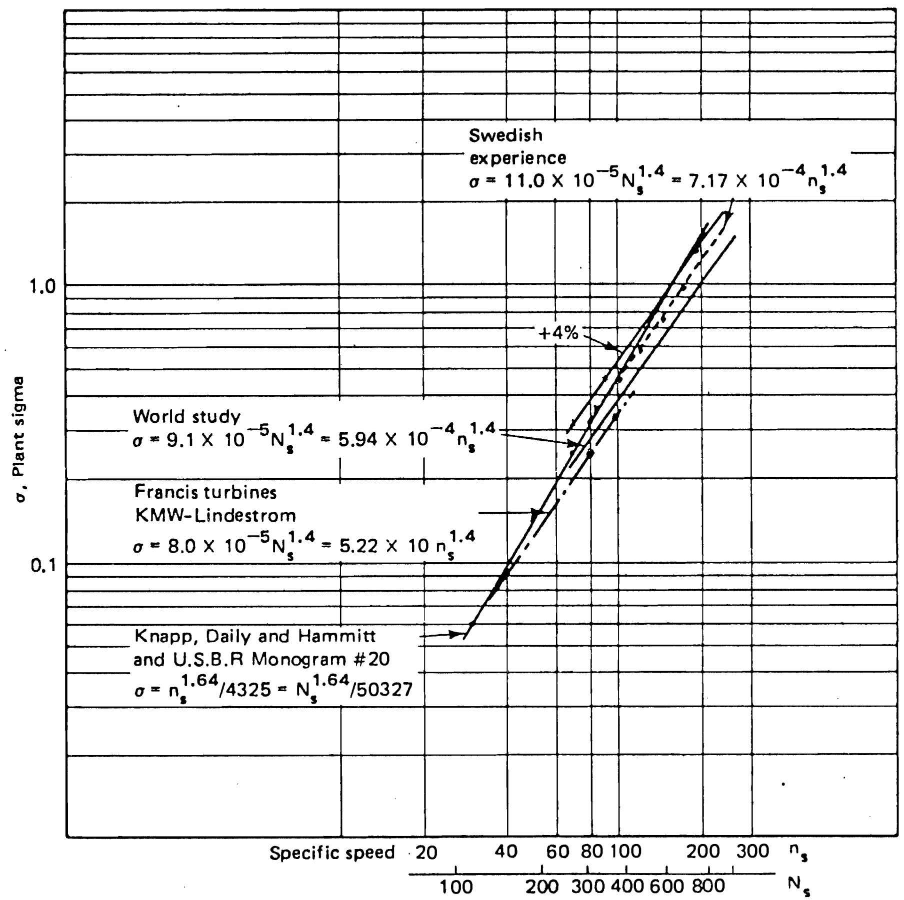


Figure 7.8 Comparison of experience curves for cavitation coefficient.

(1976) for comparison. The manufacturing companies all caution that final design and the decision for setting elevation, together with the assignment for an admissible value of sigma, σ_{adm}, should be carried out by the manufacturer since the manufacturer must be responsible for the guarantee for cavitation performance.

The common procedure at J. M. Voith GmbH (1976) is to choose a setting lower by a safety margin according to the estimate of

$$|\Delta h_s| = \Delta\sigma(H) \tag{7.17}$$

This mathematical check is made in addition to the determination of H_s according to the inception of cavitation indicated by model test results. Such a precaution is to ensure that the prototype turbine will be largely cavitation free. The safety margin $|\Delta H_s|$ is chosen to compensate for uncertainty resulting from inaccuracies of manufacturing. The amount of $|\Delta H_s|$ is chosen with consideration of turbine speed,

n, and the rated and operating heads. J. M. Voith GmbH recommends that a value of $\Delta\sigma = 0.1$ for low-specific-speed turbines and a $\Delta\sigma = 0.2$ for high-specific-speed turbines be used. For an operating head of 50 m, this results in $|\Delta H_s| = 10$ m.

For Francis runners, J. M. Voith GmbH indicates that it is possible, without serious trouble, to apply the principle that $\sigma_p > \sigma_{\text{beg}}$. The σ_{beg} is the first visual indication of cavitation.

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PROBLEMS

7.1. Using the data in Example 7.1, determine the satisfactory turbine setting centerline elevation if a synchronous speed of 277 rpm is chosen instead of $n = 257.1$ rpm. Comment on what variation in speed does to project cost.

7.2. A hydro installation is planned for which the full-gate output of the turbine would be 6000 kW under a rated net head of 18 ft. It is proposed to have a diameter of 203 in. and to operate at a speed of 60 rpm. Assume that the turbine is to be homologous to the model for which curve data are available in Fig. 7.7. Select a satisfactory turbine setting with a safety margin of 8 ft if the minimum tailwater elevation is 1000 ft MSL and the maximum water temperature is 60°F. Consider that the model turbine diameter, d, is 12 in.

7.3. Using a method other than the U.S. Corps of Engineers procedure, check the setting elevation determined in Problem 7.2. For that problem, what would be the turbine discharge at full gate?

7.4. Obtain data from a power plant in your vicinity and determine the plant sigma at minimum tailwater and maximum power output. Compare your results with limiting values as indicated by Fig. 7.5 or Fig. 7.8.

WATER PASSAGES 8

Necessary components of the hydraulic turbines in a hydropower installation are specially designed water passages and gates for controlling and directing the water as it flows to, through, and from the turbines. The principal features to consider in engineering feasibility and design studies are the flumes, penstocks, gates and valves, spiral cases, and draft tubes.

OPEN FLUMES

In very simple low-head installations the water can be conveyed in an open channel directly to the runner. Open flume settings of turbines do require a protective entrance with a trash rack. The principal problem to be solved is to provide inlet conditions to the turbine that are relatively free from swirling and vortex flow as the water approaches the turbine runner. An example of an open flume setting of a hydro installation is shown in Fig. 8.1. A usual upper limit of the use of open flume settings for hydraulic turbines is that heads should not exceed 20 ft. Flumes and canals are also used to convey water to penstocks for turbine installations with higher heads.

PENSTOCKS

A penstock is the conduit that is used to carry water from the supply sources to the turbine. This conveyance is usually from a canal or reservoir.

Penstocks can be classified as to operational type and as to the type of con-

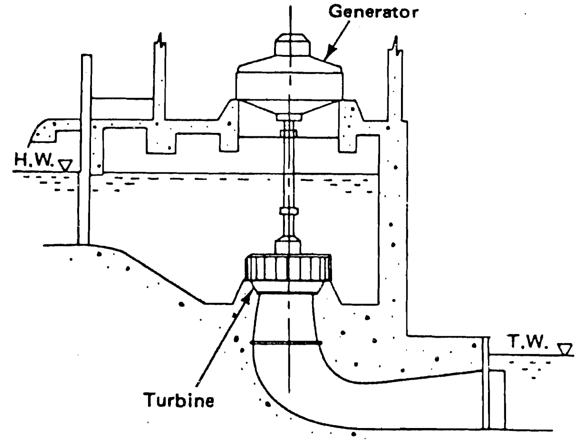


Figure 8.1 Diagram of open-flume, low-head turbine installation.

struction. Two operational types are the pressure penstock and the siphon penstock. The *pressure penstock* requires that the water discharging to the turbine always be under a positive pressure (greater than atmospheric pressure). The *siphon penstock* is constructed in such a way that at points in the penstock the pressure may be less than atmospheric pressure and sections of the conduit act as a siphon. This requires that a vacuum pump or some other means for initiating the siphon action must be used to fill the conduit with water and to evacuate air in the conduit. Figure 8.2 shows a simple diagram of a siphon penstock that has been installed in Finland.

Penstocks may be classified according to type of construction, for example:

1. Concrete penstock
2. Fiberglass or plastic pipe
3. Steel penstock
4. Wood stave pipe

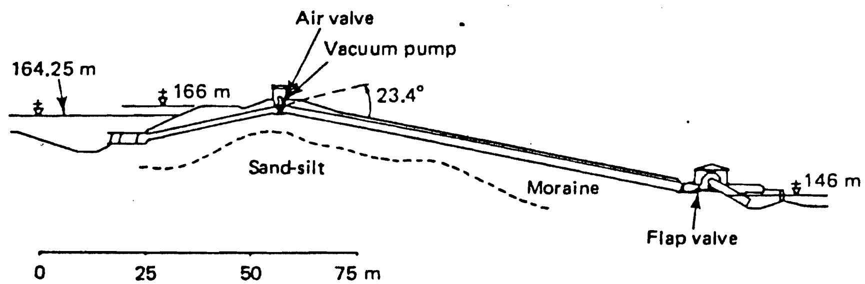


Figure 8.2 Diagram of siphon penstock type of hydropower installation (Kaarni Power Station, Finland). SOURCE: Imatran Volma OY.

Cast-in-place or precast reinforced concrete pipe can be used for penstocks. Very large diameters are somewhat impractical. Cast-in-place concrete pipes are usually limited to heads of less than 100 ft. According to Creager and Justin (1950), precast reinforced concrete penstocks can be used up to 12.5 ft in diameter and under heads up to 600 ft by using a welded steel shell embedded in the reinforced concrete.

Fiberglass and polyvinyl chloride (PVC) plastic pipe have proven to be useful for penstocks. A penstock at the Niagara Mohawk plant uses a fiberglass pipe 10 ft (3 m) in diameter.

Wood stave pipes have been used in diameters ranging from 6 in. up to 20 ft and utilized at heads up to 600 ft with proper design. Useful information for the design of wood stave pipes is contained in the handbook by Creager and Justin (1950).

Steel penstocks have become the most common type of installation in hydropower developments due to simplicity in fabrication, strength, and assurance that they will perform in a wide variety of circumstances. Normal practice is to use welded steel pipe sections. An excellent U.S. Department of the Interior monograph (1967) treats the topic of steel penstocks. This covers the many details of making selection of size and design considerations as to stresses and structural mounting.

For purposes of engineering feasibility and preliminary design, there are three major considerations that need engineering attention: (1) the head loss through the penstock, (2) the safe thickness of the penstock shell, and (3), the economical size of the penstock. Another consideration might be the routing of the penstock.

Head Loss in Water Passages

The head losses consist of the following:

1. Trash rack losses
2. Entrance losses
3. Stop log, gate slot, and transition losses
4. Friction losses in the pipe
5. Bend losses

According to the U.S. Department of the Interior (1967), losses in trash rack and entrance can be taken at 0.1, 0.2, and 0.5 ft of head, respectively, for velocities of 1.0, 1.5, and 2.5 ft/sec at the penstock entrance. *Engineering Monograph No. 3* further indicates that entrance losses for bell-mouthed entrances would be 0.05 to 0.1 times the velocity head at entrance and for square-mouthed entrances the loss would be 0.2 times the velocity head at entrance.

Pipe friction losses can be determined by several reliable equations, monographs, and tables that are available in hydraulic literature. Equations that are suggested for use are presented here in their usual from:

Scobey equation:

$$h_f = K_s \frac{V^{1.9}}{D^{1.1}} \tag{8.1}$$

where h_f = head loss, ft per 1000 ft of pipe
K_s = loss coefficient
V = velocity of flow in pipe, ft/sec
D = diameter of pipe, ft

The K_s varies from 0.32 for new smooth pipe to 0.52 for rough pipe. More precise values for K_s can be obtained from various handbooks.

Manning equation:

$$V = \frac{1.49}{n} R^{2/3} S^{1/2} \tag{8.2}$$

and

$$S = \frac{h_f}{L}$$

where R = hydraulic radius, A/P, ft
A = cross-sectional area of pipe, ft^2
P = wetted perimeter of pipe, ft
S = slope of hydraulic grade line
h_f = head loss in pipe, ft
L = length of pipe, ft
n = roughness coefficient.

The n varies from 0.010 for smooth pipes to 0.017 for rough pipes. More precise values for n can be obtained from various handbooks and suppliers' literature for fiberglass and plastic pipe.

Darcy-Weisbach equation:

$$h_f = \frac{fLV^2}{2Dg} \tag{8.3}$$

where h_f = head loss in pipe, ft
f = a numerical friction factor
g = acceleration of gravity, ft/sec^2.

According to the U.S. Department of the Interior (1977), Colebrook and White have developed an equation that relates the friction factor to pipe characteristics and Reynolds number. The equation is difficult to use to get the friction factor, f, directly. For more practical use of the equation, a diagram has been developed for obtaining the friction factor, f. This is often referred to as a Moody or Stanton diagram (Moody, 1944; Rouse, 1943). A practical presentation of this type of diagram for determining head loss in pipes is presented by the U.S. Department of the Interior (1977). This is presented at a good scale for concrete pipes, steel and cast iron pipes, riveted steel pipes, and wood stave pipes. An often useful initial approach can be to use $f = 0.02$ in the Darcy-Weisbach equation to get a beginning estimate of head loss. A more recent sophisticated approach to solving for head loss utilizing the equations of Colebrook and White is the work of Barr (1976).

Head loss in bends is given by the following formula:

$$h_b = C \frac{V^2}{2g} \tag{8.4}$$

where h_b = head loss in bend, ft

C = experimental loss coefficient.

The U.S. Department of the Interior (1967) presents two experimental curves for determining values of C for various ratios of radius of bend over pipe diameter and the deflection angles of the bends. These curves are reproduced in Fig. 8.3.

The head loss in valves is given by the equation

$$h_v = K \frac{V^2}{2g} \tag{8.5}$$

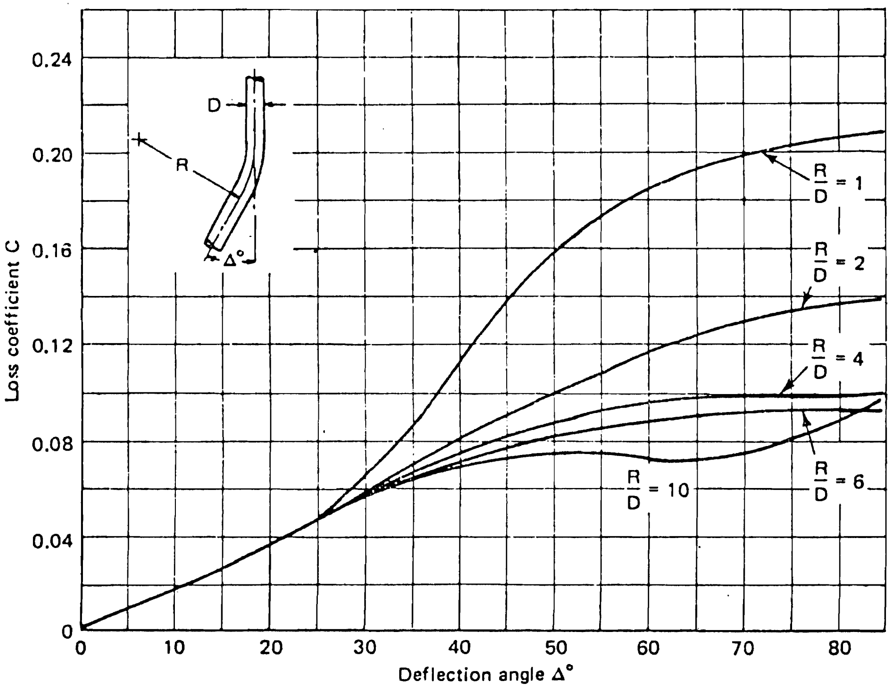


Figure 8.3 Head loss coefficient for pipe bends. SOURCE: Engineering Monogram No. 3, U.S. Bureau of Reclamation.

where h_v = head loss in valve, ft

K = loss coefficient.

The U.S. Department of the Interior (1967) gives the following values for K: for large gate valves, $K = 0.10$; for needle valves, $K = 0.20$; and for medium-size butterfly valves with a ratio of leaf thickness to diameter of 0.2, $K = 0.26$.

Safe Penstock Thickness

The thickness of the pipe shell for penstocks should be determined by using the following equation:

$$t = \frac{pD'}{2ef_t} \tag{8.6}$$

where t = penstock shell thickness, in.

p = internal pressure, lb/in^2

D' = pipe diameter, in.

e = joint efficiency of welded or riveted joint

f_t = allowable unit stress of hoop tension, lb/in^2.

Minimum thickness, based on need for stiffness, corrosion protection, and strength requirements, is indicated by the U.S. Department of the Interior (1967) to be

$$t_{\min} = \frac{D' + 20}{400} \tag{8.7}$$

The allowable equivalent unit stress for hoop tension will vary with the type of steel used in the penstock. The ASME Code for Unfired Pressure Vessels gives maximum allowable stresses for various types of steels used in penstocks. If the efficiency of the welded joints is assumed to be 0.9 or more, all the longitudinal welded joints of the penstock must be 100% radiographed. Circumferential welded joints are stressed to one-half the value for the longitudinal joints. See the ASME Code for the Unfired Pressure Vessels, Section VIII, for other instructions.

Size Selection of Penstocks

Various experience curves and empirical equations have been developed for determining the economical size of penstocks. Some of these equations use very few parameters to make initial size determinations for reconnaissance or feasibility studies. Other more sophisticated equations use many variables to obtain more precise results which may be necessary for final design. Economical size varies with type of installation and materials, as well as whether used above ground or buried.

Gordon and Penman (1979) give a very simple equation for determining steel penstock diameter for small hydropower installations:

$$D_p = 0.72Q^{0.5} \tag{8.8}$$

where D_p = penstock diameter, m
Q = water flow, m^3/sec.

Sarkaria (1979) developed an empirical approach for determining steel penstock diameter by using data from large hydro projects with heads varying from 187 ft (57 m) to 1025 ft (313 m) and power capacities ranging from 206,000 hp (154 MW) to 978,000 hp (730 MW). He reported that the economical diameter of the penstock is given by the equation

$$D = \frac{4.44P^{0.43}}{h^{0.63}} \tag{8.9}$$

where D = economical penstock diameter, ft
p = rated turbine capacity, hp
h = rated net head, ft

The study verified his earlier study reported in 1958. The two empirical studies giving Eqs. (8.8) and (8.9) were for periods before present energy crunch conditions and did not take into account penstock length or the cost of lost power in penstock flow. Therefore, the equations should be used with caution.

The U.S. Department of the Interior (1967) gives a very sophisticated graphical and empirical approach to determining the economical diameter of steel penstocks which includes the following variables:

Cost of pipe per pound, installed, in dollars
Value of power lost, in dollars per kWh
Plant efficiency
Pipe joint efficiency
Weighted average head, including water hammer effect, in feet (based on design head)
Friction coefficient in Scobey formula (0.34)
Ratio of overweight to weight of pipe shell
Flow at design head, in ft^3/sec
Allowable hoop tension stress, in lb/in^2
Weighted average plate thickness, in inches

This approach is presented in a series of graphs, with a step-by-step example.

Penstocks that are not encased in the concrete of a dam or buried in earth and rock require support. Details on the engineering analysis of the forces acting on and the structural design of support piers and anchors are available from the U.S. Department of the Interior (1967). An excellent reference on structural design temperature problems and construction requirements of penstocks is a paper by Eberhardt (1975). A rule of thumb used by some engineers in penstock design and selection is that the maximum water velocity should not exceed 35 ft/sec.

SPIRAL CASES AND DISTRIBUTOR ASSEMBLIES

Distributor assemblies consisting of spiral cases, head cover, bottom ring, and wicket gates are used to control the amount and direction of the water entering into the rotating turbine. For low heads, below 20 ft (6.1 m) an open-flume or pressure case setting may be used. These are simply rectangular or cylindrical chambers. Register and cylindrical gates were used in early hydropower installations; they were basically a cylinder with rectangular openings that could be moved circumferentially to block or permit the flow into the turbine or a cylinder that was moved axially to permit flow. Normally, now to obtain better part load efficiency, wicket gates control flow into the turbine within the open flume or within the pressure case.

For large vertical-axis turbines, semispiral or reinforced concrete spiral cases are used. A simplified drawing showing the arrangement for a semispiral concrete case is shown in Fig. 8.4. The advantage of the semispiral case is lower head loss. The head loss is also lower for an open-flume arrangement than for more sophisticated spiral casings. For large-, medium-, and high-head turbines the spiral case is usually fabricated from steel plate. Older installations sometimes had cast steel or

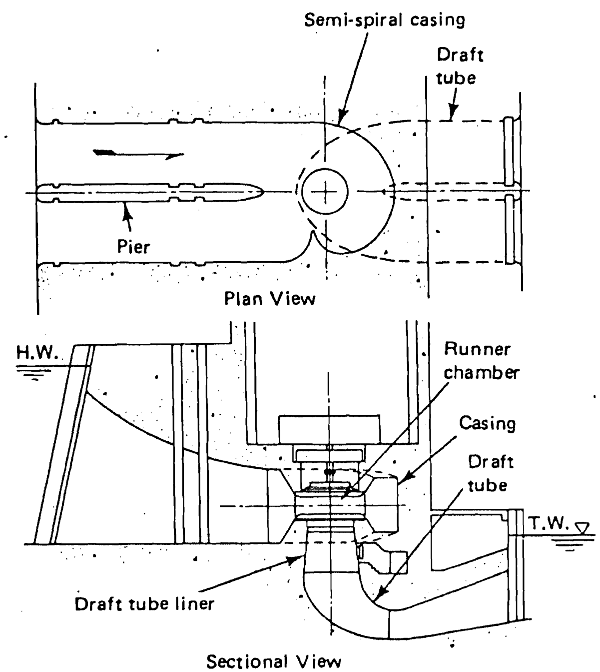


Figure 8.4 Simplified drawing of typical semi-spiral concrete case. SOURCE: Allis-Chalmers Corporation.

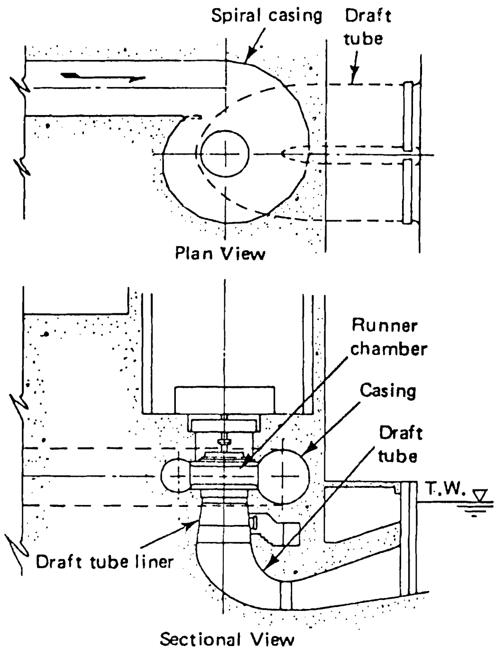


Figure 8.5 Simplified drawing of typical spiral case. SOURCE: Allis-Chalmers Corporation.

cast iron spiral cases. Figure 8.5 is a simplified drawing showing the characteristics of a spiral case. The water passageways are different for the various types of turbines. For very preliminary planning, the sketches in Fig. 8.6 are useful to indicate the relative dimensions and sizes.

Water velocities and uniformity of flow are primary engineering concerns. Water velocities in the spiral casing as a rule of thumb, according to Brown (1970), should, for low-specific-speed turbines, be approximately

$$v = 0.14\sqrt{2gh} \tag{8.10}$$

and, for high-specific-speed turbines,

$$v = 0.20\sqrt{2gh} \tag{8.11}$$

where v = water velocity at cross sections normal to the radius and at entrance to spiral case, ft/sec

h = design net head, ft

Generally, there should be no deceleration of water as it flows from the penstock to and through the spiral case. Special requirements as to shape, dimensioning, and velocity are needed for different kinds of turbines. Water passageways for each type of turbine will be discussed separately as to particular requirements and characteristics.

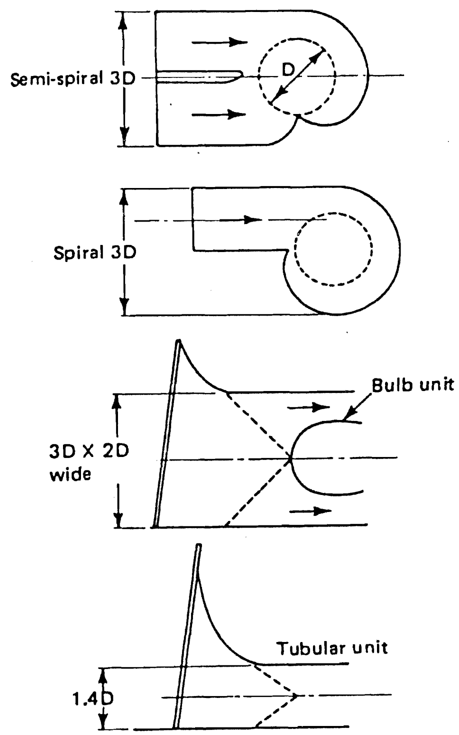


Figure 8.6 Relative dimensions for intakes and cases for different types of turbines and cases. SOURCE: Allis-Chalmers Corporation.

Spiral Cases and Manifolds for Impulse Turbines

The work of deSiervo and Lugaresi (1978) provides excellent experience curves and empirical equations for engineering design of the setting and penstock for impulse turbines. The dimensions of the casing for housing a vertical shaft impulse turbine depend primarily on the outside diameter, D_3, of the turbine runner. Figure 8.7, a drawing for dimensioning, identifies the controlling dimensions for the casing of an impulse-type turbine. For a circular-type casing, the plan diameter, L, is given by the empirical equation

$$L = 0.78 + 2.06D_3 \tag{8.12}$$

where L = casing diameter, m

D_3 = outer diameter of turbine runner, m

The other dimensions in the drawing that provide controlling size characteristics are indicated in the following equations:

$$G = 0.196 + 0.376D_3 \tag{8.13}$$

$$F = 1.09 + 0.71L \tag{8.14}$$

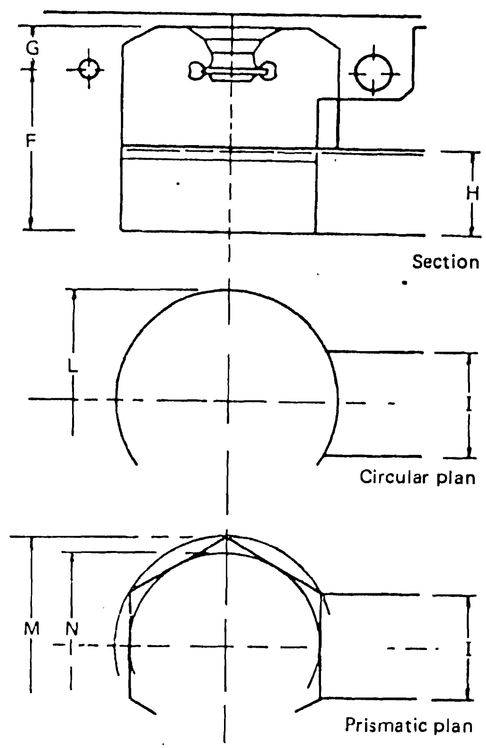


Figure 8.7 Primary dimensions for casings for Pelton-type turbines.

$$H = 0.62 + 0.513L \tag{8.15}$$

$$I = 1.28 + 0.37L \tag{8.16}$$

where G = distance between turbine runner centerline and top of casing, m
F = distance between turbine runner centerline and bottom of casing pit, m
H = height of tailrace exit, m
I = width of tailrace exit, m.

A prismatic layout shape of casing with dimensions M and N can be determined from properties of a hexagon (see Fig. 8.7). An impulse turbine covered with a steel casing also requires a penstock extension or manifold that supplies water to the nozzles. The water velocity at the entrance to the spiral casing, according to deSiervo and Lugaresi (1978), is given as a function of design head, H_n, by the equation

$$v = 0.82 + 0.358\sqrt{H_n} \tag{8.17}$$

where v = entrance velocity of the spiral scroll case, m/sec
H_n = net head on turbine, m.

Normally, a maximum velocity of 30 ft/sec is specified. Figure 8.8 presents a dimensioning layout for a typical four-nozzle spiral case for supplying water to the nozzles of an impulse turbine. Shown in the drawing are the principal controlling

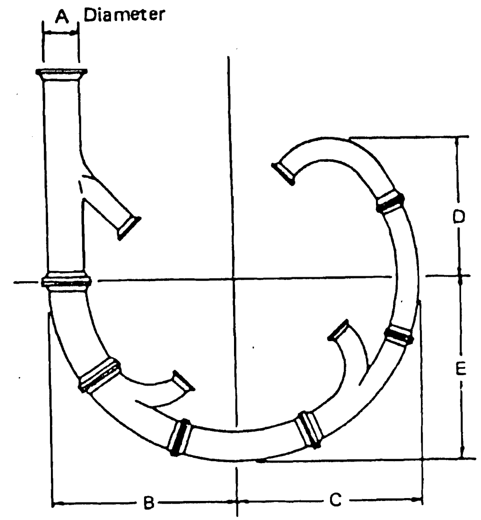


Figure 8.8 Dimensioning layout for typical four-nozzle spiral casing for Pelton turbines.

dimensions. The dimension A, supply pipe diameter, is determined by using Eq. (8.17) and relating it to the design discharge for the turbine. The values for the respective layout dimensions B, C, D, and E can be determined in relation to the turbine casing diameter, L, using the following empirical equations developed by deSiervo and Lugaresi:

$$B = 0.595 + 0.694L \tag{8.18}$$

$$C = 0.362 + 0.68L \tag{8.19}$$

$$D = -0.219 + 0.70L \tag{8.20}$$

$$E = 0.43 + 0.70L \tag{8.21}$$

The letter symbols are identified dimensionally in Fig. 8.8.

Spiral Cases for Francis Turbines

The work of deSiervo and deLeva (1976) provides excellent experience curves and empirical equations for engineering design of spiral cases for Francis-type turbines. The empirical curves relate D_3, the runner diameter, through Eq. (4.31) to the net head H_n, specific speed N_s, and runner speed n. Figure 8.9 is a dimensioning layout that identifies the various controlling dimensions that are characteristic of this type of spiral case. Once the discharge diameter, D_3, is determined by using Eq. (4.31), all the other dimensions can be empirically related to that D_3 and N_s as developed by deSiervo and deLeva (1976). The absolute velocity of the water at the inlet section to the spiral casing, which has a diameter A as shown in Fig. 8.9, is given by the equation

$$v = 844N_s^{-0.44} \tag{8.22}$$

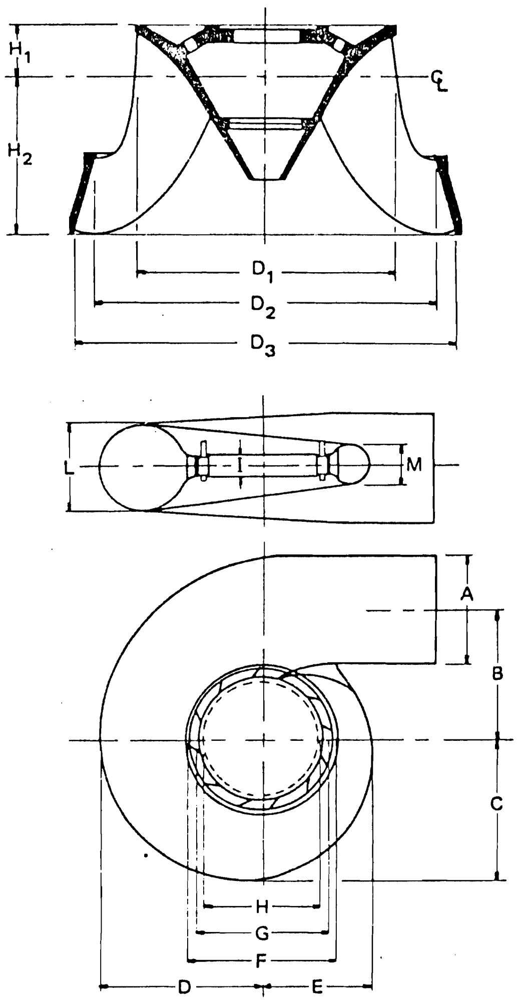


Figure 8.9 Dimensioning layout for typical steel spiral case for Francis turbines.

where v = water velocity at the inlet to the spiral casing, m/sec

N_s = turbine specific speed.

According to deSiervo and deLeva, the dimensions of the spiral casing depend on velocity v and variation in discharge proceeding through progressive radial sections along the spiral case, as given by the equation

$$Q_\gamma = Q_0 \frac{1 - \gamma}{2\pi} \tag{8.23}$$

where Q_γ = discharge through a radial section rotated an angle γ with respect to the inlet section, m^3/sec

Q_0 = turbine rated discharge, m^3/sec.

The cross-sectional area of the spiral case must decrease so that the turbine runner is supplied with water uniformly around its circumference. The following equation must also be satisfied:

$$V_u r_1 = k \tag{8.24}$$

where V_u = peripheral velocity of the water in the scroll case, m/sec

r_1 = radial distance of a point in the spiral case from the turbine axis, m

k = constant term.

Equation (8.24) reflects the irrotationality of the water flow. The characteristic dimensions of Francis turbine runners, as shown in Fig. 8.9, necessary for planning the size and layout of the spiral cases, are determined in terms of D_3 and N_s by the following empirical equations from deSiervo and deLeva (1976):

$$\frac{D_1}{D_3} = 0.4 + 94.5N_s \tag{8.25}$$

$$\frac{D_2}{D_3} = \frac{1}{(0.96 + 0.000386N_s)} \tag{8.26}$$

$$\frac{H_1}{D_3} = 0.094 + 0.000025N_s \tag{8.27}$$

$$\frac{H_2}{D_3} = -0.05 + 42N_s \qquad \text{for } (50 < N_s < 110) \tag{8.28}$$

$$\frac{H_2}{D_3} = \frac{1}{(3.16 - 0.0013N_s)} \qquad \text{for } (110 < N_s < 350) \tag{8.29}$$

The D_1, D_2, D_3, H_1, and H_2 are identified in Fig. 8.9.

The various other controlling dimensions for the spiral cases for Francis turbines related to D_3 and N_s are given in the following empirical equations from deSiervo and deLeva (1976):

$$\frac{A}{D_3} = 1.2 - \frac{19.5}{N_s} \tag{8.30}$$

$$\frac{B}{D_3} = 1.1 + \frac{54.8}{N_s} \tag{8.31}$$

$$\frac{C}{D_3} = 1.32 + \frac{49.25}{N_s} \tag{8.32}$$

$$\frac{D}{D_3} = 1.50 + \frac{48.8}{N_s} \tag{8.33}$$

$$\frac{E}{D_3} = 0.98 + \frac{63.60}{N_s} \tag{8.34}$$

$$\frac{F}{D_3} = 1 + \frac{131.4}{N_s}. \tag{8.35}$$

$$\frac{G}{D_3} = 0.89 + \frac{96.5}{N_s} \tag{8.36}$$

$$\frac{H}{D_3} = 0.79 + \frac{81.75}{N_s} \tag{8.37}$$

$$\frac{I}{D_3} = 0.1 + 0.00065N_s \tag{8.38}$$

$$\frac{L}{D_3} = 0.88 + 0.00049N_s \tag{8.39}$$

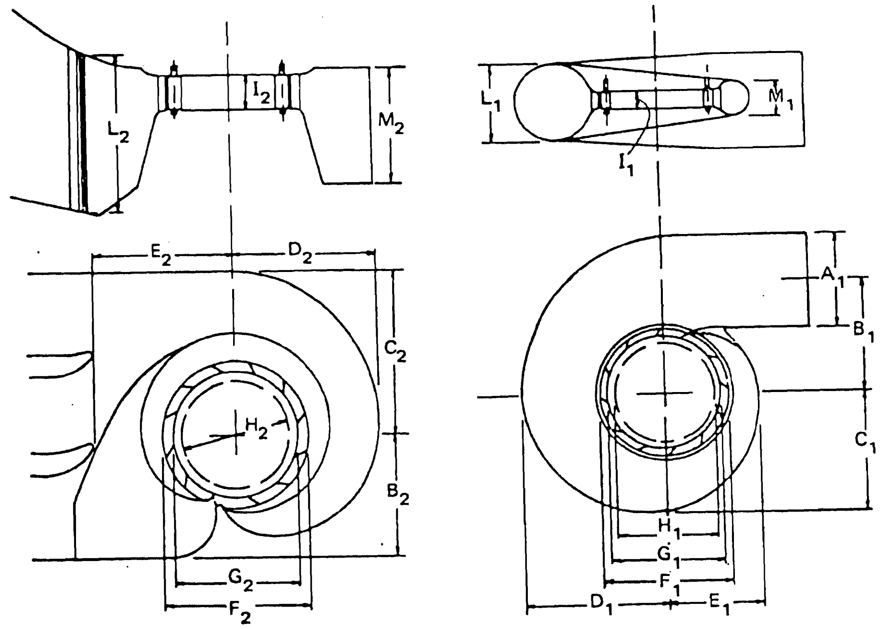
$$\frac{M}{D_3} = 0.60 + 0.000015N_s. \tag{8.40}$$

The various letter symbols are identified in Fig. 8.9.

The U.S. Department of the Interior (1976) also gives graphs and interpretive dimensional drawings for determining the characteristic size and shape of steel spiral cases for reaction-type turbines. Standardized dimensioning procedures are also presented in Allis-Chalmers Corporation's *Publication 54X10084-01* (n.d.). This publication has excellent detailed drawings that are especially labeled to show how the spiral case is connected to the stay ring and positioned with respect to the wicket gates.

Spiral Cases for Kaplan Turbines

Further work of deSiervo and deLeva (1978) provides excellent experience curves and empirical equations for engineering design of semispiral cases and spiral cases for Kaplan turbines. The various controlling dimensions are related to the discharge diameter D_M and its relation with net head H_n, specific speed N_s, and runner speed n. The value of D_M is obtained by applying Eq. (4.34). Figure 8.10 is a dimensioning layout that identifies the various controlling dimensions that are characteristic of spiral cases used for encasing Kaplan turbines. Once the discharge diameter D_M (the outer diameter of the blades) has been determined by Eq. (4.34), all the other dimensions can be determined in terms of D_M and N_s by using relations developed by deSiervo and deLeva (1978). The water velocity at the inlet to steel spiral cases for Kaplan turbines is given by the empirical equation



Concrete semi-spiral Steel spiral

Figure 8.10 Dimensioning layout for steel spiral and concrete semi-spiral case for Kaplan-type propeller turbines.

$$V_1 = 3.17 + \frac{759.21}{N_s} \tag{8.41}$$

where V_1 = water velocity at steel spiral case inlet, m/sec
N_s = turbine specific speed.

The corresponding water velocity at a concrete semispiral case inlet for Kaplan turbines is given by the empirical equation

$$V_2 = 2.44 - 1.19 \times 10^{-3}N_s \tag{8.42}$$

A means for finding the various controlling dimensions for steel spiral cases for Kaplan turbines related to discharge diameter D_M and specific speed N_s is given in the following equations developed by deSiervo and deLeva (1978):

$$\frac{A_1}{D_M} = 0.40N_s^{0.20} \tag{8.43}$$

$$\frac{B_1}{D_M} = 1.26 + 3.79 \times 10^{-4}N_s \tag{8.44}$$

$$\frac{C_1}{D_M} = 1.46 + 3.24 \times 10^{-4} N_s \tag{8.45}$$

$$\frac{D_1}{D_M} = 1.59 + 5.74 \times 10^{-4} N_s \tag{8.46}$$

$$\frac{E_1}{D_M} = 1.21 + 2.71 \times 10^{-4} N_s \tag{8.47}$$

$$\frac{F_1}{D_M} = 1.45 + \frac{72.17}{N_s} \tag{8.48}$$

$$\frac{G_1}{D_M} = 1.29 + \frac{41.63}{N_s} \tag{8.49}$$

$$\frac{H_1}{D_M} = 1.13 + \frac{31.86}{N_s} \tag{8.50}$$

$$\frac{I_1}{D_M} = 0.45 - \frac{31.80}{N_s} \tag{8.51}$$

$$\frac{L_1}{D_M} = 0.74 + 8.7 \times 10^{-4} N_s \tag{8.52}$$

$$\frac{M_1}{D_M} = \frac{1}{2.06 - 1.20 \times 10^{-3} N_s} \tag{8.53}$$

Identification of the respective dimensions is indicated in Fig. 8.10 with all terms in meters; N_s is the metric form of specific speed.

Controlling dimensions for concrete semispiral cases for Kaplan turbines related to D_M and N_s can be found by using the following empirical equations from the work of deSiervo and deLeva (1978):

$$\frac{B_2}{D_M} = \frac{1}{0.76 + 8.92 \times 10^{-5} N_s} \tag{8.54}$$

$$\frac{C_2}{D_M} = \frac{1}{0.55 + 1.48 \times 10^{-5} N_s} \tag{8.55}$$

$$\frac{D_2}{D_M} = 1.58 - 9.05 \times 10^{-5} N_s \tag{8.56}$$

$$\frac{E_2}{D_M} = 1.48 - 2.11 \times 10^{-5} N_s \tag{8.57}$$

$$\frac{F_2}{D_M} = 1.62 - 3.18 \times 10^{-5} N_s \tag{8.58}$$

$$\frac{G_2}{D_M} = 1.36 + \frac{7.79}{N_s} \tag{8.59}$$

$$\frac{H_2}{D_M} = 1.19 + \frac{4.69}{N_s} \tag{8.60}$$

$$\frac{I_2}{D_M} = 0.44 - \frac{21.47}{N_s} \tag{8.61}$$

$$\frac{L_2}{D_M} = 1.44 + \frac{105.29}{N_s} \tag{8.62}$$

$$\frac{M_2}{D_M} = 1.03 + \frac{136.28}{N_s} \tag{8.63}$$

The letter symbols are identified in Fig. 8.10; all linear terms are expressed in meters and N_s is the metric form of specific speed. Comparison of these equations shows that for the same-size turbines the width of the spiral case is less for the concrete semispiral case than for the steel spiral case, mainly because of cross-sectional shape of the spiral form.

The U.S. Department of the Interior (1976) also provides graphs and typical proportional labeled drawings for designing concrete semispiral cases. *Engineering Monograph No. 20* indicates that the following basic criteria should govern design of semispiral cases:

1. The velocity (V_2) at entrance to the semispiral case, just upstream from the stay-vane foundation cone, should be 14% of spouting velocity at design head, but in no case should V_2 be less than 5 ft/sec (1.5 m/sec).
2. Water passage sections should approach a square to minimize friction loss. The height of the entrance section should be approximately one-third the width of the intake. All interior corners should have 12-in. (0.3-m) fillets to minimize eddies.
3. The baffle vane should be in the downstream quadrant approximately 45° from the transverse centerline. To induce equal distribution of flow in the entrance, the turbine centerline is offset from the centerline of the intake passage. This offset will place one-third of the stay ring intake opening in one-third of the area of the entrance. The remaining two-thirds of the flow enters the stay ring directly and through the large semispiral passage.
4. The large semispiral is designed for uniform angular velocity (V_2) of flow around the spiral. $V_a = q/a$ and is a constant in which q is diminished in proportion to the remaining stay-ring arc and a is the radial cross-sectional area.
5. The entrance wall upstream from the large semispiral section should lie in the same vertical plane with the corresponding sidewalk of the draft tube for structural economy.

6. The intake is laid as a simple rectangular intake without intermediate piers, with a 0.7 coefficient of contraction from intake opening to casing entrance. The change of area is similar to that found in a jet issuing from an orifice. The intake opening should have a minimum depth of water above it equivalent to 0.3 the height of the opening, to avoid entraining air and to assure uniform vertical distribution of flow in the entrance section of the case.

Special structural problems with the civil works of the installation may dictate piers in the transition approaching the spiral case. Cautions for this and representative details for preparing the design are available from the U.S. Department of the Interior (1976).

DRAFT TUBES

Draft tubes are the final components of the water passages of hydropower plants and are necessary in carrying the water away from the turbine runner to the tailrace, where the water rejoins the stream channel or receiving body of water. The proper engineering design of draft tubes provides for the recovery of a portion of the velocity head as it leaves the turbine proper to recover energy and improve the efficiency of the turbine unit. The draft tube also permits utilization of the runner discharge head if the runner is set above the tailwater level. The discussion in Chapter 7 emphasized the importance of limiting the setting of the runner with respect to the tailwater.

Draft tubes usually consist of steel sections which change shape from circular to rectangular in cross section. The sections expand in cross-sectional area to decrease the water velocity with a minimum occurrence of vortexes and maintain a nearly uniform velocity at any section. The draft tube is usually formed in reinforced concrete. The principal engineering problems are determining the water velocity at the exit to the draft tube and determining the controlling dimensions of the draft tube.

Manufacturers consider the draft tube as a part of the turbine when determining an efficiency guarantee. Therefore, it is customary for the manufacturer to furnish the final dimensions for draft tube shape as limited within certain civil works restraints of the structural components of a hydropower plant.

For preliminary and reconnaissance planning the sketches of Fig. 8.11 are useful to indicate the relative dimensions and size for spacing of units and to make estimates of excavation requirements. The D_3 in these sketches refers to the discharge diameter of the turbine runner. Note that these can cover different types of turbines, including tubular turbines. Impulse turbines do not require draft tubes, but must be set above the tailwater. Open-channel-flow capacity in the pit connecting to the tailrace is necessary to carry away the water discharging from the turbine unless compressed air is provided. Like the problems of spiral case design, special requirements and characteristics prevail for the different types of turbines.

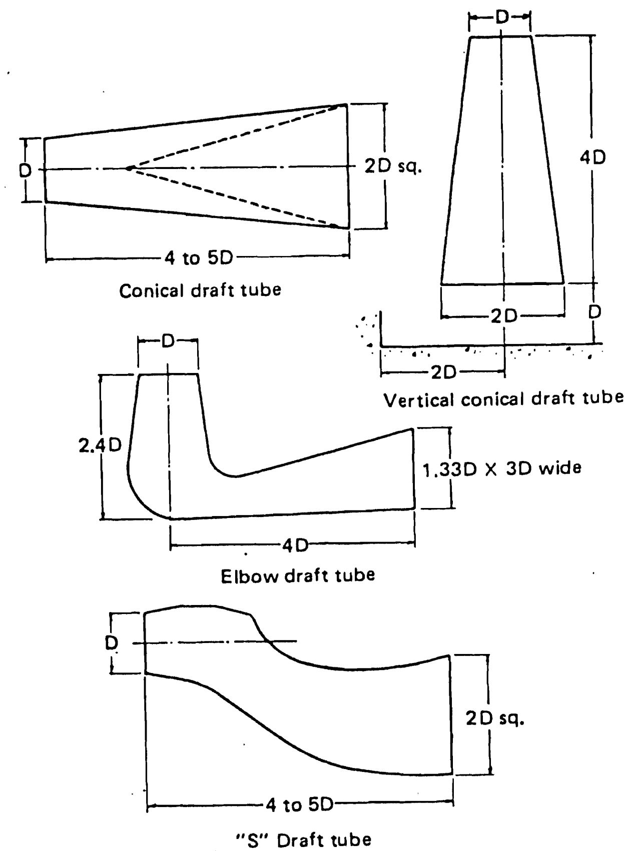


Figure 8.11 Relative dimensions for different types of draft tubes. SOURCE: Allis-Chalmers Corporation.

Draft Tubes for Francis Turbines

Empirical relations and experience curves have been developed to make preliminary determinations for draft tube design. The turbine discharge diameter, D_3, and specific speed, N_s, are used as reference parameters for developing the appropriate controlling dimensions according to deSiervo and deLeva (1976). The absolute velocity at the inlet to the draft tube is given by the following equation:

$$V_3 = 8.74 + \frac{248}{N_s} \tag{8.64}$$

where V_3 = water velocity at the draft tube inlet section, m/sec
N_s = turbine specific speed.

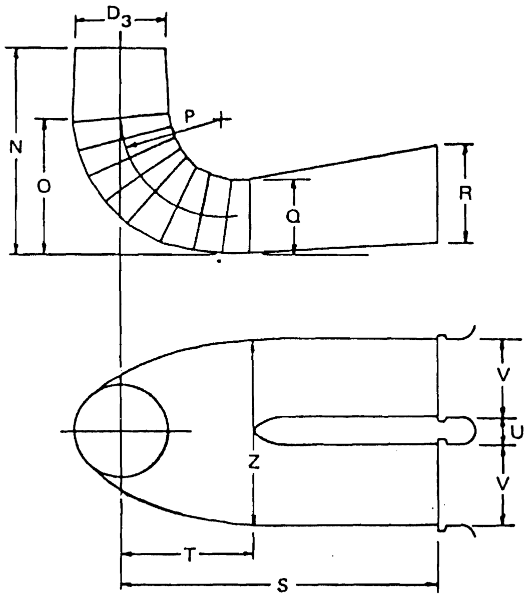


Figure 8.12 Dimensioning layout for typical draft tube for Francis turbines.

Figure 8.12, a drawing for dimensioning, identifies the controlling dimensions for the draft tube for Francis turbines. Empirical equations for determining the controlling and characteristic dimensions in terms of turbine runner diameter, D_3, and specific speed, N_s, are given by deSiervo and deLeva (1976) as follows:

$$\frac{N}{D_3} = 1.54 + \frac{203.3}{N_s} \tag{8.65}$$

$$\frac{O}{D_3} = 0.83 + \frac{140.7}{N_s} \tag{8.66}$$

$$\frac{P}{D_3} = 1.37 - 0.00056N_s \tag{8.67}$$

$$\frac{Q}{D_3} = 0.58 + \frac{22.6}{N_s} \tag{8.68}$$

$$\frac{R}{D_3} = 1.6 - \frac{0.0013}{N_s} \tag{8.69}$$

$$\frac{S}{D_3} = \frac{N_s}{-9.28 + 0.25N_s} \tag{8.70}$$

$$\frac{T}{D_3} = 1.50 + 0.00019N_s \tag{8.71}$$

$$\frac{U}{D_3} = 0.51 - 0.0007N_s \tag{8.72}$$

$$\frac{V}{D_3} = 1.10 + \frac{53.7}{N_s} \tag{8.73}$$

$$\frac{Z}{D_3} = 2.63 + \frac{33.8}{N_s} \tag{8.74}$$

Identification of the letter symbols as dimensions is indicated in Fig. 8.12.

As the specific speed increases, there is a tendency for controlling dimensions to be increased because the amount of kinetic energy within the draft tube relative to the potential energy is larger, but countering that is the higher cost for the civil works to accommodate the larger dimensions. The experience curves of deSiervo and deLeva (1976) tend to indicate that the civil works cost limitations control the relative size at higher values of specific speed, N_s.

Draft Tubes for Kaplan Turbines

The later work of deSiervo and deLeva (1978) provides the following empirical equation for determining the water velocity at the inlet to the draft tube for Kaplan turbines:

$$V_4 = 8.42 + \frac{250}{N_s} \tag{8.75}$$

where V_4 = water velocity at draft tube inlet section, m

N_s = turbine specific speed.

Figure 8.13 identifies the controlling dimensions for the draft tubes commonly used for Kaplan turbines. DeSiervo and deLeva (1978) have given empirical equations relating the controlling dimensions and characteristic dimensions to turbine runner discharge diameter, D_M, and specific speed, N_s, as follows:

$$\frac{H_t}{D_M} = 0.24 + 7.82 \times 10^{-5}N_s \tag{8.76}$$

$$\frac{N}{D_M} = 2.00 - 2.14 \times 10^{-6}N_s \tag{8.77}$$

$$\frac{O}{D_M} = 1.40 - 1.67 \times 10^{-5}N_s \tag{8.78}$$

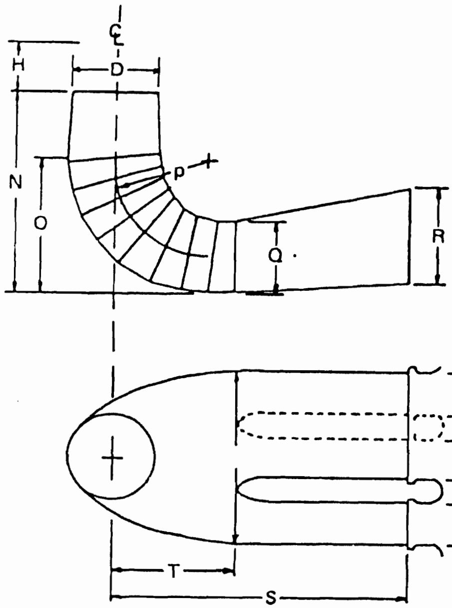


Figure 8.13 Dimensioning layout for typical draft tube for Kaplan-type propeller turbines.

$$\frac{P}{D_M} = 1.26 - \frac{16.35}{N_s} \qquad (8.79)$$

$$\frac{Q}{D_M} = 0.66 - \frac{18.40}{N_s} \qquad (8.80)$$

$$\frac{R}{D_M} = 1.25 - 7.98 \times 10^{-5} N_s \qquad (8.81)$$

$$\frac{S}{D_M} = 4.26 + \frac{201.5}{N_s} \qquad (8.82)$$

$$\frac{T}{D_M} = 1.20 + 5.12 \times 10^{-4} N_s \qquad (8.83)$$

$$\frac{Z}{D_M} = 2.58 + \frac{102.66}{N_s} \qquad (8.84)$$

The letter symbols as dimensions are identified in Fig. 8.13. The tendency is for the dimensions to remain relatively constant with respect to N_s and vary only with the turbine discharge diameter D_M for Kaplan turbines.

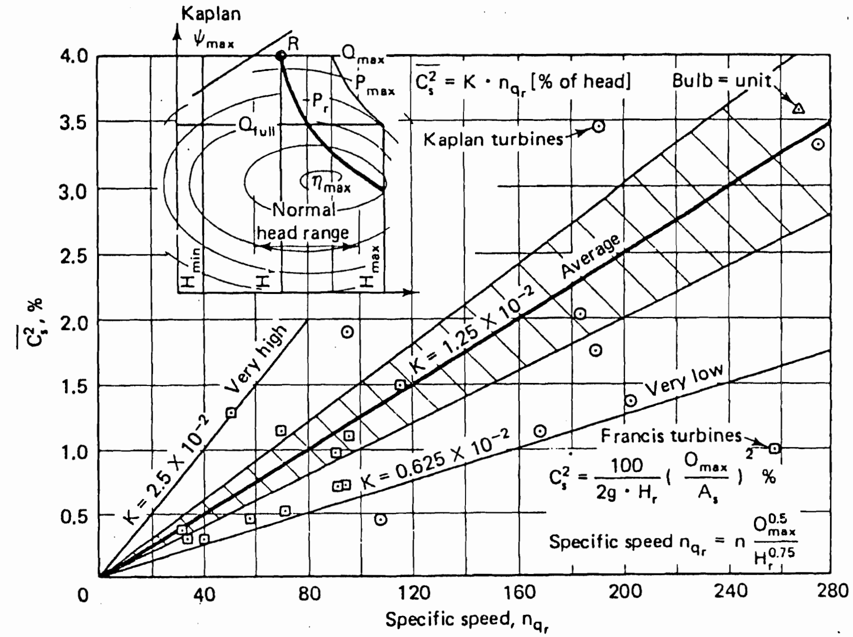


Figure 8.14 Experience curve for determining velocity head at draft tube outlet. SOURCE: Voith.

The U.S. Department of the Interior (1976) has available preliminary plant layout drawings with generalized dimensions for four different types of draft tubes: (1) double pier, (2) single pier, (3) cone and elbow, and (4) conical. Dimensions for these different types of draft tubes are all related to the turbine discharge diameter D_3. The same presentation indicates a useful limitation of the requirement for plate steel draft tube liners, specifying that the liner extend from the turbine discharge diameter D_3 to the following points:

1. To the point at which the draft tube area is twice the circular area calculated by using the runner diameter (at D_3) for all propeller-type turbines and for Francis turbines if the design head is less than 350 ft (107 m)
2. To include the full elbow with pier nose if the design head exceeds 350 ft (107 m)

The draft tube outlet size limits and governs the velocity at the outlet. An experience curve and equation for guidance in planning and design for draft tubes has been prepared by Ulith (1974). This relates the velocity head at the draft tube outlet to a special specific speed, $N_{s_{qr}}$. Figure 8.14 presents the experience curve.

The necessary definitive equations are

$$\bar{C}_s^2 = KN_{s_{qr}} \tag{8.85}$$

and

$$\bar{C}_s^2 = \frac{100}{2gH_r}\left(\frac{Q_{max}}{A_s}\right)^2 \tag{8.86}$$

where C_s = percent the draft tube outlet velocity head is of the full gate velocity head at runner discharge of the turbine

K = a constant related to the slope of the experience curve.

$$N_{s_{qr}} = \frac{n(Q_{max})^{0.5}}{H_r^{0.75}} \tag{8.87}$$

where n = turbine speed, rpm

Q_{max} = turbine discharge at full gate, m^3/sec

H_r = net head, m

A_s = outlet area of draft tube, m^2.

Selection of a higher-than-normal specific speed runner (small runner and higher-than-normal velocity at the entrance to the draft tube) results in higher-than-normal exit velocity. The high exit velocity results in the loss of effective head, but it is not included as the net head under which the turbine output is guaranteed. Nevertheless, there is an energy loss. This loss can be a high percentage in low-head installations. For example, the water velocity leaving the draft tube can be 80% of the spouting velocity for a low-head turbine. If the draft tube reduces velocity as much as 20% of the spouting velocity (as is normal for draft tubes), the exit loss is 20% of gross head. The importance of this excessive loss, which occurs throughout the life of the plant, was pointed out by Purdy (1979).

For example, consider a site where the head is 16 ft; the runner discharge velocity and draft tube entrance velocity would be 25.6 ft/sec ($0.8\sqrt{2gh}$). The draft tube exit velocity would be 6.4 ft/sec. The exit loss would be $(6.4)^2/2g$ = 0.637 ft or 4% of the original head for the life of the project.

The U.S. Army Corps of Engineers used to specify 24 ft/sec velocity for the exit velocity for the runner exit velocity. Since the draft tube exit area was four times the runner exit area, the velocity of exit at the draft tube was 6 ft/sec. The runner exit velocity for a recently constructed turbine such as at Grand Coulee is 48 ft/sec and the draft tube exit velocity is about 12 ft/sec, resulting in an exit loss of four times what used to be considered normal loss.

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PROBLEMS

8.1. A hydropower plant has been designed to have a design discharge of 1200 ft^3/sec (34 m^3/sec) and a gross head of 68 ft (20.7 m). The space requirements indicate that the length of the penstock is 190 ft (57.9 m). Recommend a suitable type and size of penstock.

8.2. A hydropower plant is to include three units that each have a design capacity of 9000 kW, a full-gate discharge of 1970 ft^3/sec (55.8 m^3/sec), and a rated net head of 60 ft (18.3 m). Indicate the approximate required space for accommodating the turbines.

8.3. Prepare a preliminary plan drawing for the required spiral case in Problem 8.2.

8.4. From a hydropower plant in your vicinity, obtain data to determine the various controlling dimensions of the spiral case and draft tube similar to those listed in Figs. 8.9 through 8.13 and check at least two dimension values with the empirical equations presented for spiral case and draft tube size determination.

8.5. Prepare a neat line drawing for generalized dimensioning of draft tubes for a bulb turbine unit or tubular turbine unit and prepare a flow diagram of how you would develop the empirical equations and diagrams similar to the work of deSiervo and deLeva.

8.6. A very small hydropower site has a rated potential capacity of 250 kW and is to have two penstocks. If the length of the penstock is 3700 ft (1128 m) and the gross head is 405 ft (123.4 m), select suitable penstocks assuming the turbine efficiency will be 85%. Make all necessary assumptions to determine a suitable penstock.

9 ELEMENTARY ELECTRICAL CONSIDERATIONS

SYNCHRONOUS GENERATORS

Converting water energy to electric energy at hydropower plants is possible through the operation and functioning of electrical generators. The phenomenon of producing an electrical current in a conductor, discovered by Michael Faraday, involves moving a copper coil through a stationary magnetic field or moving a magnet through a copper coil. In the practical generator, an induced voltage is caused by the magnetic field of a rotor sweeping by the coils of the stator. The rotor of an electrical generator in the case of hydropower developments is driven by the rotation of the turbine. Usually, the turbine and generator are directly connected on a common shaft. Most generators used in hydropower developments are alternating-current (ac) synchronous generators. These require excitation current which is usually provided by a small auxiliary generator that supplies direct current to create the magnetic field of the rotor.

Generator Components

The basic components of generators as used with hydraulic turbines are (1) supporting frame, (2) shaft that transmits the rotating motion of the turbine, (3) exciter, (4) assembly of the built-up rotor, (5) rotor poles, (6) collector rings and brushes or solid-state rectifiers, (7) stator and its component coils, (8) stator coil supports, (9) air cooler, (10) thrust bearings for vertical shaft machines, and (11) brake. These various components are indicated in the cutaway diagrammatic drawing of Fig. 9.1.

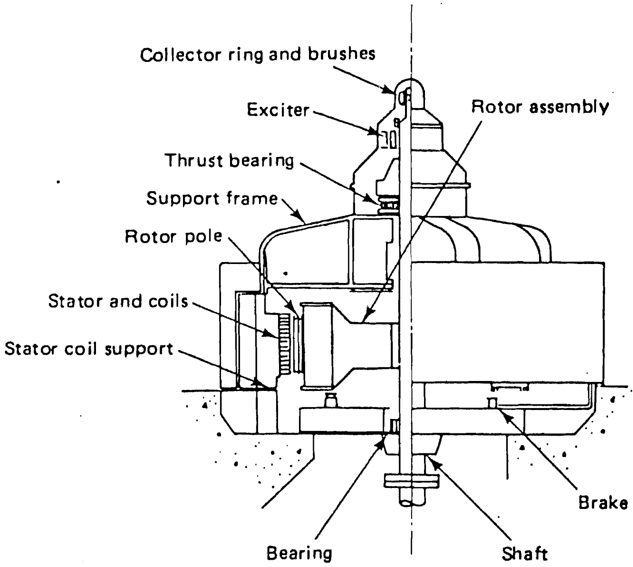


Figure 9.1 Cutaway diagrammatic drawing of synchronous electrical generator.

The stator contains the armature in conventional ac generators and consists of windings of coils pressed into slots in a symmetrical pattern. The core of the stator is composed of laminated steel sheets to reduce power losses by hysteresis and eddy currents.

The rotor contains the coils that make up the electromagnets or field winding. The windings surround the individual poles that are mounted on a structure that makes up a wheel attached to the rotary shaft. If the windings surround each pole in a symmetrical fashion and are wound individually around a pole that extends out from a cylindrical surface, they are termed *salient-pole* fields.

The number of coils, the wire size and number of turns in a coil, and the number of slots in a stator are design considerations by which size and capacity of the generator are varied. Sometimes a double layer of windings is pressed into the slots. These windings may be connected in series or parallel to achieve the desired voltage or current ratings. For actual functioning of the generator, it is then necessary to have twice the number of coils per phase per pair of poles.

The *phase* refers to the manner in which the electrical current is taken from the windings. A single-phase current is a situation or arrangement in which electric current is taken from the generator with one armature winding and delivers electric current to two wires. A two-phase generator has two armature windings generating two single phases of current 90 electrical degrees apart in phase, and the output electric current is connected to a four-wire system. A three-phase generator has three armature windings, delivering single-phase alternating currents, 120 electrical degrees apart in phase, to a three-wire system. Different kinds of connections are

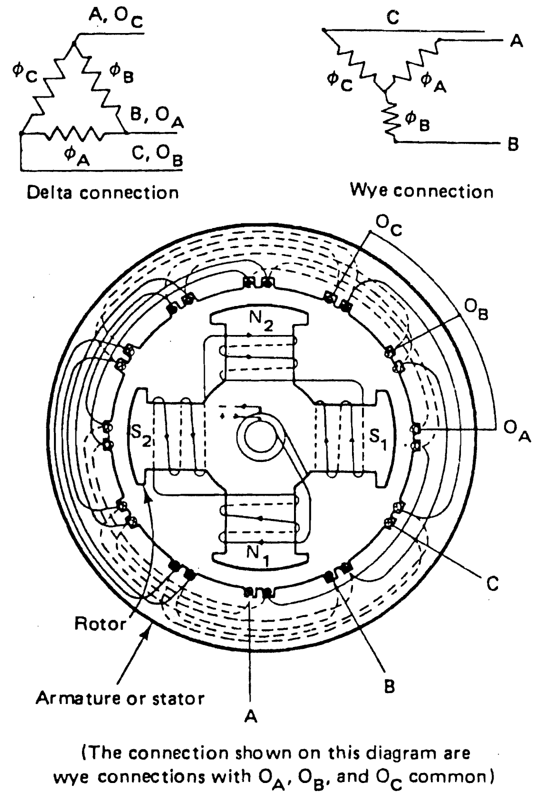


Figure 9.2 Schematic diagram of a four-pole AC generator with connection diagrams.

used to connect the windings. A delta, Δ, connection has each winding connected to the line wire and also connected to the next winding. A wye, Y, connection has one end of each winding connected to similar ends of the other windings with the other ends connected to the line wires. The common connection may be used as a grounding point, or as a neutral for a four-wire system. Figure 9.2 shows a schematic diagram of a simple four-pole generator with line diagrams showing the two types of connections.

To illustrate the functioning of the generator, consider phase A windings starting at location O_A. The symbol x denotes that the winding is going downward through the armature. The symbol · indicates that the coil winding is coming upward through a slot in the armature. Referring to Fig. 9.2, the phase A coil conductor starts at point O_A and passes downward through a slot in the armature next to rotor pole S, goes across the bottom of the armature (lines shown dotted), and comes up in a slot next to rotor pole N_2, then goes across the top of the armature (lines shown solid) to a slot next to rotor pole S_2, and then downward. This continues until the coil winding ends up back at the next slot marked O_A. The slot spacing and number of windings are design variables for changing the output of the generator. The B and C phases take a similar relative path.

The field winding magnetic circuit and rotor consists of poles that are duplicates of each other except that they are arranged alternately north and south magnetically. A full cycle of alternating current is developed for each pair of magnetic poles swept by the winding, that is, one cycle per two poles. The fixed number of poles is provided in a full circle and must be an even-integer number of poles, because a north pole must exist for each south pole. The following fundamental formula must be met:

$$f = \frac{N_P n}{120} \tag{9.1}$$

where f = frequency, Hz (= cycles/sec)
N_P = number of poles
n = speed, rpm

or, if rotative speed, ω, is in radians per second,

$$f = \frac{P\omega}{4\pi} \tag{9.2}$$

There are only a limited number of frequencies used for ac power frequencies. The usual ones are 25, 50, 60, and 400 Hz. The most common frequency used with hydropower generators in North America is 60 Hz.

Power Factor

In most operations of generators it is important to recognize the concept of power factor. In usual generator circuits there is inductive reactance and the load current lags or leads the voltage by an angle, θ, measured as a part of the 360° electrical cycle. The value of $\cos\theta$ is the power factor, PF. To illustrate this, consider the simple ac diagram in Fig. 9.3.

To understand the effect of power factor on synchronous ac generators, it is necessary to consider three conditions: (1) unity power factor load, (2) lagging power factor load, and (3) leading power factor load. Additional knowledge of the characteristics of ac generators is needed to explain fully the importance of these three conditions of power factor. The strength of the magnetic field varies directly with the current in amperes passing through the rotor winding, and generator voltage output will depend on the amperes passing through the rotor coils. Figure 9.4 illustrates how generator voltage for a 20,000-kVA generator varies with the rotor current at different power factors. If the rotor current of the generator is considered to be constant at 270 A while the load increases from no-load to full-load, the voltage at the generator terminals will decrease. Referring to Fig. 9.4, it is noted that the decrease is from 13,200 V at point A, to 9000 V at point B, for

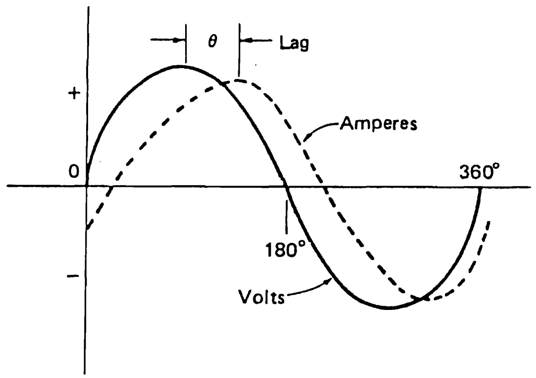


Figure 9.3 AC current diagram showing alternating sign and electrical lag.

the unity, or 100% power factor. This change in voltage is called the *regulation* of the generator. This is usually expressed as a percentage:

$$\text{Change in voltage} = \frac{(13{,}200 - 9{,}400) \times 100}{13{,}200} = 28.8\%$$

This voltage drop in the armature coils caused by resistance and reactance gives the impedance drop of the armature at the particular load being carried. The impedance drop for an armature can be expressed mathematically by the following formula:

$$IZ = \sqrt{(IR)^2 + (IX)^2} \tag{9.3}$$

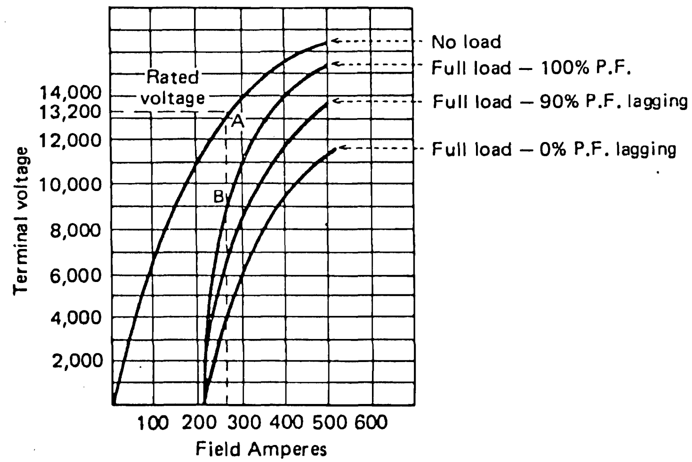
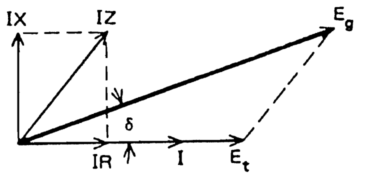


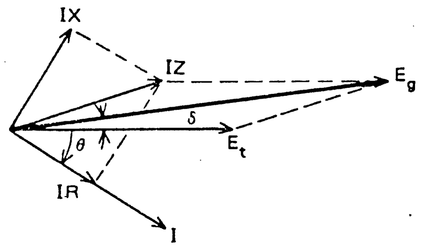
Figure 9.4 Characteristic curves for 20,000 $\overline{\text{KVA}}$ generator. SOURCE: U.S. Bureau of Reclamation.

where IZ = impedance drop, V
I = armature current, A
Z = armature impedance, Ω
R = armature effective resistance, Ω
X = armature reactance, Ω.

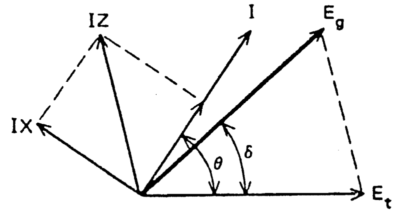
The armature effective resistance drop is in phase with the armature current. Inductive reactance drop is at right angles, or 90°, out of phase with (and leading) the armature current. Figure 9.5 shows how the three conditions of power factor load



(a) Generator with unity power factor load



(b) Generator with lagging power factor load



(c) Generator with leading power factor load

E_t = Terminal voltage
E_g = Generator voltage
I = Armature current
IR = Effective resistance drop
IX = Reactance drop
IZ = Impedance drop
θ = Power factor angle
δ = Power angle

Figure 9.5 Vector diagrams showing generator voltage variation with load conditions.

variation cause changes in the generated voltage. The magnitude of reactance and resistance drop depend on design factors such as slot depth, conductor size, spacing of conductors, and the type of coils.

Additional understanding of generator performance, generator losses, and power factor can be obtained by referring to Fig. 9.6. Here it is noted that the power of the generator output is computed by multiplying the voltage, V, times that component of the current that is in phase with the voltage, $I \cos\theta$, so that

$$P_g = \begin{cases} VI\cos\theta & \text{(single phase)} \\ \sqrt{3}\,VI\cos\theta & \text{(three phase)} \end{cases} \tag{9.4}$$

where P_g = generator output, W (1 W = 1 V × 1 A)
V = voltage, V
I = current, A
$\cos\theta$ = power factor.

The reactive power, P_{GR}, is computed by multiplying the voltage, V, by the other component of current, so that

$$P_{GR} = VI\sin\theta \quad \text{(single-phase situation)} \tag{9.5}$$

Frequently, reactive power, P_{GR}, is defined as the quantity of power in ac circuits obtained by taking the square root of the difference between the square of the volt-amperes, VI, and the square of the watts:

$$P_{GR} = \sqrt{(VI)^2 - (VI\cos\theta)^2} \tag{9.6}$$

Reactive power is expressed as reactive volt-amperes or *vars*. The inductive reactance is expressed as positive vars and capacitive reactance is expressed as negative vars.

The losses in the generator are given by the following equation:

$$P_L = \sqrt{3}\,VI\cos\theta - 3I_aR_a \tag{9.7}$$

where P_L = power loss for three-phase generator, W
V = line-to-line voltage, V
I = line current, A
$\cos\theta$ = power factor
I_a = armature current per phase if Y connected $I_a = I$, A
R_a = stator effective ac resistance per phase, Ω.

Note. It is conventional that synchronous generators be rated by electri-

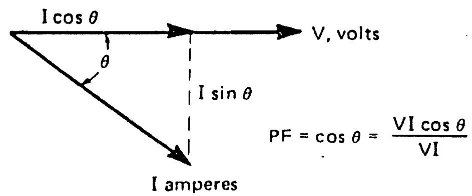


Figure 9.6 AC current vector diagram showing lag and power factor angle.

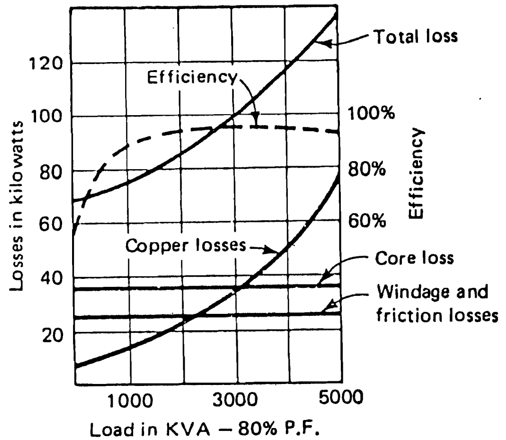


Figure 9.7 Loss and efficiency for an AC generator. SOURCE: U.S. Bureau of Reclamation.

cal output, VI, and expressed as kilovolt-amperes (kVA), not as kilowatts, unless a power factor is listed. In general, it is good to list it both ways to avoid misunderstanding.

Power losses in ac generators are characterized by the following:

1. Copper losses in the armature and rotor windings
2. Friction and windage losses
3. Core losses in the magnetic circuits of the generator

The copper losses are those losses due to resistance in the windings of the armature and rotor. The windage and friction losses are due to bearing friction and resistance of the air on rotation of the rotor and are reasonably constant. The core losses are due to hysteresis and eddy currents in the cores and metal parts of both the rotor and the stator. Figure 9.7 shows how the losses vary and gives a typical way in which generator efficiency varies for a 5000-kVA generator. Further detail on the technical aspects of generators can be found in books by Richardson (1980) and Kosov (1964).

EXCITATION

Important in electrical considerations for synchronous generators is the control of power output. This can be accomplished in two ways. One method of controlling output is provided by changing the position of the wicket gates, thus changing the water energy input to the turbine. The other method of power-output control is the direct-current (dc) field excitation provided by a separate dc supply or excitor.

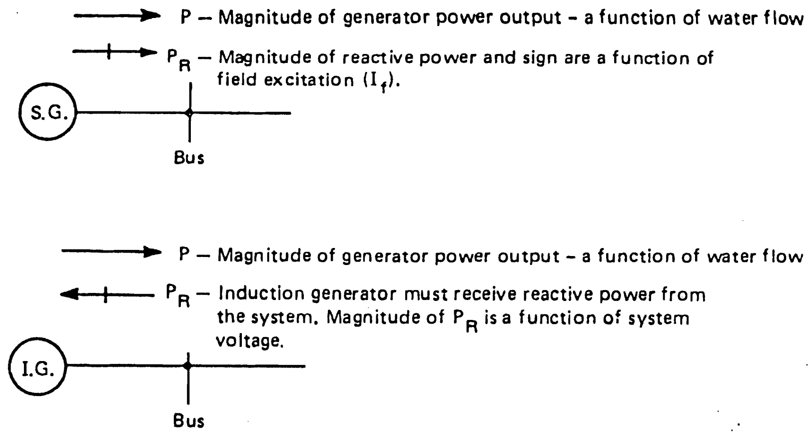


Figure 9.8 Reactive and real power flow for synchronous and induction generators.

Through this excitation there is an increase in field current and magnetic strength by the rotor field poles. The increased magnetic field strength results in an increased generator internal voltage. This will cause more reactive power to flow from the generator armature to the electric system. Figure 9.8 shows a simple single-line diagram of how real power and reactive power flow from a synchronous generator and an induction generator.

In the operation of a generator at different conditions of lagging current or leading current, the real power output and reactive power flow can be understood by considering the reactive capacity curve of Fig. 9.9. In this case it is seen that at

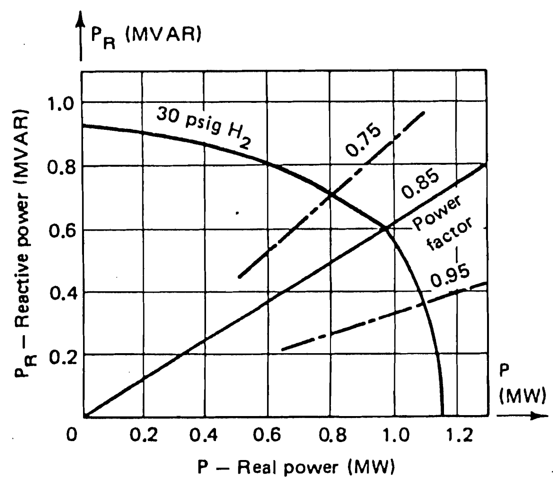


Figure 9.9 Representative reactive-capability curves for a hydropower generator.

unity power factor there would be zero reactive power and the rated real power output would be greatest. As the power factor decreases, the value of magnitude of reactive power increases and the field current has to increase. As the power factor reaches zero, the reactive power becomes a maximum. In the case of leading current situation for a synchronous generator the same relative increase and reactive power and decrease in real power output can be represented by a similar curve indicated in a lower right quadrant of a coordinate system. As the magnetic field current increases, the reactive power increases.

Synchronous ac generators require that the field cores of the alternator receive excitation from an external direct current (dc) source. The exciter is designed to provide a flow of direct current to the rotor to produce the desired voltage and var loading. The excitation systems in general use are:

1. Direct-connected shaft-driven dc generator, conventional or brushless
2. Separate prime mover or motor-driven dc generator
3. Alternating-current supply through static or mercury-arc rectifiers

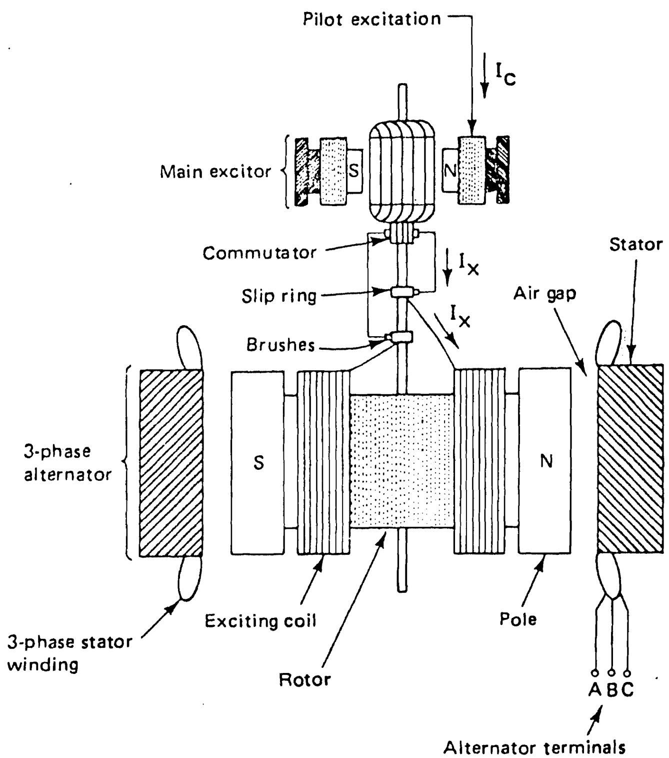
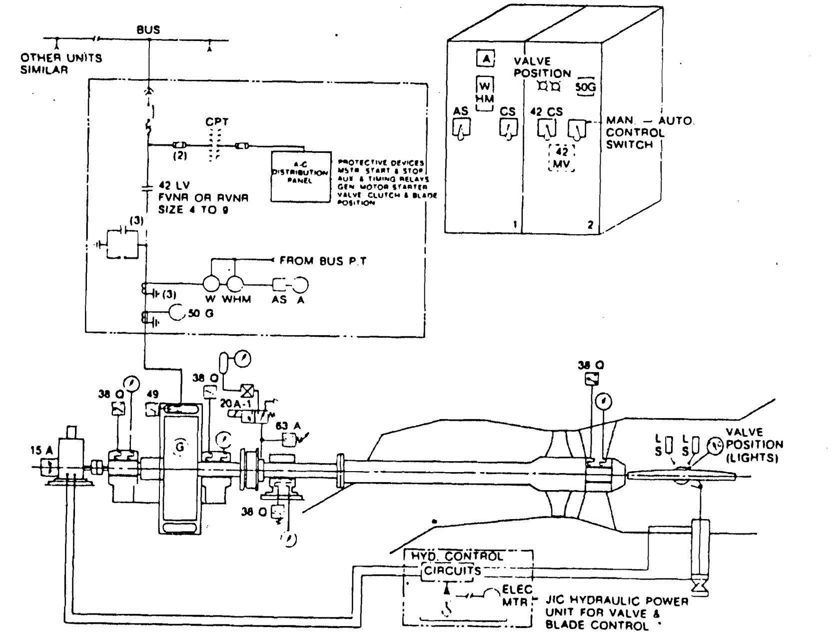


Figure 9.10 Simplified diagram of excitation for a synchronous generator.

Figure 9.10 shows a simplified diagram of how excitation is provided for a synchronous generator. Sometimes the excitation system is designed to serve an entire power plant with several turbines. At other times each turbine and generator has its own excitation equipment. Auxiliary power station needs for electricity should not be operated off the excitation equipment.

INDUCTION GENERATORS

An induction generator has no source of excitation and must draw reactive power from the electrical system or transmission grid. The result of this fact is the tendency to draw down the system voltage at the location of the generator. This is then a disadvantage of induction generators. One means of overcoming this is to provide a capacitor at the induction generator bus. The capacitors draw leading



Induction generators may be used for

1. Operation in conjunction with a large system
2. Low capacity and high generator speeds
3. Applications not requiring power factor correction

Figure 9.11 Line diagram of electrical needs for induction generator. SOURCE: Allis-Chalmers Corporation.

current from the system, or in other words the capacitors are a source of lagging current required by the induction generator. A unique requirement of the induction generator is that it requires lagging reactive power at all times even when it is not generating real power. Figure 9.8 shows in a comparative way in a simple single-line diagram how real power flows away from the generator but reactive power flows to the induction generator.

Induction generators are composed of a rotor with single squirrel-cage unshielded windings. The stator can be standard induction motor design. Connection to a bus or a system requires no excitation or synchronizing equipment. The machine can be brought up to synchronous speed and the breaker closed to put it on line. Power factors for induction generators are higher at higher speeds, so the tendency is to favor small units with higher speeds. A rule of thumb would be to use induction generators on applications below 600 hp and above 600 rpm. Induction generators would not be useful in isolated small systems, in plants requiring plant factor correction, in large-capacity plants, or in plants at the end of a system. A line diagram for an induction generator type of installation is shown in Fig. 9.11.

SPECIFYING GENERATOR EQUIPMENT

It is customary for the turbine manufacturer to work closely with the generator manufacturer to furnish and specify the necessary engineering information for generators. The various specified items of concern are listed in a checklist in Table 9.1.

TABLE 9.1 Checklist of Specification Items for an AC Generator

1. General considerations: number of units, inspection requirements, and working environment for erection, weight of major components, critical and outside dimensions
2. Rating characteristics: kilovolt-amperes, frequency phase, voltage, speed, amperes, and power factor
3. Type: synchronous or induction, horizontal, inclined or vertical shaft, direct connected, geared or belt drive, simple or complex load
4. Excitation: excitation voltage, requirements for direct-connected exciter, voltage control, and switch requirement
5. Temperature requirements: allowable temperature rise in degrees Celsius of stator, rotor, rings, and bearing at critical load conditions, and temperature detectors
6. Connections and terminals: wye or delta connections, size of armature terminals, location of armature terminals, needs for field terminals, and temperature-detection terminals
7. Ventilation and air-conditioning requirement: type of system, inlet and outlet locations
8. Mechanical requirements: number and types of bearings, weight carried by bearings, oil and water requirements, fire-protection needs, overspeed requirements, time to stop, rotation shaft size, and flywheel effect

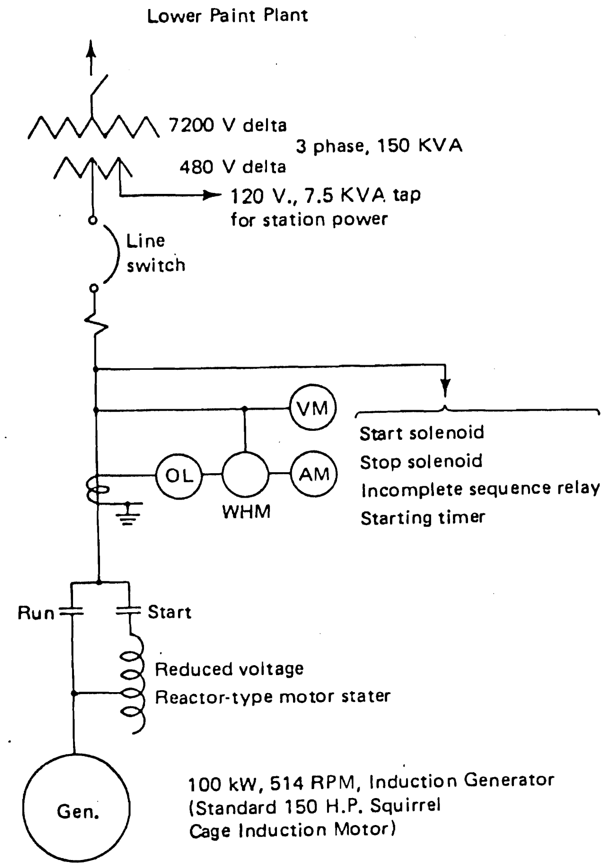


Figure 9.12 Line diagram of electrical switching and control system for a very small hydropower plant. SOURCE: Allis-Chalmers Corporation.

SWITCHING, SAFETY, AND ELECTRICAL CONTROL EQUIPMENT

The information on switch gear, equipment protection, and plant safety is very specialized and is mentioned only briefly here. The equipment can vary from simple to very complex, depending on the size of the hydroplant and whether the plant is to be automatically or manually operated. Two line diagrams showing the electrical and control needs are presented in Figs. 9.12 and 9.13. The example in Fig. 9.12 is a small hydropower plant, Lower Paint Plant, of the Wisconsin-Michigan Power Company. Figure 9.13 shows a more sophisticated line diagram for a more complex medium-size hydro installation. A good reference for detail on design and planning for electrical switching, plant protection, electrical monitoring, and transmission is Brown (1970).

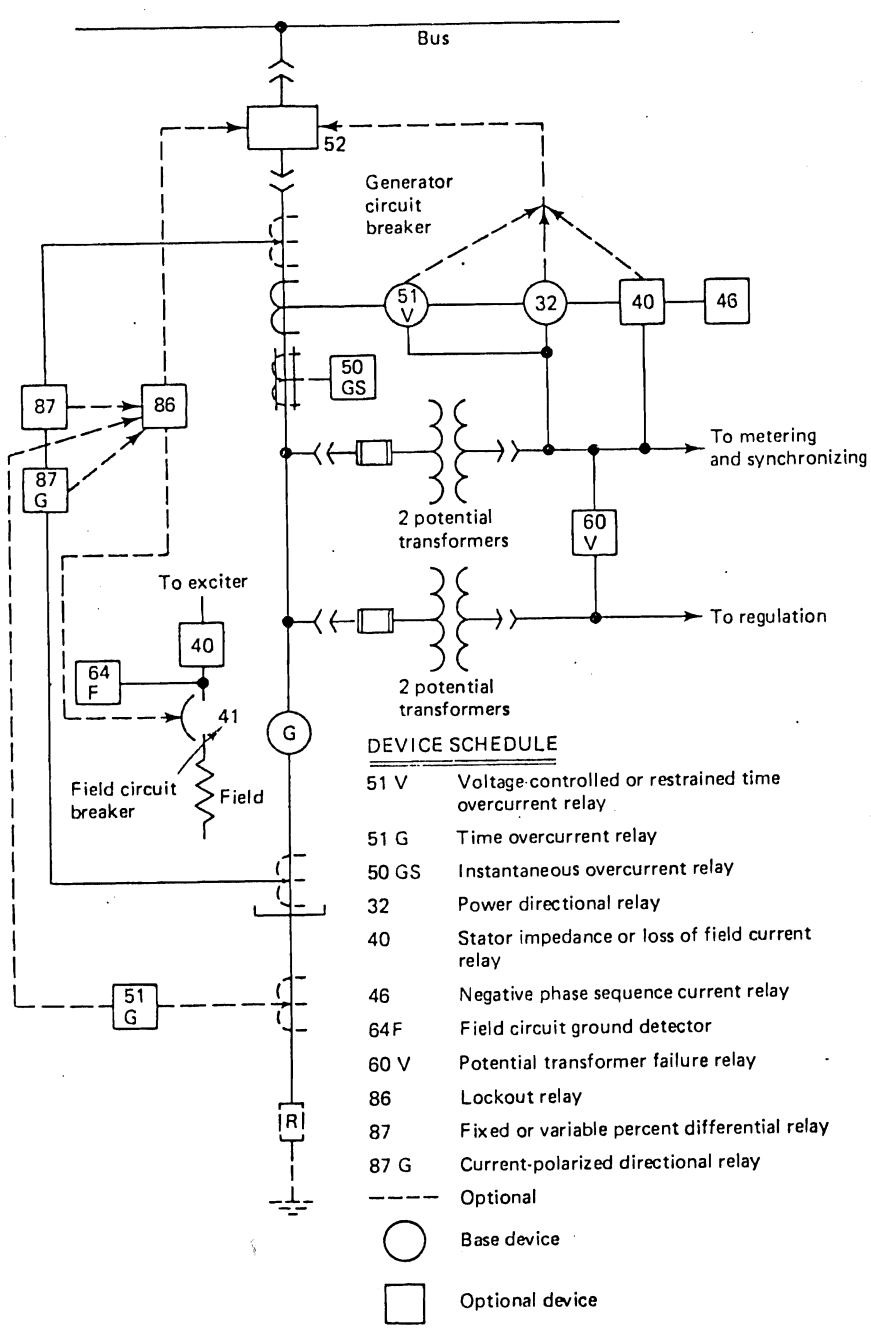


Figure 9.13 Line diagram of electrical switching and control system for a medium-sized hydropower plant.

REFERENCES

Brown, J. G., *Hydro-electric Engineering Practice*, Vol. 2, 2d ed. London: Blackie & Son, Ltd., 1970.

Kosov, I. L., *Electrical Machinery and Control.* Englewood Cliffs, N.J.: Prentice-Hall, Inc., 1964.

Richardson, D. V., *Handbook of Rotating Electric Machinery.* Reston, Va.: Reston Publishing Company, Inc., 1980.

PROBLEMS

9.1. Visit a hydroplant and obtain detail on as many electrical items as possible on the checklist in Fig. 9.9.

9.2. A three-phase 5000-kVA 13.8-kV 60-Hz 133.3-rpm Y-connected synchronous generator for a hydro plant has losses as shown in Fig. 9.7. The per phase winding reactance is 43.8 Ω. If the generator is operating at 0.8 power factor lagging, find:
(a) Electrical power output in kW
(b) Reactive power in kvar
(c) Input mechanical power in kW
(d) Line current in amperes
(e) Effective resistance (assume that the rotor copper loss is 1% machine rating, 50 kW)
(f) Number of poles
(g) Generator internal electromotive force E_g in volts on a line-to-line basis
(h) Percent voltage regulation (loaded to unloaded with constant field current)

9.3. The terminal voltage of the machine of Problem 9.2 is held constant by adjusting field current as power factor goes to unity with the same output power. Assume the same values of armature reactance (43.8 Ω) and resistance [0.2 Ω, part (e) of Problem 9.2]. Find the generator internal voltage E_g on a line-to-line basis. Find the percent voltage regulation.

PRESSURE CONTROL AND SPEED REGULATION 10

Sudden shutdowns of hydroelectric plants or changes in water flow through hydraulic turbines may cause problems ranging from rupture of penstocks due to water hammer to runner speed changes that cause the line current of the generators to vary from the desired frequency. Regulating the water flow and coping with sudden closure of gates and valves require special equipment such as governors, pressure relief valves, and surge tanks. Solving the problems of pressure control and speed regulation requires an understanding of the basic theory of water hammer.

WATER HAMMER THEORY AND ANALYSIS

Water hammer is a phenomenon of pressure change in closed pipes caused when flowing water in a pipeline is decelerated or accelerated by closing or opening a valve or changing the velocity of the water rapidly in some other manner. The phenomenon is accompanied by a series of positive and negative pressure waves which travel back and forth in the pipe system until they are damped out by friction. The various components of a hydropower installation must be capable of withstanding these changes.

When a valve in a pipe or penstock carrying water is closed, the pressure head immediately upstream of the valve is increased, and a pulse of high pressure is propagated upstream to the nearest open water surface. On the downstream side of the valve a lowered pressure moves in a downstream direction to the nearest open water surface. If the valve closure is rapid enough, a decrease in pressure may be sufficient to cause a vapor pocket to form on the downstream side of the valve.

When that vapor pocket collapses a high-pressure wave moves downstream.

The sequence of events following a sudden valve closure is presented graphically in Fig. 10.1(a). When a valve in a pipeline is suddenly closed, the element of water, dE, nearest the valve is compressed by the water flowing toward it and the pipe is stretched by the action. In the next time frame the element dE_2 is stopped and it too is compressed. The water upstream of dE_2 continues to move at the original velocity and successive elements of water are compressed. The action of compression moves upstream as a wave until it reaches the open water surface and the last element dE_n is compressed and the entire conduit of water is under the increased pressure head, $h + \Delta h$. The pressure pulse or wave moves at a velocity, a, which is essentially the velocity of sound in water. Thus the pressure wave reaches the open water surface in time, $t = L/a$, where L is the pipeline length from the valve to the open water surface. At that time the kinetic energy of the moving water has been converted to elastic energy in compressing the water and stretching the pipe. At the open water surface the last element, dE_n, expands to its original state, followed by other elements, causing a reverse or negative pressure wave. As this wave travels downstream, conditions change from zero velocity to a negative velocity of minus V_0 and from the increased water pressure head, $h + \Delta h$, back to the normal pressure head, h. When the pressure wave reaches the valve, the pressure in the pipeline has returned to normal and a time $t = 2L/a$ has elapsed. The water moving away from the valve and back upstream now causes a reduction in pressure and a negative pressure wave moves upstream to the open water surface. This peri-

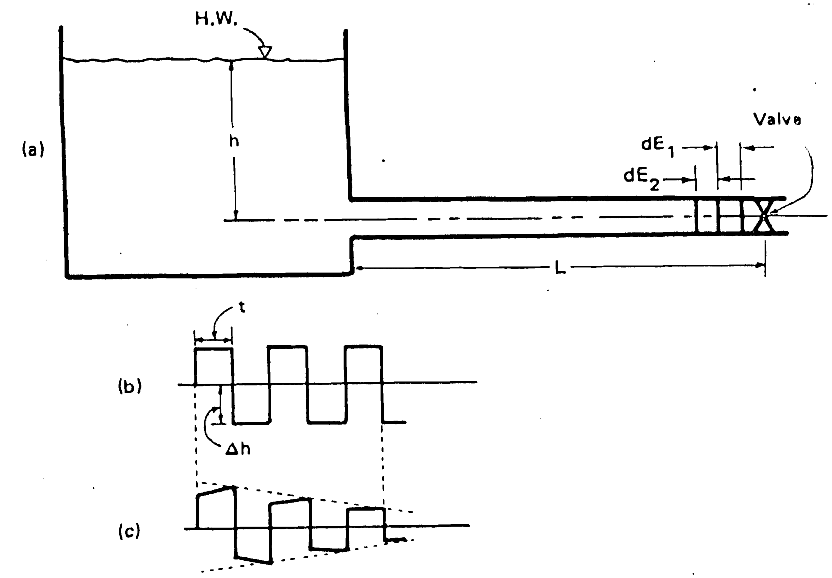


Figure 10.1 Diagram of pressure wave action due to water hammer from sudden valve closure.

odic fluctuation is shown schematically in Fig. 10.1(b) as if the water did not have friction acting. In reality, friction does act within the water and at the boundaries so that the pulses of pressure change have a decreasing amplitude as shown in Fig. 10.1(c).

Pressure Wave Velocity

Engineering analysis of water hammer necessitates use of the term, a, the velocity of the pressure wave. According to Parmakian (1955), the velocity of the pressure wave in a pipe is given by the following formula:

$$a = \sqrt{\frac{1}{(\gamma/g)[(1/K) + (dC_1/eE)]}} \tag{10.1}$$

where a = velocity of the pressure wave, ft/sec
γ = specific weight of water, lb/ft^3
g = acceleration of gravity, ft/sec^2
K = volume modulus of water = 43.2×10^6 lb/ft^2
d = diameter of pipe, in.
e = thickness of pipe, in.
E = Young's modulus of elasticity, lb/ft^2
for steel = 4.32×10^9 lb/ft^2
for cast iron = 2.30×10^9 lb/ft^2
for transite = 0.49×10^9 lb/ft^2
C_1 = factor for anchorage and support of pipe
C_1 = 0.95 for pipe anchored at upper end and without expansion joints
C_1 = 0.91 for pipe anchored against longitudinal movement
C_1 = 0.85 for pipe with expansion joints.

Parmakian (1955) gives details on how to determine a value for the pressure wave velocity, a, for composite pipes and reinforced concrete pipes. If a penstock or pipe is embedded in mass concrete, it is customary to consider the velocity of the pressure wave, a, equal to a value of less than 4660 ft/sec, the velocity of sound in water. When a steel penstock is embedded in concrete, it is embedded with a compressible membrane, called a *mastic blanket*, around it or it is embedded under full hydrostatic pressure plus the expected increase in pressure caused by water hammer. Thus the steel penstock will expand under transient pressure to prevent internally loading the surrounding concrete under tension stresses.

Two approaches to the theory of water hammer have evolved: the rigid water column theory and the elastic water column theory.

Rigid Water Column Theory

Early investigators showed that for pipes that do not stretch, water that is incompressible, the pipeline full at all times, hydraulic losses negligible, velocity uniform in the direction of the pipe axis, a uniform pressure over the transverse

cross section of the pipe, and no fluctuation in the open headwater level, the following equation applies for uniform rates of closure of the valve or gate:

$$\frac{(h_a)_{max}}{h_0} = \frac{K_1}{2} + \sqrt{K_1 + \frac{K_1^2}{4}} \tag{10.2}$$

where $(h_a)_{max}$ = maximum rise in pressure head at gate due to uniform closure, ft
h_0 = initial steady pressure head at gate, ft.

$$K_1 = \left(\frac{LV'}{gh_0T}\right)^2 \tag{10.3}$$

where L = length of pipe, ft
V' = difference between initial and final steady velocity, ft/sec
g = acceleration of gravity, ft/sec^2
T = time for gate closure or opening, sec.

The maximum drop in head at a gate or valve due to uniform gate opening is given by the equation

$$\frac{(h_a')_{max}}{h_0} = \frac{K_1}{2} - \sqrt{K_1 + \frac{K_1^2}{4}} \tag{10.4}$$

where $(h_a')_{max}$ = maximum drop in head at gate due to uniform gate opening, ft.

This approach is limited to slow-gate movement and according to Parmakian (1955) should be used only when T is greater than $L/1000$. For rapid gate movements, the elastic water column theory and analysis should be used.

Elastic Water Column Theory

The physical significance of water hammer is shown diagrammatically in Fig. 10.2. Parmakian (1955) has shown that the following equations apply for the case where the pipe is considered to deform under pressure increase due to water hammer and the water is considered to be compressible. The equation of equilibrium for the element of water is

$$\frac{\partial h}{\partial x_1} = -\frac{1}{g}\left(\frac{\partial V}{\partial t} + V\frac{\partial V}{\partial x_1}\right) \tag{10.5}$$

and the equation of continuity is

$$\frac{\partial h}{\partial t} + V\frac{\partial h}{\partial x_1} = -\frac{a^2}{g}\frac{\partial V}{\partial x_1} \tag{10.6}$$

where h = pressure head in pipe, ft or m
V = velocity of water in pipe, ft/sec or m/sec (V is considered positive moving from the headwater)
g = acceleration of gravity, ft/sec^2 or m/sec^2

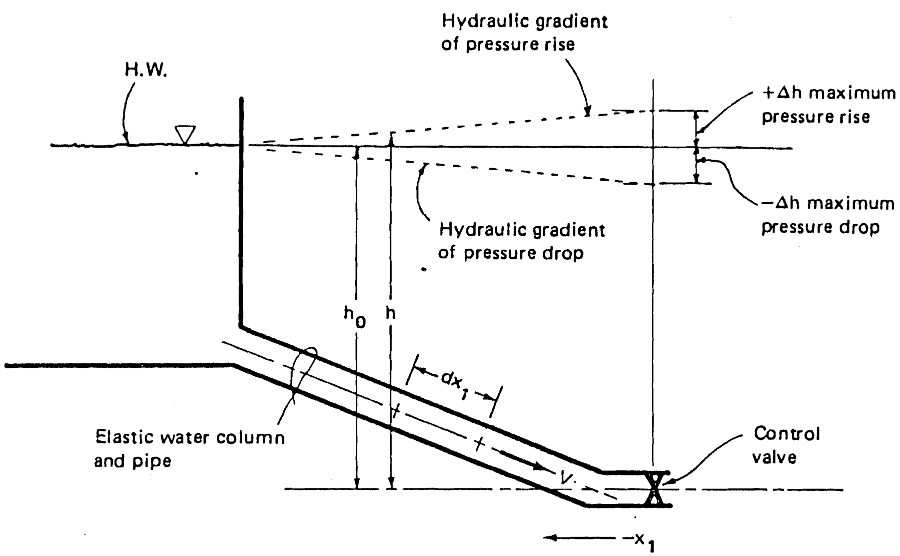


Figure 10.2 Diagram of water hammer action for elastic water column theory.

x_1 = distance along pipeline measured negative upstream from valve, ft or m
a = velocity of pressure wave, ft/sec or m/sec
t = time, sec.

If $V(\partial V/\partial x)$ is small compared with $\partial V/\partial t$ and $V(\partial h/\partial x)$ is small compared with $\partial h/\partial t$, then

$$\frac{\partial h}{\partial x_1} = -\frac{1}{g}\frac{\partial V}{\partial t} \tag{10.7}$$

$$\frac{\partial h}{\partial t} = \frac{a^2}{g}\frac{\partial V}{\partial x_1} \tag{10.8}$$

Parmakian (1955) has simplified the two differential equations into the following:

$$h - h_0 = F\left(t - \frac{x}{a}\right) + f\left(t + \frac{x}{a}\right) \tag{10.9}$$

and

$$V - V_0 = -\frac{g}{a}\left[F\left(t - \frac{x}{a}\right) - f\left(t + \frac{x}{a}\right)\right] \tag{10.10}$$

where h_0 = normal pressure head before closure, ft
V_0 = normal velocity before closure, ft/sec
x = distance from discharge end of conduit, ft, measured positive in the upstream direction.

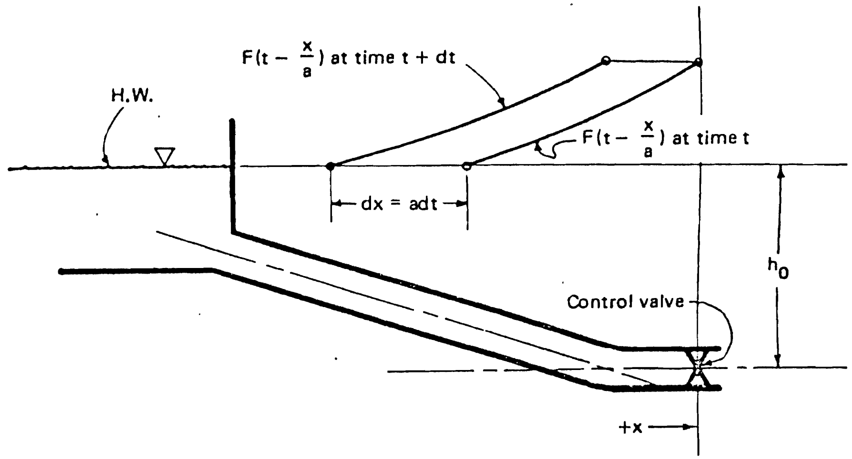


Figure 10.3 Diagram for defining mathematical terms in pressure wave action.

$F(t - x/a)$ and $f(t + x/a)$ are functional representations of resulting pressure head expressed in feet of water.

F signifies a wave traveling from the valve or gate to the reservoir, and f a wave traveling from the reservoir toward the valve. All other values are expressed as indicated in Eqs. (10.5) and (10.6) except that $-x = x_1$, where x is positive in the upstream direction from the valve, as shown diagrammatically in Fig. 10.3. As a generated F wave reaches the reservoir, $x = L$, $h - h_0 = 0$, and from Eq. (10.9),

$$F\left(t - \frac{L}{a}\right) = -f\left(t + \frac{L}{a}\right) \tag{10.11}$$

Substituting into Eq. (10.10), we have

$$V - V_0 = \frac{-2g}{a} F\left(t - \frac{L}{a}\right) \tag{10.12}$$

When an F-type wave reaches the reservoir an f-type wave of equal magnitude is reflected. When this wave reaches the valve at time $t = 2L/a$,

$$f(t) = -F\left(t - \frac{2L}{a}\right) \tag{10.13}$$

At a partially open gate, substituting F_1 for $F(t - a/x)$ and f_1 for $f(t + a/x)$ into Eq. (10.9) gives

$$h - h_0 = F_1 + f_1 \tag{10.14}$$

and into Eq. (10.10) gives

$$V - V_0 = \frac{-g}{a}(F_1 - f_1) \tag{10.15}$$

The gate or valve discharge is

$$Q = AV = (C_d A_d)\sqrt{2gh} \tag{10.16}$$

where $C_d A_d$ = effective area of valve, ft^2
A = cross-sectional area of pipe or penstock, ft^2
V = velocity in the pipe, ft/sec

and

$$V = B\sqrt{h} \tag{10.17}$$

where

$$B = \left(\frac{C_d A_g}{A}\right)\sqrt{2g} \tag{10.18}$$

Solving Eqs. (10.14), (10.15), and (10.17) for V gives

$$V = -\frac{aB^2}{2g} + \frac{B}{2}\sqrt{\left(\frac{aB}{g}\right)^2 + 4\left(h_0 + \frac{aV_0}{g} + 2f_1\right)} \tag{10.19}$$

This equation defines the velocity of water adjacent to the valve in terms of valve opening B and the f_1 wave. The F_1 wave, from Eq. (10.10), is

$$V_1 = -\frac{a}{g}(V - V_0) + f_1 \tag{10.20}$$

Instantaneous Gate Movements

The instantaneous closure of a gate in a pipeline results in a sudden pressure rise at the gate. At the instant of closure, an F-type wave is initiated and from Eqs. (10.9) and (10.10),

$$h - h_0 = F\left(t - \frac{x}{a}\right)$$

$$X - V_0 = \frac{-g}{a}F\left(t - \frac{x}{a}\right)$$

Solving these equations gives

$$h - h_0 = -\frac{a}{g}(V - V_0)$$

or

$$\Delta h = -\frac{a}{g}\Delta V \tag{10.21}$$

The head rise is therefore proportional to the change in velocity. This is the limit of the head rise for a constant rate of closure where the closing time is less than or equal to $2L/a$. Equation (10.21) has been suggested and developed by Joukovsky (1904) and Parmakian (1955).

Araki and Kuwabara (1975) have written differential equations for elastic theory water hammer and introduced a term for considering head loss. Simplification of the equations using finite difference methods permits solving the equations by inserting appropriate boundary conditions. This is normally done with special computer programs. Typical of these computer programs is WHAMO, a special program that was prepared for the U.S. Corps of Engineers (n.d.). The official designation is "Water Hammer and Mass Oscillation Simulation Program Users Manual." This is available by contacting the Office of the Chief of Engineers, Washington, D.C., or the Waterways Experiment Station, Vicksburg, Mississippi.

Streeter and Wylie (1979) present similar equations for solving water hammer transients in simple pipelines with an open reservoir upstream and a valve at a downstream point. They have used the method of characteristics to develop a basic computer program printed in FORTRAN IV language in their book. Escher Wyss published a treatise by Nemet (1974) on mathemathical models of hydraulic plants in which he presented a theory on water hammer and the structure for a digital computer program to solve for the pressure increase of water hammer. This publication also gives the format for the computer system data input.

For understanding engineering applications of water hammer theory. four cases of gate or valve closure need to be considered: (1) instantaneous valve closure, (2) valve closure time, $T = 2L/a$, (3) valve closure time, T, is less than $2L/a$, and (4) valve closure time, T, is greater than $2L/a$. In cases 1, 2, and 3 no reflected wave returns to the valve soon enough to alter the pressure head rise at the valve. In those three cases the maximum pressure head rise, Δh, can be computed by using Eq. (10.21). For analyzing the more usual problem of valve closure time, T greater than $2L/a$, a more complex analysis is required. Mathematical models have been developed with digital computer approaches by Araki and Kuwabara (1975), Streeter and Wylie (1979), Nemet (1974), and the U.S. Army Corps of Engineers (n.d.). To illustrate the stepwise mathematical approach to analyzing the pressure rise for slow closure of the valve, Parmakian (1955) presented the following example.

Example 10.1

Given: The pipe system is shown in Fig. 10.4, where L = 3000 ft, h_0 = 500 ft, d = 10 ft, Q_0 = 843 ft^3/sec, and a = 3000 ft/sec. The characteristics of a valve operating at partial opening is given in Fig. 10.5.
Required: If the valve closes in 6 sec, determine the maximum pressure rise at the valve.

Analysis and solution: The following equations apply:

$$h - h_0 = F + f \qquad \text{from Eq. (10.9)}$$

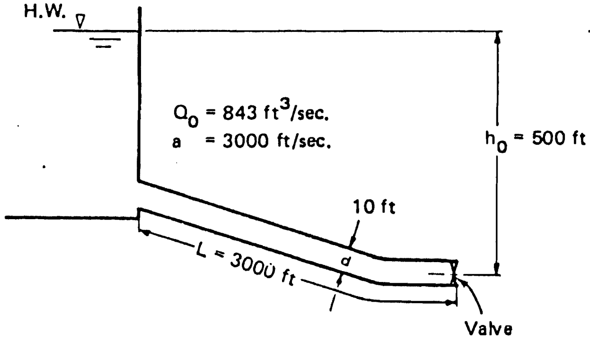


Figure 10.4 Pipe system diagram for Example 10.1.

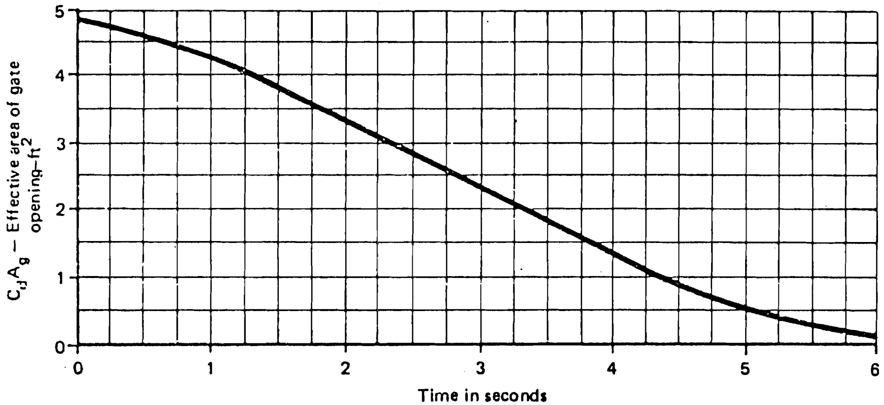


Figure 10.5 Valve operating characteristics for Example 10.1.

$$V - V_0 = -\frac{g}{a}(F - f) \qquad \text{from Eq. (10.10)}$$

and

$$Q = AV - C_d A_d \sqrt{2gh_0} \qquad \text{from the orifice equation and continuity equations}$$

so that

$$B = \frac{C_d A_d}{A}\sqrt{2g} \qquad \text{for } V = B\sqrt{h_0} \tag{10.22}$$

$$A = \frac{\pi D^2}{4} = 25\pi$$

and

$$V = \frac{-aB^2}{2g} + \frac{B}{2}\sqrt{\left(\frac{aB}{g}\right)^2 + 4\left(h_0 + \frac{aV_0}{g} + 2f\right)} \tag{10.23}$$

$$F = -\frac{a}{g}(V - V_0) + f \tag{10.24}$$

and

$$f(t) = -F\left(t - \frac{2L}{a}\right) \tag{10.25}$$

all from Eqs. (10.7) and (10.8).

Now if a time increment of $t = 1$ sec is chosen, a series of computations can be made to obtain the sum of $F + f$ or $h - h_0$ for each increment of time, t. The results of these computations are shown in Table 10.1. The maximum pressure head in the pipe would be

$$272 + 500 = 772 \text{ ft} \quad \text{ANSWER}$$

It should be noted that this calculation gives only the pressure rise at the valve. Parmakian (1955) shows that at intermediate locations along the pipe, pressure rise can be determined by recognizing that the F and f pressure wave components are displaced by an increment of time, Δt.

Special nomographs have been developed for solving the maximum pressure rise for closing or opening of valves that are considered to close and open at a uniform rate. An example presentation of this is made in an article by Kerr and

TABLE 10.1 Computation Table for Water Hammer Analysis for Example 10.1

Time, t (sec)	$C_d A_d$ (ft²)	B (ft$^{0.5}$ sec)	V (ft/sec)	F (ft)	f (ft)	$h - h_0$
0	4.70	0.480	10.738	0	0	0
1	4.23	0.432	10.166	53.3	0	53
2	3.29	0.336	8.781	182.3	0	183
3	2.35	0.240	6.673	325.5	-53.3	272
4	1.41	0.144	3.984	447.0	-182.3	265
5	0.47	0.048	1.296	554.2	-325.5	229
6	0	0	0	553.5	-447.0	107
7	0	0	0	446.3	-554.2	-108
8	0	0	0	447.0	-553.5	-107
9	0	0	0	354.2	-446.3	+108
10	0	0	0	553.5	-447.0	+107
11	0	0	0	446.3	-554.2	-108
12	0	0	0	447.0	-553.5	-107

SOURCE: Parmakian (1955).

Strowger (1933), based on nomographs published by Allievi (1925) and Quick (1927). Figures 10.6, 10.7, and 10.8 are reproductions of those nomographs that are useful in solving for maximum pressure rise in penstocks. These charts use four characteristic terms as parameters in the graphical solutions: pressure rise, P; pipeline constant, K; time constant, N; and Z^2, a pressure rise term defined as follows:

$$P = \frac{h}{h_{max}} = \frac{gh}{aV_0} \tag{10.26}$$

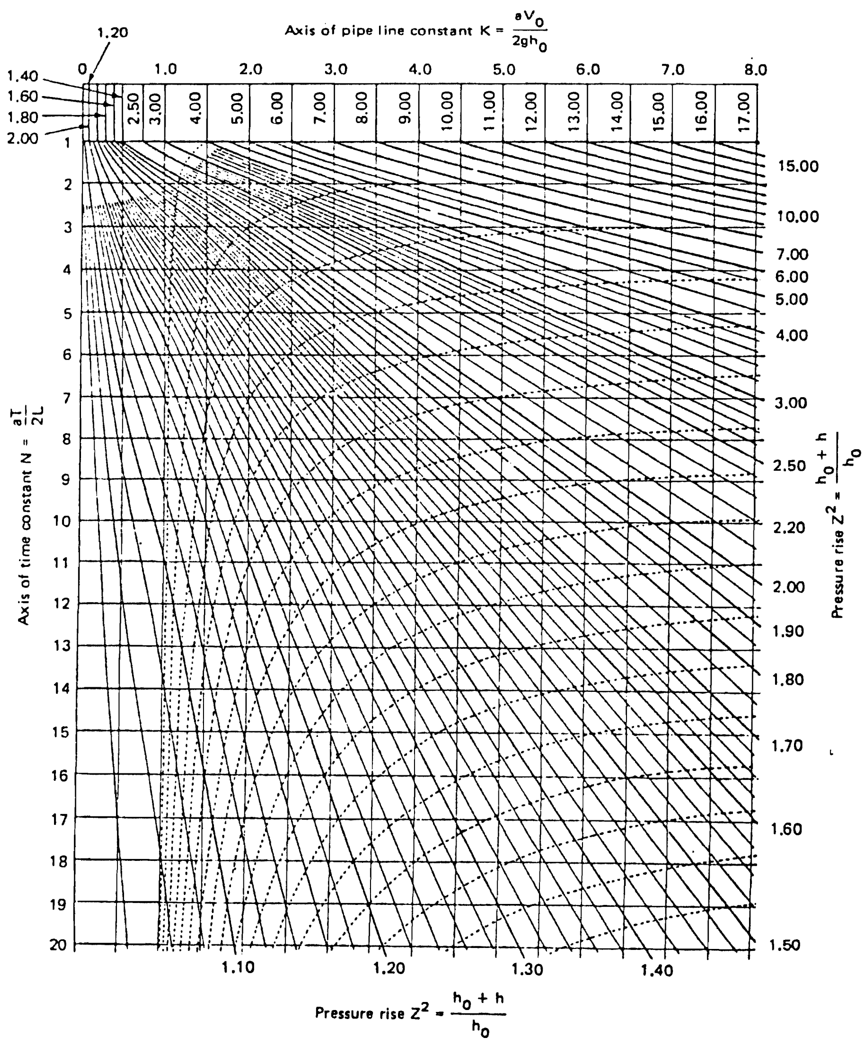


Figure 10.6 Allievi chart for determining maximum pressure rise in penstocks.

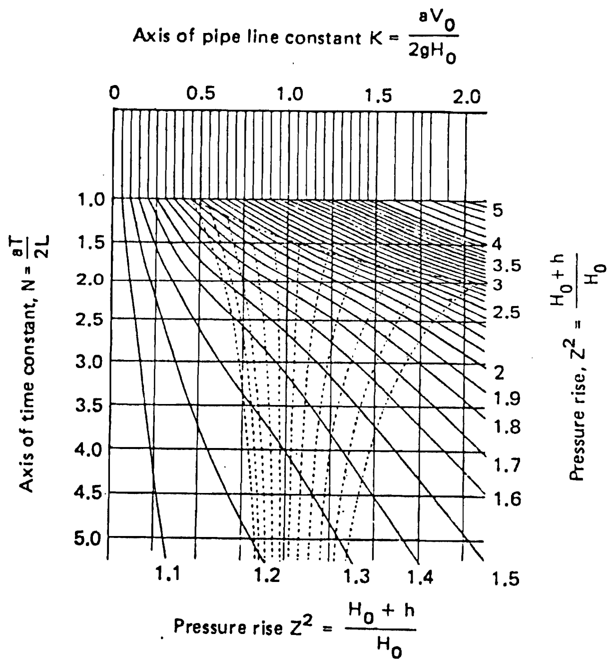


Figure 10.7 Allievi chart for determining maximum pressure rise, low values of pipeline constant.

$$K = \frac{aV_0}{2gh_0} \tag{10.27}$$

$$N = \frac{aT}{2L} \tag{10.28}$$

$$Z^2 = \frac{h_0 + h}{h_0} \tag{10.29}$$

where P = ratio of increased pressure or pressure head rise to maximum pressure head, h_{max}
h = pressure head rise, ft
V_0 = initial velocity in the penstock, ft/sec
h_0 = initial pressure head at the gate, ft
N = time constant, number of $2L/a$ intervals in the time closure
L = length of the penstock, ft
T = time of valve closure, sec
Z^2 = pressure rise term, dimensionless.

The application and use of these diagrams is presented next in an example problem.

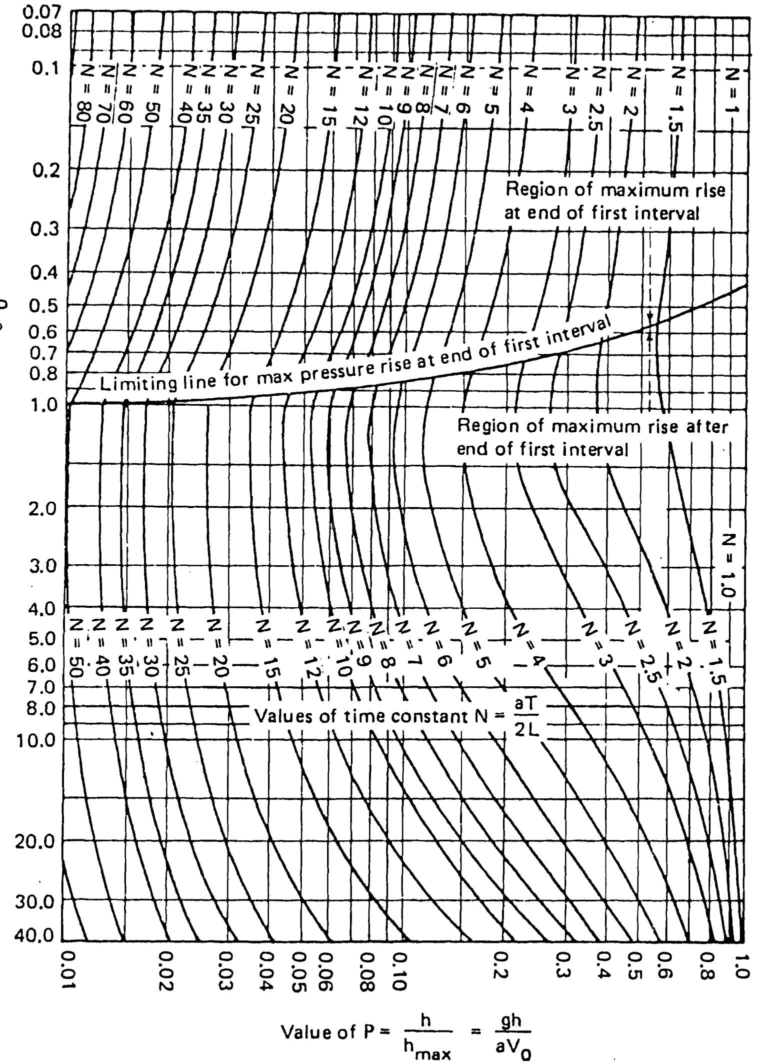


Figure 10.8 Chart from Quick (1927) for determining maximum pressure rise.

Example 10.2

Given: The pipe system is shown in Fig. 10.9 where pipe length $L = 820$ ft; full initial velocity, $V_0 = 11.75$ ft/sec; initial normal head at valve is 165 ft; time of closure, $T = 2.1$ sec; velocity of pressure wave, $a = 3220$ ft/sec; critical time, $2H/a = 0.50932$ sec; and gate-opening constant, $B = 0.9147$, from Eq. (10.22).

Required: Determine the expected total pressure caused by water hammer action.

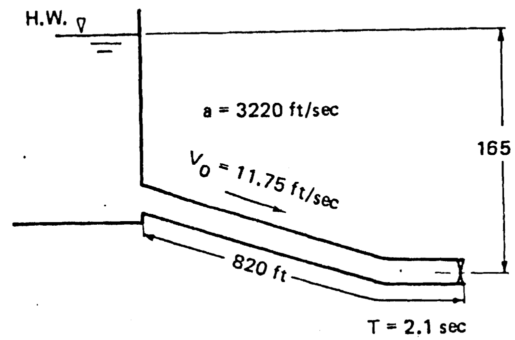


Figure 10.9 Pipe system diagram for Example 10.2.

Analysis and solution:

K = pipeline constant from Eq. (10.27)

$$K = \frac{aV_0}{2gh_0} = \frac{3220(11.75)}{2(322)(165)} = 3.56$$

N = time constant from Eq. (10.28)

$$N = \frac{aT}{2L} = \frac{3220(2.1)}{2(820)} = 4.12$$

On Fig. 10.6, the Allievi chart, with ordinate $N = 4.17$ and $K = 3.56$, read $Z^2 = 2.33$, on the right-hand scale. Then using Eq. (10.29), we have

$$Z^2 = \frac{h_0 + h}{h_0}$$

Solve for h:

$$h = 2.33h_0 - h_0 = (1.33)(165)$$

$$= 219.5 \text{ ft} \quad \text{ANSWER}$$

On Fig. 10.8, the Quick chart, with abscissa value of $K = 3.56$, intercept the $N = 4.12$ line and read on the ordinate scale $P = 0.187$. Using Eq. (10.26), we obtain

$$P = \frac{gh}{aV_0}$$

$$h = P\frac{aV_0}{g} = \frac{(0.187)(3220)(11.75)}{32.2}$$

$$= 219.7 \, ft \quad \text{ANSWER}$$

Thus using the Quick chart, the same result, within the accuracy of graphical interpolation, is obtained as by using the Allievi chart.

The symposium proceedings that included the paper by Kerr and Strowger (1933) also contained articles on more complex water hammer problems, including branching pipes and pipes of different diameter. More details for specialized cases of water hammer are also presented in the book by Parmakian (1955). McNown (1950) also gives a graphical approach to solving water hammer problems. The U.S. Department of the Interior (1967) has available a Quick-type chart for solving water hammer problems and outlines basic conditions that should govern in considering the effects of water hammer in the design of penstocks for hydropower installations.

An additional problem with water hammer occurs when the turbine is shut down and there is no flow into the draft tube while flow continues to exit the draft tube. Jordan (1975) treats the analysis of this problem and shows that damping of this negative water hammer can be accomplished by admission of air into the draft tube.

PRESSURE CONTROL SYSTEMS

Hydraulic transients and pressure changes such as water hammer can be controlled in several ways. Gate controls and governor regulation can limit the gate or valve closure time so that there is no damaging pressure head rise. Pressure regulator valves located near the turbine can be used. The relief valve can be connected to the turbine spiral case and controlled by the turbine gate mechanism to prevent excessive pressure by maintaining a nearly constant water velocity in the penstock. The relief valve may be designed to close at a rate which limits pressure rise to an acceptable value. One type of free discharge relief valve is called a Howell-Bunger valve. Howell-Bunger-type relief valves, 8.5 ft in diameter at the discharge, were used on eight turbines at Tarbela dam in Pakistan. The discharge rating of each relief valve was sufficient to prevent the pressure rise from exceeding 21%. The governor was designed to operate the relief valves in three modes: (1) pressure relief, (2) synchronous bypass, and (3) irrigation bypass. The planning and design of relief valves is beyond the scope of this book. Specific valve manufacturers should be contacted for assistance.

In the case of impulse turbines, a jet deflector linked to the turbine governor is used to prevent increased pressure rise by permitting slower movement of the needle valve. In some cases the control can be accomplished by adding additional flywheel effect to the turbine runner and generator so that the rate of acceleration to runaway speed is slowed. In a later section of this chapter further explanation is given regarding the interrelation between pressure control and speed control of hydropower units, including a discussion of runaway speed.

Surge Tanks

Another way that control can be accomplished is with surge tanks. Surge tanks are vertical standpipes that act as a forebay and shorten the distance for relief from the pressure wave of water hammer. A surge tank serves a threefold purpose

in a hydropower plant. It provides (1) flow stabilization to the turbine, (2) water hammer relief or pressure regulation, and (3) improvement of speed control. In a practical sense, a rule of thumb that might be applied to determine whether a surge tank or a relief valve may be needed is that extra caution should be exercised to evaluate pressure rise or decrease in systems where the water conduit total length equals or exceeds the head by a factor of 3. Surge tanks are usually not economical unless most of the drop in elevation in the penstock occurs near the turbine. Three slightly different types of surge tanks are used. These are shown schematically in Fig. 10.10. A *simple surge tank* is a vertical standpipe connected to the penstock with an opening large enough so that there is no appreciable loss in head as the water enters the surge tank. This is the most efficient surge tank to provide a ready water supply to the turbine when it is being accelerated, and especially when the initial loading is being applied. However, it is the most hydraulically unstable. A *restricted-orifice surge tank* is connected in such a way that there is a restricted opening between the tank and the penstock that develops appreciable head loss in

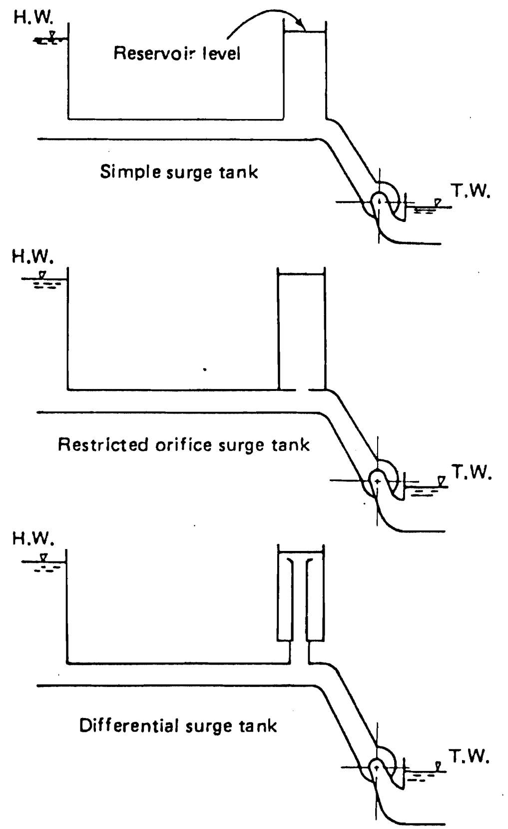


Figure 10.10 Types of surge tanks.

the water that flows into or out of the tank. Thus the orifice tank does not supply or accept excess penstock flow, but it is more hydraulically stable. The *differential surge tank* is a combination of a simple tank and a restricted-orifice tank. An internal riser of smaller diameter than the full connection to the penstock is built to extend up through the tank while an outer tank is connected by a simple pipe connection to the penstock. The riser may also have a flow restrictor or orifice inside. Thus one part of the tank responds with a minimum of head loss while the outer tank offers resistance to rapid flow into the tank.

Simple Surge Tank: Theory and Analysis

For simple surge tanks it is important to have analytical procedures for computing the upsurge. Parmakian (1955) developed three fundamental equations based on the assumptions that hydraulic losses are negligible in the simple surge tank, the velocity head in the pipe can be neglected, and that rigid water column theory of water hammer is sufficient. This is justified if the pressure rise is small and there is not appreciable stretching of the pipe nor compressing of the water. The equations are

$$S = \frac{q_0}{A_T}\sqrt{\frac{A_T L}{A_p g}}\sin\sqrt{\frac{A_p g}{A_T L}}\,t \tag{10.30}$$

$$S_{\max} = \frac{q_0}{A_T}\sqrt{\frac{A_T L}{A_p g}} \tag{10.31}$$

$$T = \frac{\pi}{2}\sqrt{\frac{A_T L}{A_p g}} \tag{10.32}$$

where S = upsurge in surge tank above static level, ft
q_0 = water discharge in the pipe before closure, ft^3/sec
A_T = cross-sectional area of surge tank, ft^2
L = length of pipe from surge tank to open water surface, ft
A_p = cross-sectional area of pipe, ft^2
g = acceleration of gravity, ft/sec^2
t = time from instantaneous closure of valve, sec
T = time required to reach maximum upsurge in surge tank, sec

These last three equations can be useful for preliminary analysis and in cases where the magnitude of hydraulic losses is small in the portion of the penstock extending upstream from the surge tank. For other cases where head loss is considered significant, the theory presented by Nechleba (1957) is necessary. Definition diagrams for defining the theory are presented in Fig. 10.11. Nechleba presents the following basic equations:

$$\frac{L\,dV}{g'\,dt} = Y - kV^2 \tag{10.33}$$

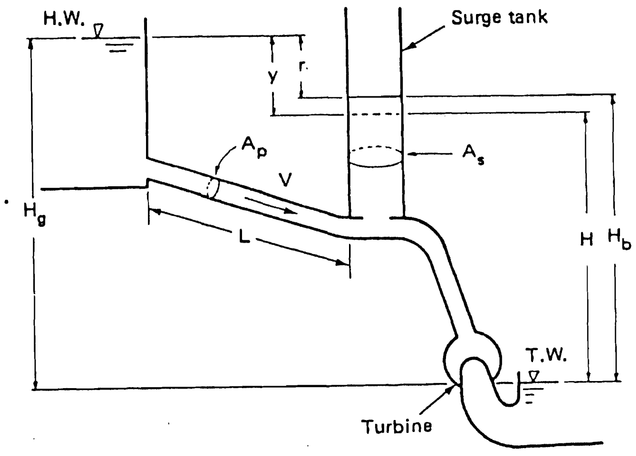


Figure 10.11 Definition diagram for surge tank theory.

$$A_s\frac{dY}{dt} = Q_r - A_p V \tag{10.34}$$

where L = length of pipe from surge tank to the open water reservoir, m
$\frac{dV}{dt}$ = change of water velocity in the pipe, m/sec/sec
g' = acceleration of gravity, m/sec^2
Y = elevation difference between headwater level and surge tank level, m
k = a coefficient related to the losses in the pipe and surge tank
V = velocity of water in pipe, m/sec
A_s = cross-sectional area of surge tank, m^2
A_p = cross-sectional area of pipe, m^2
Q_r = water discharge in pipe and system to turbine in steady flow condition, m^3/sec.

From Eq. (3.9), $Q_r = P_r\,1000/(H_g - Y)\rho' g'\eta$, so that Eq. (10.34) may be written as

$$A_s\frac{dY}{dt} = \frac{1000P_r}{(H_g - Y)\rho' g'\eta} - A_p V \tag{10.35}$$

Nechleba (1957) states that if fluctuations in the surge tank are small in comparison to the gross head, the following two limits apply:

$$KV_s^2 = r < \frac{H_b}{2}\left(\frac{\eta_s}{\eta_s - iH_b}\right) \tag{10.36}$$

$$\left(\frac{2kA_s g'}{A_p L} - \frac{i}{\eta_s}\right)H_b > 1 \tag{10.37}$$

where V_s = medium velocity of flow in the pipe upstream of the surge tank, m/sec
H_b = pressure head to top surface of surge tank, m
η_s = medium efficiency of the hydropower system at flow rate considered
$i = d\eta/dH$ = change in system efficiency created by fluctuating head in surge tank
r = head loss in penstock upstream of surge tank, m.

Equation (10.36) dictates that losses in the pipe upstream of the surge tank, labeled r in Fig. 10.11, should be smaller than $H_b/2$. Equation (10.37) provides a means of determining the required cross-sectional area of the surge tank. The change in efficiency, i, is normally small with respect to fluctuations in the head in the surge tank so that the following limits apply:

$$kV_s^2 = r < \frac{H_b}{2} = \frac{H_g}{3} \tag{10.38}$$

$$\frac{2kA_s g' H_b}{LA_p} > 1 \tag{10.39}$$

Using the limit of Eq. (10.39) as equal to 1 gives what has been referred to by Nechleba as Thomas's area:

$$A_{s\,\mathrm{Th}} = \frac{LA_p}{2kg'H_b} \tag{10.40}$$

This can be used for determining the cross-sectional area of the surge tank. In planning the physical conditions there may be a limit to height of the surge tank. A trial-and-error solution can be used with economic analysis to determine an optimum surge tank size by varying height and the cross-sectional area of the surge tank and comparing the different costs.

Nechleba (1957) gives corresponding equations to Eqs. (10.31) and (10.32) developed by Parmakian that can be used to find maximum surge in the surge tank as follows:

$$S_{\max} = \frac{\Delta Q}{\sqrt{A_p A_s}} \sqrt{\frac{L}{g'}} \sqrt{\frac{H_b}{H_b - 2r}} \tag{10.41}$$

$$T = 2\pi \sqrt{\frac{A_s H_b L}{A_p g' (H_b - 2r)}} \tag{10.42}$$

where $S_{\max}$ = maximum pressure head rise in surge tank, m
ΔQ = sudden change in discharge through turbine (for total maximum $S_{\max\,T}$, the ΔQ would be Q_r)
T = time of oscillation rise in surge tank, sec.

Parmakian (1955) has also given equations for solving the maximum surge in a simple surge tank where head loss in the penstock is significant and there are head losses within the surge tank. Figure 10.12 is presented to help in defining the

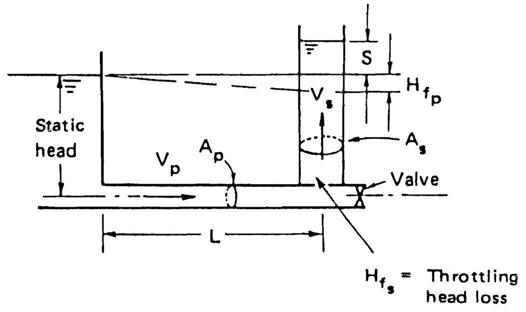


Figure 10.12 Definition sketch for terminology for surge tank.

necessary terms. The equations are for the case of upsurge caused by closure of turbine gates (situation 1) and for the case of downsurge caused by sudden opening of turbine gates (situation 2). In situation 1, $V_p > 0$ and $V_s > 0$:

$$H_a = -S - C_p V_p^2 - C_s V_s^2 \tag{10.43}$$

In situation 2, $V_p > 0$ and $V_s < 0$, being a downsurge:

$$H_a = -S - C_p V_p^2 + C_s V_s^2 \tag{10.43a}$$

Other applicable equations similar to those of Nechleba (1957) were proposed by Parmakian (1955) in differential form:

$$\frac{dV_p}{dt} = \frac{gH_a}{L} \tag{10.44}$$

$$V_o = \frac{A_p V_p - Q}{A_o} \tag{10.45}$$

$$\frac{ds}{dt} = \frac{A_p V_p - Q}{A_T} \tag{10.46}$$

where the following are terms defined for Eqs. (10.43) through (10.46):

H_a = upsurge or pressure head rise in surge tank, ft
S = pressure head change in surge tank, ft
$C_p V_p^2 = H_{f_p}$ = pipe friction head loss plus velocity head plus other head losses between the surge tank and the reservoir
$C_s V_s^2 = H_{f_s}$ = head losses in the surge tank, ft (this could include a restricted orifice-type connection)
V_p = velocity of water in penstock upstream of surge tank, ft/sec
V_o = velocity of water in surge tank orifice, ft/sec
g = acceleration of gravity, ft/sec^2
L = length of penstock upstream of surge tank, ft
A_p = cross-sectional area of the penstock, ft^2

A_o = cross-sectional area of surge tank orifice, ft^2
A_T = cross-sectional area of surge tank, ft^2
Q = discharge initially flowing in penstock, ft^3/sec

Solution charts for solving these equations were developed by Jacobson (1952). These charts are reproduced as Fig. 10.13 for solving maximum upsurge in surge tanks due to gate closure at the turbine and Fig. 10.14 for solving maximum downsurge in surge tanks. Use of these charts requires determination of values for the head loss terms H_{f_p} and H_{f_s}. Values for these terms can be determined from standard head loss equations for pipes and orifice loss equations.

Restricted-Orifice Surge Tank Analysis

The introduction of a restricted orifice in the connection between the penstock and the surge tank complicates the problem of water hammer analysis and design for pressure control. Mosonyi and Seth (1975) have developed equations to solve problems that arise when the restricted-orifice surge tank operates and water hammer causes significant pressure head rises in the penstock upstream of the surge tank. Mosonyi and Seth developed the theory and tested it in a laboratory in Germany for a particular cross-sectional area of surge tank and particular cross-

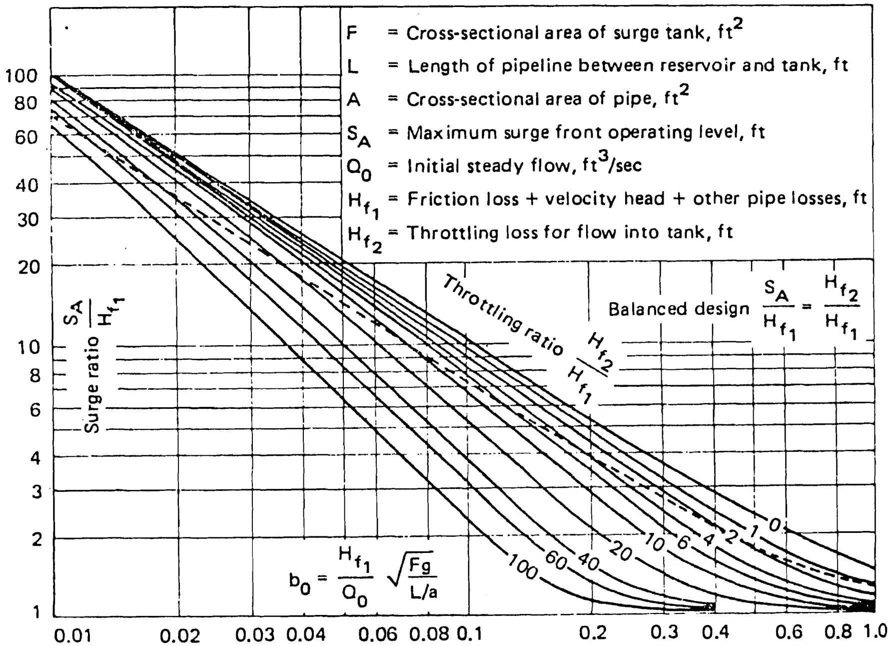


Figure 10.13 Chart for solving maximum upsurge in surge tanks. SOURCE: U.S. Bureau of Reclamation.

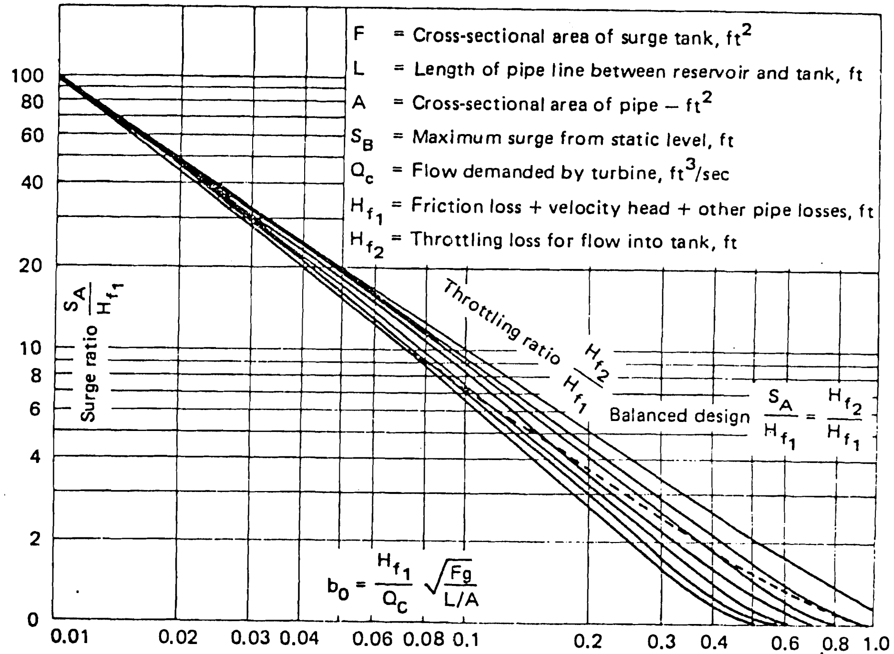


Figure 10.14 Chart for solving maximum downsurge in surge tanks. SOURCE: U.S. Bureau of Reclamation.

sectional area of penstock. Treatment of the complex analysis of restricted-orifice surge tanks is beyond the scope of this book.

SPEED TERMINOLOGY

In Chapter 4 the speed of the turbine runner and specific speed, unit speed, and synchronous speed were discussed and expressed mathematically. In engineering considerations other speed terms are used and knowledge of these speed terms and speed control is necessary in planning for safe operation of a hydropower system. A speed term that is important in the safety of hydroplants is the *runaway speed.* Runaway speed is the speed attained by turbine and generator, if directly connected, after a load rejection, if for some reason the shutdown mechanism fails to shut down the unit or if the rate of shutdown is not fast enough.

Runaway speed may reach 150 to 350% of normal operating speed. The magnitude of runaway speed is related to turbine design, operation, and setting of the turbine and will vary with the windage and friction that the runner and generator rotor offer as a revolving mass. For some adjustable blade propeller turbines the maximum runaway speed may be as high as 350% of normal operating speed. In the case of Kaplan turbines, there are two runaway speeds, on cam and off cam. The latter is the higher of the two. That is, since the flattest blade angle causes the highest runaway speed, if the governor malfunctions while the gates are fully open and puts the blades flat, a higher runaway speed will occur. For this reason most Kaplan blades are designed to be balanced so that the blades will move into a steep position if governor blade control pressure is lost.

Accurate numbers for defining runaway speed values must be based on model tests conducted by the turbine manufacturers. The U.S. Department of the Interior (1976) gives empirical equations for estimating runaway speed based on several model tests of units installed by the Bureau of Reclamation. The equations are:

$$n_r = 0.85nn_s^{0.2} \tag{10.47}$$

$$n_s = 0.63nN_s^{0.2} \quad \text{in metric units} \tag{10.48}$$

$$n_{max} = n_r\left(\frac{h_{max}}{h_d}\right)^{0.5} \tag{10.49}$$

where n_r = runaway speed at best efficiency head and full gate, rpm
n = normal rotational speed, rpm
n_{max} = runaway speed at maximum head
n_s = specific speed based on full gate output at best efficiency head
h_d = design head, ft
h_{max} = maximum head, ft.

Another term is *overspeed.* It is the speed attained under transient conditions by a turbine after a load rejection, while the governing and gate closure mechanisms are going into action. The rapidity with which the shutoff operates controls how much the overspeed will be. In very slow moving gate mechanisms overspeed can approach runaway speed.

SPEED CONTROL AND GOVERNORS

Much equipment connected to a hydroelectric system is sensitive to frequency variation. Therefore, speed control of the system is a necessity. Regulating the quantity of water admitted to the turbine runner is the usual means of regulating and maintaining a constant speed to drive the generator and to regulate the power output. This is done by operating wicket gates or valves. Such action requires a mechanism to control the wicket gates, which is the *governor* or *governor system.* At decreasing load, the speed tends to rise, and the governor has to close the wicket gates to such an extent that the torque created by the turbine equals the torque offered by the electrical load on the generator and the speed returns to the desired synchronous speed. As the wicket gates open, the speed adjustment is lowered and the inherent tendency of the turbine unit is to pick up additional load in response to a decrease in system speed.

The function of the governor is to detect any error in speed between actual and desired values and to effect a change in the turbine output. This is done so that

the system load is in equilibrium with the generating unit output at the desired speed.

The governor system of the turbine acts as an opening, closing, and gate-setting mechanism for starting, stopping, and synchronizing the turbine which allows for matching output to the system load to maintain the system frequency and creates the necessary adjustment on Kaplan turbine blade angles for optimum operation at synchronous speed.

Governor Types and Characteristics

Governor systems can be classified as either mechanical-hydraulic governors or electrohydraulic governors. The three elements of operation are (1) the speed-responsive system for detecting changes in speed, (2) the power component for operating the wicket gates, and (3) the stabilizing or compensating element that prevents runaway speed in the turbine and holds the *servomotor* in a fixed position when the turbine output and the generator load are equalized. The servomotor is the oil pressure system and piston arrangement used to operate the wicket gates.

A *mechanical-hydraulic governor* is a unit in which a mechanical centrifugal pendulum acts as a sensor. The flyballs of the pendulum move outward with increasing speed and inward with decreasing speed. The operation of the flyball mechanism is shown schematically in Fig. 10.15. The movement of the flyballs is transferred by means of a rod and links to the pilot valve of the governor, which in turn operates the servomechanism for changing the position of the wicket gates. In a Kaplan turbine, the propeller blades are controlled by a separate servomotor with a series of cams or a microprocessor to maintain a position relative to the head and gate position. The flyball mechanism does not have power output necessary to move the wicket gates on a hydraulic turbine. The power for moving the wicket gates is normally supplied by a hydraulic system controlled by the flyball action.

The power component of the entire governor system and gate operating system consists of (1) the servomotor, a fluid-pressure-operated piston or pistons to move the wicket gates; (2) the oil pressure supply, which furnishes the power for the action of the servomotor; and (3) a control valve, which regulates oil pressure and flow of oil in the servomotor.

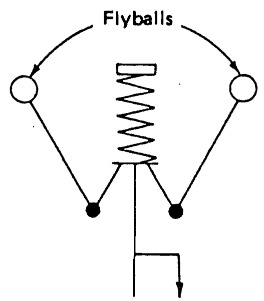


Figure 10.15 Schematic drawing of speed sensing flyball mechanism.

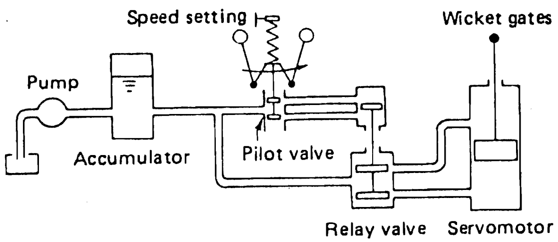


Figure 10.16 Schematic drawing of isochronous governor.

The sequence of governor operation consists of the action of the flyballs, which respond to the speed change and transmit motion through the system of floating levers to the pilot valve. The pilot valve equipped with relay valves causes oil pressure to be transmitted to one side or the other of the servomotor. The action of the piston of the servomotor in turn opens or closes the turbine wicket gates and regulates the flow of water to change the speed.

The speed of a turbine will deviate from normal synchronous speed for a certain percentage of load change. The amount of deviation of speed will depend on (1) the time required to alter the flow of the hydraulic oil in the governor system to correspond with action necessitated by the change in load, (2) the amount of flywheel effect of the entire rotating mass of the turbine and generator, and (3) the time required for the water flow to respond to action caused by the change in the turbine operating point.

The simple governor and control system is shown in Fig. 10.16. The system shown in Fig. 10.16 is known as an *isochronous governor.* The isochronous governor is inherently unstable, and although some stabilization results from the characteristics of the turbine and connected load, these are generally inadequate and an additional means of stabilization is required to overcome the unstabilizing effect of the inertia of the water flow in the penstock. The necessary stability is provided by feedback from the servomotor, which, by means of the dashpot, temporarily restores the control valve toward the null position, and thus dampens the servomotor movements. The dashpot functions with a spring-loaded valve that is mechanically linked to the servomotor and to the controls from the pilot valve and the flyball mechanism. A diagram of the operation of a dashpot governor is shown in Fig. 10.17.

Speed Sensing

To control the turbine speed, the governor must sense the system speed and compare it to a standard. In the case of a mechanical governor, the flyball mechanism is driven by a permanent magnet generator (PMG) attached to the generator shaft, or by a potential transformer (PT) so that any change in system speed results in a change in the flywheel mechanism's position. In an electronic governor, the line frequency may be sensed directly from a potential transformer or the output

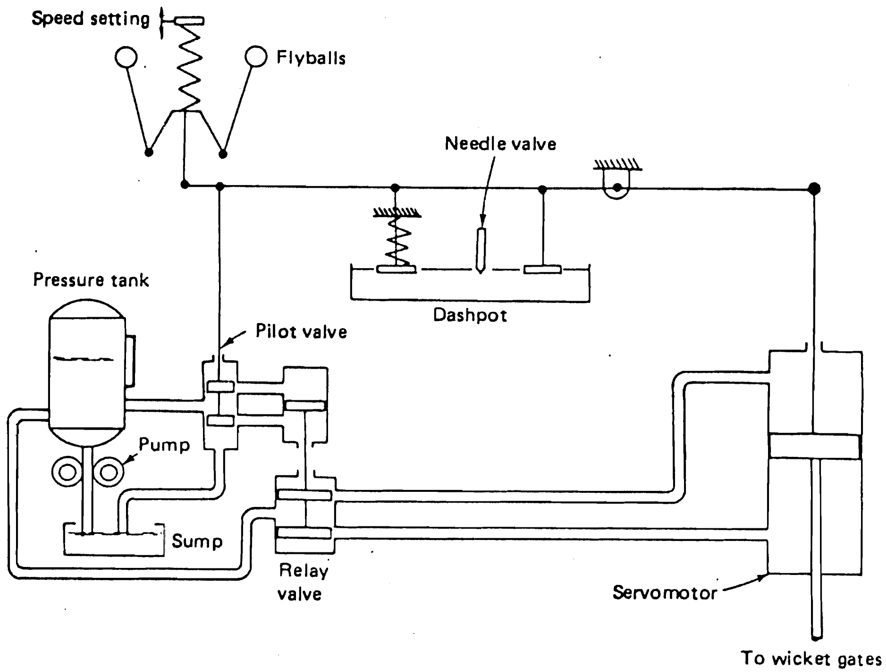


Figure 10.17 Schematic drawing of dashpot governor.

frequency of a speed signal generator (SSG) attached to the generator. The output frequency is compared to a standard and the difference is processed electronically to drive a transducer-operator control valve.

Power Unit

The power unit to operate the speed control is supplied by the oil pressure system, which consists of a sump, oil pumps, accumulator tank or tanks, and an air compressor. Oil is pumped from the sump into an air-over-oil accumulator tank(s) as needed to maintain the required pressure (see Fig. 10.17). The accumulator tanks store the oil until needed in the control system. Pressure in the tank is maintained by the air compressor which admits air to maintain the oil at the required level.

There are many auxiliary controls which are normally a part of the modern governor. Two of these which are necessary for proper operation are the speed level and the gate limit controls.

Speed level. The speed level allows for manually adjusting the governor control speed from the normal setting. This adjustment is usually limited to +10 to −15% and serves to load the unit. If the speed control is set below the line frequency, the unit will drop load. If set above the line frequency, the unit will pick up load.

Gate limit. The gate limit control allows for manually setting the maximum gate opening that the governor will allow. With the gate limit set at any intermediate value, and the speed adjustment at a positive setting, the gates will remain at the set point. The turbine is then operating at what is known as *blocked load.*

Speed Droop

If two or more generating units are operating to govern a system, it is impossible for them to maintain exactly the same speed. The governor with the higher speed will try to bring the system up to its speed, and take on load until it either achieves this or the turbine reaches full-gate position. If it is able to raise the system frequency, the other governor will sense too high a speed and begin to drop load. The governors will therefore oppose each other. To avoid this situation, governors are built with a feature called *speed droop.* Speed droop reduces the governor sensing speed as the gate opening or load increases. It is defined as the difference in speed in percent permitted when the units are operating between zero gate opening and 100% gate opening.

The action of speed adjustment and speed droop may best be visualized by referring to Fig. 10.18. If the speed adjustment is set at 5% and the speed droop at 10%, the governor sensing speed is represented by line AC. Points to the left of

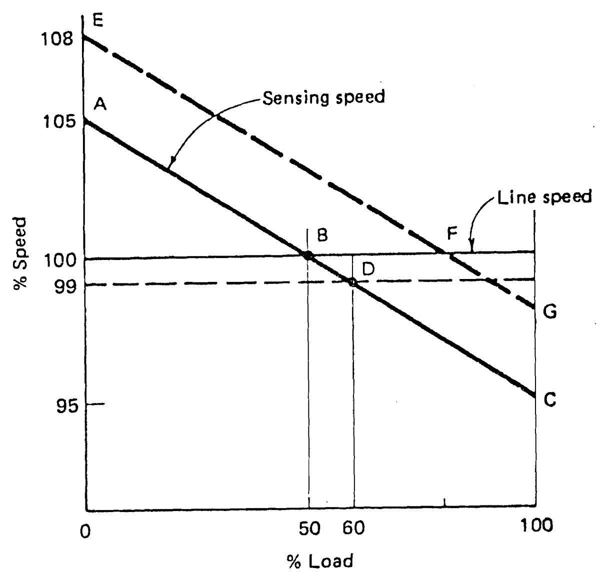


Figure 10.18 Speed adjustment and speed droop schematic diagram.

point B are seen as an underspeed condition requiring an increase in turbine output, and those to the right are seen as an overspeed requiring a decrease in output. At point B the turbine is operating at 50% load. If the system deviates to 99% of normal speed, the turbine output will increase 10 percentage points to 60% load. Any change in the speed adjustment shifts the governor sensing speed without changing its slope (see Fig. 10.18). Raising the speed adjustment three percentage points from 105 to 108% shifts the turbine operating point to F, an output increase of 30%.

With several units supplying the load, all set with a significant speed droop, the system frequency would vary greatly with load changes. It is normal therefore to set one unit with zero or very small speed droop. This unit would then effectively govern the system. If the load change exceeds the capacity of the unit with a zero speed droop setting, other units would pick up or drop load in response to their settings. In a large system, units may be operated with blocked load, and with speed droop, and one unit with zero speed droop for governing the system. The governing unit (with zero droop) should be of sufficient capacity to handle the expected load variations.

To explain the speed droop concepts, the following example is presented.

Example 10.3

Given: A power plant containing four 25-MW turbine generators is supplying a load of 80MW. Three units are operating at three-fourths of capacity and are set with an effective speed droop of 10% and a speed adjustment of plus 7.5%. Unit 4 is operating at zero droop and zero speed adjustment and is carrying the remaining load. The system load increases to 85 MW. Speed droop characteristics for the three units is shown in Fig. 10.19.

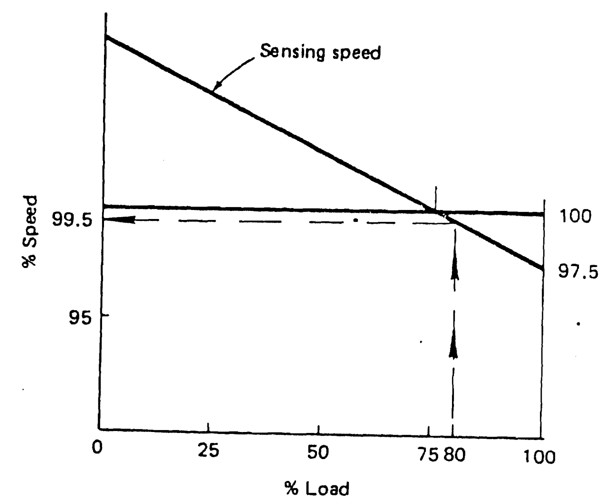


Figure 10.19 Speed adjustment and speed droop for Example 10.3.

Required: Determine the change in the system frequency.

Analysis and solution: The initial load on units 1 to 3 = 0.75 X 25 = 18.75 MW. The initial load on unit 4 = 80 - 3(18.75) = 23.75 MW. An increase of 5 MW cannot be taken by unit 4, 23.75 + 5 = 28.75 > 25. The final load on units 1 to 3 = 85 - 25 = 60 MW. The percent load on units 1 to 3 is

$$\frac{60 \text{ MW}}{3 \times 25} \times 100 = 80\%$$

With a 10% speed droop, the load change for units 1 to 3 is from 75% to 80%, resulting in a 0.5% speed decrease (see Fig. 10.19).

0.5% change ANSWER

100 - 0.5 = 99.5% speed

System Governing

For proper frequency control the setting of the governor must conform to electrical grid system requirements. In an isolated system, with one generating unit, drop in speed would be inappropriate, as the system frequency would vary greatly with load changes. In this case the speed droop and the speed adjustment should be set at zero to maintain the proper speed and line frequency.

Equipment is available for sharing the load between several units. This is called *joint control* and allows all the units in a power plant to govern the system (with zero speed droop) by electronically controlling the governors to share the load equally. As it is necessary to maintain close average frequency over significant time periods for electric clocks, a system must include equipment to integrate the number of cycles and signal a correction for any deviation from the frequency-time value.

Governor Theory and the Flywheel Effect

The basic mechanics of turbine speed change caused by change in load recognizes that the work done in the closing operation must equal the change in kinetic energy of the rotating masses, which is given by the following equation:

$$\int_0^t (\eta_t P_t - P_r)\,dt = \tfrac{1}{2} I(\omega_1^2 - \omega_0^2) \qquad (10.50)$$

or

$$k\,\Delta P_0 \tau = \tfrac{1}{2} WR^2\left(\frac{2\pi}{60}\right)^2 (n_1^2 - n_0) \qquad (10.51)$$

where η_t = efficiency of turbine
P_t = input of turbine, lb-ft/sec or kg-m/sec

P_r = power demand of system, including generator losses, lb-ft/sec or kg-m/sec
dt = elemental unit of time, sec
$I = WR^2/g$ = moment of inertia of the rotating mass, lb-ft sec^2 or kg-m sec^2
W = weight of rotating mass, lb or kg
R = radius of gyration of the rotating mass, ft or m
g = acceleration of gravity = 32.2 ft/sec^2 or 9.81 m/sec^2
ω_0 = initial speed of turbine before load change, rad/sec
ω_1 = highest (or lowest) speed due to the load change, rad/sec
τ = time during change of speed or time of closing or opening of turbine gates, sec.

Note that

$$k = \int_0^t \frac{(\eta_t P_t - P_r)}{\Delta P_0 \tau} dt \qquad (10.52)$$

where ΔP_0 = magnitude of change of load in the speed change, lb-ft/sec or kg-m/sec. At steady state when output from the turbine equals the generator load, $P_0 = \eta_t P_t = P_r$ and $\partial\omega/\partial t = 0$. When turbine output is smaller than the demand load on the generator, $\eta_t P_t < P_r$ and $\partial\omega/\partial t < 0$, and the turbine speed decreases. When the turbine output is greater than the demand load on the generator, $\eta_t P_t > P_r$ and $\partial\omega/\partial t > 0$ and the turbine speed increases.

It is common practice to use a term, C, called the flywheel constant, which is given by Jaeger (1970) as

$$C = \frac{WR^2 n_0^2}{P_0} \qquad (10.53)$$

Some authors define the flywheel constant as

$$C = \frac{WR^2 n_0^2}{N_0} \qquad (10.54)$$

where N_0 = power output, hp.

Jaeger (1970) simplifies Eq. (10.51) to the following forms:

$$\frac{\Delta n}{n} = \sqrt{\left(1 + \frac{1790}{C} k \frac{\Delta P_0}{P_0} \tau\right)} - 1 \quad \text{for load rejection} \qquad (10.55)$$

$$\frac{\Delta n}{n} = 1 - \sqrt{1 - \frac{1790}{C} k \frac{\Delta P_0}{P_0} \tau} \quad \text{for load increase} \qquad (10.56)$$

Jaeger points out that the k depends on pressure rise and pressure change resulting from water hammer action. For rejection of load when the time of speed change τ is smaller than the pressure pipe period $\mu = 2L/a$,

$$k = 0.5 + 0.5 \frac{aV_0}{2gH_0} - 0.1\left(\frac{aV_0}{2gH_0}\right)^2 \qquad (10.57)$$

and for time of closure or speed change where $\tau > 2L/a$,

$$k = 0.5 + 0.75\frac{LV_0}{gH_0\tau} - 0.125\left(\frac{LV_0}{gH_0\tau}\right)^2 \qquad (10.58)$$

where L = penstock lengths, ft or m
V_0 = water velocity in pipeline, ft/sec or m/sec
a = velocity of pressure wave, ft/sec or m/sec
H_0 = steady-state head, ft or m.

The flywheel effect can be related to accelerating time, T_m, for the rotating masses of the turbine by the following formula:

$$T_m = \frac{WR^2}{g'}\frac{n^2}{P_0} \qquad (10.59)$$

where P_o is in kg-m/sec. The U.S. Department of the Interior (1976) gives the flywheel effect equation in the following form:

$$T_m = \frac{WR^2}{1.6 \times 10^6}\frac{n^2}{P_{hp}} \qquad (10.60)$$

where W is in pounds, R is in feet, n is in rpm, and P_{hp} is in hp. In metric form the equation is

$$T_m = \frac{WR^2}{6.7 \times 10^4}\frac{n^2}{P_{kW}} \qquad (10.61)$$

where W is in kg, R is in meters, n is in rpm, and P_{kW} is in kW.

For the purpose of preliminary planning it is important to define two other terms. The first is T_w, the starting time of water column related to the water hammer action, which is given by the formula

$$T_w = \frac{LV_0}{g'H_0} \qquad (10.62)$$

where T_w = starting time of water column, in seconds, and other terms are as defined for Eq. (10.58). The second parameter of time is T_g, the equivalent servomotor opening or closing time. Hadley (1970) has shown that these terms can be related to each other and to pressure rise in the turbine water passage by a curvilinear relation. The ratio T_w/T_g is a function of pressure rise or pressure decrease. This relationship is shown in Fig. 10.20.

For practical use Hadley (1970) has shown that a simplification can be made of Eq. (10.55) and written in terms T_g and T_m as follows:

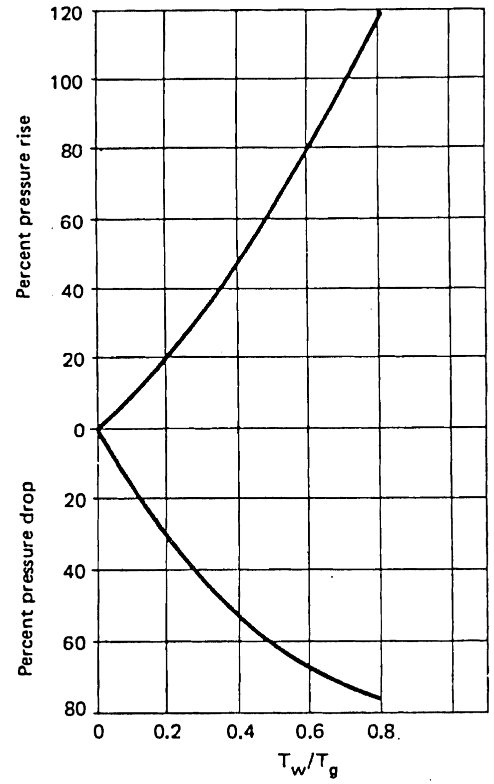


Figure 10.20 Variation in pressure rise and pressure drop with servomotor opening and closing time. SOURCE: Hadley (1970).

$$\frac{\Delta n}{n} = \sqrt{1 + \frac{KT_g(1 + \Delta H)^{3/2}}{T_m}} \qquad (10.63)$$

where K is a coefficient that is related to turbine characteristics and has been empirically expressed by Hadley as a function of the specific speed, as indicated in Table 10.2.

For normal speed-sensitive governing the flywheel effect required has been estimated by Hadley (1970) to have the following relationship between T_m, mini-

TABLE 10.2 Variation of the Governor Coefficient K with Specific Speed

n_s, specific speed	20	35	50	75	100	125	150	175	200
N_s, specific speed (metric)	76.3	133.6	190.8	286.3	381.7	477.1	572.5	671.8	763.4
K	0.98	0.96	0.92	0.84	0.77	0.71	0.66	0.61	0.57

mum starting time for the rotating of the turbine units, and T_w, the starting time of the water column.

$$\left.\begin{array}{l}\text{Pelton turbines}\\ \text{Francis turbines with pressure relief valve}\end{array}\right\} \quad T_m = 2.5T_w$$

$$\text{Francis and Kaplan turbines} \qquad T_m = 3.0T_w$$

Hadley also gives a rule of thumb for estimating the starting time or accelerating time of the rotating mass as follows:

$$T_m = \sqrt{\frac{\text{kVA}}{n}} \tag{10.64}$$

Empirically, the experience record of units furnished to the U.S. Bureau of Reclamation reported by the U.S. Department of the Interior (1976) indicates that WR^2 can be estimated by the following formulas:

$$\text{Turbine } WR^2 = 23{,}800\left(\frac{P_d}{n^{3/2}}\right)^{5/4} \quad \text{lb-ft system} \tag{10.65}$$

or

$$= 24{,}213\left(\frac{P_d}{n^{3/2}}\right)^{5/4} \quad \text{kg-m system} \tag{10.66}$$

$$\text{Normal generator } WR^2 = 356{,}000\left(\frac{\text{kVA}}{n^{3/2}}\right)^{5/4} \quad \text{lb-ft system} \tag{10.67}$$

or

$$= 15{,}000\left(\frac{\text{kVA}}{n^{3/2}}\right)^{5/4} \quad \text{kg-m system} \tag{10.68}$$

The flywheel effect is a stabilizing influence on the turbine speed and speed change. Most of the flywheel effect for direct-connected turbine-generator units is in the rotor of the generator. In practice, the flywheel effect is the design responsibility of the generator manufacturer guided by the governor design and the turbine design. It is possible for the generator manufacturer to add mass to the generator rotor to help in providing better speed control.

Governor design is beyond the scope of this book. However, Jaeger (1970) provides an excellent explanation of the basic mechanics, mathematics, and movement requirements. Another excellent reference on the topic of speed control and governors is a training manual of the U.S. Department of the Interior (1975). Details on the requirements and design for the servomotor and oil pressure requirements are given in Kovalev (1965). To understand the usefulness of the various relationships presented on speed control of turbines, the following example problem is presented.

Example 10.4

Given: A hydropower development is to have a design output, P_d = 2680 hp, at an estimated efficiency of η = 0.91, a design head of h_d = 36 ft, and a synchronous speed n = 257 rpm. The penstock serving the unit is 600 ft long and has a cross section of A_p = 50 ft^2.

Required:

(a) Determine the minimum starting time T_m for accelerating the rotating mass to a stable condition and the minimum flywheel effect WR^2 to give stable governing conditions.

(b) If the design closing time, T_g, for the servomechanism is 30 sec, estimate the expected pressure rise and the accelerating time T_m for accommodating the expected pressure rise when a speed rise of 10% for full-load rejection is considered. What will be the corresponding minimum flywheel effect to accommodate the 10% speed rise?

Analysis and solution:

(a) First the design discharge must be determined in order to calculate the starting time T_w. Utilizing the fundamental power equation, Eq. (3.8),

$$P_0 = \frac{q_d \gamma h_d \eta}{550}$$

solve for the design q_d:

$$q_d = \frac{P_0(550)}{\gamma h_d \eta} = \frac{2680(550)}{(62.4)(36)(0.91)}$$
$$= 721.06 \text{ ft}^3/\text{sec}$$

Velocity in the penstock:

$$V_0 = \frac{q_d}{A_d} = \frac{721.06}{50} = 14.42 \text{ ft/sec}$$

Using Eq. (10.62), solve for T_w:

$$T_w = \frac{LV_0}{gH_0} = \frac{600(14.42)}{(32.3)(36)} = 7.46 \text{ sec}$$

Find the minimum starting time for stable governing using the relation indicated by Hadley (1970), $T_m = 3.0T_w$, so that

$$T_m = (3.0)(7.46) = \mathit{22.38\ sec} \quad \text{ANSWER}$$

The required minimum WR^2 is obtained from Eq. (10.60):

$$T_m = \frac{WR^2 n^2}{1.6 \times 10^6}$$

Transposing gives

$$WR^2 = \frac{(1.6 \times 10^6)P_0 T_m}{n^2} = \frac{(1.6 \times 10^6)(2680)(22.38)}{(257)^2}$$

$$WR^2 = 1.453 \times 10^6 \text{ lb-ft}^2 \quad \text{ANSWER}$$

(b) It is necessary to characterize the turbine and find specific speed n_s. Using Eq. (4.17) gives us

$$n_s = \frac{n\sqrt{p}}{h_d^{5/4}} = \frac{257\sqrt{2680}}{(36)^{5/4}} = 150.9$$

To keep the relative speed change $\Delta n/n$ less than 0.10, Eq. (10.63) will apply.

$$\frac{\Delta n}{n} = \sqrt{1 + \frac{KT_g(1 + \Delta h)^{3/2}}{T_m}} - 1$$

From Table 10.2 for $n_s = 150.9$, $K = 0.66$. For $T_g = 30$, $T_w/T_g = 7.46/30 = 0.25$, and using Fig. 10.20 for $T_w/T_g = 0.25$, $h = 0.27$. Then

$$T_m = \frac{KT_g(1 + \Delta h)^{3/2}}{[(\Delta n/n) + 1)^2] - 1} = \frac{(0.66)(30)(1.27)^{3/2}}{(0.1 + 1)^2 - 1}$$

$$= 135 \text{ sec} \quad \text{ANSWER}$$

Using Eq. (10.62) again, we obtain

$$WR^2 = \frac{1.6 \times 10^6 P_o T_m}{(257)^2} = \frac{(1.6 \times 10^6)(2680)(135)}{(257)^2}$$

$$= 8.76 \times 10^6 \text{ lb-ft}^2 \quad \text{ANSWER}$$

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PROBLEMS

10.1. A turbine installation has a normal design head of 50 ft, a velocity in the penstock of 6 ft/sec, and a length of penstock from closure valve to an open water surface of 500 ft. Assume that the pressure wave velocity, a, is 3000 ft/sec. Determine the relative pressure rise if closure is less than 0.1 sec. What is the critical time of pressure wave travel? If the time of closure is 3 sec, what will be the expected pressure rise and pressure drop?

10.2. Visit a hydropower plant and determine the method of controlling water hammer. Report on closure time of closing valves or gates.

10.3. A simplified problem for a hydro plant involves a penstock of length 2500 ft, the velocity of the penstock is 6 ft/sec, the penstock area is 100 ft^2, the surge tank area is proposed to be 2000 ft^2, and the friction head loss in the penstock is 3 ft. Determine the approximate maximum pressure surge in the surge tank and time from beginning of flow until maximum surge if the gate valve is suddenly closed.

10.4. Three turbine generators of 20 MW capacity are supplying a load of 48 MW and the load suddenly increases to 50 MW. Two of the units are set with a speed droop of 9% and speed adjustment of 6%. The third unit is operating at zero speed droop and zero speed adjustment. Determine the change in system frequency due to the sudden load change. What will the speed change be if the load increases to 54 MW?

10.5. For a small hydropower plant with a proposed servomotor opening time for the gates of not to exceed 12 sec, select a suitable penstock diameter if the penstock is 1000 ft long and the turbine operates under a 60-ft head with an expected output of 5 MW. Determine a suitable design starting time, T_m, expected pressure rise, and the minimum flywheel requirement. Make any necessary assumptions.

11 POWERHOUSES AND FACILITIES

TYPES OF POWERHOUSES

Powerhouses for hydroplants usually consist of the superstructure and the substructure. The superstructure provides protective housing for the generator and control equipment as well as structural support for the cranes. The superstructure may provide for an erection bay that protects component assemblies during inclement weather.

The substructure or foundations of the powerhouse consists of the steel and concrete components necessary to form the draft tube, support the turbine stay ring and generator, and encase the spiral case. A control room is also included in the powerhouse to isolate the control systems from generator noise and to provide a clean and comfortable environment for operators.

Conventional Installations

Conventional powerhouses differ depending on how they are oriented or connected with the dam. Some are structurally connected to the dam; others are located some distance from the dam and a penstock carries the water from the intake to the powerhouse. Figure 11.1 is a simplified sketch of the general arrangement for powerhouses of the conventional type. Different ways of connecting the powerhouse to the impounding dam or the tunnel penstock are illustrated. Figure 11.2 is a plan layout and cross section for a typical powerhouse of the conventional type showing the arrangement of the various components. The shaft of the runner and generator is vertical and water flow is normal to the line of turbine units. Some

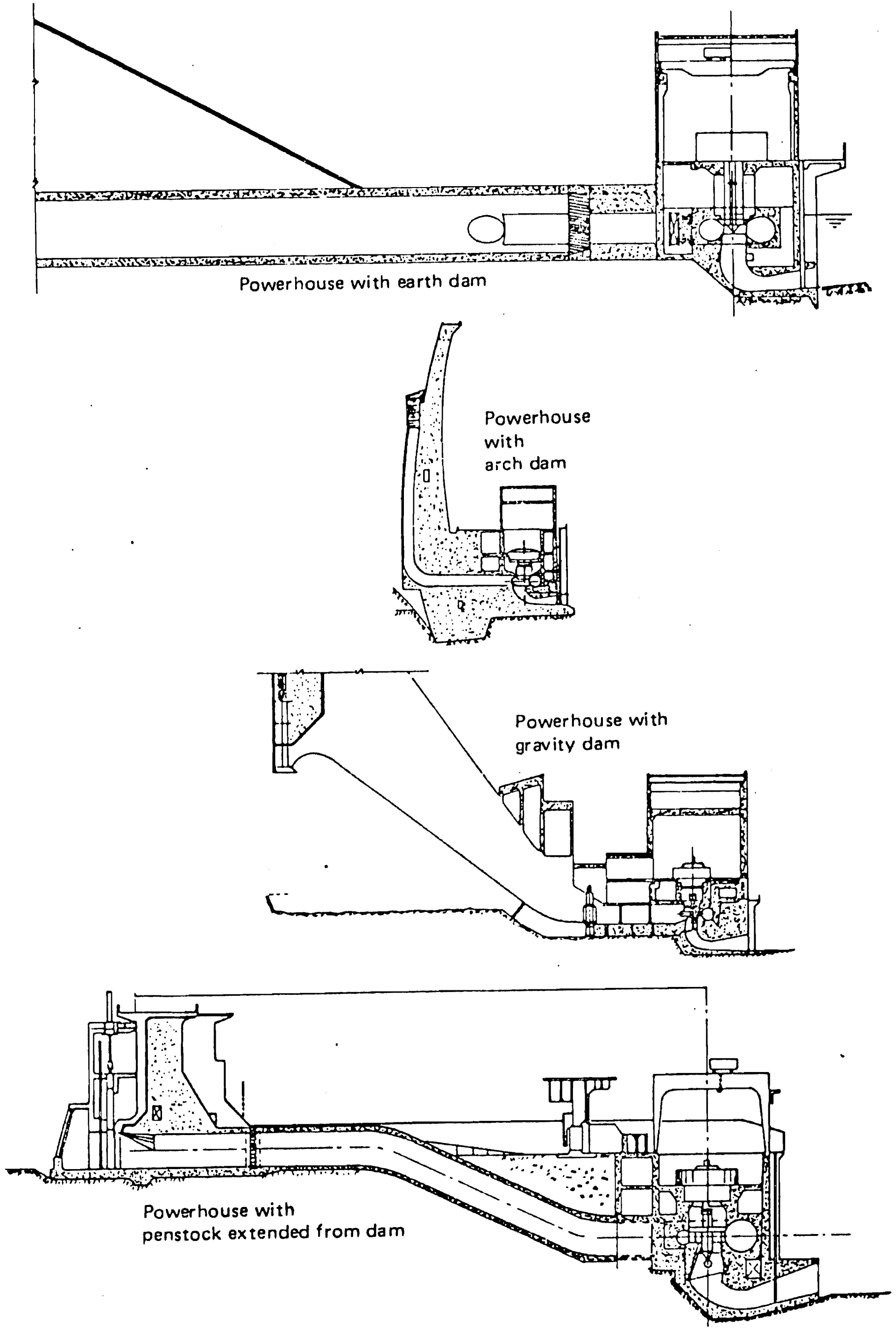
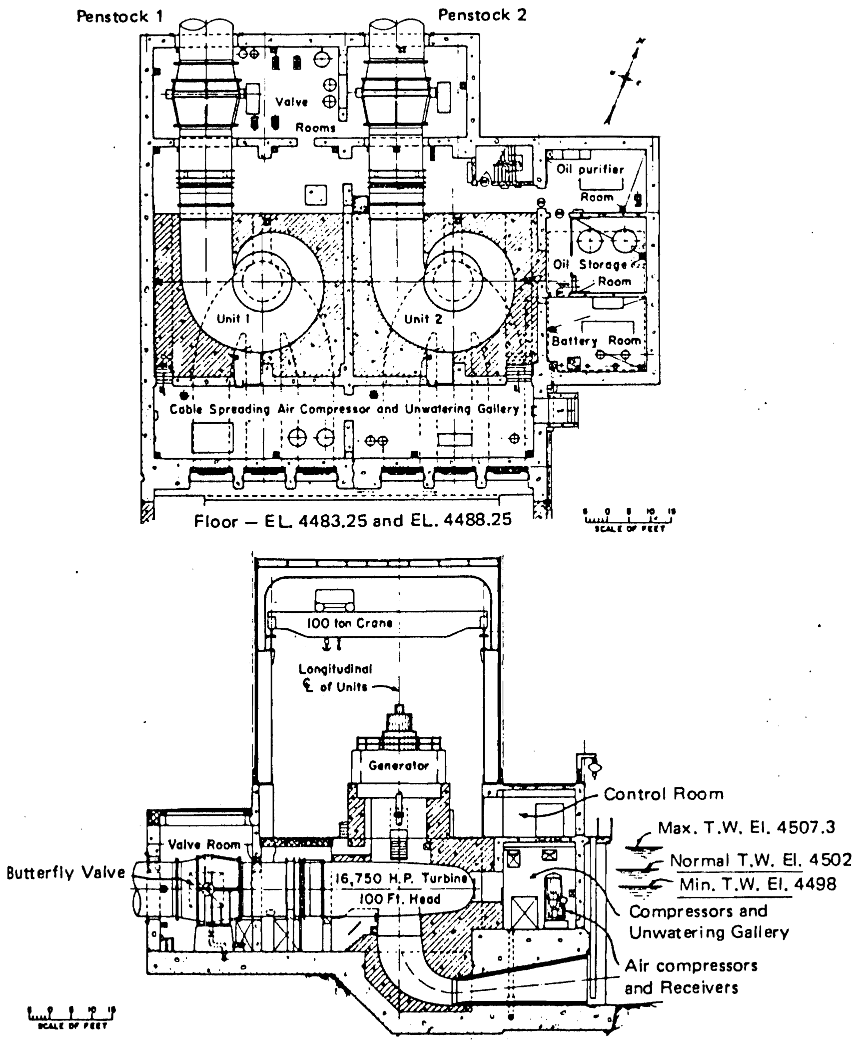


Figure 11.1 General arrangements for powerhouses for conventional hydropower installations.

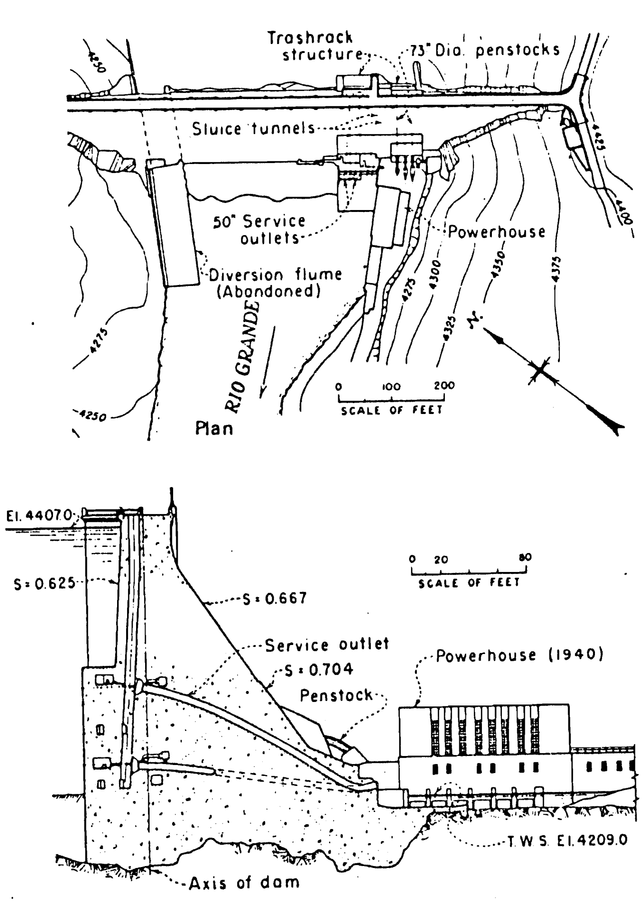


Transverse section thru ℄ unit 1

Glendo Powerplant

Figure 11.2 Plan layout and cross-section of conventional powerhouse. SOURCE: U.S. Bureau of Reclamation.

powerhouses are oriented differently to accommodate excavation and site preparation problems. Figure 11.3 shows how the powerhouse at Elephant Butte Dam in New Mexico is arranged, and Fig. 11.4 shows how the third powerhouse at Grand Coulee Dam in Washington was added to an existing dam to accommodate the discharge of water from the dam to the turbines. Each development may require



Section at Service Outlets

Figure 11.3 Unique layout of powerhouse at Elephant Butte Dam, New Mexico. SOURCE: U.S. Bureau of Reclamation.

special arrangements to accommodate the geology and topography at the site in order to gain the best economy of construction. Design of the powerhouse is primarily a structural and architectural problem. The size of the building is governed by the requirements to accommodate the generator, the spiral casing, and the outlet area of the draft tube. It is the usual practice for larger capacity installations to have vertical shafts; the layout dimensioning presented in Chapter 8 gives controlling dimensions that can be used in feasibility studies. For final design it is customary for the turbine and generator manufacturers to furnish dimensions for the interiors of the spiral casing, draft tube, and generator assembly.

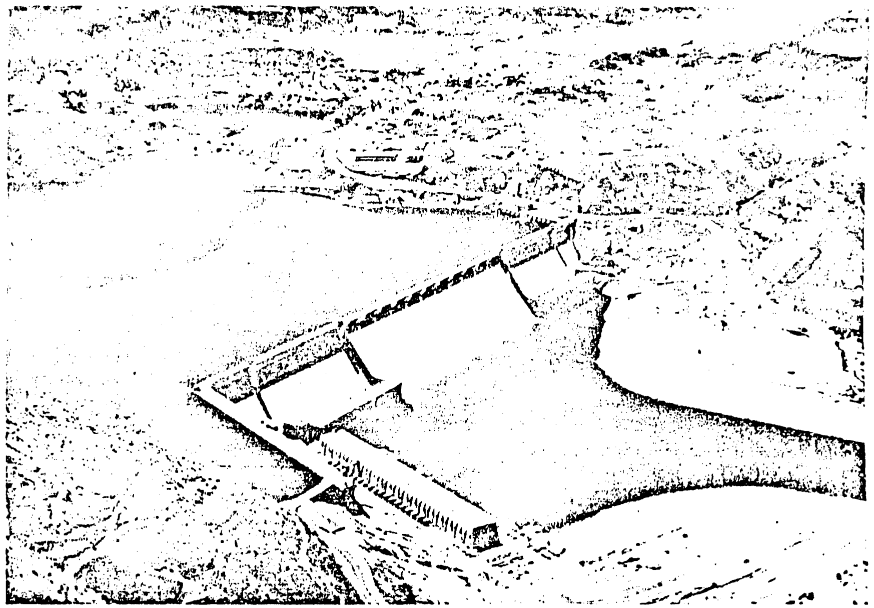


Figure 11.4 Powerhouse arrangement at Grand Coulee Dam, Washington. SOURCE: U.S. Bureau of Reclamation.

Outdoor and Semi-Outdoor Installations

In some hydropower installations it is economical to dispense with superstructure and protect the generator with a weatherproof covering of metal. This is called an outdoor powerhouse. The substructure supports the turbine and generator assembly. An outdoor crane for large units and a mobile crane for smaller units is likewise included in an outdoor installation. Figure 11.5 is a photograph of a typical outdoor installation. The weather conditions dictate whether outdoor installations are practical.

A semi-outdoor installation involves locating all equipment in the substructure levels and providing exterior or mobile cranes that move the unit components through hatches or openings in the substructure roof. Such installations have the advantage of providing a more aesthetic exposure of the power plant and frequently can reduce costs of low-head power plants. Figure 11.6 shows a cross-sectional sketch of a semi-outdoor type of hydropower plant.

Horizontal-Shaft Installations

With low-head hydropower installations utilizing bulb units and tubular units, the turbine shafts are mounted horizontally. This allows a design for water

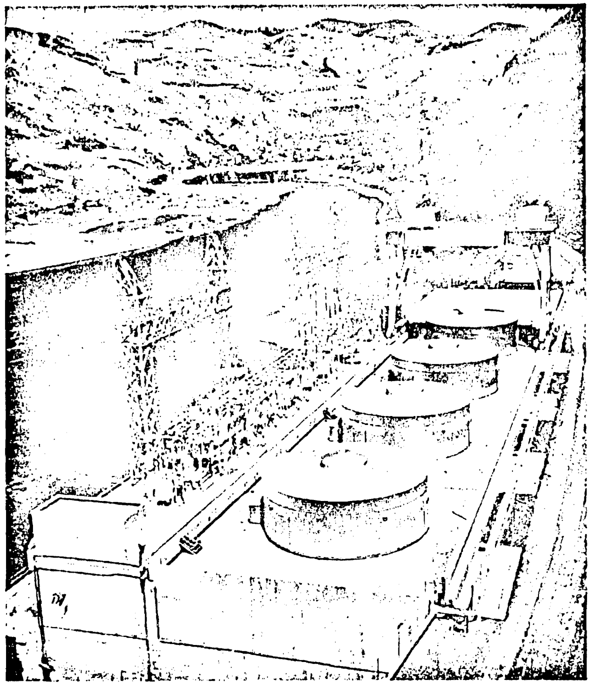


Figure 11.5 Outdoor-type powerhouse arrangement. SOURCE: Idaho Power Co.

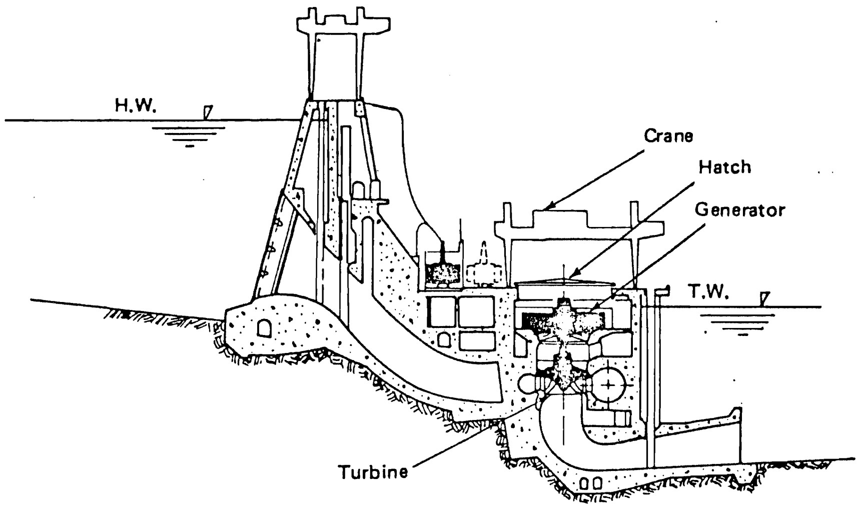


Figure 11.6 Semi-outdoor-type powerhouse arrangement.

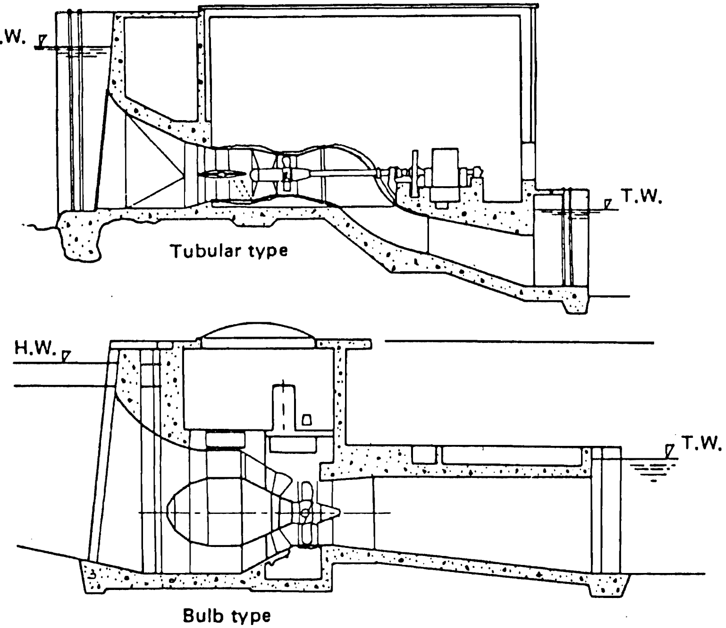


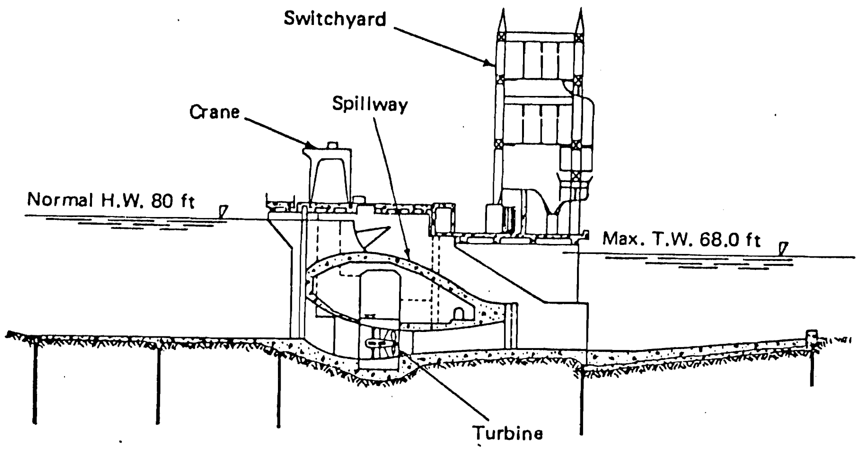
Figure 11.7 Low profile, powerhouse arrangements for tubular and bulb-type hydro developments. SOURCE: KMW, Sweden.

passages that can be accommodated in low-profile powerhouses to protect the turbines and generators. For some installations no powerhouse has been required. Figure 11.7 shows a cross-sectional sketch of two different types of turbines with representative powerhouse designs. Normally, the powerhouse and civil works are longer in the direction of flow and narrower than conventional vertical-shaft mounted turbines and require less excavation depth to accommodate the turbine equipment and water passages.

In special cases where space for powerhouses and spillway is critical, the turbines and generators can be housed in the overflow spillway portion of a dam. The gantry cranes and electrical facilities are accommodated on decks bridging across the overflow spillway. Figure 11.8 shows a cross-sectional sketch of such an installation.

Underground Installations

To accommodate narrow canyon locations and to reduce environmental impact, powerhouses can be built underground using tunnel construction. This requires access tunnels to deliver equipment. Figure 11.9 gives an example of an underground installation. The excavated volume should be kept to a minimum and all functions of air conditioning and ventilation adequately planned. Underground powerhouses are very specialized in their design and thus beyond the scope of this



Brahmaputra Barrage

Figure 11.8 Spillway-type powerhouse arrangement.

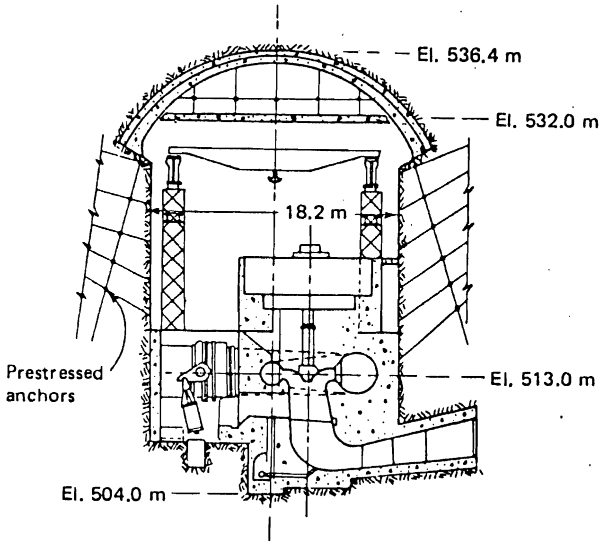


Figure 11.9 Underground-type powerhouse arrangement.

book. An excellent reference on the topic is Jaeger (1970). Another useful reference is the U.S. Army Corps of Engineers (1960).

FACILITIES AND AUXILIARY EQUIPMENT

Water Bypass and Drainage

Important in planning for hydropower installations are the necessary arrangements for bypassing water. Separate sluices or channels that operate independently of the penstock or intake that supplies water to the turbine may be required. Need for bypass facilities will depend on the arrangement and requirements for releasing water when not discharging water through the turbines. Usually, the separate bypass facilities are a part of the spillway and outflow facilities of the dam. The planning must be done in connection with the flood design analysis and planning for other releases for irrigation or water supply purposes. Figure 11.10 shows how

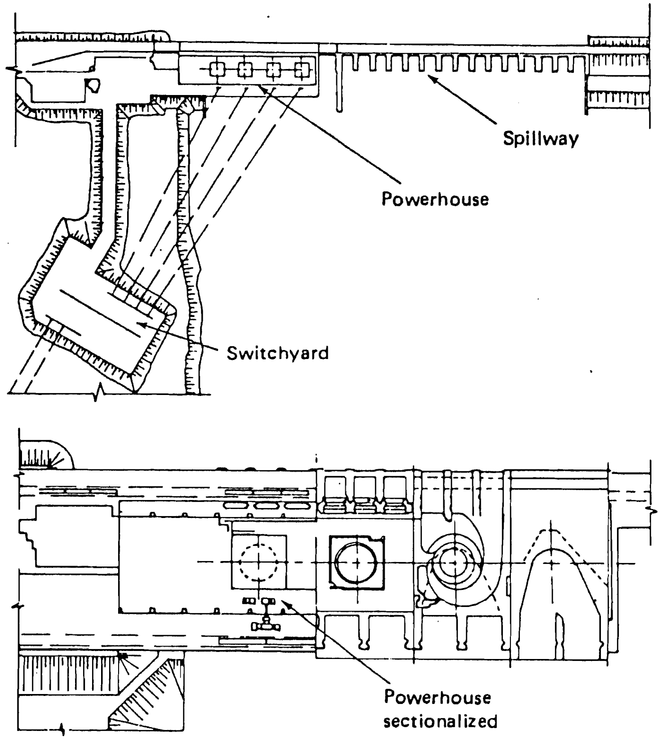


Figure 11.10 Plan layout for powerhouse.

the spillway and outlet works might be arranged relative to the powerhouse of the hydropower plant. The planning must accommodate different dam designs and variable geologic conditions. A useful reference on this subject is provided by the U.S. Department of the Interior (1977).

Arrangements must always be made for dewatering or draining water from the civil works to allow access to the turbine and the draft tube. Provisions are usually made to have gates or stop logs upstream of the spiral case that can be closed to allow water to drain from the intake area, the spiral case, and the turbine area itself. These can usually drain by gravity down to the level of the tailwater.

Draft tube dewatering requires sump pumps and a stop log arrangement at the outlet to the draft tube. The discharge outlets should be above the maximum tailwater level to prevent water from getting back into the draft tube. Pumps are necessary to dewater the draft tube area and pits may be necessary to accommodate the suction arrangement for the pumps. Design of walls and structural components should allow for pressure conditions when passages are full of water and also when water is evacuated, as well as for any subatmospheric pressure that can result from hydraulic transients. The pumps and pump sumps may need to be designed to accommodate leakage and have an auxiliary power source that can operate when station power or network system power is not available. Powerhouses with high tailwater must be checked for flotation.

Electrical Protection Equipment and Switchyard

The design effort incorporating the electrical equipment into the powerhouse is specialized work of the electrical engineer. Special requirements for electrical cable passages, trenches, and ducts must be accommodated in final design. Important provisions are space for switching, the location of the transformers, and auxiliary equipment. Batteries for standby electricity in event of station power outages and accommodating electrical equipment for alarm systems, communications, and lighting may also be required.

Transformer locations vary greatly. If the transformers are mounted outdoors, they are frequently located upstream of the powerhouse, somewhat hidden from public view. Sometimes location of overhead transmission lines from the power plant may make it desirable to locate the transformers on a deck above the draft tubes. In other locations it may be desirable to locate the transformer yard some distance away from the powerhouse. In all cases, adequate spacing must be provided to minimize fire hazard with the large volume of oil involved in the transformers. Protective isolation walls between adjacent units may be necessary and oil drainage facilities must be provided. The type of cooling may also dictate an inside or protective housing of the transformer units. Auxiliary transformers will often be necessary depending on the auxiliary power supply that is provided. Space must be provided for these auxiliary units. Air-cooled transformers are commonly used and these are located out-of-doors.

Switchgear is a term that includes the current breakers that are necessary for

making and breaking the current continuity of the electricity generated. The switchgear assembly is necessary to safeguard and protect equipment and personnel, to connect the generators to the transmission system, to provide for withdrawal of current for circuits to supply electrical service to the plant itself, and to provide automatic disconnection of any faulty equipment. Factors and information necessary for designing switchgear are transmission voltage, number of circuits, size and capacity of generators, load reliability required, busbar voltage, interconnection with other plants, land available, and distance to location of switchgear. Planning the electrical equipment associated with powerhouses is primarily an electrical engineering problem. A useful reference is the National Electrical Manufacturers Association (NEMA) (1979).

Cranes

Handling of the large components of hydroplants, including the turbine runner, the shaft of the unit, and the rotor and stator of the generator, requires large cranes to maneuver the equipment during installation and to remove components or entire assemblies for maintenance. If the cranes are housed in the superstructure, they are usually bridge cranes spanning the generator bay. They operate on rails and have trolleys to move transversely along the direction of the supporting beam. Planning for powerhouses requires information such as the center-to-center span of runway rails, building clearance from the wall and supporting roof components, hook travel and elevation, high and low distance limits of movement, operating cage location, and maximum spatial size of equipment, plus maximum weight. Figure 11.2 shows a typical arrangement for a powerhouse operating crane. These cranes may need to have capacities of 300 to 500 tons in large power plants.

Gantry cranes have legs that support the lifting mechanism and a hoisting trolley that operates on a bridge between the legs. Usually, the legs are mounted on four two-wheeled trucks that operate on rails. In some cases one leg is a stub or abbreviated leg that operates along an elevated rail on a portion of the dam. Gantry cranes are usually enclosed to some extent to provide protection from the weather. The gantry crane is customarily used with outdoor-type powerhouses. Sometimes it is necessary to have a separate gantry crane to operate and lift the gates and stop logs on large power plants.

Jib cranes with a swing book are available as fixed or mobile units and for smaller hydropower installations are used to economic advantage.

A useful reference in planning for power station cranes is available from the U.S. Army Corps of Engineers (1968).

Auxiliary Systems

Some of the auxiliary systems required for hydro powerhouses are:

Dewatering and filling system

Cooling-water system
Service-water system
Flow, pressure, and level measuring system
Sanitary system
Station drainage system
Fire alarm and protection system
Lubricating and insulating oil systems
Compressed air system
Heating, ventilating, and air-conditioning systems (HVAC)
Machine shop equipment
Emergency power system

The auxiliary systems which are most important from an engineering standpoint will be discussed.

Ventilation systems. Provisions must be made for cooling the generator and for maintaining satisfactory temperatures in the powerhouse. Neville (1970) gives an equation determining the V_A, volume of air in cubic feet per minute, for ventilating generators as

$$V_A = 1650 \frac{G_L}{T_c} \tag{11.1}$$

where G_L = generator losses, kW
T_c = temperature rise of air, °C.

For example, if a 5000-kVA generator with an 80% power factor as shown in Fig. 9.7 has losses of 128 kW and the temperature rise is limited to 10°C, the volume of air per minute required would be 21,120 ft^3/min.

Different arrangements are possible for supplying the ventilation air. In a small powerhouse with adequate space and sufficient windows and louvered openings, it may be possible to take the air from the generator and discharge it into the station. In medium-size powerhouses it may be necessary to supply air from ducts with fans either taking air from outside and discharging it into the powerhouse after passing around the generator or taking air from the powerhouse and discharging it to the outside. In either case, care must be taken that the air is free of dust. Duct air velocities should be kept below 1500 ft/min.

In large power installations of the vertical-shaft type, the ventilation system will require recirculating air that is water-cooled using heat exchangers. This has the advantage that the air can be kept free of dust and contaminants and less cleaning of the generator will be required. Similarly, the hazard of fire can be lessened due to the air being kept under controlled conditions.

Layout of equipment for cooling and piping for the water line must be provided in the design of the powerhouse. In colder climates, heat will be necessary to prevent freezing. Heat from the generator may be utilized in the space heating if careful consideration is given to a backup heat supply in case of downtime with the power units. Design of air diversion and air exchange into and out of the powerhouse should minimize hazard from fires that might occur in the generator or the generator area.

Where humidity is high, condensation may occur on the generator windings. This may be particularly troublesome when generator units are being started. Space heaters may be needed to dry the air. A simple means of drying out the air in the generators when restarting them is to run the machine on a short-circuit at about half speed and half rated current.

Water systems and fire protection. Arrangements must be provided in the powerhouse for water for various uses: (1) as a domestic supply for operating personnel, (2) as cooling water, and (3) as a fire-protection system.

The domestic water supply system is normally independent of the water flowing through the turbine and will require a higher quality that is potable. There may be need for showers, wash basins, toilets, water fountains, and washing facilities for cleaning the powerhouse and equipment.

The quantity of cooling water will depend on the amount of air used in cooling, the equipment to be cooled, the temperature rise allowed in the powerhouse, and the seasonal ambient air temperatures.

Fire-protection water will need to be at a high pressure to be able to reach all equipment and must be able to operate in emergencies when the station power is not available. The piping system will need to be connected to the detector and sprinkler system and should normally be independent of the domestic water system. Planning should include fire-protection water for the transformer area which might be somewhat removed from the powerhouse.

Present-day practice for fire protection is to use carbon dioxide fire extinguisher systems as much as possible for enclosed areas. The carbon dioxide helps to quench the fire, cool the area, and the foam covering prevents oxygen from getting to the endangered area or equipment. Precautions must be taken to limit having personnel in areas where carbon dioxide is discharged. Some designs for fire protection in enclosed spaces, such as in bulb units, use Halon gas instead of carbon dioxide. Halon extinguishes fire by a chemical reaction rather than by oxygen deprivation, thus allowing for safer évacuation of personnel. Design arrangements for locating oil supply lines and isolating the storage of oil make it less likely for oil fires to occur. Separate portable CO_2 extinguishers must be provided at strategic locations in all powerhouses.

Oil systems. Two types of oil are used in a hydropower installation. Lubricating oil is used in the load and speed control system, the bearings, and sometimes in hydraulic operation of valves and gates. The system is a high-pressure oil system and includes the necessary storage reservoirs, pumps, and piping. Transformer oil is used in the transformers of the electrical system.

The high-pressure oil system for load and speed control requires maintaining oil pressure up to more than 70 atm (1000 psi) under nearly constant-pressure conditions. The oil pressure is used to activate pistons that supply force and movement to blade and gate operations. Pumps used for this purpose are gear or rotary screw pumps.

Figure 10.18 in an earlier chapter shows a schematic drawing of the arrangement for a hydropower oil system. A good reference covering details and suggestions for oil system piping and physical arrangements is presented by the U.S. Department of the Interior (1971). This contains useful information on piping and arrangements for large unit water needs and fire-protection systems.

Other Planning Considerations

Access roads. Planning for hydropower plants must include transportation access roads that will facilitate all construction activities including dewatering of the construction area, movement of excavated material, movement of materials of construction, and movement of equipment into place. Provision must be made for the movement of personnel and equipment during operation and maintenance of the plant. This might include mobile cranes and necessary large vehicles to move large pieces of equipment for maintenance plus equipment and components for operating the gates used for water control. Normally, these roads need to be hard surfaced to assure durability in all kinds of weather conditions. Caution should always be taken that the roads are located above flood levels. The terrain and geology normally dictate the location of access roads.

Fish passage. Most hydropower plants will involve some accommodating facilities for handling fish. If the impoundment dam blocks the stream, fish ladders for passing fish over or around the dam may be required. These are necessary on streams with migrating fish, especially on streams having anadromous fish runs.

A frequent problem is handling the passing of fish or controlling the movement of fish through the turbines. Present environmental requirements demand some control measures. Four different concepts have been listed by the American Society of Civil Engineers (1981). These are (1) fish collection and removal, (2) fish diversion, (3) fish deterrence, and (4) physical exclusion. The first control measure implies there must be some screening system, usually a traveling screen, that collects and removes the fish. The second measure employs a design to remove the fish from the intake of the turbine without the fish being impinged on a screen. The fish are guided to a possible means of bypassing turbines through water that does not flow through the turbines. The other two methods are merely variations of this concept.

In most cases there will still be some fish passing through the turbines. Studies by Cramer and Oligher (1964) and Turbak, Riechle, and Shriner (1981) show that the maximum survival of fish in prototype hydropower units was associated with relatively low runner speed, high efficiency of the turbine, and relatively deep

setting of the turbine below tailwater to minimize negative pressure. This should be used as a guide in planning and design of new developments. Recent tests indicate very low mortality in tubular-type turbines.

Most hydropower developments with need for downstream fish passage will be species specific and site specific and will require specialized skills in fisheries science. Contact should be made with either the State Fish and Game Departments, the U.S. Fish and Wildlife Service, or the U.S. Bureau of Commercial Fisheries. The work of Long and Marquette (1964) is particulary significant and applicable in this area.

Frequently, it is necessary to provide for migrant fish passage upstream over or around a dam. Four types of facilities are used: (1) fishways, (2) fish locks, (3) fish lifts or elevators, and (4) fish traps with trucks. In a *fishway,* fish swim up a series of pools each of which is slightly higher in elevation than the preceding pool. These are often referred to as *fish ladders.* In a *fish lock,* the fish are crowded into a chamber, the chamber is lifted to the upstream headwater elevation, and fish are allowed to swim out. The *fish lift* is similar to the fish lock chamber, except that the fish lift uses a mechanical hopper to raise the fish above the dam. This is a specialized element of design for powerhouses and requires special fisheries skills. A good generalized reference for this is the work of Hildebrand et al. (1980). Further information is contained in Loar and Sale (1981).

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PROBLEM

11.1. Obtain a topographic map of a proposed hydropower site. Make a preliminary layout for the powerhouse and prepare a list of facilities that will be required.

12 ECONOMIC ANALYSIS FOR HYDROPOWER

INTRODUCTION AND THEORY

Economic analysis of hydropower projects concerns measuring the benefits from the development and the costs expended. In the context of hydropower planning, *benefits* are the goods and services produced by the development and *costs* are the goods and services used in constructing and maintaining the development. An economic analysis is necessary to determine whether the project is worth building and to determine the most economical size of the development or components of the development. Because both benefits and costs come about at different times, it is necessary to evaluate the benefits and costs in equivalent monetary terms, considering the time value of the expenditures and revenues involved. This is primarily an engineering economics problem.

Cash Flow Calculations

Cash flow is the expenditure (costs) and receipts or values obtained (benefits) over time that are directly related to a given enterprise or project development. A cash flow diagram is a graphic representation of cash flow with magnitude of expenditures and receipts plotted vertically and time represented on the horizontal scale. Figure 12.1 is a typical cash flow diagram. The benefits (receipts) are represented by upward arrows and costs (expenditures) are represented by downward arrows. The distance along the horizontal scale represents time.

The basic idea for economic equivalency calculations is to convert the value of benefits and costs that occur at different times to equivalent monetary amounts,

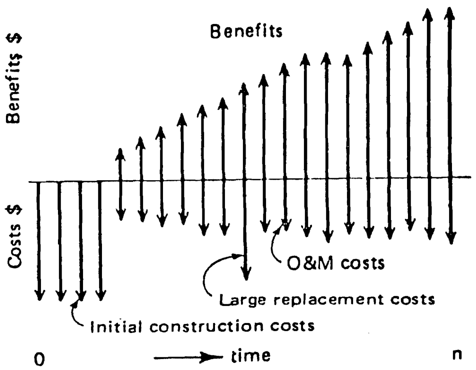

Figure 12.1 Cash-flow diagram.

recognizing the time value of money. Terms used in cash flow analysis are present worth, future worth, interest, and equivalent annual payment.

Present worth is a sum of money at the present, the value of an investment at the present, or the value of money expended in the future discounted back to the present. *Future worth* is a sum of money at a future time, the value of a future investment, or the value of an expenditure at present discounted out to that future time. *Interest* is the price paid for borrowing money expressed as a percentage of the amount borrowed or the rate of return (discount rate) applied in computing the equivalency of present worth and future worth. *Equivalent annual payment* is a discounted uniform annual amount expended or paid that is equal to a present invested amount to cover some given activity over a fixed period of time.

The process of mathematically obtaining the present worth of benefits and costs is called *discounting.* This recognizes the time value of money in the form of the willingness to pay interest for the use of money.

The interest formulas to facilitate equivalence computation are presented in equations as a number of discounting factors. Nomenclature and definitions of terms are:

P = a present sum of money, expressed as dollars in this book

This term represents a present worth; it can be either a cost or a benefit.

F = a future sum of money. The future worth of P is an amount n interest periods from the present that is equivalent to P with interest at rate i.

i = interest rate per interest period (decimal form used in equations). This is referred to as the *discount rate.*

n = number of interest periods (not always years).

A = an end-of-period cash amount in a uniform series continuing for n periods. The entire series is equivalent to P or F at interest rate i.

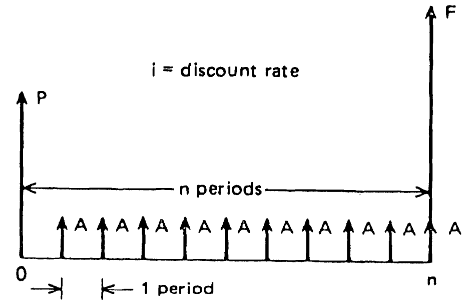

Figure 12.2 Definition sketch of discounting terms.

Figure 12.2 is a graphic representation of the terms present worth, future worth, and equivalent annual payment in a cash flow diagram of positive values, benefits.

Single-payment compound-amount factor. An initially invested amount P after one interest period will have accumulated an amount F for every dollar invested at i interest:

$$F = P(1 + i) \tag{12.1}$$

and after n interest periods:

$$F = P(1 + i)^n \tag{12.2}$$

The term $(1 + i)^n$ is known as the single-payment compound-amount factor (SPCAF). A functional designation for the factor is $(F/P, i, n)$:

$$(F/P, i, n) = (1 + i)^n = F/P \tag{12.3}$$

Single-payment present-worth factor. The inverse of Eq. (12.3) provides a means of determining the present value of an amount F that would have accumulated in n interest periods if the interest rate paid is i compounded for each period. The single-payment present-worth factor (SPPWF) is then $1/(1 + i)^n$ noted functionally as $(P/F, i, n)$, so that

$$(P/F, i, n) = \frac{1}{(1 + i)^n} = P/F \tag{12.4}$$

Uniform-annual-series factors. Discounting and present-worth calculation can be done using the foregoing equations, but it is useful to determine benefits and costs in the form of equivalent annual payments. This provides a means of calculating a series of equal annual payments that is equivalent or equal to a present-worth, P, or a future-worth value, F, based on a defined interest rate for discounting. Four terms are frequently used: (1) sinking-fund factor, (2) compound-amount factor, (3) capital recovery factor, and (4) uniform-series present-worth factor.

The *sinking-fund factor* (SFF) is functionally noted as $(A/F, i, n)$, so that

$$(A/F, i, n) = \frac{i}{(1+i)^n - 1} = A/F \tag{12.5}$$

where A is the magnitude of equal payments at the end of each period for n periods that must be invested at interest rate i to accumulate an amount F at the end of n periods. Usually, the periods are one year, so the equal payments are referred to as uniform annual payments, A. A sinking fund is a separate fund into which payments are made to accumulate some desired amount in the future.

The *compound-amount factor* is the inverse of the sinking fund factor and is also known as the uniform-series compound-amount factor (USCAF), which is functionally noted as $(F/A, i, n)$, so that

$$(F/A, i, n) = \frac{(1+i)^n - 1}{i} = F/A \tag{12.6}$$

Compound amount is the value a series of payments compounded at rate i will have in the future.

The *capital recovery factor* (CRF) is concerned with the capital recovery amount that is the value of uniform payments that are made and discounted at the rate i from a present worth. The factor (CRF) is obtained by combining Eqs. (12.2) and (12.5) and is functionally noted as $(A/P, i, n)$, so that

$$(A/P, i, n) = \frac{i(1+i)^n}{(1+i)^n - 1} = A/P \tag{12.7}$$

The *uniform-series present-worth factor* (USPWF) is concerned with the present-worth value of a series of equal payments made over some specified period of time and discounted at rate i. The factor is functionally noted as $(P/A, i, n)$, so that

$$(P/A, i, n) = \frac{(1+i)^n - 1}{i(1+i)^n} = P/A \tag{12.8}$$

Uniform-gradient-series factors. Useful also are equations that permit calculation of present-worth or annual-equivalent amounts wherein the periodic payments are uniformly increasing. These are termed gradient-series factors and the present-worth uniform-gradient series factor (PWUGSF) is functionally noted as $(P/G, i, n)$, so that

$$(P/G, i, n) = \frac{(1+i)^{n+1} - (1 + ni + i)}{i^2(1+i)^n} = P/G \tag{12.9}$$

where G is a uniformly increasing payment for each period of 1 G per period. This uniform-gradient discounting situation is shown graphically on a cash flow diagram in Fig. 12.3.

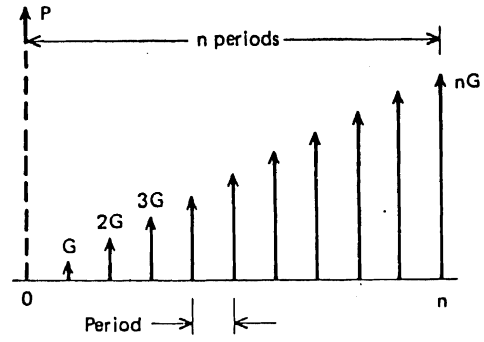

Figure 12.3 Gradient series cash-flow diagram.

Further elaboration giving additional inverse relations and factors for geometric progressions of growth for discounting payments can be found in standard engineering economics texts such as Newnan (1976).

METHODOLOGY FOR ANALYSIS

Basically, economic comparisons and equivalency evaluations can be made by the following methods:

1. Present-worth comparison
2. Annual-worth comparison
3. Future-worth comparison
4. Rate-of-return comparison
5. Net benefit comparison
6. Benefit-cost ratio comparison

Present-Worth Comparison

Present-worth comparison requires converting all cash flows to an equivalent present-worth value. This can involve three important concepts in discounting practice: (1) salvage value, (2) future required replacement costs, and (3) project life or discounting period of anlaysis. *Salvage value* involves an estimate that identifies the future value of the original investment, say a penstock or dam at some point of time in the future. Similarly, a *replacement cost* is a required payment sometime in the future that must be estimated and that payment discounted back to the present. Most important is the *project life* or *discounting period* when comparing two alternatives. The period of analysis must be equivalent. If the project lives of two alternatives are different, then a least common multiple of lives must be used and identical replacement consideration made. This requires a calculation of periodic reinvestment for the least common time. To illustrate the principle of present-worth comparison, a brief example of an economic analysis of a component portion of a hydropower plant is presented.

Example 12.1

Given:

Alternative A, long penstock:

Initial investment	\$510,000
Useful life	30 yr
Salvage value at 30 yr	\$50,000

Alternative B, canal and short penstock:

Initial investment cost	\$520,000
Useful life	45 yr
Canal gate replacement cost every 15 yr	\$30,000
Salvage value at 45 yr	\$70,000

Required: Make a present-worth comparison of the two alternatives. Consider interest rate $i = 10\%$.

Analysis and solution: The least common multiple of lives is 90 years, so that the cash flow diagrams for the two alternatives would be as indicated in Fig. 12.4. $P_{A,1}$ is the initial investment, $S_{A,1}$ is the salvage value at the end of the first investment period, $P_{A,2}$ is the investment at start of the second period, and $S_{A,2}$ is the salvage value at the end of the second investment period, and so on. R_B is the replacement cost at the end of each 15-year period. The calculation would be as follows, using Eq. (12.4):

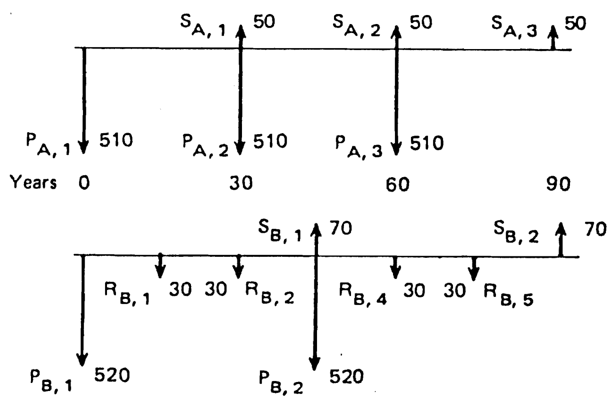

Figure 12.4 Cash-flow diagram for Example 12.1.

$$(P/F, i, n) = \frac{1}{(1+i)^n}$$

$$\begin{aligned}
\text{PW Cost alt. A} &= P_{A,1} - S_{A,1}(P/F, i, 30) \\
&\quad + P_{A,2}(P/F, i, 30) - S_{A,2}(P/F, i, 60) \\
&\quad + P_{A,3}(P/F, i, 60) - S_{A,3}(P/F, i, 90); \qquad i = 0.10 \\
&= 510{,}000 - 50{,}000(0.05731) + 510{,}000(0.05731) \\
&\quad - 50(0.003284) + 510{,}000(0.003284) - 50{,}000(0.0001882) \\
&= \textit{\$537,863} \quad \text{ANSWER}
\end{aligned}$$

$$\begin{aligned}
\text{PW Cost alt. B} &= P_{B,1} + R_{B,1}(P/F, i, 15) + R_{B,2}(P/F, i, 30) \\
&\quad - S_{B,1}(P/F, i, 45) + P_{B,2}(P/F, i, 45) + R_{B,4}(P/F, i, 60) \\
&\quad + R_{B,5}(P/F, i, 75) - S_{B,2}(P/F, i, 90); \qquad i = 0.10 \\
&= 520{,}000 + 30(0.02394) + 30{,}000(0.05731) \\
&\quad - 70{,}000(0.01372) + 520{,}000(0.01372) + 30{,}000(0.003284) \\
&\quad + 30{,}000(0.000786) - 70{,}000(0.0001882) \\
&= \textit{\$536,070} \quad \text{ANSWER}
\end{aligned}$$

Alternative B has the lesser present-worth cost.

Annual-Worth Comparison

Annual-worth comparison requires that all expenditures and revenues be converted to equivalent annual cash flows. For the cost component comparison the annual cost is the equivalent first cost considering salvage and is termed capital recovery cost. Several correct formulas that are interchangeable can apply. In functional form the formulas are

$$\text{EUAC} = P(A/P, i, n) - F(A/P, i, n) \tag{12.10}$$

$$\text{EUAC} = (P - F)(A/F, i, n) + P(i) \tag{12.11}$$

$$\text{EUAC} = (P - F)(A/P, i, n) + F(i) \tag{12.12}$$

where EUAC = equivalent uniform annual cost, dollars
P = initial investment cost, dollars
F = salvage value at time n, dollars

A required feature of the calculation is that the analysis period is to be the least common multiple of project lives. Example 12.2 illustrates the use of this method of comparison for two pumps being considered for a hydropower installation.

Example 12.2

Given: Two pumps:

	Pump A	Pump B
First cost	$7000	$5000
Salvage value	$1500	$1000
Useful life	12 yr	6 yr

Assume that interest rate $i = 7\%$.
Required: Determine the most economical purchase.

Analysis and solution: Using Eqs. (12.7) and (12.12), we have

$$\begin{aligned}
\text{EUAC}_A &= (P - F)(A/P, i, n) + F(i) \\
&\qquad n = 12, \quad i = 0.07 \\
&= (7000 - 1500)(0.1259) + 1500(0.07) \\
&= \$797 \quad \text{ANSWER} \\
\text{EUAC}_B &= (P - F)(A/P, i, n) + F(i) \\
&\qquad n = 6, \quad i = 0.07 \\
&= (5000 - 1000)(0.2098) + 1000(0.07) \\
&= \$909 \quad \text{ANSWER}
\end{aligned}$$

For an $n = 12$ periods of analysis for pump B, use Eqs. (12.4) and (12.7):

$$\begin{aligned}
\text{EUAC}_B &= [P - F(P/F, i, n) + P(P/F, i, n)] \\
&\qquad i = 0.07, \quad n = 6 \qquad i = 0.07, \quad n = 6 \\
&\qquad - F(P/F, i, n)]\,[(A/P, i, n)] \\
&\qquad i = 0.07, \quad n = 12 \\
\text{EUAC}_B &= [5000 - (1000)(0.6663) + 5000(0.6663) \\
&\qquad - 1000(0.4440)]\,[0.1259] \\
&= \$909 \quad \text{ANSWER}
\end{aligned}$$

The two approaches give the same result for pump B for 6-year and 12-year analyses as long as there is identical replacement of the pump at the end of each 6-year period. If that assumption is not valid, it is necessary to use least common multiple lives or lives that are coterminous with appropriate recognition of respective salvage values.

Future-Worth Comparison

Future-worth comparisons require that a future date be selected for alternatives to be terminated or compared on and the cash flows projected into future dollars of value, giving due regard for intermediate replacement costs, salvage values, and periods of useful life. The usual discounting factors can be used.

Rate-of-Return Comparison

The rate-of-return (ROR) method of economic comparison or evaluation calculates the interest rate on unrecovered investment such that the payment schedule (schedule of return or project analysis period) makes the unrecovered investment equal to zero at the end of the useful life of the investment. In public investment economics the rate of return refers to internal rate of return. Also, the rate of return may refer to rate of return over investment:

$$\frac{\text{Annual net (Benefits - Costs)}}{\text{Investment}}$$

This is often used by public service commissions to set permitted rates. *ROR is the interest rate that makes the sums of the present worths of expenditures and payments equal to zero, or* $\Sigma PW = 0$, *and it is the interest rate at which benefits equal costs or the net benefits equal zero.* The method requires a trial-and-error approach. The computations may be carried out in either present-worth configuration or in the annual equivalent value configuration. Figure 12.5 gives a graphic representation of the significance of rate of return when utilizing a present-worth type of computation.

In a publication of the U.S. Department of the Interior, Glenn and Barbour (1970) give an excellent example of how to apply the ROR method. A modified example of Glenn and Barbour's work is presented in Example 12.3 to show how to utilize ROR, applying it with the annual equivalent type of calculation.

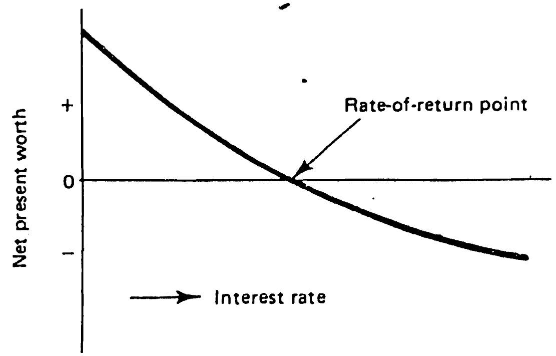

Figure 12.5 Graphic representation of rate-of-return.

Example 12.3

Given: Water project with:

Construction costs	= \$160,000,000
Interest during construction (IDC) = construction cost × construction period (4 yr) divided by 2, and multiplied by the interest rate, i:	
$\$160{,}000{,}000\,\dfrac{4i}{2} =$	= 320,000,000i
Annual operation, maintenance, and replacement (OM&R) cost	= 1,000,000
Annual benefits	= 15,000,000

Required: Determine the rate of return, ROR, if project life is considered 100 years, $n = 100$.

Analysis and solution:

First trial at $i = 0.05$:

Construction cost	= \$160,000,000
IDC: 320,000,000 × 0.05	= 16,000,000
Total investment cost (TIC)	= 176,000,000
Annual equivalent investment cost (AEIC) = TIC × $(A/P, i, n)$ = 176,000,000(0.0504)	= 8,870,000
Annual OM&R cost	= 1,000,000
Total annual cost	= 9,870,000
Annual benefits	= 15,000,000
Net benefits	= \$ 5,130,000

Because the computed benefits are greater than the costs, the interest rate 0.05 is not the correct rate of return for the investment, so that the i must be increased.

Second trial at $i = 0.07$:

Construction cost	= \$160,000,000
IDC: 320,000,000 × 0.07	= 22,400,000
TIC	= 182,400,000
AEIC = TIC × $(A/P, i, n)$ = 182,400,000(0.0701)	= 12,786,000

Annual OM&R cost	= 1,000,000
Total annual cost	= 13,786,000
Annual benefits	= 15,000,000
Net benefits	= \$ 1,214,000

Benefits still exceed cost, so the ROR is greater than $i = 0.07$.

Third trial at $i = 0.08$:

Construction cost	= \$160,000,000
IDC: 320,000,000i	= 25,600,000
TIC	= 185,600,000
AEIC = TIC$(A/P, i, n)$	= 14,848,000
Annual OM&R cost	= 1,000,000
Total annual cost	= 15,848,000
Annual benefits	= 15,000,000
Net benefits	= \$ −848,000

Benefits are now less than costs. Therefore, the ROR has been bracketed between $i = 0.07$ and $i = 0.08$. Linear interpolation gives the following:

$$\begin{aligned} \text{At } i = 0.07 \text{ net benefits} &= +1{,}214{,}000 \\ \text{At } i = 0.08 \text{ net benefits} &= \underline{-(-848{,}000)} \\ & \;\; 2{,}062{,}000 \end{aligned}$$

$$\frac{1{,}214{,}000}{2{,}062{,}000} = 0.59$$

so that

$$\text{ROR} = 7.59\% \quad \text{ANSWER}$$

Some economists claim that the ROR analysis has the advantage of not having to preselect an interest rate for discounting purposes. However, in actual practice most companies, industries, or even government agencies have a limit below which they will not continue to invest in development. This limiting interest rate is often called the "minimum attractive rate of return," MARR, and represents the lowest rate the decision-making entity will accept for expending investment capital. Care should always be exercised when making ROR comparisons to see that the analysis periods are compatible. Occasionally, there can be multiple roots of the ROR, where the net positive worth changes more than once due to special characteristics of the cost and benefits. The ROR comparison method is good for making comparisons having dissimilar alternatives and can help in ranking alternatives without preselecting a MARR.

Newnan (1976) points out that incremental analysis of the change in rate of return for different alternatives is a good practice. The steps pointed out by Newnan are to first compute the rate of return for the alternatives and normally reject those alternatives where the ROR is less than the agreed MARR. Then rank the remaining alternatives in order of increasing present worth of their costs. Next, compare by two-alternative analysis the two lowest-cost alternatives by computing the incremental rate of return (ΔROR) on the cash flow representing the differences between the alternatives. If the ΔROR ⩾ MARR, retain the higher-cost alternative. If the ΔROR < MARR, retain the lower-cost alternative and reject the higher-cost alternative. Proceed systematically through the entire group of ranked alternatives on a challenger–defender basis and select the best of the multiple alternatives. This is often referred to as the *challenger–defender concept* of incremental analysis. Excellent simple examples of how to apply this methodology are given in Newnan (1976).

Net Benefit Comparison

The benefit-cost analysis has already been mentioned in discussing elements of the other methods. The computation for net benefit analysis can be in either the present-worth configuration or the annual-equivalent-value configuration. Two

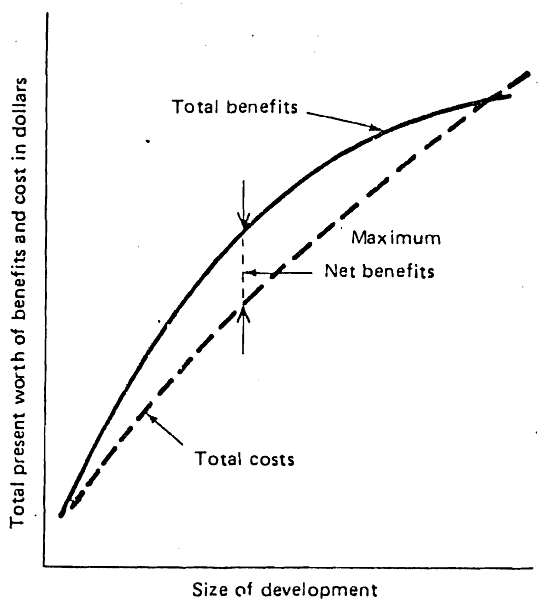

Figure 12.6 Graphic representation of net benefit.

graphical ways of representing the present-worth of benefits are available for expressing the net benefits. The first case is to use two curves plotting on a common axis the total present-worth of costs and benefits against a scale of size or alternatives for development. The technique lends itself to an analysis for determining the best size of projects. Figure 12.6 is a graphic representation of net benefit wherein total benefits and costs on a present-worth basis are plotted against the scale of the projected development in increments of possible sizes. The vertical distance between the curves represents the net benefit. The slope of the benefit curve is known as the marginal benefit and the slope of the cost curve is the marginal cost. When the two curves have the same slope, or marginal benefit equals marginal cost, the maximum net benefit is reached. This is normally the optimum size or scale to develop the project being analyzed. Under private investment policy the choice may be made to develop to a different scale based on some expected changes in economy, taxing policy, or inflation trends. In any case, net benefit can be used to compare different sizes of projects or alternatives.

Another way of expressing net benefits is to plot the present-worth of benefits against the present-worth of costs for different scales of development or different alternatives. Figure 12.7 shows graphically the significance of such a benefit-cost analysis. Note that the 45° line represents the point where net present worth is zero and the marginal acceptable rate of return, MARR, is the i value that is used in

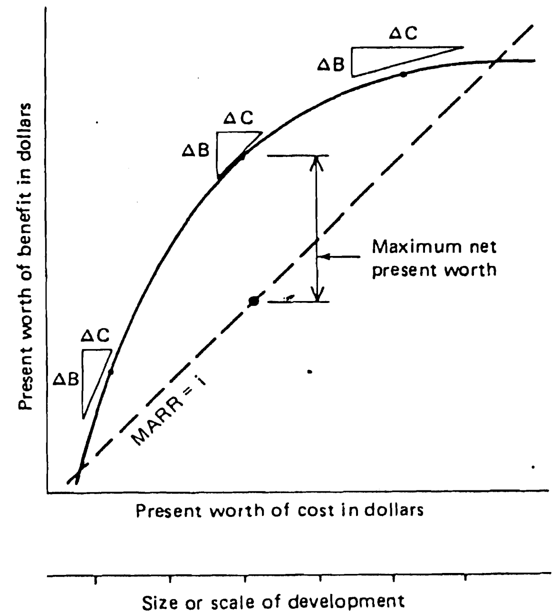

Figure 12.7 Graphic representation of benefits versus costs for varying size of development.

the discounting of the benefits and costs. For the example illustrated, above the point of maximum net present worth the unit return (benefit) from an increase in size of development is less than the unit expenditure or unit cost for that increase in size.

To apply this type of analysis and comparison requires that the discount rate be defined. Often defining the discount rate is a problem under various restraints that might be in force during planning and actual development. A frequent problem is also encountered as to what to treat as appropriate costs and what values to assign to the benefits. Howe (1971) gives an excellent discussion of problems of choice of discount rate, appropriate identification of costs and benefits, and suitable planning period or project life.

Benefit-Cost Ratio Comparison

Utilization of the benefit-cost ratio method of comparison may also be done in the present worth configuration or the annual-equivalent-value configuration. The comparison is made utilizing the relation

$$\text{Benefit-cost ratio, } \frac{B}{C} = \frac{\text{PW of benefits}}{\text{PW of costs}}$$

$$= \frac{\text{EUAB}}{\text{EUAC}} \geqslant 1 \tag{12.13}$$

Newnan (1976) indicates that when comparisons are made for developments where the inputs and outputs are not fixed, an incremental analysis should be made by pairs where the incremental benefit-cost ratio, $\Delta B/\Delta C$, is computed on the difference between the alternatives. If the $\Delta B/\Delta C \geqslant 1$, choose the higher-cost alternative; otherwise, choose the lower-cost alternative. The usual criteria for justifying economic feasibility are that the benefit-cost ratio is equal to or greater than 1. Sometimes separable components of development are required to meet a benefit-cost ratio of 1 or greater. However, sometimes social conditions may dictate criteria that will permit a ratio less than 1 and an overall public point of view may dictate a consideration that has not been accounted for in the economic analysis. This might involve political decisions that desire a given feature of benefit even if it is not profitable, so in order to accomplish some socially acceptable goal a particular feature may be included that is not cost effective. Environmental considerations have become important and some of those considerations are not easily quantified in economic terms. Appropriate for evaluation of environmental alternatives is an accounting of benefits and costs in some form of quantification.

OTHER ECONOMIC CONSIDERATIONS

In applying the foregoing methods of comparison, there are further considerations that must be made and evaluated on the cost side of the analysis. These include cost

of money, depreciation and amortization, interim replacement, insurance, and taxes. On the benefit or value side of power economics, consideration must be given to the capacity value of the power and the simple energy value. If there is a differential in the inflation rates of the components of costs or benefits, the inflation effect will need to be treated.

Cost of Money

Various rates of interest are applied in determining the cost of money for different entities developing a project. One rate might be the interest rate that the government uses in economic analysis, which for water development projects is specified by the U.S. Water Resources Council (1973) and is an interest rate of long-time borrowing of the Federal Treasury as limited by administrative decision. In private financing, and thus for analyses of private developments by investor-owned utilities, it is an interest rate of long-term debt, preferred stock, and common equity of the company. The U.S. Department of Energy (1979b) reports for 1979 that this cost of raising new money was approximately $i = 0.105$. For municipalities, irrigation districts, and rural electric cooperatives, this rate may be different and usually of lesser magnitude because of the tax-exempt nature of their funding. This cost of money, interest rate, must be used in discounting investment costs.

Amortization or Depreciation

Amortization or depreciation provides for future financing and the usual practice in power economics is to apply sinking-fund depreciation plus the cost of money rate to the total investment in deriving the annual cost of capital recovery. Most utilities compute annual depreciation with straight-line depreciation accounting principles. In any one year, the cost of capital recovery for a particular plant would include the annual depreciation cost plus the cost of money rate applied to the net depreciation investment.

Interim Replacement

The interim replacement accounts provide money for the components of the hydroplant that need to be replaced during the project life and should be discounted appropriately. Sometimes this item is included in the operations, maintenance, and replacement annual charge. If the overall economic life of a project properly reflects the weighted service life of each of the components, an interim replacement allowance would not be required.

Insurance

A necessary annual cost is the cost of insurance to protect the utility against losses and damage. This varies between 0.001 and 0.002 times the capital investment in normal cases for hydropower developments. This item is usually higher for other types of power plants such as fossil fuel steam plants and nuclear power plants.

Taxes

Investor-owned utilities annually pay a variety of taxes, including federal income tax, property tax, state income tax, electric power tax, corporate license tax, and sales tax. An excellent discussion explaining different taxes is included in a report by Bennett and Briscoe (1980). The total tax percentage will vary from state to state and normally ranges from 3 to 5% of investment costs on an annual basis.

Table 12.1 gives an example of these additional economic considerations for hydropower plants. Table 12.1 includes annual operation and maintenance costs.

TABLE 12.1 Example of Annual Costs Considered for a Privately Financed Hydropower Plant (Average Annual Plant Factor of 55%)

			Dollars per Net Kilowatt
(A) Plant investment, excluding step-up substation			\$1000.00
(B) Annual capacity cost			
I. Fixed charges		Percent	
a. Cost of money		10.50	
b. Depreciation (10.50%, 50-year sinking fund)		0.07	
c. Insurance		0.10	
d. Taxes	Percent	5.00	
1. Federal income	2.25		
2. Federal miscellaneous	0.10		
3. State and local	2.65		
Total fixed charges		15.67	156.70
II. Fixed operating costs			
a. Operation and maintenance[a]			2.50
b. Administrative and general expense (35% of \$2.50/kW-yr)			0.88
Total fixed operating costs			3.38
Total annual capacity cost (B - I) + (B - II)			160.08
			Mills per net kWh
(C) Energy–variable operating costs			
a. Energy fuel			0.00
b. Operation and maintenance[b]			0.00
Energy cost–total variable operating costs			0.00

[a]Based on an estimated fixed operating and maintenance cost of 0.5 mill/kWh.

[b]All operating and maintenance costs considered to be fixed.

SOURCE: U.S. Department of Energy (1979b).

Capacity Value and Energy Value Adjustment

The dependable capacity of an electric grid system is changed as new plants are added. The magnitude of this change depends on the degree of coordination between plants in a system, maintenance schedules, the general reliability of different modes of electrical production, and the relative sizes of the plants. Added value for hydroplants is warranted because of the rugged nature of the plants and the fast loading characteristics to give an annual capacity value credit. Usually, the credit value per kilowatt of capacity will range from 5 to 10% of the cost of thermal-electric capacity, but will vary from region to region.

The energy value of a hydroelectric plant introduced into a system having thermal-electric generation may change the average cost of energy, or introduction of new thermal-electric capacity can change the average cost. The FERC "Hydroelectric Power Evaluation" report (U.S. Department of Energy, 1979b) indicates that an energy value can be obtained by making simulation studies of the operating system that is affected. Such studies involve making detailed comparative analyses of annual system production expenses, first with the hydroelectric project and then with an alternative electric capacity. Due consideration would be given for the variable cost of fuel.

The difference between the total system costs with the hydroelectric project and the total system with the most likely thermal-electric alternative, divided by the average annual energy output of the hydroelectric project, gives an adjusted energy value for the particular year considered. Successive evaluation of ensuing years, and the use of present-worth procedures, should be used to determine the equivalent levelized energy value applicable over the economic life of the hydropower plant.

The FERC "Hydroelectric Power Evaluation" report gives an approximate method suitable for computing the capacity value and energy value adjustment for any one year by the following formulas:

$$E_n = \frac{\mathrm{P.F.}_{\cdot t} - \mathrm{P.F.}_{\cdot h}}{\mathrm{P.F.}_{\cdot h}}(\Delta C) \tag{12.14}$$

and

$$\mathrm{CP}_n = (\mathrm{P.F.}_{\cdot t} - \mathrm{P.F.}_{\cdot h})(\Delta C)(8.76) \tag{12.15}$$

where E_n = energy value adjustment for year n, mills/kWh of hydroelectric generation

CP_n = capacity value adjustment for year n, dollars per kilowatt-year of dependable production

$\mathrm{P.F.}_{\cdot t}$ = plant factor of the alternative thermal-electric plant

$\mathrm{P.F.}_{\cdot h}$ = plant factor of the hydroelectric plant

$\Delta C = \mathrm{EC}_t - \mathrm{EC}_d$, energy cost (mills/kWh) of thermal electric alternative (EC_t) minus the average energy costs of those plants which the thermal-electric alternative might reasonably be expected to displace (EC_d).

TABLE 12.2 Example of the Computation of Energy and Capacity Value Adjustments

Year of Analysis	Thermal Plant Factor, P.F.$_t$ (%)	Hydro Plant Factor, P.F.$_h$ (%)	Cost, C (mills/kWh)	Adjustment[a] Energy Value, E_n (mills/kWh)	Capacity Value, CP_n ($/kW)	Present-Worth Factor (at 10%)	Present Worth Energy, E_n (mills/kWh)	Capacity, CP_n ($/kW)
0–5	65	20	4.00	9.00	15.77	3.791	34.119	59.784
6–10	60	20	3.30	6.60	11.56	2.353	15.530	27.201
11–15	50	20	2.60	3.90	6.83	1.462	5.702	9.985
16–20	40	20	1.90	1.90	3.33	0.908	1.725	3.024
21–25	25	20	1.20	0.30	0.53	0.565	0.169	0.298
26–30	10		0.50	–0.25	–0.44	0.350	–0.088	–0.154
							57.157	100.138

[a]Average annual equivalent adjustment, interest at 10%, 30-year period:

Energy value adjustment = 57.157 (capital recovery factor, 10%, 30 years) = 57.157 × 0.10608 = 6.06 mills/kWh

Capacity value adjustment = 100.138 (capital recovery factor, 10%, 30 years) = 100.138 × 0.10608 = $10.62/kW

SOURCE: U.S. Department of Energy (1979b).

Table 12.2 gives an example of how the computation can be made over time for the project life.

Inflation

Inflation is the economic phenomenon where prices of goods and services increase with time. This general upward movement of prices affects both costs and benefits; unfortunately, the price increases are not uniform. The prices of different items in the group of goods and services involved in hydropower projects do not remain constant or change uniformly. Since value of electrical energy is often computed on the cost of alternative thermal-electric supplies, the fact that the cost of fuel is escalating faster than normal inflationary trends makes this accounting for inflation particularly important. An important issue is what interest rate is appropriate when adjusting for inflation. Similarly, in after-tax economic analyses of private-power investment, the impact of inflation or deflation cannot be eliminated by simply adjusting future before-tax benefits. If appropriate values for the cost of fuel reflect the energy cost of the alternative thermal-electric supply, then the energy value adjustment procedure mentioned in the discussion of Eq. (12.14) can account for inflation. Inflation benefits long-time borrowing that is characteristic of hydropower development because the debt is paid with dollars having reduced purchasing power.

Newnan (1976) gives an excellent treatment of how to handle the inflation effect, especially as it concerns after-tax calculations of the effect on income tax. Abramowitz (1977) has developed curves and multipliers for reflecting the influence of inflation based on differentials of value of alternative sources of thermal-electric power with escalating fuel prices that appear to be increasing faster than normal prices. A good source for information on price escalation for hydropower development parameters is *Construction Cost Trends* of the U.S. Department of Interior, Bureau of Reclamation. This periodical is published semiannually.

Table 12.3 shows the effect of including inflation in an example of a net present worth evaluation of the stream of costs and benefits for a hydropower project. In this particular example the effects of income tax are excluded.

Allocation of Costs

The foregoing discussion on costs and economic analysis has assumed that the only benefits are production of power. It is rare that for a hydropower development there are not other purposes served or benefits derived. There are often water supply benefits, flood control benefits, recreation benefits, and other benefits. When a multipurpose project is involved some procedure must be followed to allocate the portion of the costs pertaining to power to the separable purpose of power. It is beyond the scope of this book to treat that facet of cost accounting and economic analysis. Methodology for this is treated in a manual of the U.S. Army Corps of Engineers (1958). A method commonly used in the past is known as

TABLE 12.3 Example of the Inflation Effect on the Net Present Value of a Hydropower Project

(1) Year	(2) Capital Costs	(3) Other Costs	(4) Benefits	(5) Net Annual Benefits, (4) - (2) - (3)	(6) Present-Value Factor	(7) Present Value, (5) X (6)
		0.0% Price Escalation, 10.0% Interest				
0	$600,000			$-600,000	1.000	$-600,000
1	900,000			-900,000	0.909	-818,181
2		45,000	245,000	200,000	0.826	165,289
3		45,000	245,000	200,000	0.751	150,262
4		45,000	245,000	200,000	0.683	136,602
5		45,000	245,000	200,000	0.620	124,184
6		45,000	245,000	200,000	0.564	112,894
7		45,000	245,000	200,000	0.513	102,631
8		45,000	245,000	200,000	0.466	93,301
9		45,000	245,000	200,000	0.424	84,819
10		45,000	245,000	200,000	0.385	77,108
11		45,000	245,000	200,000	0.350	70,098
12		45,000	245,000	200,000	0.318	63,726
13		45,000	245,000	200,000	0.289	57,932
14		45,000	245,000	200,000	0.263	52,666
				Net present value of project =		$-126,662
		7.0% Price Escalation, 10.0% Interest				
0	$600,000			$-600,000	1.000	$-600,000
1	963,000			-963,000	0.909	-875,454
2		51,520	280,500	228,980	0.826	189,239
3		55,126	300,135	245,008	0.751	184,078
4		58,985	321,145	262,159	0.683	179,058
5		63,114	343,625	280,510	0.620	174,174
6		67,532	367,678	300,146	0.564	169,424
7		72,260	393,416	321,156	0.513	164,803
8		77,318	420,955	343,637	0.466	160,309
9		82,730	450,422	367,691	0.424	155,937
10		88,521	481.952	393,430	0.385	151,684
11		94,718	515,688	420,970	0.350	147,547
12		101,348	551,786	450,438	0.318	143,523
13		108,443	590,412	481,969	0.289	139,609
14		116,034	631,740	515,706	0.263	135,801
				Net present value of project =		$619,738

SOURCE: U.S. Army Corps of Engineers (1979a).

the separable-costs-remaining-benefits method. In that method each project purpose is assigned its separable cost plus a share of joint costs proportionate to the remainders found by deducting the separable costs of each purpose from the justifiable expenditure for that purpose. A good reference on evaluation of benefits and costs of other water resource functions is a text by James and Lee (1971).

Even though the U.S. Water Resources Council and its procedures for water resource analysis have been discontinued, the methodology presented has value, especially for government-sponsored hydropower projects, in the analyses of the water power benefits. The procedures are presented in the *Federal Register* as "Principles and Standards for Planning Water Resources and Related Land Resources" (U.S. Water Resources Council, 1973). Subsequent government regulations for current procedures should be referrred to when making new analyses involving hydropower planning for government-sponsored projects.

COST ESTIMATION

The economic analysis of project studies is dependent on orderly and accurate cost estimation. The type of study, whether a reconnaissance study, a feasibility study, or a final design study, will tend to dictate the precision with which cost estimates are made. A good pattern to follow is the Uniform System of Accounts taken from the FERC "Hydroelectric Power Evaluation" report (U.S. Department of Energy, 1979b).

Hydroelectric Plant Accounts

The Federal Energy Regulatory Commission's Uniform System of Accounts, prescribed for public utilities and licensees, includes the following electric plant accounts relating to hydroelectric developments:

330. *Land and land rights.* Includes the cost of land and land rights used in connection with hydraulic power generation.

331. *Structures and improvements.* Includes the in-place cost of structures and improvements used in connection with hydraulic power generation.

332. *Reservoirs, dams, and waterways.* Includes the in-place cost of facilities used for impounding, collecting, storing, diverting, regulating, and delivering water used primarily for generating electricity.

333. *Water wheels, turbines, and generators.* Includes the installed cost of water wheels and hydraulic turbines (from connection with the penstock or flume to the tailrace) and generators driven by them to produce electricity by water power. In the case of pumped/storage projects, this account includes the cost of pump/turbines and motor/generators.

334. *Accessory electric equipment.* Includes the installed cost of auxiliary generating apparatus, conversion equipment and equipment used primarily in con-

nection with the control and switching of electric energy produced by hydraulic power and the protection of electric circuits and equipment.

335. *Miscellaneous power plant equipment.* Includes the installed cost of miscellaneous equipment in and about the hydroelectric generating plant which is devoted to general station use.

336. *Roads, railroads, and bridges.* Includes the cost of roads, railroads, trails, bridges, and trestles used primarily as production facilities. It also includes those roads, etc., necessary to connect the plant with highway transportation systems, except when such roads are dedicated to public use and maintained by public authorities.

Functionally, the foregoing accounts may be combined into the following major categories:

Power plant:—Accounts 331, 333, 334, and 335.

Reservoirs and dams:—Account 332, clearing, dams, dikes, and embankments.

Waterways:—Account 332, intakes, racks, screens, intake channel, pressure tunnels, penstocks tailrace, and surge chambers.

Site:—Account 330 and 336.

Transmission Plant Accounts

In addition to the hydroelectric plant accounts, the following transmission plant accounts may need to be included in cost estimates of hydroelectric power developments:

350. *Land and land rights.* Includes the cost of land and land rights used in connection with transmission operations.

352. *Structures and improvements.* Includes the in-place cost of structures and improvements used in connection with transmission operations.

353. *Station equipment.* Includes the installed cost of transforming, conversion, and switching equipment used for the purpose of changing the characteristics of electricity in connection with its transmission or for controlling transmission circuits.

354. *Towers and fixtures.* Includes the installed cost of towers and appurtenant fixtures used for supporting overhead transmission conductors.

355. *Poles and fixtures.* Includes the installed cost of transmission line poles, wood, steel, concrete, or other material together with appurtenant fixtures used for supporting overhead transmission line conductors.

Table 12.4 is an example of a cost accounting for a completed project. A screening curve for overall costing of hydropower projects has been developed by FERC and is shown as Fig. 12.8. A problem with all cost-estimating curves is that they become obsolete due to price escalation and due to technological changes in design and manufacture of equipment and changes in construction techniques. Another problem is that it is frequently impossible to determine from published curves what items are included and what are not.

Many public and private entities have published cost estimating curves and

TABLE 12.4 Example of Costs for Power Planning

Line		Licensed Project 2146
1	Name of project	Lay dam redevelopment
2	Owner	Alabama Power Co.
3	State, river	Alabama, Coosa River
4	River	Coosa River
5	Installed capacity, kW (no. units)	177,000 (6-C)
6	Gross head, ft	83
7	Type of development	Storage-flowage; integral powerhouse
8	Type of dam	Earthfill
9	Construction period	11/4/65–5/14/68
10	*Cost of development* ($1000) FERC accts.	
11	Land and land rights (330)	6.161
12	Structures and improvements (331)	756
13	Reservoirs, dams, and waterways (332)	11.703
14	Equipment (333-4-5)	8,362
15	Roads, railroads, and bridges (336)	0
16	Total direct costs	$26,982
17	Total indirect costs	864
18	Subtotal	$27.846
19	Total overhead costs	4,807
20	Total project cost (hydraulic production)	$32.653
21	Cost per kW	$ 184 (12/31/68)

Line	Items of Work (FERC Accts)	Unit	Unit Price Actual[a]	Unit Price Adjusted	Quantity
22	*Reservoirs, dams, and waterways* (332)				
23	Reservoirs clearing and debris disposal	acre	273.82	726.00	6,159
24	Diversion	job	1,126,272.00	2,620,000.00	1
25	Dam	yd^3	20.95	46.90	338,024
26	Foundation excavation				
	Earth	yd^3	2.94	6.44	44,968
28	Rock	yd^3	14.69	32.20	13,147
29	Mass concrete	yd^3	20.21	44.50	297.866
30	Earthfill	yd^3	2.41	5.40	40,158
31	Spillway excavation	yd^3	NR		
32	Spillway structure	yd^3	NR		
33	Spillway gates, guides and hoists	lb	0.99	2.30	866,128
34	Power intake	job	539,959.00	1,340,000.00	1
43	Tailrace				
44	Excavation	yd^3	1,846.00[b]	4,170.00	1

TABLE 12.4 cont.

	Items of Work (FERC Accts)	Unit	Unit Price Actual[a]	Unit Price Adjusted	Quantity
45	Appurtenances	lb	0.20	0.50	73,024
47	*Power plant structures and improvements* (331)				
53	Superstructure	yd^3	5.88	13.00	109,234
54	Mass concrete	yd^3	NA		
55	Steel	lb	0.32	0.71	236,000
56	Station yard	job	44,248.00	95,100.00	1
57	Operators village	job	66,662.00	154,000.00	1
58	Recreational structures and improvements	job	NA		
59	*Waterwheels, turbines, and generators* (333)				
60	Turbines	hp	14.30	31.20	240,000
61	Generators	kW	17.87	39.00	177,000
62	*Accessory Electric equipment* (334)	kW	6.27	13.50	177,000
63	*Misc. power plant equipment* (335)	kW	1.34	2.88	177,000
64	*Total power plant structures and equipment* (331-4-5)	kW	51.51	118.00	177,000

[a]NR, not reported; NA, not applicable.
[b]Unit is a job.
SOURCE: U.S. Department of Energy (1979b).

equations for hydropower evaluations. Only a few of the most recent and most applicable are here mentioned.

Gordon and Penman (1979) recently published a series of empirical equations that are useful for quick estimation purposes. The costs are reported in 1978 U.S. dollars and require certain physical size information to estimate a second component of size or quantity. Typical of this type of calculation is an estimation for quantity of concrete for a powerhouse proposed by Gordon and Penman (1979). The equation and needed explanation are

$$V = KD_T^{2.5}(N + R) \tag{12.16}$$

where V = volume of concrete in power house structure, m^3
K = coefficient varying with head, varies from 80 to 250
D_T = turbine throat diameter, m
N = number of turbine units
R = repair bay factor; it varies from 0.3 to 1.2 depending on repair bay size.

For costs of power units, Gordon and Penman (1979) propose

$$C_T = 40{,}000(\text{kW}/H_R)^{0.53} \tag{12.17}$$

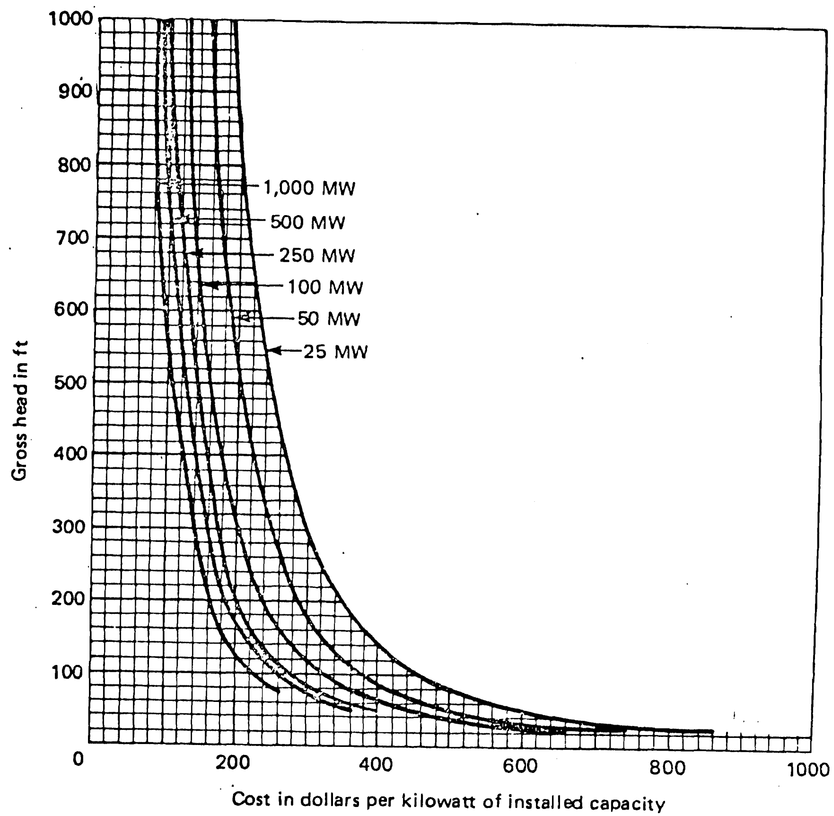

Figure 12.8 FERC screening guide for estimating the costs of conventional hydropower plants. SOURCE: FERC (1979).

where C_T = cost of a package turbine-generator unit and controls at factory, 1978 U.S. dollars
kW = turbine capacity, kW
H_R = valid head, m.

Equation (12.17) is based on Swedish experience. A formula for North American experience is

$$C_T = 9000(\text{kW})^{0.7}H_R^{-0.35} \tag{12.18}$$

For adding power to an existing dam, the following formula applies:

$$C_P = 9000S(\text{kW})^{0.7}H_R^{-0.35} \tag{12.19}$$

where C_P = total cost of project, excluding interest during construction and escalation, in 1978 U.S. dollars
S = site factor values

Values of S are as follows:

	Installed Capacity	
	Below 5000 kW	Above 5000 kW
No penstock	3.7	2.6
With penstock	5.5	5.1
New units in powerhouse	1.5	1.5

SOURCE: Gordon and Penman (1979).

For converting to annual cost the following formula applies:

$$C_A = 0.125 C_P \tag{12.20}$$

where C_A = cost per annum in 1978 U.S. dollars.

Energy costs for quick estimates based on head, discharge, and site factor are given by the following formula:

$$C_E = 122.6 S Q_M^{-0.30} H_R^{-0.65} \tag{12.21}$$

where C_E = cost of energy, U.S. mills/kWh

Q_M = mean discharge available to plant, m^3/sec.

A simplified cost curve based on statistics from 39 plants having capacities of less than 10 MW prepared by Imatra Voima OY of Finland gives an interesting means of estimating the cost of the mechanical and electrical equipment for hydropower plants as shown in Fig. 12.9. The costs are in 1978 U.S. dollars and the cost

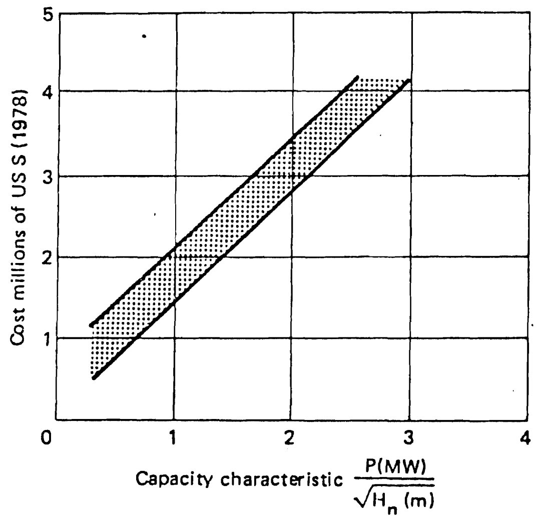

Figure 12.9 Cost estimating curve from Finnish experience.

is expressed as a function of capacity P in MW and the net head, H_n, in meters as a ratio $P/\sqrt{H_n}$.

Several recent publications and manuals primarily from government agencies have published various kinds of curves for making cost estimates of various features of hydroplants. Figure 12.10 is typical of cost curves available. Similar curves are presented in the U.S. Department of the Interior (1980) publication. Table 12.5 gives a summary of current publications, indicating the type of cost information that is available. These curves should be useful in making feasibility-level studies. In making design studies it is preferable to get cost estimates from manufacturers to ensure that escalations in prices are treated realistically and that technological advances have been included. Some manufacturers refer to a detailed analysis by Sheldon (1981) of the cost of purchasing and installing Kaplan and Francis turbines that includes escalation features.

Comparative Costs

Important in cost estimating and economic analysis is information on alternative costs of other modes of electrical energy production and the relative value of

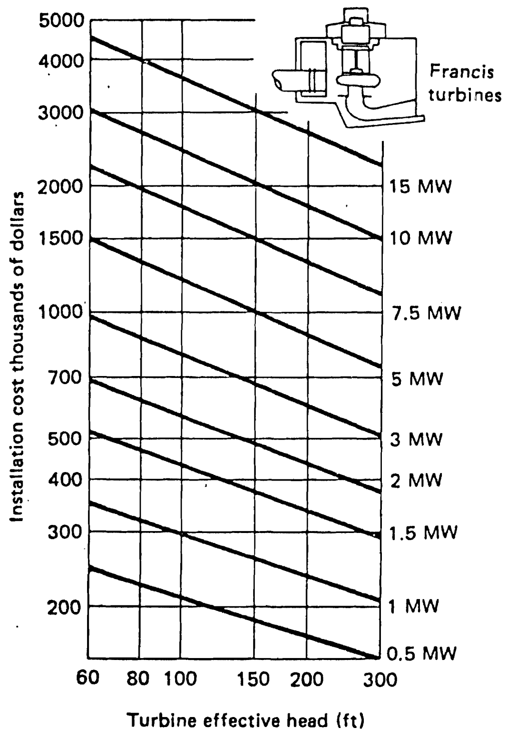

Figure 12.10 Sample cost estimating curves. SOURCE: U.S. Army Corps of Engineers.

TABLE 12.5 Summary of Useful Manuals and Publications for Estimating Costs for Hydropower Developments

1. Ver Planck, W. K., and W. W. Wayne, "Report on Turbogenerating Equipment for Low-Head Hydroelectric Developments, U.S. Department of Energy" Stone and Webster, Boston, 1978.
   Limited to cost of turbine equipment, station electrical equipment, and annual operating costs.
2. New York State Energy Research, "Site Owners' Manual for Small Scale Hydropower Developments," New York State Energy Resource and Development Authority, Albany, N.Y., 1979.
   Limited to general cost of equipment.
3. U.S. Army Corps of Engineers, "Hydropower Cost Estimating Manual," Portland District, U.S. Army Corps of Engineers, Portland, Oreg., 1979.
   Limited to turbine costs, station electrical costs, intakes and outlets, spillways, and headworks.
4. U.S. Army Corps of Engineers, "Feasibility Studies for Small Scale Hydropower Additions, A Guide Manual" U.S. Army Corps of Engineers Water Resource Institute, Ft. Belvoir, Va., 1979.
   Fairly complete coverage, except for penstock costs, dam costs, and spillway costs.
5. U.S. Department of the Interior, Bureau of Reclamation, "Reconnaissance Evaluation of Small, Low-Head Hydroelectric Installations," U.S.B.R., Denver, Colo., 1980.
   Fairly complete coverage of all components.
6. Electric Power Research Institute, "Simplified Methodology for Economic Screening of Potential Low-Head Small-Capacity Hydroelectric Sites," EM-1679, EPRI, Palo Alto, Calif., 1981.
   Fairly complete coverage of all components.

the various parameters influencing cost of electrical energy. Figure 12.11 shows a projection of energy costs with varying plant capacity factors and different types of thermal-electric plants, compared to expected cost of hydropower. This figure projects relative costs for the year 1985 based on escalation of prices at a uniform rate of 5% per annum and fuel costs escalated at 7% per annum. Another source of useful cost information is the annual publication of the U.S. Department of Energy entitled *Hydroelectric Plant Construction Cost and Annual Production Expenses* (issued annually). This gives actual data on costs of constructing plants and actual operation and maintenance costs to check against estimating curves. Also useful is a report by the U.S. Departments of Labor and Energy (1979). The FERC "Hydroelectric Power Evaluation" report (U.S. Department of Energy, 1979b) also gives example cost data for alternative sources.

Many agencies and most hydroelectric consulting firms now have computerized programs for making economic analyses. The FERC "Hydroelectric Power Evaluation" report gives detailed flow diagrams of the computer program and format details for the necessary computer cards. Broadus (1981) has published "Hydropower Computerized Reconnaissance (HCR) Package Version 2.0," which computes necessary economic analysis for reconnaissance level studies using data from the publications mentioned in Table 12.5.

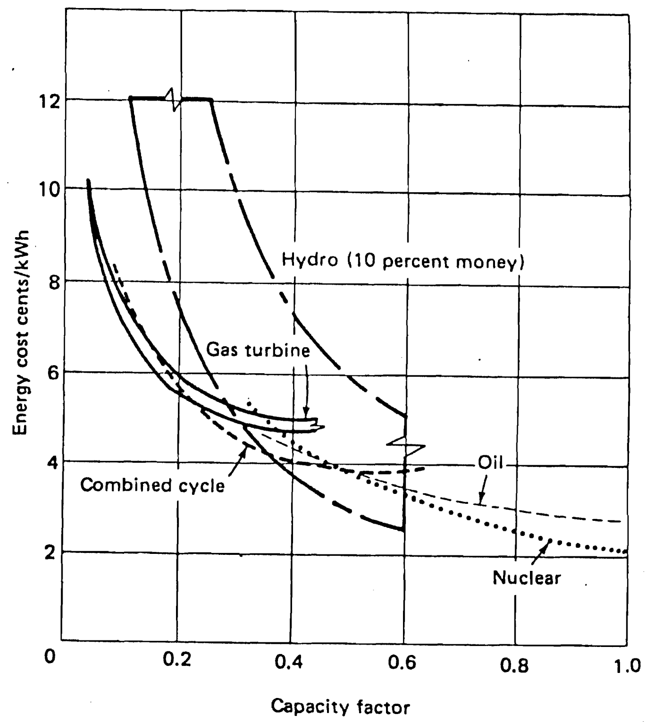

Figure 12.11 Comparison of energy cost for generation alternatives, 1985. SOURCE: Lawrence (1979).

APPLICATION OF ANALYSIS

The numerous formulas, steps in making analyses, and the different procedures used in hydropower economic analysis are best understood by proceeding through some example problems.

Example 12.4

Given:

Plant capacity	108,000 kW.
Capital cost of plant (PIC)	\$105,800,000
Construction period	5 yr
Project life	50 yr
Interest during construction	PIC $\times\ i \times \frac{5}{2}$
Cost of money, i	10.5%

Periodic replacement cost (PRC) replacement every 10 yr	4,100,000
Plant factor (from flow duration curve)	0.50
Taxes and insurance, % of capital investment	5.2%
Operation and maintenance (estimated),	$17,200(MW)^{0.543}$
Capacity value of power	0.1(kW)($30/kW)
Value of energy	35 mills/kWh

Required: Find annual net benefits assuming no difference in escalation of prices for various costs and the energy benefts.

Analysis and solution:

Project initial cost (PIC)	= $105,800,000
Interest during construction: PIC(i)5/2 = 105,800,000(0.105)5/2	= 27,772,500
Total initial project cost (TIPC)	= 133,572,500

Annual equivalent cost:

Annual fixed cost of investment =
(TIPC)(A/P, i, n)
$i = 0.105$, $n = 50$

AFCI = 133,572,500(0.1057)	= 14,718,613
Annual replacement cost (ARC) ARC = PRC(A/F, i, n) = 4,100,000(0.612) $i = 0.105$, $n = 10$	= 250,920
Annual taxes and insurance (ATI) ATI = PIC(0.052) = 105,800,000(0.052)	= 5,501,600
Annual operation and maintenance cost (AOMC) AOMC = $17{,}200(\text{MW})^{0.543}$ = $17{,}200(108)^{0.543}$	= 218,614
Total annual variable costs	$ 5,971,134
Total annual equivalent costs (TAEC)	$20,689,747

Annual equivalent benefits:

Annual capacity benefit (ACB) ACB = 0.10(capacity in kW)(30) = 0.10(108,000)(30)	= 324,000
Annual energy value (AEV) AEV = (kW)(P.F.)(8760)(0.035) = (108,000)(0.5)(8760)(0.035)	= 16,556,400
Total annual equivalent benefits (TAEB)	= $16,880,400
Annual net benefits TAEB − TAEC = $16,880,400 − $20,687,747	= $-3,809,347 ANSWER

This indicates that the net annual benefits are negative for the cost of money interest used, with no accounting for the likely escalation of power benefits faster than escalation of variable costs. As exercises, Problems 12.3 and 12.4 are proposed for extension of this example.

An example from the studies of Goodman and Brown (1979) uses a similar approach but shows what happens when an accounting is made for the expected difference in price escalation of the value of energy and the annual variables costs involved in production: namely, costs of replacements, costs of taxes, and operation and maintenance costs.

Example 12.5

Given:

Installed capacity (IC)	1,500 kW
Dependable capacity = 0.10 IC	150/kW
Unit cost of construction	$800/kW
Construction cost	$1,200,000
Completed project cost	$1,411,100
Plant factor	0.62
Annual output	$8,146,800 kWh

Required:

(a) Find the net value of power if an average value of energy is assumed to be $0.05/kWh, the capacity value is $30/kW-yr. The multiplier for the capital recovery factor by a public entity is to be 0.125 times the complete project cost to obtain the total annual costs.

(b) Find the life cycle returns if the annual variable costs are 0.02 times completed fixed cost and these costs are assumed to escalate at 7% but the value of power begins at 0.20 mills/kWh and escalates at 8.5%; assume a 20-year payout period and a capital recovery factor of 0.105 to include amortization and depreciation.

Analysis and discussion:

(a) First determine the total annual cost of the project (TAC):

TAC = 12.5% of completed capital cost
= 0.125(1,411,100) = $176,389

Second determine value of capacity and energy:

Capacity value dependable capacity × 30/kW-yr

= 150 × 30 = \$4,500

Energy value = 0.05(1500)(0.62)(8760) = \$407,340

Total annual benefits = \$411,840

Therefore net benefit = \$411,840 − \$176,389 = \$235,451/yr

NAB/total kWh = net value added per kwH

$$= \frac{235{,}451}{8{,}146{,}800} = 0.029/kWh \quad \text{ANSWER}$$

(b) For the second part a computational table, Table 12.6, is presented based on fixed cost equal to completed capital cost times capital recovery factor of 0.105.

TABLE 12.6 Computational Table for Example 12.5 (Life Cycle Economic Evaluation)

Year	Fixed Costs[a] (\$)	Variable Costs[b] (\$)	Total Costs (\$)	Value per kWh[c] (\$)	Benefits (\$)	Net Benefits (\$)	Present Value-Factor[d]	Present Value (\$)
1	148,167	28,222	176,389	0.020	162,936	(13,453)	0.935	(12,579)
2	148,167	32,312	180,479	0.023	187,376	6,897	0.873	6,021
3	148,167	34,573	182,740	0.025	203,670	20,930	0.816	17,079
4	148,167	36,994	185,161	0.027	219,964	34,803	0.762	26,520
5	148,167	39,583	187,750	0.030	244,404	56,654	0.712	40,338
6	148,167	42,354	190,521	0.032	260,698	70,177	0.666	46,738
7	148,167	45,317	193,484	0.035	285,138	91,654	0.623	57,100
8	148,167	48,491	196,658	0.038	309,578	112,920	0.582	65,719
9	148,167	51,885	200,052	0.041	334,019	133,967	0.544	72,878
10	148,167	55,517	203,684	0.045	366,606	162,922	0.508	82,764
11	148,167	59,404	207,571	0.049	399,193	191,622	0.475	91,020
12	148,167	63,562	211,729	0.053	431,780	220,051	0.444	97,707
13	148,167	68,011	216,178	0.057	464,368	248,190	0.414	102,751
14	148,167	71,772	220,939	0.062	505,101	284,162	0.388	110,255
15	148,167	77,865	226,033	0.067	545,835	319,802	0.362	115,768
16	148,167	83,317	231,484	0.073	594,716	363,232	0.338	122,772
17	148,167	89,149	237,316	0.080	651,744	414,428	0.316	130,959
18	148,167	95,389	243,556	0.086	700,624	457,068	0.295	134,835
19	148,167	102,066	250,233	0.094	765,799	515,566	0.276	142,296
20	148,167	109,211	257,378	0.102	830,973	573,595	0.258	147,988
							Net present value =	\$1,598,925

[a]Fixed costs equal 0.105 × 1,411,100.

[b]Variable costs equal 0.02 × 1,411,100; these costs are escalated at 7%.

[c]These values have been levelized. The value of energy is escalated at 8.5%.

[d]The present-value factors are generated with a discount rate of 7%.

SOURCE: Goodman and Brown (1979).

Variable cost is escalated by 7% by using the first year's variable cost = 0.02 times initial capital cost times $(1.07)^n$, where n is the end-of-year time. The benefits are calculated by the following formula:

$$\text{Benefit year } n = 0.02(\text{kWh})(1.085)^n$$

Net benefits then equal the difference between annual benefit and annual cost for a given year. The present value of the net benefit is calculated by multiplying each net benefit by an appropriate discounting factor $(F/A, i, n)$ from Eq. (12.6), where $i = 0.07$ and n is the appropriate year from 1 to 20. This gives a net present value of benefits of \$1,598,925.

FINANCIAL CONSIDERATIONS

Government Financing and Development

Financial analysis or evaluation has to do with the manner in which the funding is obtained and repaid. In the case of federal financing it is usual practice to obtain funding from a given year's congressional appropriations. The method of payment for the expenditures is worked out on long-term contracts. This is the manner of financing for the extensive federal developments in the Pacific Northwest United States. Such entities as Bonneville Power Administration are responsible for developing the power sales agreements that repay the federal expenditures.

There is a definite difference between the accounting for financial analysis and the economic analysis that is done for feasibility studies. In financial analysis consideration must be given to the actual payment dates, contractual interest rates, and problems of balance of payments that must be considered in developing countries. A forthcoming book by A. S. Goodman (1983) treats this problem in a more specific manner.

Other Public Financing

Municipalities, irrigation districts, public power supply districts, and electric cooperatives normally get financing for hydropower projects by general obligation bonds and revenue bonds. The general obligation bond is secured by the taxing power of the entity issuing the bonds. If at any time the revenue of the hydro development is inadequate to cover the bond repayment and other annual costs, the public entity may take steps to correct the situation by increasing taxes to make up the deficiency. Because the general obligation is a legal obligation, it is normal to require an approval vote of the qualified electors before issuance of the bonds.

Revenue bonds, on the other hand, are usually restrictive in that their security for payment is the revenue from the hydroelectric project. Issuing revenue bonds usually requires voter approval but generally only a simple majority vote. As repayment of revenue bonds is not as well secured, the margin of safety required for

TABLE 12.7 Values of Financial Parameters for Different Types of Hydropower Financing, Southeast United States

Type of Financing[a]	Short-Term Interest Rate[b] (%)	Long-Term Interest Rate[c] (%)	Real Discount Rate[d] (%)	Debt Period (years)	Percent Debt Financing	Rate for Non-income-Related Taxes (% of initial project cost)[e]	Insurance Rate (% of initial project cost)	Real Return on Equity (%)	Payback Requirement[f] (years)	Break-even Requirement[g] (years)
Municipal	6.75	6.75	2.0	30	100	0.73	0.2	–	30	5
Electric cooperative	12.25	9.0	2.5	35	100	0.83	0.2	–	35	5
Investor-owned utility (IOU)	11.75	10.0	3.0	25	50	2.57	0.2	5.0	25	20
Private entrepreneur	12.75	11.0	4.0	25	80	2.57	0.2	5.0	25	3

[a]Municipal and electric cooperative data also apply to the municipal and cooperative bulk power suppliers (i.e., joint operating agencies).

[b]The short-term interest rate is set at the prime rate (July–August, 1980) for IOUs and a percentage point above that for private entrepreneurs. Municipalities are able to obtain a tax-exempt rate. These nominal interest rates are converted to real interest rates in order to calculate real interest on funds expended during construction.

[c]Rates are those currently available to each developer type. IOU and private interest rates are subject to greater fluctuation than the subsidized municipal and cooperative rates.

[d]Real discount is inflation-free discount.

[e]Initial cost of project = total capital cost + licensing cost + real interest during construction.

[f]In the private sector, 5 years is a commonly used payback constraint for investment in capital equipment, while 10 years is more appropriate to construction projects. The payback periods for a municipal developer and not-for-profit organizations such as electric cooperatives are not as binding: conceivably, a municipality or cooperative could accept a payback period as long as its debt period. Similarly, an IOU would be less concerned with recovering its investment quickly than having the investment included in its rate base. A payback period no longer than the utility's debt period–the time allowed for repayment of a loan or for bond maturity–is assumed to be acceptable.

[g]For the municipality, electric cooperative, and private entrepreneur a projected positive annual cash flow from the small scale hydropower project within the stated number of years is necessary in order to obtain financing. In addition, a municipality might find itself politically vulnerable if it invested in a SSH project expected to lose money for more than 5 years–roughly the term of office of municipal leadership. An electric cooperative's management would face similar pressure from its members. The IOU could tolerate much longer periods of theoretical negative cash flow than the private developer, municipality, or cooperative, since the costs of the SSH plant would be passed on to its customers. In other words, even if the SSH plant could not produce power as cheaply as an alternative investment available to the utility and thus theoretically was losing money, its costs would be covered by the utility's allowed revenue. The IOU would have to convince the state Utilities Commission that the project was a desirable one and that its higher cost per kWh would be offset by saving fossil fuels and avoiding a certain amount of environmental pollution.

SOURCE: Bennett and Briscoe (1980).

bond payment is higher. Municipalities may in some cases get loans or grants from either federal or state agencies to help in the financing. It is usually practice for public entities to issue tax-free bonds, that is, the interest received by the bond-holder is exempt from federal and state income tax. Electric cooperatives and public utility districts operate under these general rules of financing, but different rules as to operation, bonding limits, and tax exemptions apply in different states.

Investor-Owned Utility Financing

Investor-owned utilities, sometimes referred to as private utilities, are financed from bonds, preferred stock, or common stock. The mix of funding to be used for bonds, stocks, and short term loans will vary with corporate financial structure, debt rate, and financial conditions. Hence it is quite common to find that the interest rate used in financing of power projects is variable depending on the composite mix of outstanding bonds, stocks, and new financing. Financing for investor-owned utilities is usually at a higher interest rate than public utilities. Private entrepreneur developments are normally financed as corporations and follow the general rules of corporate financing.

Bennett and Briscoe (1980), in a study of economic restraints in North Carolina, South Carolina, and Georgia, prepared an interesting table showing the relative values of the financial parameters for different types of financing. Table 12.7 shows the results of that study of southeastern United States and the various terms. This may be useful as a guide, but specific application in different regions and with different entities will require professional advice from financial consultants and bonding specialists. There does appear to be an advantage to financing under public entity sponsorship, but other institutional restraints may make that advantage minor in consequence. The advantage may appear to be favorable to the local consumers under a particular project's economic analysis but not necessarily so in terms of national economic income accounting.

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PROBLEMS

12.1. An electric utility experienced a net return of $250,000 at the end of last year. If the forecast for sales indicate a growth in those net returns of $37,000 per year over the next 20 years, what will be the total present worth of the stream of net returns over the 20-year period if the discount rate is 9%? What will be the equivalent uniform annual value of that stream of net benefits?

12.2. Prepare a checklist for economic analysis of a hydro development, including a cost estimate form for capital costs and develop a flow diagram for a typical economic analysis.

12.3. For Example 12.4 determine when the net benefits will become positive if the variable costs are considered to escalate at a 7% annual rate and the energy benefits escalate at 8.5% annual rate due to the cost of thermal-energy alternatives' increasing fuel costs.

12.4. For Example 12.4 determine the rate of return that would make the development have total annual benefits equal to total annual costs.

12.5. A proposed site at Grimes Pass on the South Fork of the Payette River has available a full-gate capability of 16,919 kW. The estimated average annual production is 62,912,329 kWh. Following is a tabulation of the estimated hydropower plant costs.

Powerhouse	$ 809,000
Turbines and generators	3,425,000
Electrical powerhouse equipment	635,000
Gates, hoists, and miscellaneous	617,000
Subtotal	$ 5,486,000
Mobilization and access	300,000
Diversion of the river	350,000
Excavation	912,000
Concrete in place	3,735,000
Subtotal	$ 5,297,000

Contingencies on construction, 10%	630,000
Mobile crane, operations quarters, transmission lines, and common share costs of the four-dam complex	383,000
Total construction costs	$11,696,000
Engineering and administration, 12%	1,404,000
Total investment cost	$13,100,000
Interest during construction (15.6% of investment)	2,044,000
Total capital cost	$15,144,000

(a) What are the annual revenue requirements for such a site on a before-tax basis (dollars/yr)?
(b) What is the annual cost of generating energy (dollars/kwh)?
(c) If 75% of the investment cost must be obtained from the sale of 25-year serially maturing bonds yielding 7%, what is the maximum yearly cost of debt service?

State clearly any assumptions you make as you solve these problems.

PUMPED/STORAGE AND PUMP/TURBINES 13

BASIC CONCEPTS

Pumped/storage hydropower developments are energy-storing systems. Water is pumped from a lower reservoir to a higher one, utilizing low-cost "dump" power produced during periods of low demand by power plants which can be operated economically at a constant load. The water in the higher reservoir is then released through turbines to produce power needed during periods of peak demand. Although there is a net energy loss in the system because more energy is expended in pumping than can be produced by the turbines, the relative monetary value of "peak" power compared to "dump" power makes pumped/storage projects economically feasible. Frequently, peak power is worth at least three times as much as dump power.

Figure 13.1 schematically illustrates the basic concept of pumped/storage hydropower developments. Figure 13.2 is a graphic accounting of the energy storage cycle for a large pumped/storage plant in Europe. This shows relatively how much energy is used and produced in the pumped/storage operation. The way in which pumped/energy loads might occur is shown in Fig. 13.3 for a combined thermal-hydro energy-producing system. The excess capacity of hydro power in this case is being used for pumping power. Pumping is usually done with excess thermal power. The top portion of the curves represents the time when pumped/storage water is used for generating electricity to meet peak loads.

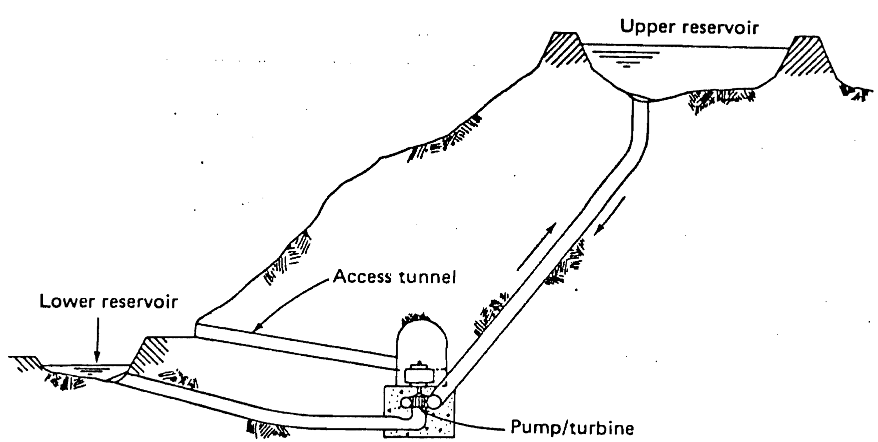


Figure 13.1 Schematic drawing of pumped/storage hydropower development.

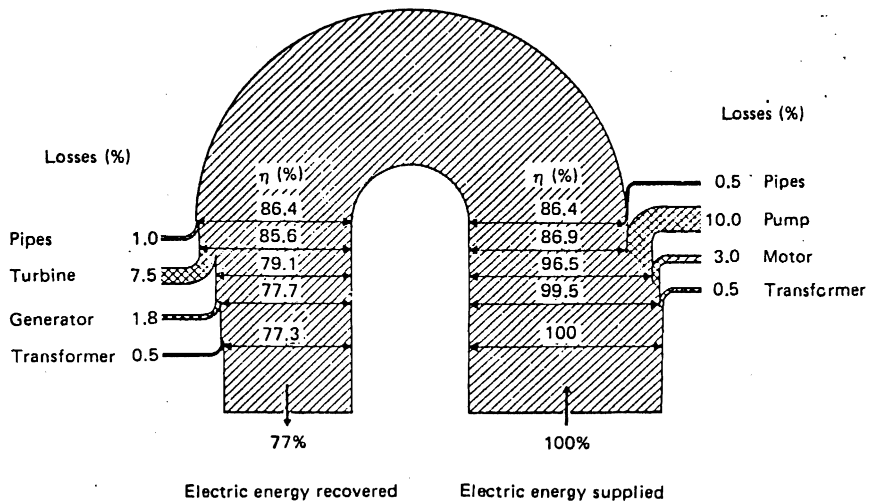


Figure 13.2 Graphic accounting of energy storage cycle for pumped/storage hydropower development. SOURCE: Escher Wyss.

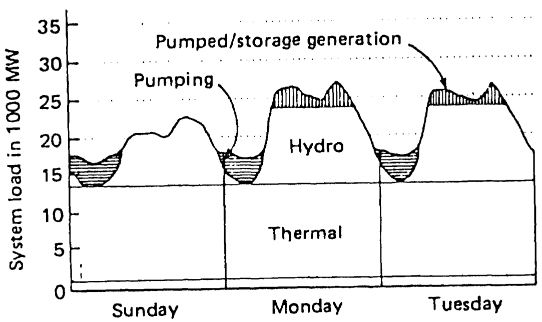


Figure 13.3 Graphic representation of energy load use with a pumped/storage power operation.

APPLICATION SITUATIONS

Three basic situations should be considered for planning purposes: (1) short-term peaking, (2) weekly peaking, and (3) seasonal peaking. Short-term peak operations would involve turbines operating for a few hours each day to meet daily peaks with the recovery pumping during the early morning hours when energy loads are lowest and cheap dump power is available. Weekly peaking operation requires operating turbines in generating mode to meet several peak loads during the workweek, refilling the upper reservoir to a degree each day but gradually drawing down the upper reservoir during the week and then major recovery pumping during the weekend when loads are low. This requires relatively large reservoirs. Weekly operation is shown graphically in Fig. 13.4. Air conditioning is now the peak demand in a

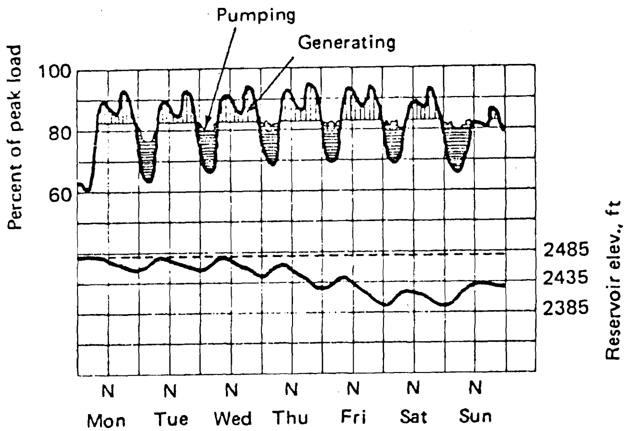


Figure 13.4 Weekly electric load variation and pumped/storage operation.

number of electric systems, so peaking is not necessarily tied to the workweek. Seasonal peaking would require very large upper reservoir storage to provide longer generator operation during seasonal peaks of energy demands such as summer air-conditioning loads. The recovery pumping would take large blocks of dump power.

ARRANGEMENT OF UNITS

Early pumped/storage systems had a separate pump and penstock system for the pumping portion of the operation and separate penstock, turbine, and draft tube for the generating portion of the operation. Later, reversible systems were designed.

Arrangements that have been used are

1. Complete pump and motor units and separate but complete turbine and generator units, involving four machines
2. Multistage pump and impulse turbine with a common motor/generator, involving three machines as shown in Fig. 13.5
3. Multistage pump and Francis turbine with a common motor/generator, involving three machines
4. Single-stage pump and Francis turbine with a common motor/generator involving three machines, as shown in Fig. 13.6 (this is a unique installation at the Waldeck II station built by Escher Wyss in Austria where the turbine draft tube is at the top of the line of equipment).
5. A single-stage pump/turbine and a common motor/generator involving two machines in which units have reversible directions for operation
6. A multistage pump/turbine

Experience in Europe has favored combinations involving three machines, whereas American practice has tended to favor two machine systems with units similar to Francis-type turbines as reversible units. The turbines and pumps can be mounted on vertical shafts or on horizontal shafts for smaller units.

A unique development in Europe reported by Mühlemann (1971) is the Isogyre pump/turbine, which consists of a double runner with pump impeller on the upper level of the shaft and the turbine runner on the lower level. The runners are back to back and fixed to a common shaft. The turbine runner is equipped with movable guide vanes and the pump has fixed guide vanes. The closing valves for both units are sleeve valves (cylinder gates) on the outside of the runner and the impeller, which isolate the runner and impeller chambers that are filled with air during the idling operation phase of either unit. A common spiral casing is used for distributing water to the runners. The runner direction of rotation is not reversed in changeover from pumping to generating, so the changeover time is relatively fast. The double-wheel design tends to accommodate optimum efficiency in both pump-

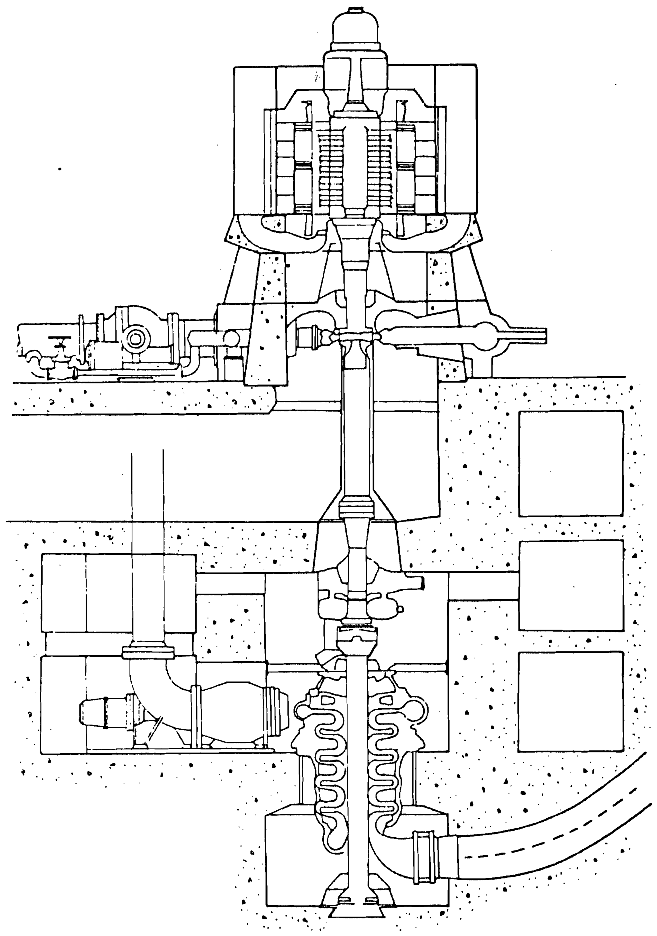


Figure 13.5 Multistage pump and Pelton turbine for pumped/storage hydropower development.

ing and generating operations. Cost of the two-runner arrangement is greater than a single unit, but costs of casings and penstock bifurcation is reduced over the three-unit systems. Figure 13.7 is a sectional drawing of the Isogyre pump/turbine.

Another type of arrangement is the development of pumped/storage units using an underground reservoir. This would be possible and most economical where an existing surface reservoir is near an excavation that has been created for some other purpose, such as a mining operation. It may be that the underground reservoir could be especially made to use the space for a specific application but sharing the cost of creating the cavern is likely to make the development economically more feasible. A suggestion has been made to use space being developed for storm water storage in large tunnels beneath the city of Chicago. Future mining for oil shale and

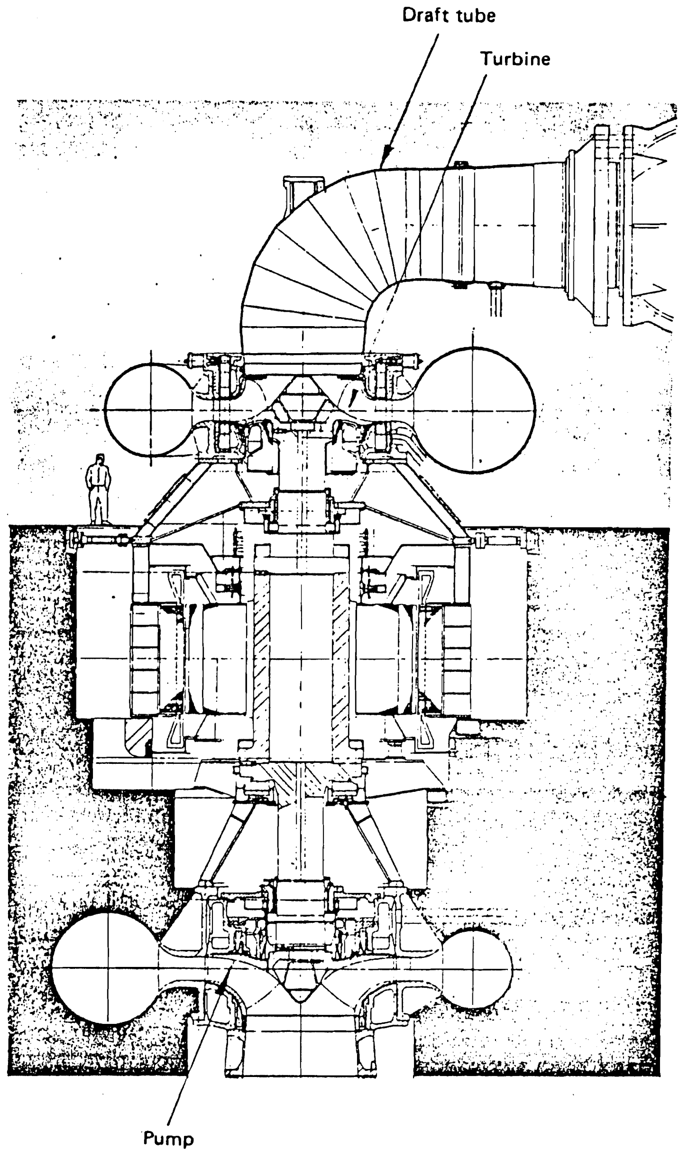


Figure 13.6 Single stage pump and Francis turbine for pumped/storage hydropower development. SOURCE: Escher Wyss.

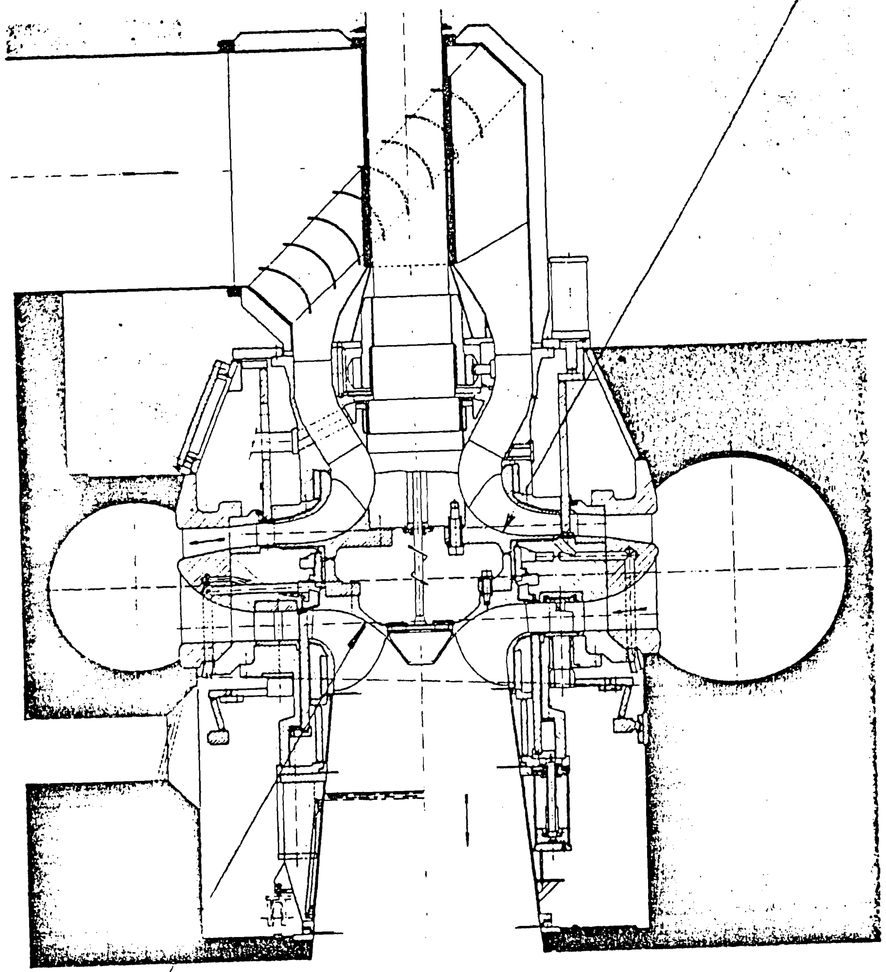


Figure 13.7 Sectional drawing of Isogyre pump/turbine. SOURCE: Vevey-Charmilles.

insitu development of oil from oil shale have been discussed as possibilities for locating underground reservoirs that will serve two purposes. Advantages of such developments would be:

1. Short, direct, minimum-length penstocks
2. Possible use of excavated material for other construction purposes

3. Location away from live streams
4. Minimization of the environmental impact of open impoundments

A possible disadvantage may be increase in the temperature of the water used in the operation. Excavation costs may be significantly higher than simple surface reservoirs.

Table 13.1 gives a summary of the relative advantages of the different arrangements for developing pumped/storage. The tendency is for three-unit systems to be more expensive but to have greater flexibility and higher efficiency. Trends in development of pump storage, according to Whippen and Mayo (1974) and Graeser and Walther (1980), have continued to favor higher heads and greater capacity. In 1953 the Flatiron installation of the U.S. Bureau of Reclamation pumped against a head of 290 ft (88.4 m) and had an output capacity of 9 MW, whereas the Bath County installation ordered in 1974 was designed to operate against a head of 1260 ft (384 m) and to have an output capacity per unit of 457 MW. The Bissorte II

TABLE 13.1 Summary of Characteristics and Relative Advantages of Different Pumped/Storage Arrangements

	Head Limit (m)	Ratio of Pump Q_p to Turbine Q_T	Efficiency	Changeover Time
Single-stage pump/turbine	500–750	0.80	Compromise	Slow
Multistage pump/turbine	1000	0.80	Compromise	Slow
Single-stage pump and Francis turbine with common motor/generator	500–750	As required	High	Fast
Multistage pump and Francis turbine with common motor/generator	700	As required	High	Fast
Multistage pump and Pelton turbine with common motor/generator	1200	As required	Good	Fast
Completely separate turbine/generator and pump/motor units	Same as with common motor/generator	As required	Good	Fast

SOURCE: Carson and Fogleman (1974).

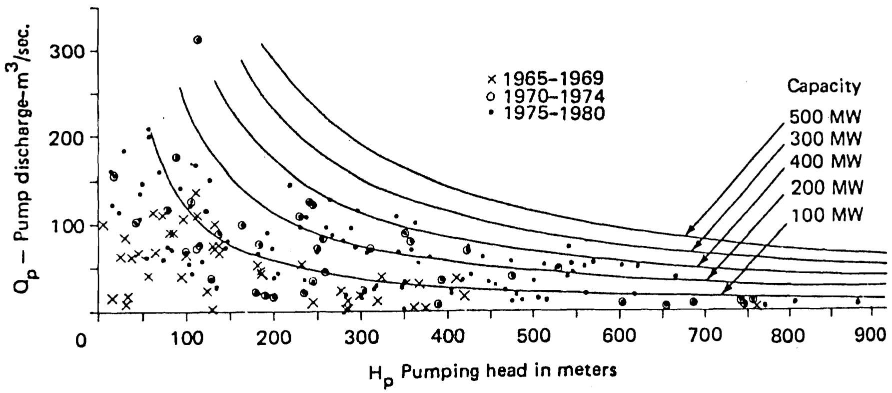


Figure 13.8 Experience curve for pump/turbines. SOURCE: Graeser & Walther (1980).

plant in France is designed to operate against a head of 3917 ft (1194 m) and to have an output capacity of 156 MW. The latter development is with a five-stage pump/turbine. Pump/turbines are produced in several plants in Europe, the United States, and Japan. Names and addresses of manufacturers are given in Appendix A.

Graeser and Walther (1980) list 198 separate pump/turbine installations that were developed as of 1980, with a total installed capacity of 75,000 MW. Figure 13.8 is an experience curve of the installed units, showing characteristic relations of head and discharge.

PLANNING AND SELECTION

Important in planning for pumped/storage hydropower development is the determination of two fundamental parameters of head(s) and discharge(s) to be utilized in sizing the plant. This may mean different values of head and discharge for the pumping and the generating modes. It is evident that the head will vary as the reservoirs are drawn down. The power capacity to be developed will be dictated by the load demands and the economics of building large plants. Sites with heads less than 1000 ft are not likely to be economically suitable for pure pumped/storage development.

An analysis must be made of the length of time the units will operate to determine the volume of water that must be stored in an upper reservoir. Figure 13.9 presents a graph that is useful in determining the live storage required. Example 13.1 illustrates how the curves of Fig. 13.9 can be used in planning.

Example 13.1

Given: A pump/turbine installation is proposed to have a power capacity of 500 MW with an average head of 300 meters.
Required: What amount of live storage would be required to provide six hours of generation?

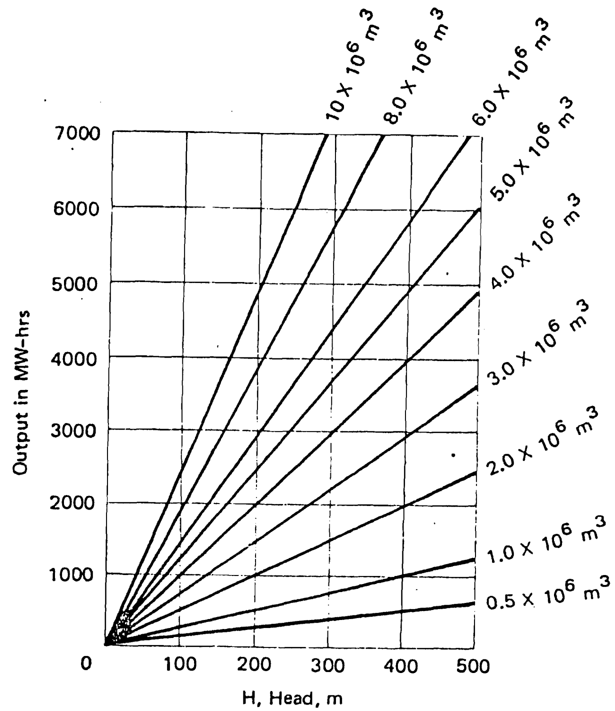


Figure 13.9 Storage requirement for power output versus head. SOURCE: Carson and Fogleman (1974).

Analysis and solution: The energy input would be $500 \times 6 = 3000$ MWh. Entering the graph of Fig. 13.9 with the 3000 MWh as an ordinate value and with an abscissa value of 300 m, average head, the live storage required would be

$$4 \times 10^6\ m^3 \quad \text{ANSWER}$$

This could vary somewhat with drawdown of each reservoir.

The curves of Fig. 13.9 have been developed by Carson and Fogleman (1974) using the power equation, Eq. (3.8), and an assumed efficiency of about 91%.

There are experience limits of how much the head can vary during operation of the pump/turbine in the pumping mode. A useful guide to these limits is presented in Fig. 13.10. This limit is expressed as the ratio (maximum total dynamic pumping head over minimum net pumping head) plotted against the maximum net head with pump/turbine operating as a turbine. In pump terminology, total head, or total dynamic head, means the energy increase per pound imparted to the liquid by the pump.

Planning and selection for the pumping and generating mode require determining speed, preliminary design diameter, and the turbine setting elevation. At preliminary design stages these must be estimated because detailed test data from

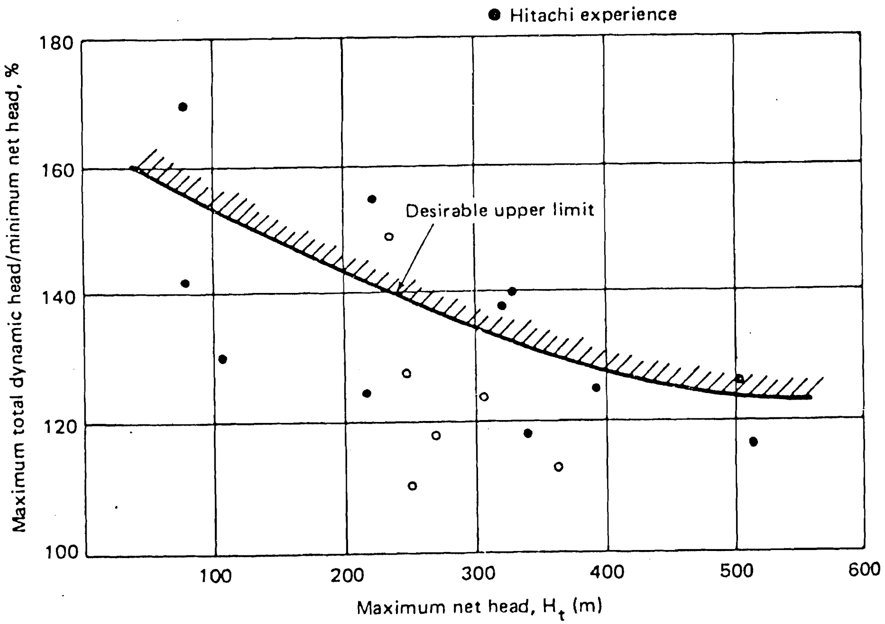


Figure 13.10 Experience limits for pumping head variation for pump/turbines. SOURCE: Kimura & Yokoyama (1973).

the model tests of the pump/turbine manufacturers are not available. This information is usually needed for cost-estimating purposes, for layout of the plant, and for preliminary design of the civil works features of the pumped/storage development.

The approach to planning, selection, and design of a reversible pump/turbine is similar to that outlined in Chapter 6 for conventional hydraulic turbines and utilizes some of the same concepts of turbine constants covered in Chapter 4. However, the approach involves different experience curves and must account for the fact that it is not possible for a reversible pump/turbine to operate at peak efficiency as a turbine and at peak efficiency as a pump under the same design arrangement and at any one synchronous speed. This is shown graphically in Fig. 13.11. This is a plot of efficiencies for various unit speeds and unit discharges from model studies of a pump/turbine. Variation in the point at which peak efficiency for pumping operation occurs with respect to peak efficiency for turbine operation is also reported in the work of Kimura and Yokoyama (1973), Stelzer and Walters (1977), and Kaufmann (1977).

Normally, the selection of pump/turbine sizes involves determining a suitable specific speed, n_{st}, for the turbine and a suitable pump specific speed, n_{sp}. From this characteristic turbine or pump constant a selection is made of the design speed of the pump/turbine, which must be a synchronous speed. Once the design speed, n, is determined an actual n_{st} and n_{sp} can be determined based on the desired rated power output p_d, rated discharge, q_d, and rated head, h_d. It should be noted that

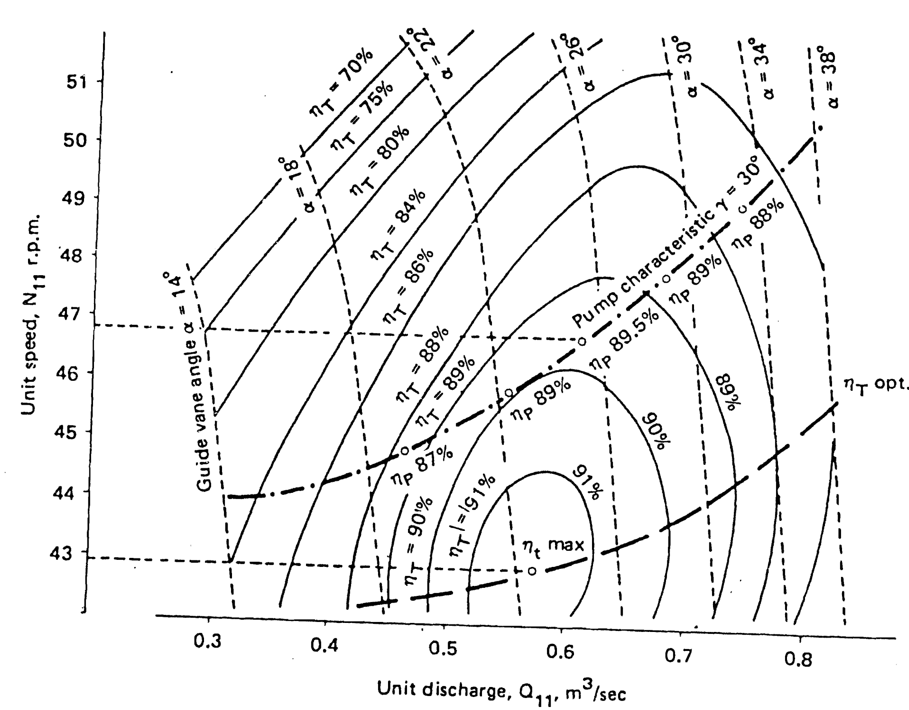


Figure 13.11 Variation in turbine efficiency and pump efficiency for pump/turbines. SOURCE: Alestig & Carlsson.

p, q and d are different for the pumping and the turbining (generating) modes. The design diameter, d_3, can then be found from the empirical experience curves of n_{st} versus d_3. Following this the limiting dimensions of the water passages (spiral case and draft tube) can be determined. Finally, the pump/turbine setting elevation can be determined using empirical experience curves that relate the acceptable cavitation coefficient or plant sigma, σ, to the n_{st}.

Using turbine constants is complicated by the fact that various forms of the specific speed term have developed using different parameters and different units. To understand the different forms of the term "specific speed," the appropriate formulas are presented. Conventionally in American literature and use, the design specific speed for turbine action of a pump/turbine is given by the equation

$$n_{st} = \frac{n\sqrt{p_d}}{h_d^{5/4}} \tag{13.1}$$

where n_{st} = specific speed for turbine (rpm, ft, hp)
n = rotational speed, rpm
h_d = turbine design head, ft
p_d = turbine full-gate capacity (at h_d), hp; however, when comparing hy-

draulic machines the rated output will be that based on the gate opening and head (h_{BE}) where best efficiency is attained.

Note. The convention here is to use lowercase letters for parameters expressed in American units of ft, hp, ft^3/sec, and rpm.

In the European convention and utilizing metric units, the design specific speed for turbine action of a pump/turbine is given by the equation

$$N_{st} = \frac{N\sqrt{P_d}}{H_d^{5/4}} \tag{13.2}$$

where N_{st} = specific speed for turbine (rpm, m, kW)
N = rotational speed, rpm
H_d = turbine design head, m
P_d = turbine full-gate capacity (at H_d), kW; however, when comparing hydraulic machines the rated output will be that based on the gate opening and head (H_{BE}) where best efficiency is attained.

In American convention for the pumping mode of pump/turbines, the specific speed for pump action is given by the equation

$$n_{sp} = \frac{n\sqrt{q_p}}{h_p^{3/4}} \tag{13.3}$$

where n_{sp} = specific speed for pump (rpm, gpm, ft)
n = rotational speed, rpm
q_p = pump best efficiency discharge, gpm
h_p = pump best efficiency head, ft.

In European convention for the pumping mode of pump/turbines, the specific speed for pump action is given by the equation

$$N_{sp} = \frac{N\sqrt{Q_p}}{H_p^{3/4}} \tag{13.4}$$

where N_{sp} = specific speed for pump (rpm, m^3/sec, m)
N = rotational speed, rpm
Q_p = pump best efficiency discharge, m^3/sec
H_p = pump best efficiency head, m.

Stepanoff (1957), using the same parameters, $Q_T = Q_p$ and $H_T = H_p$, for both machines, indicates a relation exists between N_{st} and N_{sp} as follows:

$$N_{st} = N_{sp}\eta \tag{13.5}$$

where η = hydraulic efficiency, approximately equal to the square root of the pump best efficiency.

A dimensionless specific speed could be developed and utilized as discussed in Chapter 4 and indicated in Eq. (4.22). DeSiervo and Lugaresi (1980), in reporting on characteristics of pump/turbines utilized the turbine form of the specific speed

equation, Eq. (13.2), and rated power capacity of the pump P_p in kilowatts to express specific speed for the pumping mode of pump/turbines. Caution should be exercised to be sure what units the specific speed is expressed in and at what gate capacity. The usual practice is to express specific speed at the best efficiency point of operation.

Experience Curves

As with conventional hydraulic turbines, pump/turbines are characterized by relating the specific speed to the rated head in both the turbine mode and the pumping mode. Numerous experience curves have been developed and are reported in the literature (Alestig and Carlsson, n.d.; Carson and Fogelman, 1974; deSiervo and Lugaresi, 1980; Graeser and Walther, 1980; Kaufmann, 1977; Stelzer and Walters, 1977; Warnock, 1974; Whippen and Mayo, 1974). In the experience curves shown in Fig. 13.12, the specific speed n_{sp} in the pump form is plotted against head. DeSiervo and Lugaresi (1980) have developed empirical equations for determining the specific speed of pump/turbines for both the turbine operation and for the pumping operation. These equations are as follows:

$$N_{st} = 1700H_t^{-0.481} \quad \text{(based on 1961-1970 data)} \tag{13.6}$$

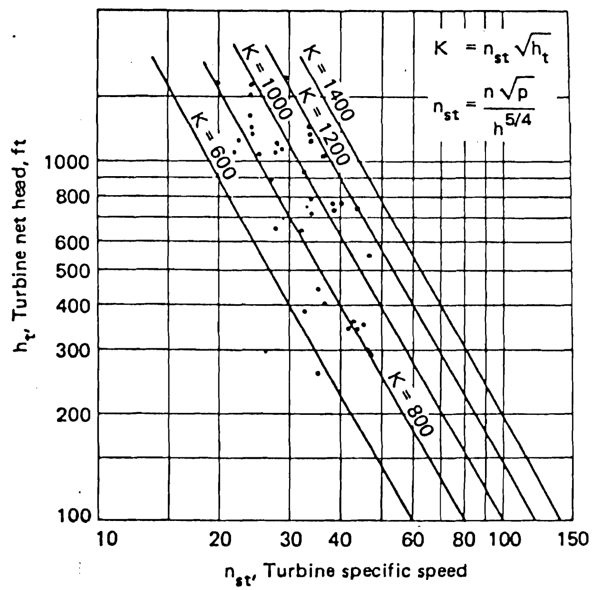


Figure 13.12 Experience curve relating specific speed of turbine, n_{st}, to head of pump/turbine. SOURCE: Allis-Chalmers Corporation.

where N_{st} = specific speed (rpm, kW, m)
H_t = rated head as turbine, m

and

$$N_{st} = 1825 H_t^{-0.481} \quad \text{(based on 1971-1977 data)} \tag{13.7}$$

For the pumping mode the empirical equations are

$$N_{sp} = 1672 H_p^{-0.480} \quad \text{(based on 1961-1970 data)} \tag{13.8}$$

where N_{sp} = specific speed (rpm, kW, m)
H_p = rated head as pump, m

and

$$N_{sp} = 1768 H_p^{-0.480} \quad \text{(based on 1971-1977 data)} \tag{13.9}$$

These empirical equations can be used to determine the design speed. Example 13.2 shows a suitable procedure for selecting the rotational speed of the unit functioning as a turbine.

Example 13.2

Given: A large pumped/storage development calls for a rated turbine output of 343 MW and a rated head as a turbine of 1710 ft (521.2 m).

Required: Determine a suitable operating speed using available experience curves.

Analysis and solution: Using Eq. (13.7), the empirical equation from deSiervo and Lugaresi (1980):

$$N'_{st} = 1825 H_t^{-0.481} = \frac{1825}{(521.2)^{0.481}} = 90.02$$

From Eq. (13.2),

$$N' = \frac{N'_{st} H_d^{5/4}}{P_d^{1/2}} = \frac{(90.02)(521.2)^{1.25}}{(343{,}000)^{1/2}} = 382.8 \text{ rpm}$$

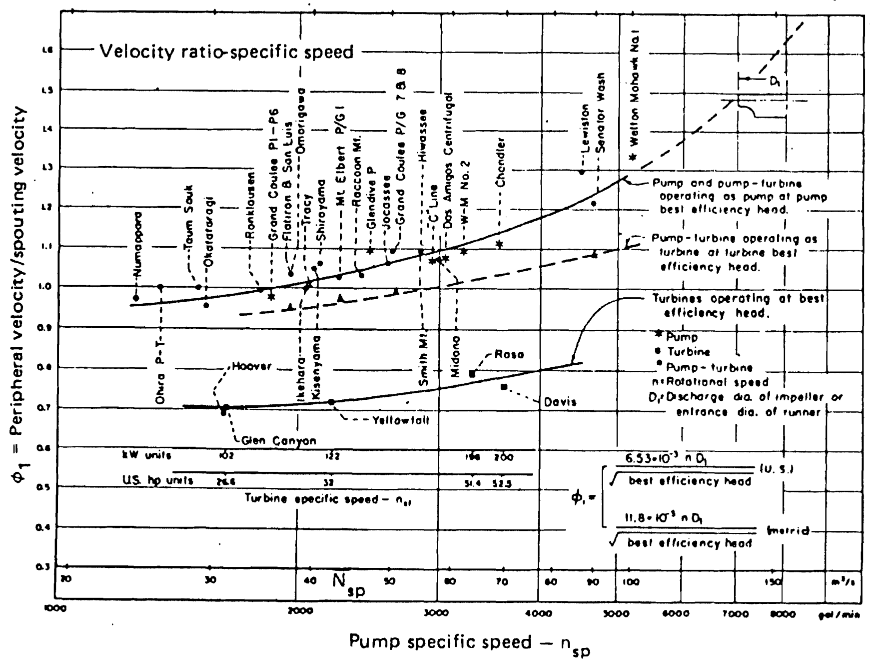
Using Eq. (4.30), we have

$$n = \frac{7200}{N_p}$$

$$N_p = \frac{7200}{382.8} = 18.8 \qquad \text{choose } p = 18 \text{ poles}$$

$$n = \frac{7200}{18} = 400 \text{ operating synchronous speed} \quad \text{ANSWER}$$

Empirical experience curves are utilized to develop various preliminary dimensions for the pump/turbine installation, including pump/turbine impeller diameter, height of the gate opening, water passage dimensions, and draft tube dimensions. These dimensions are frequently related to specific dimensions by utilizing characteristics of the peripheral impeller velocity, theoretical spouting velocity of the water, and the radial velocity of the water. A series of useful empirical experience curves are presented from Stelzer and Walters (1977). Figure 13.13 relates n_{sp} to the speed ratio of the peripheral velocity of the impeller to the theoretical spouting velocity of the water. Figure 13.14 relates n_{sp} to the ratio of maximum to minimum diameter of the impeller. Figure 13.15 relates n_{sp} to the wicket gate height in relation to the impeller diameter. Figure 13.16 relates n_{sp} to the ratio of radial velocity of the water to the theoretical spouting velocity of the water. The curves are developed so that the ordinates are ratios of like terms, so that either American units or metric units can be used. Example 13.3 illustrates how these curves can be used in practical problems of planning for the appropriate sizes of components of the pump/turbine installations.



Unit selection

Figure 13.13 Experience curve relating specific speed of pump, n_{sp}, to speed ratio. SOURCE: Stelzer and Walters (1977), U.S. Bureau of Reclamation.

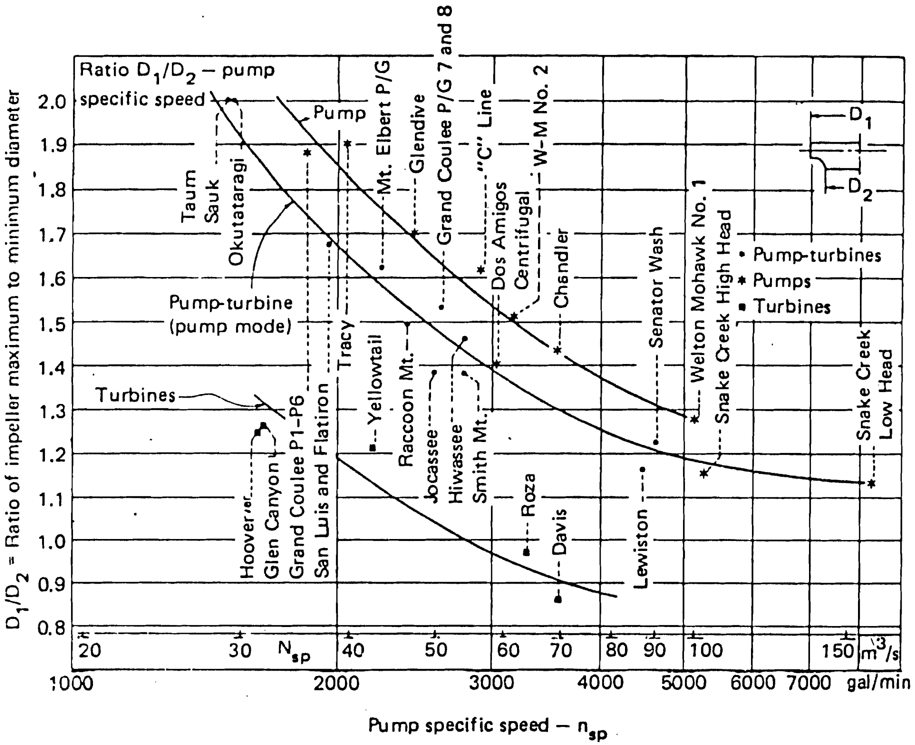


Figure 13.14 Experience curve relating specific speed of pump, n_{sp}, to turbine impeller diameter. SOURCE: Stelzer and Walters (1977), U.S. Bureau of Reclamation.

Example 13.3

Given: A pump/turbine installation is to operate as a turbine with a rated output of 100 MW with heads varying from $H_{tM} = 319$ m to $H_{tm} = 285$ with the rated head considered to be $H_t = 305$ m. The unit in the pumping mode has heads that vary from $H_{pM} = 325$ m to $H_{pm} = 291$ m. Assume that the turbine best efficiency is 0.89 and the pump best efficiency is 0.92. Consider that calculations similar to Example 13.2 have determined that the operating synchronous speed should be 600 rpm.

Required: Determine the following:

(a) Pump discharge at best efficiency
(b) Impeller diameter
(c) Wicket gate height
(d) Estimated radial velocity of the water entering the pump impeller

Analysis and solution:

(a) Using the power equation, Eq. (3.8), as a pump

$$P = \frac{Q_p \rho' g' H_p}{\eta}$$

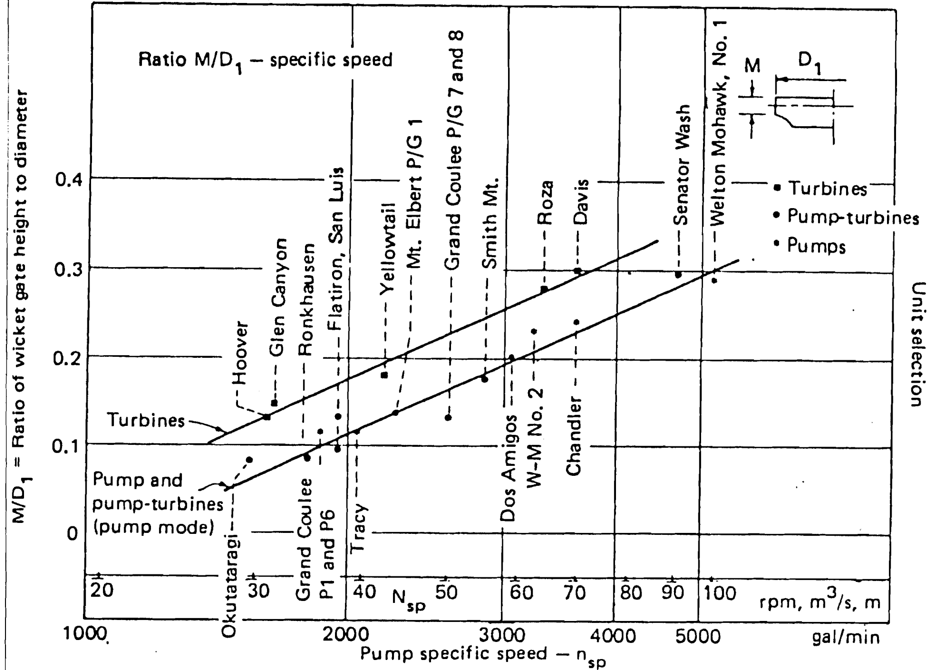


Figure 13.15 Experience curve relating specific speed of pump, n_{sp}, to wicket gate height. SOURCE: Stelzer and Walters (1977), U.S. Bureau of Reclamation.

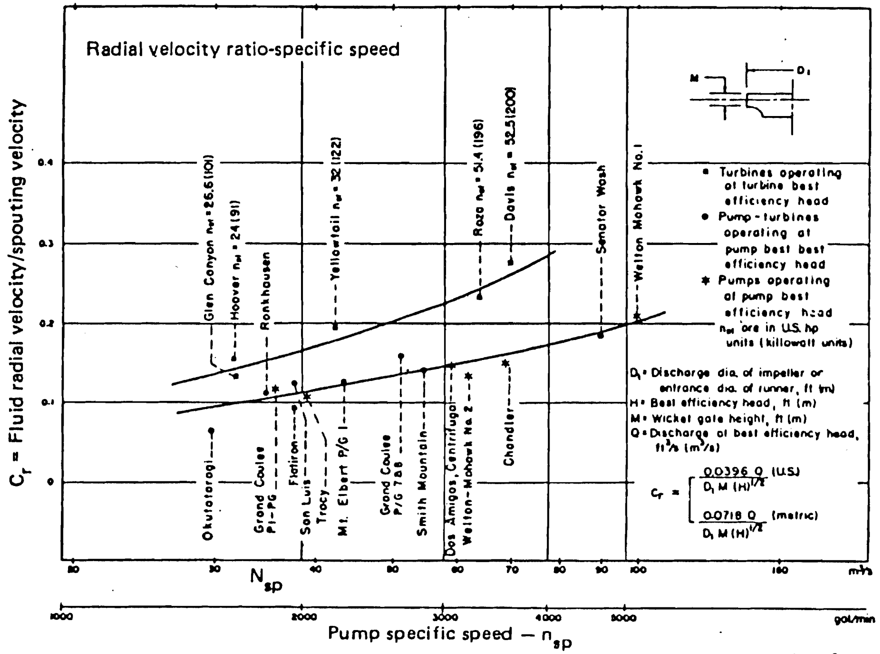


Figure 13.16 Experience curve relating specific speed of pump, n_{sp}, to ratio of radial velocity over spouting velocity. SOURCE: Stelzer and Walters (1977), U.S. Bureau of Reclamation.

Transposing gives

$$Q_p = \frac{P\eta}{\rho' g' H_p}$$

$$= \frac{100{,}000(0.92)}{(1)(9.806)(305)}$$

$$= 30.8\ m^3/sec \quad \text{ANSWER}$$

(b) Using Eq. (13.4) for specific speed of a pump/turbine in the pumping mode, we obtain

$$N_{sp} = \frac{N\sqrt{Q_p}}{H_p^{3/4}}$$

$$= \frac{600\sqrt{30.8}}{(305)^{5/4}}$$

$$= 45.6 \quad \text{(in terms of rpm, m}^3\text{/sec, m)}$$

With the known value of $N_{sp} = 45.6$, enter Fig. 13.13; the experience curve indicates for the $N_{sp} = 45.6$ a value of $\phi_1 = 1.04$. Using the equation from Fig. 13.13 yields

$$\phi_1 = \frac{(11.8)(10^{-3})ND_1}{\sqrt{\text{best efficiency head}}} \tag{13.10}$$

transposing gives

$$D_1 = \frac{\phi_1\sqrt{\text{best efficiency head}}}{(11.8)(10^{-3})N} = \frac{1.04\sqrt{305}}{(11.8)(10^{-3})(600)}$$

$$= 2.56\ m \text{ impeller diameter} \quad \text{ANSWER}$$

Solving for the impeller throat diameter, D_2, enter Fig. 13.14 with $N_{sp} = 45.6$ and find the D_1/D_2 ratio, impeller diameter to minimum diameter:

$$\frac{D_1}{D_2} = 1.54$$

so then

$$D_2 = \frac{D_1}{1.54}$$

$$= 1.66\ m \quad \text{ANSWER}$$

(c) Solving for wicket gate height, M, enter Fig. 13.15 with $N_{sp} = 45.6$ and find

$$\frac{M}{D_1} = 0.145$$

$$M = D_1(0.145)$$

$$= (2.56)(0.145)$$

$$= 0.37\ m \quad \text{ANSWER}$$

(d) Solving for radial velocity of water entering the pump, enter Fig. 13.16 with $N_{sp} = 45.6$, or by using the equation from the Fig. 13.16:

$$C_r = \frac{V_r}{\sqrt{2gH_p}} = \frac{0.0718Q_p}{D_1 M\sqrt{H_p}} \tag{13.11}$$

where C_r = ratio of radial velocity of water entering pump to the spouting velocity
0.0718 = coefficient
Q_p = pumping discharge, m^3/sec
D_1 = impeller discharge diameter, m
M = wicket gate height, m
H_p = rated pumping head, m.

$$C_r = \frac{(0.0718)(30.8)}{(2.56)(0.371)\sqrt{(305)}}$$

$$= 0.133$$

$$V_r = C_r\sqrt{2g'H_p}$$

$$= 0.133\sqrt{(2)(9.806)(305)}$$

$$= 10.29\ m/sec \quad \text{ANSWER}$$

Useful experience curves and empirical equations similar to those indicated in Figs. 13.13 through 13.17 for determining characteristic planning information for pump/turbines have been developed in the work of deSiervo and Lugaresi (1980) and Kaufmann (1977). Greater detail is given in those references as well as in Stelzer and Walters (1977). More dimension estimates and guides for characterizing the water passages and the draft tubes, as well as experience curves for estimating weights of impellers, are given in these three references.

Figure 13.17 is an empirical experience curve for relating the N_{sp} for pump/turbines to the peripheral velocity coefficient (speed ratio, ϕ_1). The term K_u, peripheral velocity coefficient, is essentially the same term as the speed ratio of American convention. K_u is obtained by the following formula:

$$K_u = \frac{D_1 N}{60\sqrt{2g'H_p}} \tag{13.12}$$

where D_1 = impeller diameter at guide vane centerline, m
N = rotational speed, rpm
H_p = rated head of pump, m.

This experience curve in Fig. 13.17 is comparable to Fig. 13.13 and can be used for making an independent check for determining the impeller diameter. The curve of Fig. 13.17 extends into higher values of N_{sp} than Fig. 13.13 and gives distinction in

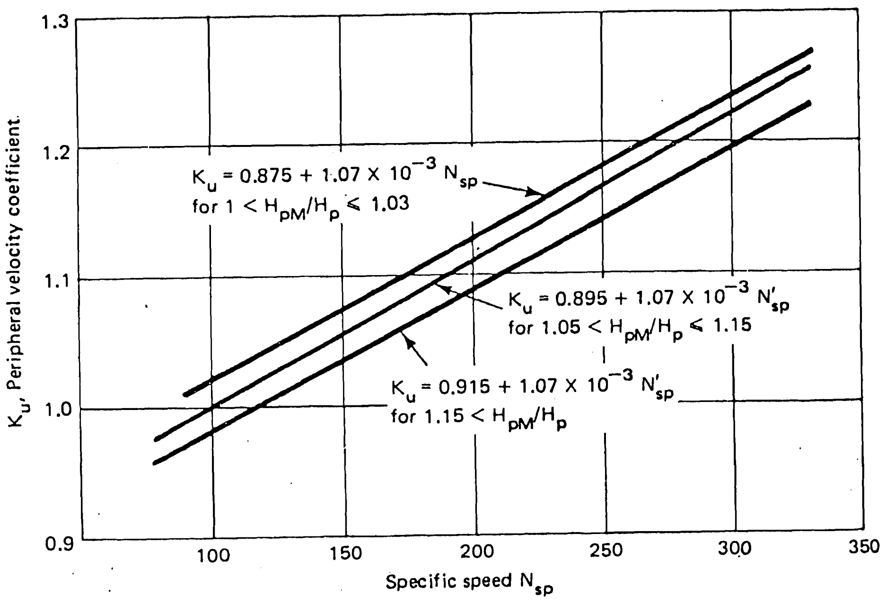


Figure 13.17 Impeller peripheral velocity coefficient versus specific speed of pump (n_{sp}, a function of P_p in kW). SOURCE: deServio & Lugaresi (1980).

the empirical equation, where variation is expected with respect to the ratio of maximum pumping head, H_{pM}, to normal rated head, H_p, for pump/turbines operating in the pumping mode. Caution should again be indicated that the specific speed for the pumping mode as used by deSiervo and Lugaresi is expressed in terms of pump output in kW. deServio and Lugaresi (1980) show the following relation for N_{st} and N_{sp}:

$$\text{For } H_p/H_t = 0.9: \quad N_{sp} = 1.272 N_{st}^{0.978} \tag{13.13}$$

$$\text{For } H_p/H_t = 1.0: \quad N_{sp} = 0.842 N_{st}^{1.033} \tag{13.14}$$

$$\text{For } H_p/H_t = 1.1: \quad N_{sp} = 0.619 N_{st}^{1.065} \tag{13.15}$$

Pump/Turbine Settings

Important in feasibility studies and preliminary planning for pump/turbine installations is the problem of determining the setting elevation for pump/turbines. Experience has dictated that the units be set at considerable submergence below the minimum elevation of the lower reservoir water elevation, the tailwater for the turbine operation of a pump/turbine. Figure 13.18 is an experience curve for making turbine setting calculations. A similar approach to determining the setting elevation for pump/turbines is used as for conventional hydraulic turbines utilizing reaction-type turbines (see Chapter 7). Example 13.4 illustrates the approach to determining the setting elevation for a pump/turbine installation.

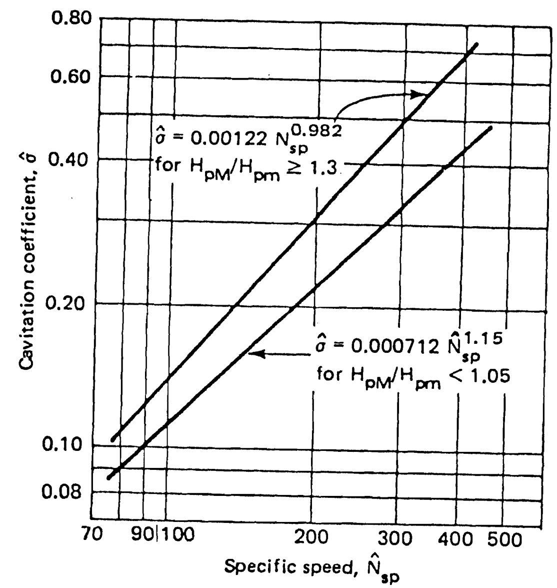


Figure 13.18 Experience curve relating statistical specific speed of pump, n_{sp}, to statistical cavitation coefficient. SOURCE: deServio & Lugaresi (1980).

Example 13.4

Given: The pump/turbine in Example 13.3 is to be located so that the lower reservoir is at elevation 1000 m MSL and with normal water temperature the barometric pressure head, H_b = 8.58 m of water.

Required: Determine the setting elevation for the pump/turbine.

Analysis and solution: Because the specific speed for pumping from Example 13.3 is not in suitable units, first use Eq. (13.2) to find the N_{st}.

$$N_{st} = \frac{N\sqrt{P_t}}{H_t^{5/4}}$$

$$= \frac{600\sqrt{100{,}000}}{(305)^{1.25}}$$

$$= 148.9 \quad \text{(in terms of rpm, kW, m)}$$

Assume that H_p/H_t = 1.0; then, using Eq. (13.13),

$$N_{sp} = 0.842 N_{st}^{1.033}$$

$$= (0.842)(148.86)^{1.033}$$

$$= 147.8$$

Referring to Fig. 13.18, it is noted that the experience curves are developed for values of $H_{pM}/H_{pm} \geqslant 1.3$ and for $H_{pM}/H_{pm} < 1.05$. Calculating H_{pM}/H_{pm} for

Example 13.13 yields

$$\frac{H_{pM}}{H_{pm}} = \frac{325}{291} = 1.12$$

The N'_{sp} can be taken as the estimated specific speed $\hat{N}_{sp}$. Entering Fig. 13.18, select a value for $\hat{\sigma}$, extrapolated between the two designated empirical curves. Choose $\hat{\sigma}$ = 0.18. At 1000 MSL the barometric pressure, H_b, acting at the tailrace of the lower reservoir, would be H_b = 8.58 m. Then using Eq. (7.9), we get

$$\sigma = \frac{H_b - H_s}{H}$$

$$\begin{aligned} H_s &= H_b - \sigma H \\ &= 8.58 - (0.18)(325) \\ &= -49.9 \text{ m} \quad \text{ANSWER} \end{aligned}$$

Note here that H is used as H_{pM} because that is the most critical head to which the pump turbine will be exposed and represents the value that will dictate a maximum submergence requirement for the pump/turbine setting. Hence the outlet to the pump/turbine should be set at

$$1000 - 49.9 = 950.1 \text{ m (MSL)} \quad \text{ANSWER}$$

Other experience curves for determining preliminary estimates of pump/turbine settings are reported in the work of Kimura and Yokoyama (1973), Jaquet (1974), Stelzer and Walters (1977), Meier, et al. (1971), and Kaufmann (1977). The final design for turbine setting of the pump/turbine will be specified by the pump/turbine manufacturer based on model tests where the cavitation phenomenon is carefully observed and related to operating conditions of gate openings, speeds, heads, and head variation as well as the temperature conditions of the water.

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PROBLEMS

13.1. Obtain data from an existing pumped/storage plant or a feasibility study and characterize the plant and determine the following:

(a) Turbine rated output
(b) Pump rated output
(c) Rated head operating as a turbine
(d) Maximum pumping head
(e) Minimum pumping head
(f) Operating rotational speed
(g) Specific speed as a turbine
(h) Specific speed as a pump
(i) Cavitation coefficient

13.2. A pumped/storage development has a maximum operating head as a turbine of 260 m and a minimum head of 240 m. If the desired capacity as a turbine is 800 MW, determine the following:

(a) The hours of peaking that a live storage volume of 6×10^6 m^3 will allow the pump/turbine to operate

(b) The specific speed of a suitable pump/turbine if three units of equal size are used
(c) A suitable operating speed of the pump/turbine
(d) The storage that would be required if the operating period were extended by 2 hours

13.3. Using the experience curve of Fig. 13.12, with $K = 1000$, check Example 13.2 to determine what rotational speed would be recommended. What does this do to the estimated design diameter of the pump/turbine impeller?

13.4. Check the results in Example 13.3 using the experience curves and empirical equations of an investigator other than Stelzer and Walters (1977).

13.5. Investigate a proposed pumped/storage site in your area and proceed through the entire design procedure to select a size, operating speed, impeller diameter, turbine setting, and characteristic water passage dimensions.

MICROHYDRO AND MINIHYDRO SYSTEMS 14

BASIC CONCEPTS

Microhydro power usually refers to hydraulic turbine systems having a capacity of less than 100 kW. Minihydro power usually refers to units having a power capacity from 100 to 1000 kW. Units this small have been in use for many years, but recent increases in the value of electrical energy and federal incentive programs in the United States have made the construction and development of microhydro and minihydro power plants much more attractive to developers. Similarly, small villages and isolated communities in developing nations are finding it beneficial and economical to use microhydro and minihydro power systems. Figure 14.1 shows graphically the relative interest in small hydropower systems throughout the United States.

The principles of operation, types of units, and the mathemathical equations used in selection of microhydro and minihydro power systems are essentially the same as for conventional hydro power developments, as discussed in Chapters 2 through 6. However, there are unique problems and often the costs of the feasibility studies and the expenses of meeting all regulatory requirements make it difficult to justify microhydro and minihydro power developments on an economic basis. Hence it is important in planning for microhydro and minihydro power developments that simplification be sought throughout the entire study and implementation process.

TYPES OF UNITS

Consideration of the standard units presently being manufactured indicates that four types of units are most adaptable to microhydro and minihydro systems. Small

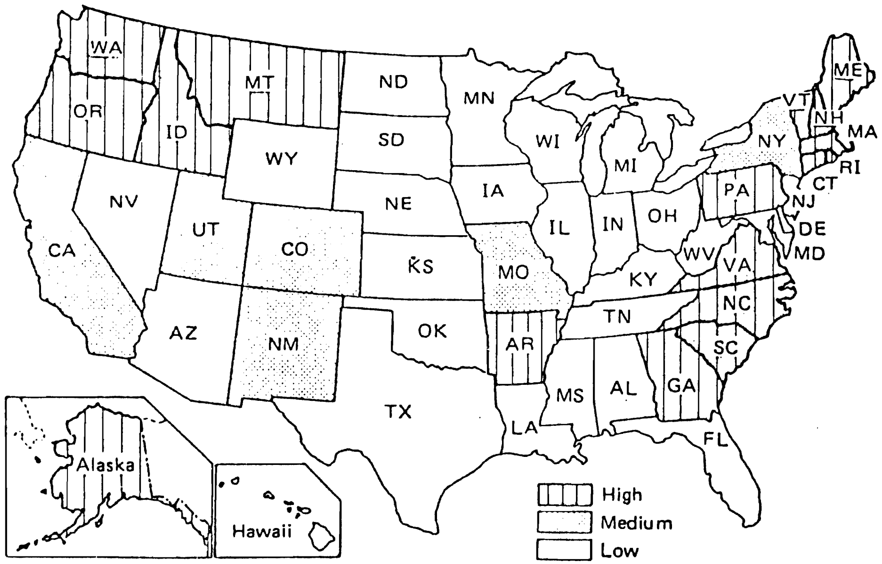


Figure 14.1 Map showing relative interest in microhydro power in the United States.

impulse-type turbines are available in numerous sizes and are being manufactured by several small companies. A list of possible sources is presented in Appendix A. The higher the head available, the better the chance for a suitable installation using a small impulse turbine. Figure 14.2 shows equipment for a typical microhydro power installation using an impulse-type turbine.

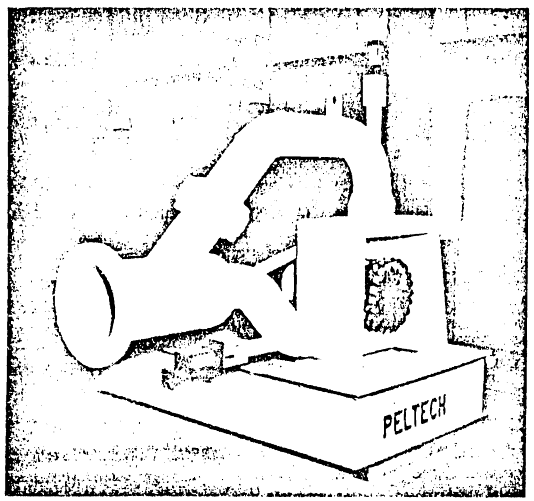


Figure 14.2 Pelton-type turbine equipment for microhydro installations. SOURCE: Small Hydroelectric Systems & Equipment.

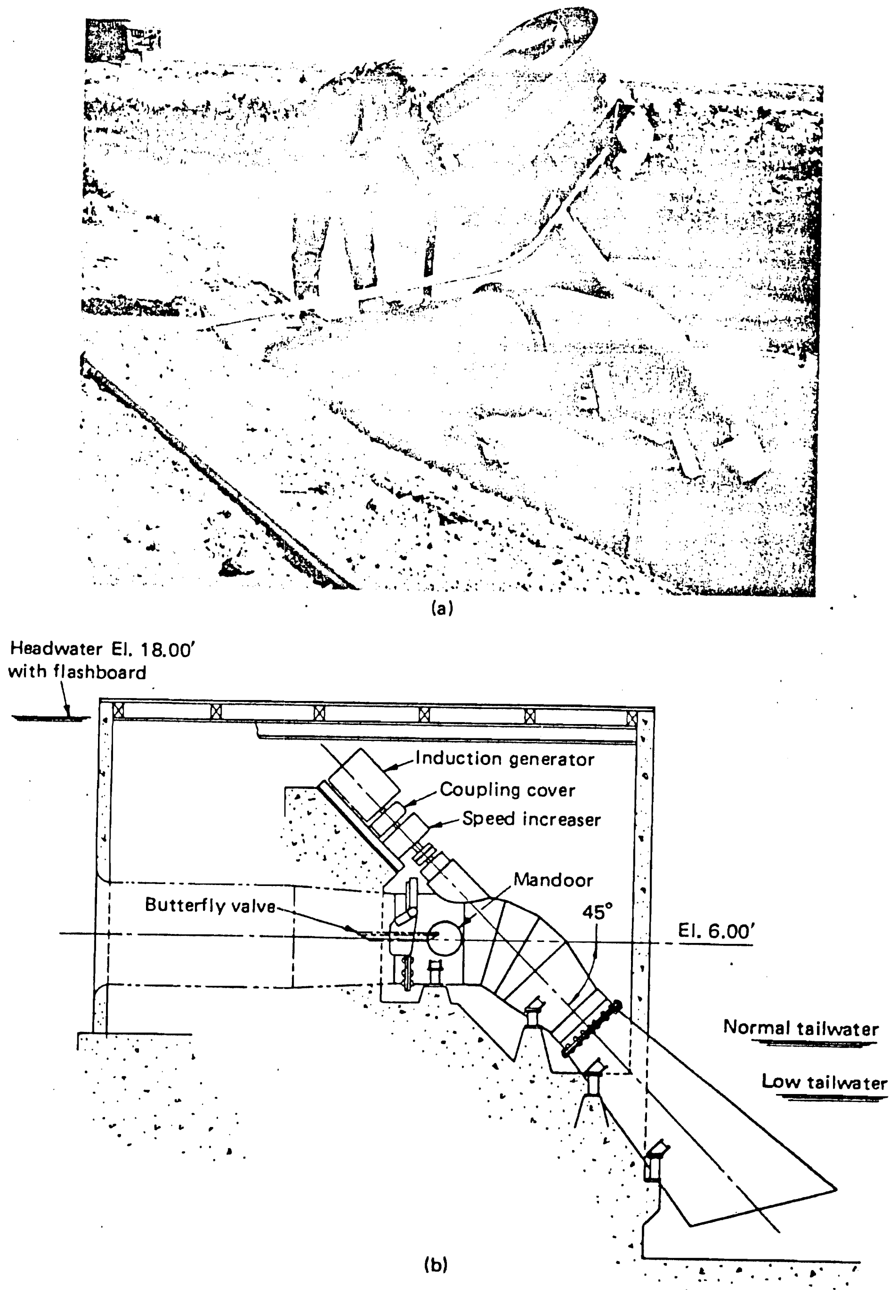


Figure 14.3 Tubular-type propeller turbine for minihydro installations. SOURCE: Allis-Chalmers Corporation.

A second standard unit is a small Francis-type turbine. Typical of these units is the vertical Samson turbine manufactured by James Leffel & Co. A third type, useful for low-head installations, utilizes small propeller-type units with the generator usually mounted outside the water passage. Some units are manufactured to operate with the generator mounted in the water passage. Figure 14.3 shows a typical propeller turbine installation of the tubular type bulb that can be used in the minihydro and microhydro ranges. A fourth type of unit is the cross-flow turbine.

Microhydro power units can utilize belt drives to accommodate to the best turbine speed of operation while allowing for selection of an economical size and type of electric generator. For the same purpose, gear drives usually provide speed increasers for the electric generators. Angle gear drives may also be used but tend to have lower efficiencies and higher costs.

DESIGN AND SELECTION CONSIDERATIONS

The common practice for microhydro and also minihydro systems is to develop standard unit sizes of equipment that will operate over a range of heads and flows. Either selection charts or selection nomographs are used to select appropriate units for site-specific applications. This is in contrast to the custom building of "tailor-made" units for conventional large hydropower installations. Figure 14.4 is a generalized selection chart showing the ranges over which the different types of units can be applicable.

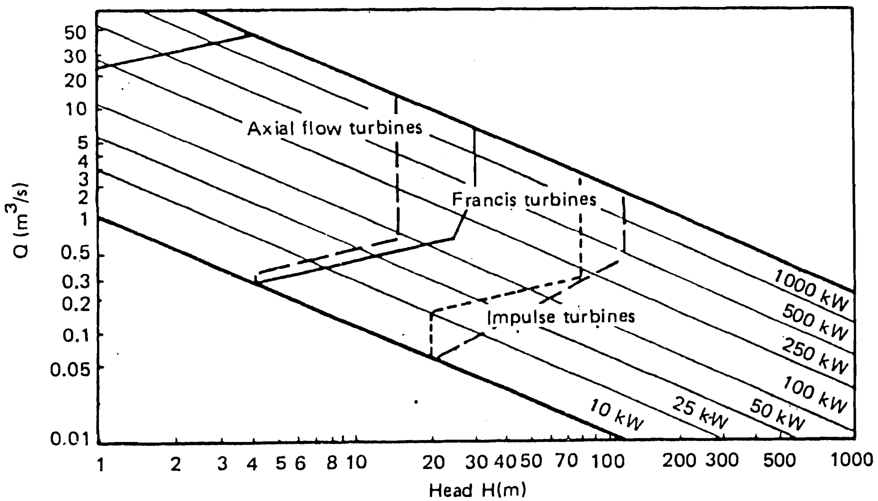


Figure 14.4 Application range of different types of turbines for microhydro and minihydro installations. SOURCE: Voest-Alpine AG (Strohmer & Walsh).

TABLE 14.1 Performance Characteristics for Pelton-Type Turbines in the Microhydro Range[a]

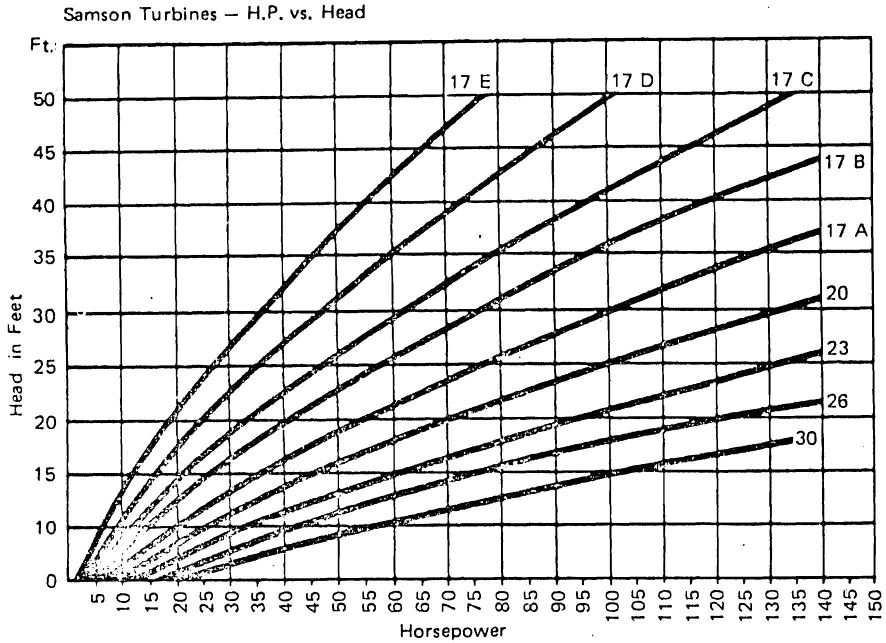
Head	Lb/in^2	Hp	Ft^3/sec	Speed
20	8.66	0.5	0.52	384
30	12.99	1.0	0.648	472
40	17.32	2.7	0.75	542
50	21.65	3.8	0.83	606
60	25.99	4.9	0.90	664
70	30.32	6.2	0.98	716
75	32.47	6.8	1.00	742
80	34.65	7.6	1.05	766
90	38.98	9.0	1.10	814
100	43.31	10.5	1.18	853
120	51.97	13.8	1.23	938
125	54.12	14.7	1.30	958
140	60.63	17.4	1.38	1014
150	64.95	19.3	1.43	1054
160	69.29	21.3	1.50	1084
175	75.77	24.3	1.55	1134
180	77.96	25.4	1.60	1150
200	86.62	29.5	1.65	1212
220	95.26	34.3	1.73	1272
240	103.92	39.0	1.78	1328
250	108.50	41.5	1.85	1354
260	112.58	44.0	1.90	1382
280	121.24	49.2	1.95	1434
300	130.20	54.6	2.03	1484
320	138.56	60.1	2.08	1534
340	147.22	65.8	2.15	1580
350	151.55	66.7	2.18	1604
360	155.88	71.7	2.23	1626
380	164.54	77.8	2.28	1670
400	173.60	84.0	2.33	1714
500	216.00	117.12	2.64	1920

[a]The data presented here are for a 9.75-in. PELTECH turbine.
SOURCE: Small Hydroelectric Systems & Equipment.

Several examples of different companies' approaches and their available selection information are presented to provide a basis for preliminary design and for feasibility analysis purposes.

Impulse-type turbine. The data in Table 14.1 show expected output for a 9.75-in.-pitch-diameter impulse turbine with a 1.625-in.-diameter jet nozzle. Similar information is available from the company for a 19.5-in.-pitch-diameter impulse turbine and for a 39-in.-pitch-diameter impulse turbine.

Francis-type turbine (James Leffel & Co.). Figure 14.5 is a nomograph for selecting Samson turbines of Francis type with design head and desired power out-



Above curves indicate maximum expected output for an open flume vertical configuration. Leffel should be consulted regarding output if a pressure case or other type of configuration is desired.

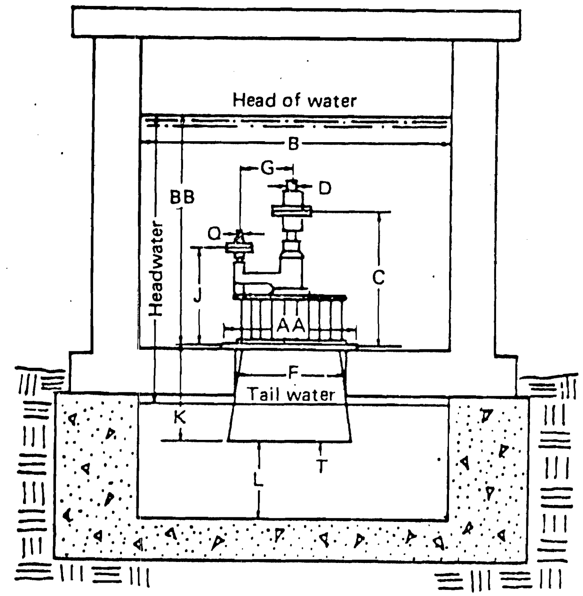
Figure 14.5 Nomograph for selecting standardized Francis turbines for microhydro installations. SOURCE: James Leffel Co.

put known. Nine different turbine sizes of units cover a range of head up to 50 ft (15.2 m) and power outputs up to 100 kW. The designations 17E, 17D, 17C, 17B, 17A, 20, 23, 26, and 30 in Fig. 14.5 refer to different sizes of standard Samson turbines. To use the nomograph, one must estimate an overall operating efficiency and compute the necessary design discharge. Figure 14.6 presents a dimension diagram for sizing microhydro installations for the designated nine standard-size Samson turbines and a table with controlling dimensions for the standard Samson turbines.

Propeller-type turbine (Hydrolic TM, Moteurs Leroy-Somer). Figure 14.7 shows different arrangements for installing microhydro units, and Table 14.2 gives the general range of use for different models of these propeller turbines. Figure 14.8 gives characteristic overall dimensions for various sizes of particular standardized models. Moteurs Leroy-Somer furnishes detailed information on all models and indicates applicable power output for maximum and minimum discharges.

Propeller-type turbine (KMW miniturbines). Figure 14.9 shows a selection chart, a dimensioning drawing for a typical installation, and a dimension table for selection and sizing of standard tubular turbines in the range 100 to 1600 kW.

Vertical Samson turbines



Size of Samson turbines	17 E	17 D	17 C	17 B	17 A	20	23	26	30
AA Flange diameter	$34\frac{1}{8}$	$34\frac{1}{8}$	$34\frac{1}{8}$	$34\frac{1}{8}$	$34\frac{1}{8}$	$38\frac{5}{8}$	$42\frac{3}{4}$	$46\frac{3}{4}$	$51\frac{3}{4}$
B Penstock width	48	48	48	48	48	60	72	84	96
BB Minimum depth	34	34	34	34	34	40	46	52	58
C Turbine shaft height	32	32	32	32	32	35	40	45	50
D Turbine coupling bore	$2\frac{1}{8}$	$2\frac{3}{8}$	$2\frac{3}{8}$	$2\frac{5}{8}$	$2\frac{5}{8}$	$2\frac{7}{8}$	$3\frac{1}{8}$	$3\frac{3}{8}$	$3\frac{7}{8}$
F Draft tube clearance	$29\frac{1}{8}$	$29\frac{1}{8}$	$29\frac{1}{8}$	$29\frac{1}{8}$	$29\frac{1}{8}$	$33\frac{5}{8}$	$37\frac{3}{4}$	$41\frac{3}{4}$	$46\frac{3}{4}$
G Center to center	12	12	12	12	12	14	16	18	20
J Gate shaft height	23	23	23	23	23	25	27	29	32
K Draft tube length	23	23	23	23	23	25	27	29	32
L Minimum required depth	27	27	27	27	27	30	36	42	48
Q Gate coupling bore	$1\frac{3}{16}$	$1\frac{3}{16}$	$1\frac{3}{16}$	$1\frac{3}{16}$	$1\frac{3}{16}$	$1\frac{7}{16}$	$1\frac{7}{16}$	$1\frac{11}{16}$	$1\frac{11}{16}$
T Minimum submergence	3	3	3	3	3	4	4	4	4
Above dimensions are in inches									

Figure 14.6 Characteristic dimensions for Francis turbines for microhydro installations. SOURCE: James Leffel Co.

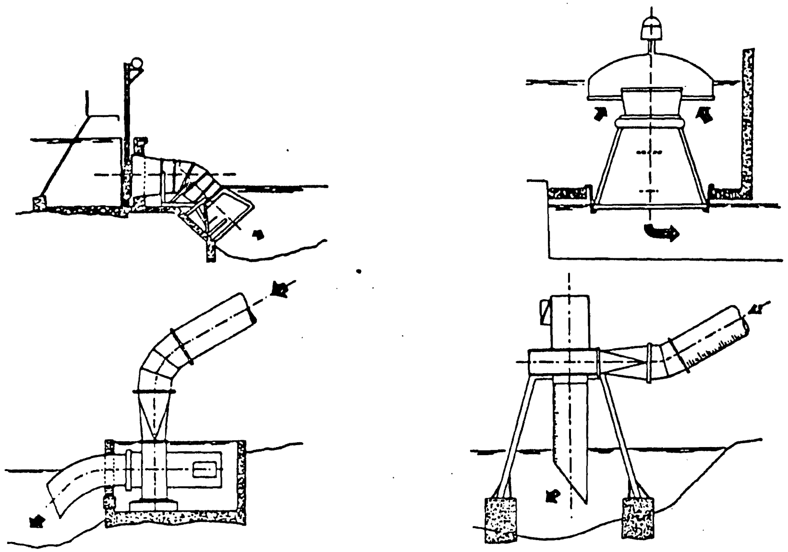


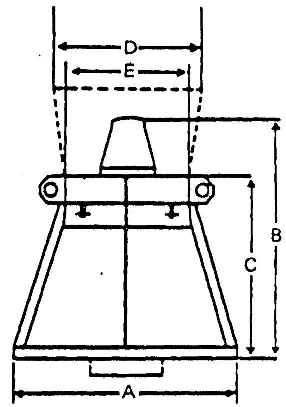
Figure 14.7 Arrangements for installing propeller turbines for microhydro installations. SOURCE: Moteurs Leroy-Somer.

TABLE 14.2 General Range of Use for Different Models of the (Hydrolec TM) Propeller-Type Turbine

Type	Flow (liters/sec)	Height (m)	Power (kW)
H4	150-400	2.5-5.50	2-12
H4H	160-400	6-10	5-22
H6-GH6	350-1100	1.5-5.50	3-34
H9-GH9	800-2300	1-4	4-51
HH-GH11	1000-2700	1-2.5	5-38
C30	60-200	3-14	1-15
C34	100-300	2-12	1-25
T15	1500-7200	1-4	10.5-220

SOURCE: Moteurs Leroy-Somer.

Propeller-type turbine (Allis-Chalmers Corporation). Figure 14.10 shows a selection nomograph for sizing plant installations in the microhydro and minihydro range. The nomograph is based on turbine installations where the runner centerline setting is one-half runner diameter above full-load tailwater level, the plant elevation is 500 ft (152 m) above mean sea level, and the water temperature does not exceed 80°F. Figure 14.11 shows a drawing of a typical installation and characteristic dimensions for various standardized unit sizes.



	A	B	C	D	E
H4	42	55	43	20	18
H6	48	55	43	32	25
H9	74	86	61	44	38
H11	79	86	61	47	43

Dimensions in inches

Figure 14.8 Characteristic dimensions for a standardized propeller turbine for microhydro installations. SOURCE: Moteurs Leroy-Somer.

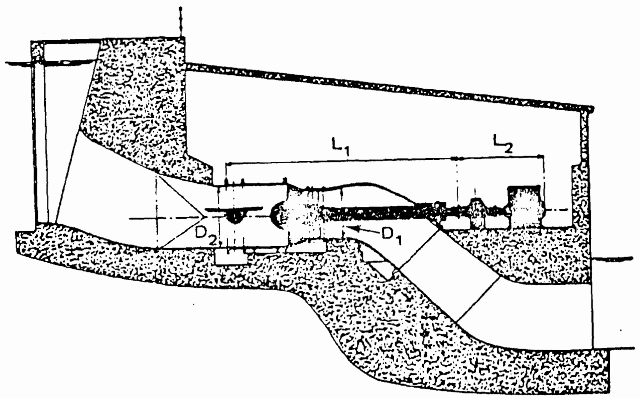
Propeller-type turbine (Voest-Alpine AG). Figure 14.12 shows a nomograph for selection of propeller or axial-flow turbines for microhydro and minihydro installations. This gives the diameter of the runner that will apply for a given unit and the range of applicable heads and turbine discharges that can be accommodated.

These particular company products are given as examples of information available including dimensions and conditions that must be known for preliminary design. Other companies produce similar units and a list of companies that furnish turbines in microhydro and minihydro range is included in Appendix A. For detailed aspects of selections and confirmation of sizing as well as to obtain estimates of costs, the manufacturers should be contacted and the latest information on performance and other characteristics obtained.

For preliminary selection on simple run-of-river installations, it is useful to consider an initial trial design capacity at the flow available 25% of the time. This will give an approximate power output that can be used for calculating energy benefits as a test of the economic viability of a project.

Simplified systems have been developed by various manufacturers to operate at variable flows. Frequently, electronic speed control equipment has been incorporated into units which are not connected to a utility system. Some very small units use dc generators and batteries to simplify the installation and reduce cost.

Figure 14.13 shows a simplified electrical diagram for the switching and con-



All measurements in millimeters

	D_1	D_2	L_1	L_2 max	min	L_3	L_4
SH 7	700	950	3700	2540	1260		
SH 9	900	1200	4700	2720	1610		
SH 11	1150	1550	6000	3120	1760		
SH 15	1500	2000	7700	3120	2250		
SV 7	700	950	4500	2920	1430	1480	1790
SV 9	900	1200	5700	3280	1770	1900	2300
SV 11	1150	1550	7300	3930	2220	2430	2940
SV 15	1500	2000	9600	3930	2420	3170	3840

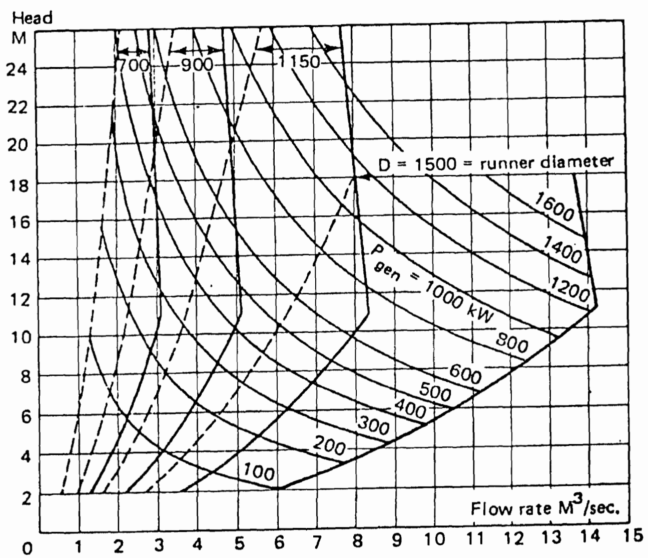


Figure 14.9 Selection chart for propeller turbines for minihydro installations. SOURCE: KMW, Sweden.

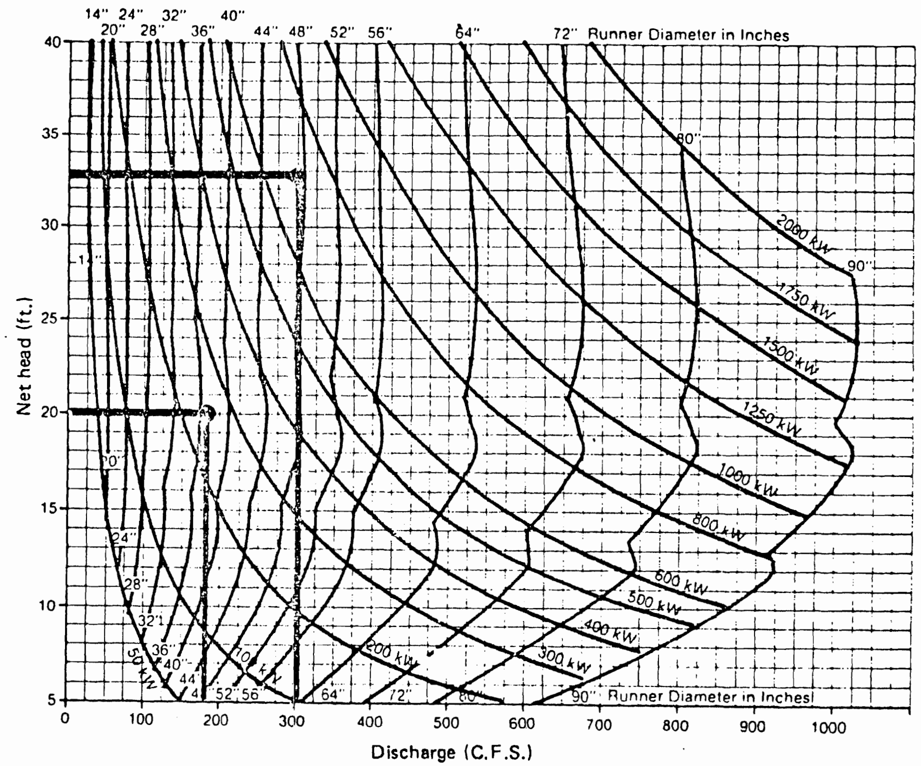
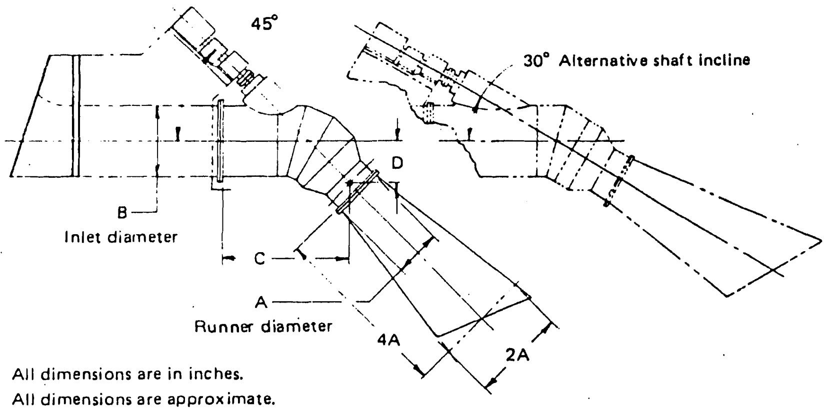


Figure 14.10 Nomograph for propeller turbines of tubular type for microhydro and minihydro installations. SOURCE: Allis-Chalmers Corporation.

nection of the electrical output that would be necessary at a microhydro or minihydro installation. Requirements for the electrical controls and safety protection for microhydro installations will vary depending on whether the unit is operated as an isolated station or whether it is connected to a distribution or transmission system that can maintain the frequency of current by being interconnected with other generating units.

PUMPS AS TURBINES

For small-capacity generating installations and particularly for microhydro and minihydro developments, it may be economically advantageous to use commercially available pumps and operate them in the reverse direction as turbines. If the flow is nearly constant, this may be particularly advantageous. Usually the flow must be at least 3% above the best efficiency flow for the pump to even operate as a turbine.



Basic dimensions for standardized tube turbine units

Dimensions	Runner size																	
A	12	14	16	18	20	24	28	32	36	40	44	48	52	56	64	72	80	90
B	18	24	24	30	30	36	42	48	54	60	66	72	78	84	96	108	120	132
C	32	38	44	47	50	58	65	80	87	95	103	110	118	125	143	158	173	192
D	11	13	14	16	18	21	25	28	32	35	39	42	46	49	56	63	70	79

Figure 14.11 Characteristic dimensions for standardized tubular-type minihydro installations. SOURCE: Allis-Chalmers Corporation.

Shafer and Agostinelli (1981) have studied the comparative performance of a pump operating as a pump and as a turbine. Their normalized performance curves are shown in Fig. 14.14. The work of Shafer and Agostinelli shows that for pumps operating as turbines the head and flow at the best efficiency point as a turbine are higher than pump head and flow at their pumping best efficiency point. The maximum efficiency tends to occur over a wider operating range of output capacity when the pump is operated as a turbine. As indicated earlier, there is a combination of head and flow when there is zero power output. Recently, renewed interest in the possibility of using pumps run in reverse to generate electricity has caused several pump companies to perform tests on pumps used as turbines.

C. P. Kittredge (1959) has given detailed procedures for estimating performance of centrifugal pumps when used as turbines. A pump performance curve is required for a constant-speed situation of a commercially available pump, information on the pump's operating speed, and the desired operating speed for turbine operation. Mayo and Whippen (1981) developed a selection diagram showing the application range of pumps used as turbines, based on the desired turbine output and the rated net head. This diagram is shown in Fig. 14.15.

Cavitation in pumps utilized as turbines has frequently not been fully investigated. Care should be taken to minimize cavitation when pumps are used as turbines by providing adequate back pressure.

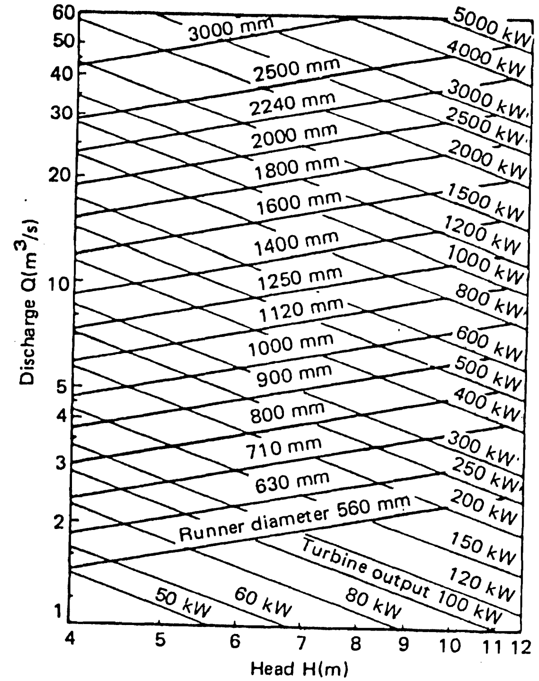


Figure 14.12 Nomograph for axial flow turbines for minihydro installations. SOURCE: Voest-Alpine, AG.

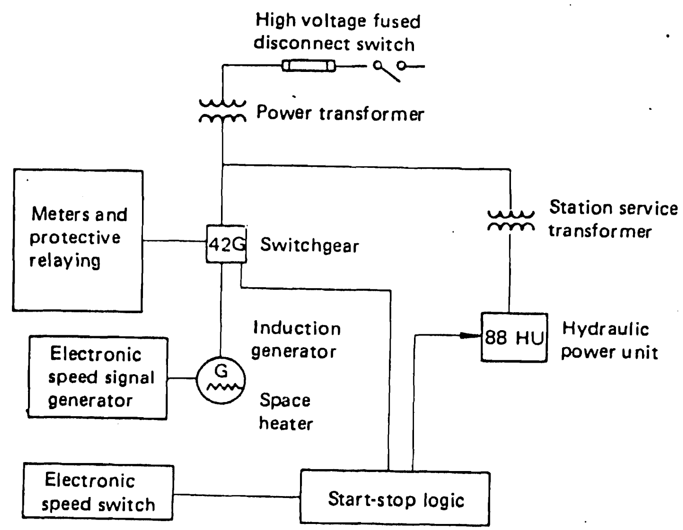


Figure 14.13 Simplified electrical diagram showing requirements for microhydro and minihydro installations. SOURCE: Allis-Chalmers Corporation.

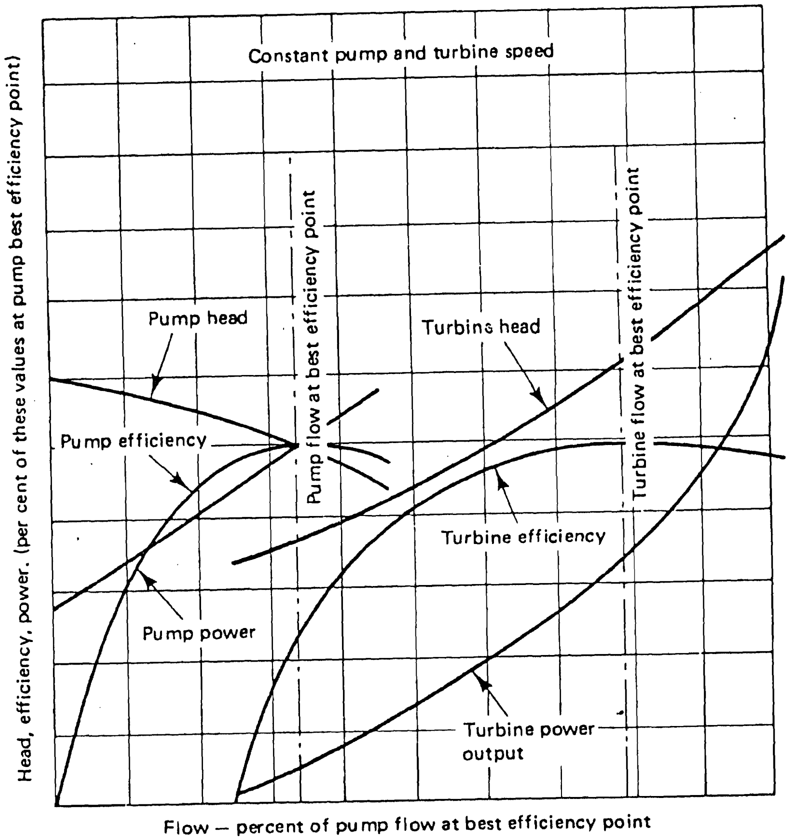


Figure 14.14 Normalized performance characteristics for a pump operating in the normal pump mode and in the turbine mode. SOURCE: Shafer & Agostinelli.

INSTITUTIONAL CONSIDERATIONS

Special attention needs to be given to the problems connected with legal and institutional considerations in the development of microhydro and minihydro installations. Recently, both federal and state governments have attempted to simplify the regulatory procedures for the developer. A useful guide is the *Microhydropower Handbook* recently published by the U.S. Department of Energy (1983). To proceed with this phase of investigation, contact should be made with the public utility commission in a particular state and with the state water resource agency, the state environmental quality agency, and the state historical society. Many states have special manuals for assisting developers. Typical is one for the state of Washington by Boese and Kelly (1981).

In the United States, federal regulations will require contacting the Federal Energy Regulatory Commission (FERC). The recent "Blue Book" publication by

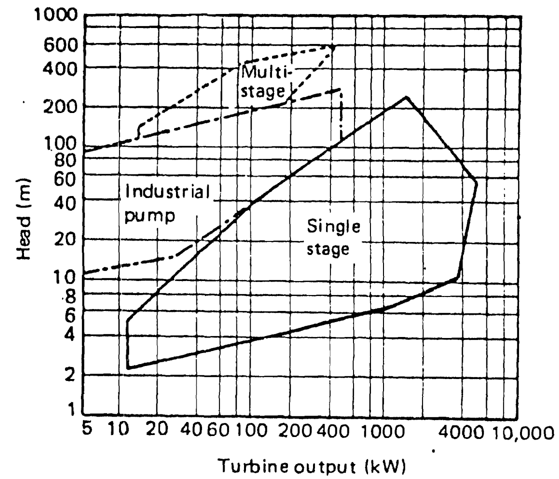


Figure 14.15 Nomograph showing application range for pumps used as turbines. SOURCE: Allis-Chalmers Corporation.

FERC (U.S. Department of Energy, 1982) defines the various requirements. The rules are being modified from time to time, so it will be necessary to check on the latest regulations. Requirements for connecting a microhydro or minihydro power plant to an electrical utility system are governed to a considerable extent by the rules of the state public utility commission, but construction of microhydro installations will also be subject to the individual requirements of the utility company, rural electric cooperative, or municipal system involved in purchasing and transmitting energy from the microhydro or minihydro installation.

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PROBLEMS

14.1. A microhydro power site has a net head of 15 ft (4.6 m) and the rated capacity is to be 70 kW. Choose two types of microhydro units that might be used and make a comparison of size and space requirements.

14.2. Recommend a turbine for an installation where the rated discharge is 250 ft^3/sec (7.08 m^3/sec) and the rated head is 13.1 ft (4.0 m). Give two different manufacturers' suggested sizes and indicate comparative advantages.

14.3. Check the diameter utilized in Table 14.1 for an operating situation of 200 ft of head and turbine speed of 1212 against the empirical equation suggested for sizing impulse turbines in Chapter 4.

14.4. Obtain flow data for a site where the head is essentially constant and proceed with preliminary selection of a microhydro or minihydro turbine unit using data presented in this chapter, Chapter 4, and Chapter 6. Make any required assumptions.

ENVIRONMENTAL, SOCIAL, AND POLITICAL FEASIBILITY 15

PRELIMINARY QUESTIONS

In assessing the feasibility of hydropower developments, it is important to consider early the social, political, and environmental feasibility at a proposed site or in a resource area that has potential sites. The purpose of such an evaluation is to determine whether there are restraints due to social concerns such as disruption of peoples' lives or the existing economy, institutional or legal restraints, and/or environmental concerns that will make proceeding with development unwise. Further, it is important to quantify the restraints to determine whether more time should be devoted to the study of social, political, or environmental acceptability and whether mitigation can be provided so that a hydroplant can be economically installed and operated.

Two questions need to be asked and answered. First, when should the evaluation be done? Second, who should make the evaluation? Assessment of social, political, and environmental feasibility should proceed concurrently with the hydrologic studies and inventorying of other pertinent physical data as well as in time sequence with the economic analysis. Necessary information to make an evaluation will often be incomplete and the evaluator will want to collect more information to make a better evaluation. Evaluators should be cautioned that collecting impact data can take several years in some cases. The decision maker may want and need to make a determination before the data collection can be completed. Who should do the evaluation? This is normally not a technological or engineering type of evaluation. However, the engineer is often responsible for this evaluation in the planning process. The engineer must depend on the judgment of professionally qualified people in the various disciplines involved, such as biologists, social scientists, and legal experts who have relevant experience qualifications.

These assessments of social, political, and environmental feasibility need to be made to screen various alternatives in certain political subdivisions, river basins, and government jurisdictions. The assessments, due to limits on time and funds, and the nature of the evaluations, often become subjective and depend on indexed representations of the various factors involved. Unlike the economic evaluation, there are no common units of measurement such as the dollar.

At present there is no established methodology that is universally accepted by planners and decision makers. The presentations in this chapter concentrate on methodologies that have been tried and on what various considerations should be included in making evaluations of social, political, and environmental feasibility.

CHECKLIST OF CONSIDERATIONS

In referring to the assessment of social, political, and environmental feasibility, the words used to refer to the variables in the appraisal include such words as factors, parameters, issues, and considerations. In this discussion, the word *considerations* will be used extensively. An example might be the effect a hydropower development might have on the migration activities of elk that inhabit the area. The consideration in this case would be an environmental impact and effect. Various means of arraying, classifying, and expressing the considerations are now in use. Important in the evaluation is first to develop a comprehensive checklist of the considerations that need to be assessed. This hopefully will ensure that none of the considerations will be overlooked. The degree of sophistication with which one weighs and determines the impact of hydropower development on various factors being considered will be quite site specific and depend on time and funding limitations. The following is a comprehensive checklist that might be used in developing and using methodologies that are referred to in later discussion.

I. Natural considerations
  A. Terrestrial
    1. Soils
    2. Landforms
    3. Seismic activity
  B. Hydrological
    1. Surface water levels
    2. Surface water quantities
    3. Surface water quality
    4. Groundwater levels
    5. Groundwater quantities
    6. Groundwater quality
  C. Biological
    1. Vegetation

    2. Fish and aquatic life
    3. Birds
    4. Terrestrial animals
  D. Atmospheric
    1. Air quality
    2. Air movement

II. Cultural and human considerations
  A. Social
    1. Scenic views and vistas
    2. Open-space qualities
    3. Historical and archaeological sites
    4. Rare and unique species
    5. Health and safety
    6. Ambient noise level
    7. Residential integrity
  B. Local economy
    1. Employment (short-term)
    2. Employment (long-term)
    3. Housing (short-term)
    4. Housing (long-term)
    5. Fiscal effects on local government
    6. Business activity
  C. Land use and land value
    1. Agricultural
    2. Residential
    3. Commercial
    4. Industrial
    5. Other (public domain, public areas)
  D. Infrastructure
    1. Transportation
    2. Utilities
    3. Waste disposal
    4. Government service
    5. Educational opportunity and facilities
  E. Recreation
    1. Hunting
    2. Fishing
    3. Boating
    4. Swimming
    5. Picknicking
    6. Hiking/biking

The specified environmental aspects that are to be weighed by federal agencies are listed in "Principles and Standards for Planning Water Resource and Related

Land Resources" (U.S. Water Resources Council, 1973). The checklist for evaluating water development projects in that document is presented below:

A. Protection, enhancement, or creation of natural areas
   1. Open and green space
   2. Wild and scenic rivers
   3. Lakes
   4. Beaches and shores
   5. Mountains and wilderness areas.
   6. Estuaries
   7. Other areas of natural beauty

B. Protection, enhancement, or preservation of archaeological, historical and geological resources
   1. Archaeological resources
   2. Historical resources
   3. Biological resources
   4. Geological resources
   5. Ecological systems

C. Protection, enhancement, or creation of selected quality aspects of water, land, and air
   1. Water quality
   2. Air quality
   3. Land quality

D. Protection and preservation of freedom of choice to future resource users

EVALUATION METHODOLOGIES

Numerous approaches have been used to systematize and quantify the assessment process. Two techniques are presented, an impact matrix and a factor profile approach.

A brief article by Carlisle and Lystra (1979) presents a discussion of the *impact matrix* approach. This technique requires the development of a matrix in which certain activities or actions are arrayed against the various considerations. If the environmental impact appraisal is very broad, it can include the social, political, and even economic issues that must be weighed. The actions or activities for planning, development, and operating a hydropower development are arrayed on the vertical scale of a matrix table and the various social, political, and environmental considerations are arrayed on the horizontal scale. Figure 15.1 shows how a comprehensive matrix might be developed.

The practice is to enter into the matrix table a symbol to indicate the extent to which a specific activity or subactivity will affect the particular consideration or subfactor. The entry can be qualitatively expressed in a scaling or rating approach such as Carlisle and Lystra did by assigning the following symbols:

- ● significant impact
- ◒ limited impact
- □ insignificant or no impact

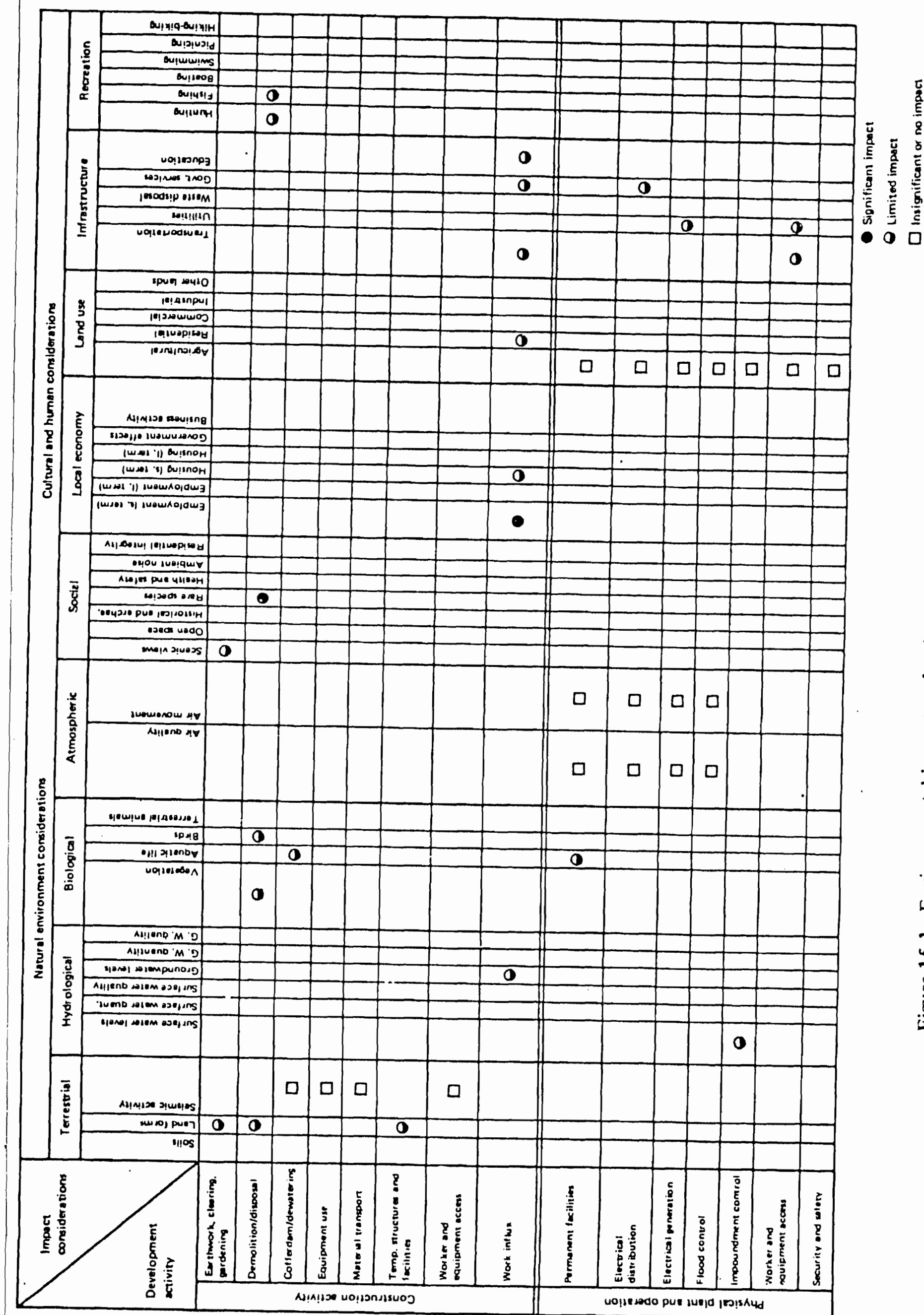

Figure 15.1 Environmental impact evaluation matrix for hydropower studies.

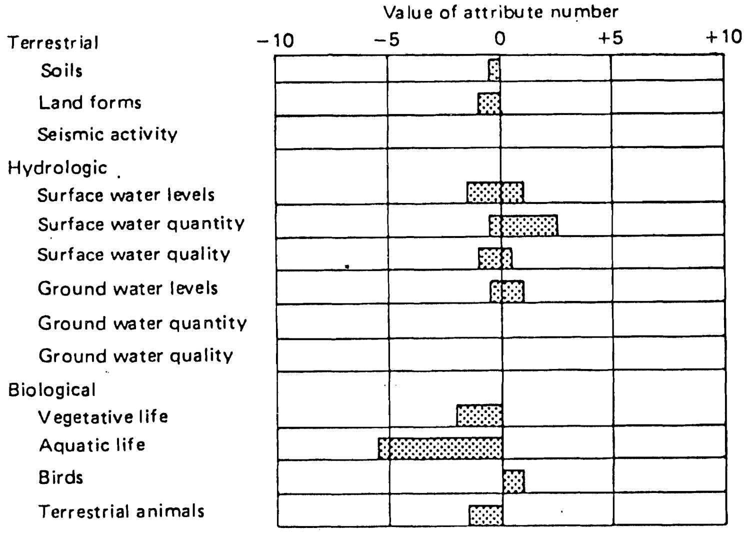

Figure 15.2 Example of factor profile for evaluating impact of hydropower development on environmental acceptability.

This implies that the evaluator has a good understanding of the base conditions as they exist or are expected to exist before construction and development proceeds. Naturally, this takes on a subjective weighing because it is not always easy to document why a particular entry was made. It implies a weighing of impact before and after development and even at stages during construction. Much literature on evaluation techniques of this type has been generated in connection with siting studies for nuclear and fossil fuel power plants. The author published a summary of these techniques that is a comprehensive reference to over twenty different studies that treated the subject of power plant siting evaluation (Warnick, 1976). Another good reference on power plant siting methodology is an exchange bibliography prepared by Hamilton (1977).

Another technique that has been used in siting highways (Oglesby, Bishop, and Willike, 1970), in a water resource planning effort (Bishop, 1972), and in an appraisal of recreational water bodies (Milligan and Warnick, 1973) is a *factor profile analysis.* This is a graphical representation of subjective scaling of the impact or importance of various considerations on the overall feasibility of development. Feasibility should be considered from four principal areas of concern: (1) engineering and technological feasibility, (2) social acceptability, (3) environmental acceptability, and (4) economic feasibility. Each of these areas of concern has subfactors as indicated earlier in developing the checklist of considerations for evaluating social, political, and environmental feasibility. For purposes of illustration, the environmental or natural considerations area in narrower context is chosen to explain the technique of factor profile analysis. Figure 15.2 arrays the considerations

for environmental evaluation in just three main categories and thirteen subfactors. In this case the atmospheric consideration has been dropped from the checklist of important considerations because rarely does a hydropower development affect air quality and air movement. In Fig. 15.2, a bar graph has been developed for each of the subfactors of the major considerations. This requires the subjective scaling of the impact the hydropower development will have either during construction or during operation, or both. A magnitude representation from 0 to -10 and 0 to +10 must be made of each of the subfactors in the factor profile. This scaling is here referred to as an *attribute number.* Note that it can be either negative or positive, or both. For instance, a hydropower development might disrupt fish habitat by decreasing flows during certain times and cause a valuation of a negative entry in the factor profile. At the same time the flow release might improve the flows at other times, making a positive entry on the factor profile. Guidelines and ways of consistently arriving at the attribute number is the challenging problem. Here is where it is important to call on the help of professionals to develop the guidelines for scaling the attribute number and actually making the assessment.

To illustrate the technique more fully, a factor profile for just one category of the cultural and human considerations has been developed and presented in Fig. 15.3. This is the social category with seven subfactors. Guidelines for assigning numerical value for the attribute numbers of two of the considerations are given.

For scenic views:

1. If a major scenic vista or attraction such as waterfall would be inundated and destroyed, assign a -10.
2. If a white-water cascading reach of stream would be inundated, assign a -7.
3. If the attractive stream bank vegetation will be partially destroyed, assign a -5.
4. If there appears to be negligible effect, assign a 0.
5. If a barren, ravaged stream channel is replaced with a mirrored lake, assign a +4.

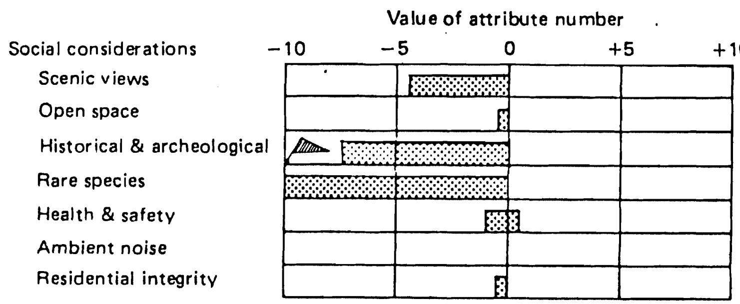

Figure 15.3 Example factor profile for evaluating impact of hydropower development on social conditions.

For open-space qualities:

1. If several thousand acres of open space is inundated and penstocks and canals cross and mar the open nature of the area, assign -10.
2. If a large area of open space is inundated, assign a -7.
3. If a limited area of open space is disrupted, assign a -3.
4. If no apparent change will occur in the open-space area, assign a 0.
5. If impoundment and control of stream allows use of open space and new vegetation creates a more open and attractive area, assign a +5.

It should be noted that a little flag symbol has been placed opposite the line for historical and archaeological sites. This expression could graphically signify that the evaluator is unwilling to make a trade-off, meaning that the impact is so negative that development should not be considered under present standards of environ-

TABLE 15.1 Rating and Scaling Form for Making an Environmental Impact Evaluation of Hydropower Development

SOCIAL IMPACTS

1. Rate the magnitude of social disruption likely to be caused by the proposal.

0	1	2	3	4	5	6	7	8	9	10
Major disruption					Moderate disruption					No disruption

2. Indicate the approximate number of people who will be directly affected in an adverse way by implementation of the proposal.

0	1	2	3	4	5	6	7	8	9	10
More than 10,000					Approximately 5,000					Very few

3. Rate the magnitude of adverse social impacts the proposal will have on areas of major national concern.

0	1	2	3	4	5	6	7	8	9	10
Major adverse social impacts					Moderate adverse social impacts					No adverse social impacts

4. Overall social assessment rating.

0	1	2	3	4	5	6	7	8	9	10
Major adverse social impacts					Moderate adverse social impacts					No adverse social impacts

5. Briefly describe any special social aspects of the proposal which were particularly influential in arriving at your judgments of the adverse social impacts. (Continue on the other side of this sheet if necessary.)

SOURCE: U.S. Department of the Interior (1977).

mental concern. More or less detail can be gained in such an evaluation by creating either more or fewer subfactors.

An example of the form actually used by the U.S. Bureau of Reclamation in the *Western Energy Expansion Study* (U.S. Department of the Interior, 1977) is presented in Table 15.1 to illustrate a similar evaluation technique.

In choosing the considerations to be used in the evaluation, a key element is to choose items or parameters which are as independent as possible. In reality it is likely that one will never get an array of considerations that are completely independent. However, the value of this type of appraisal is the focusing on systematic checking of the concerns that should be environmentally addressed.

The factor profile can give a visual representation of restraints. If desired, it is possible to sum the various values of the attribute numbers. It is also possible to give added weight to certain of the considerations by giving a weighting factor to a given consideration or subfactor. A good treatment on detail on how this might be done is presented in the literature for selection of nuclear power plants (Beers, 1974) and the studies of the author on power plant siting (Warnick, 1976).

OTHER SOCIAL AND POLITICAL CONSIDERATIONS

Land Ownership

In hydropower feasibility studies, land ownership is an important consideration. In many cases the site with the best development potential presents a problem because the entity that wants to develop the energy does not have ownership of the land, the land is in government ownership, or there are certain legal restraints on the land. Frequently, condemnation action must be taken to bring proper title to the plant site and to the flowage and impoundment areas. Land ownership problems need early attention in planning and may take on an inordinate importance in the feasibility determination and the implementation of a hydropower development.

Legal Considerations

Three legal considerations are important in the appraisal of social and political feasibility of hydropower developments: water rights, state regulatory permits, and federal licensing. Water rights are required on all hydropower developments in the United States. These are administered through state statutes. The procedures and legal approach varies greatly from state to state. The usual approach is to file as early as possible because this can prevent competitive developers from proceeding.

Depending on the state involved, there are other legal requirements that must be met and require attention even at the feasibility study level. Typical of these requirements are stream channel alteration permits, public utility certificates for study of need and convenience, state environmental impact statements, and proof of compliance with state water quality standards. Because of the direct impact of hydropower developments on the stream's fishery resources, there are always requirements and political acceptance that must be sought from the state departments of fish and game. These problems must be addressed as the planning proceeds. Variation from state to state of the particular requirements makes it difficult to give detail on how to proceed with the evaluations. A good place to start might be by conferring with the public utilities commission. Good examples of the various requirements and various state agencies involved are contained in a report to the U.S. Department of Energy by the Energy Law Institute (1980). This contains flow diagrams of several New England states' procedures.

In the federal realm of political and institutional needs for feasibility study and for proceeding to design and implementation, the key agency is the Federal Energy Regulatory Commission (FERC), formerly known as the Federal Power Commission. This commission is authorized to issue licenses for nonfederal development of hydropower projects that (1) occupy in whole or in part lands of the United States, (2) are located on navigable waters in the United States, (3) utilize surplus water or water power from government dams, and (4) affect the interests of interstate commerce. This in reality means that almost all streams of the country come under the jurisdiction of FERC.

Procedures for meeting the regulatory requirements of FERC fall into five main approaches. First, for those unsure of whether a license is necessary, a simple procedure is to obtain a jurisdictional finding of whether the proposed project requires a license and is under the jurisdictional perview of the commission (FERC). This is done by filing a declaration of intent pursuant to Title 18 of the FERC regulations (U.S. Federal Code of Regulation, current year).

The second approach that is applicable at existing dams is to apply for a categorical exemption if the site meets the necessary qualifications. A third approach is an application for exemption on a case-by-case basis where there is an existing dam or natural water feature. A fourth approach is also an exemption application where a power plant is to be developed on a conduit. This is termed a conduit exemption. The fifth approach is to proceed with a formal application. On projects over 5 MW this is required at all times. Figure 15.4 is a flow diagram showing the five options available in proceeding with meeting the FERC requirements. In the diagram a brief statement is presented on the action required and the applicable rule. These FERC requirements will change with time, so a check should be made of the current U.S. Federal Code of Regulation. Addresses for contacting FERC are listed in Appendix B.

An additional procedure that is possible is to file for a preliminary permit with FERC. This permit protects the entity during the investigation phase of the study and provides a priority advantage in obtaining a fully approved license. Figure 15.5 presents a diagram of the preliminary permit process. However, there are problems associated with this. Under the Federal Power Act, public entites are given preference to power sites, provided that they have filed an equal application or can revise an application to make it equal to one filed by a nonpublic entity.

The process for filing a formal license requires considerable information and a systematic review and hearing process. Details are covered in the FERC regulations.

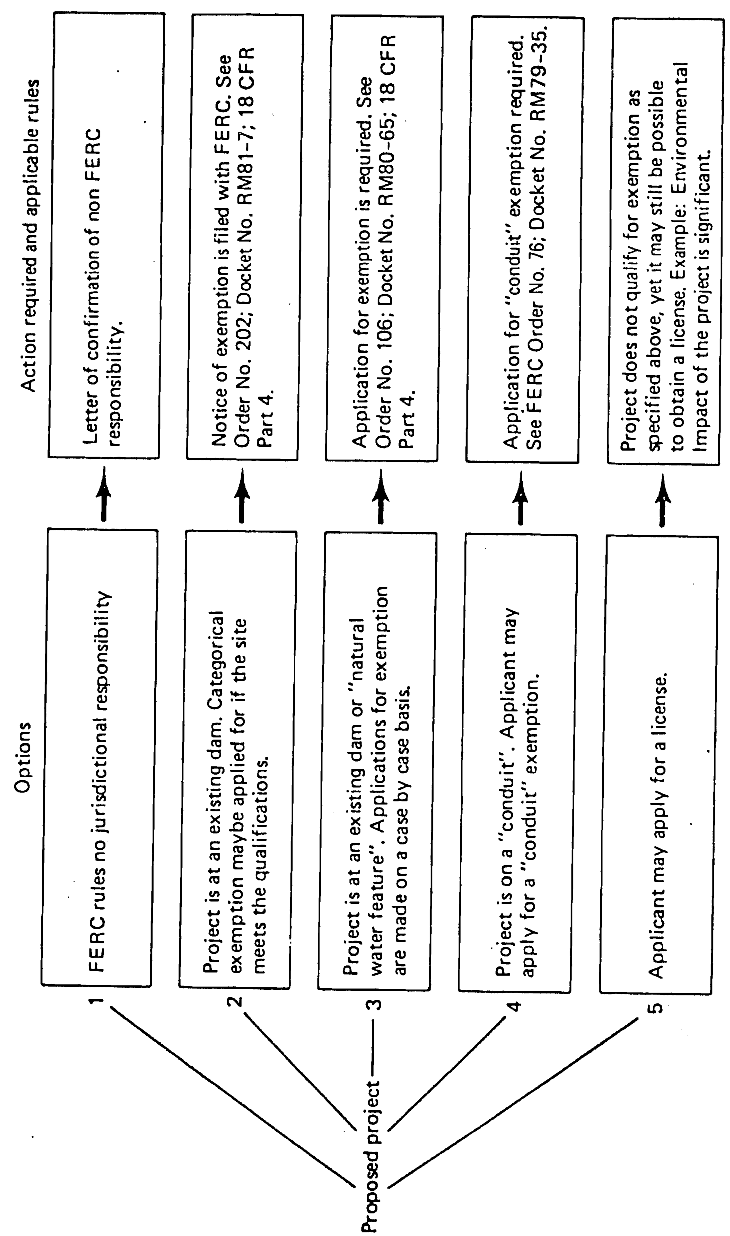

Figure 15.4 FERC regulatory options available.

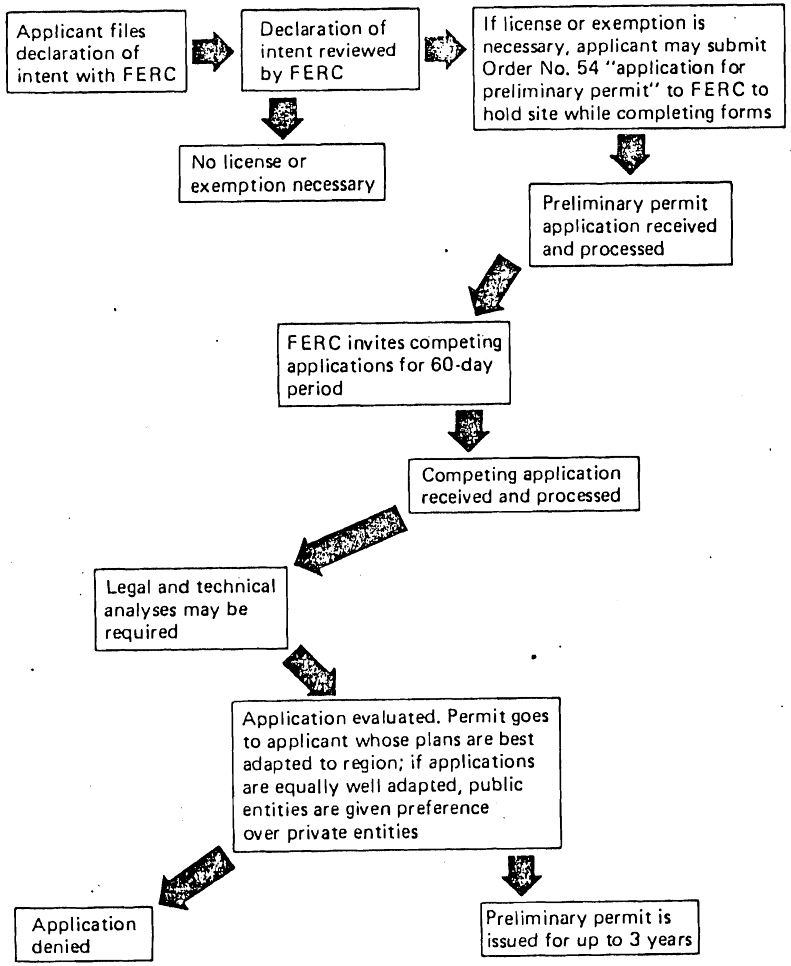

Figure 15.5 Diagram of the preliminary permit process.

The review by the commission and any hearing of that body includes a determination of the following: (1) adequacy of design, (2) economic feasibility, (3) environmental impacts, (4) financial capability of the applicants, (5) availability of power market, (6) dam safety, (7) project's adaptability to comprehensive development of the river basin, (8) potential for federal development, (9) water rights, and (10) other pertinent matters. Under those requirements for environmental impact evaluation and the part concerned with adaptability to comprehensive development of the river basin, the study must address various concerns that are jurisdictionally the responsibility of other federal agencies. This is above and beyond the state requirements already mentioned. Important in this regard is a series of important federal acts that affect the licensing process and influence how the social, political,

and environmental feasibility is evaluated. For purposes of reference these laws are listed with an appropriate public law and federal statute number:

Fish and Wildlife Coordination Act	(PL85-624) 72STAT.563
Wilderness Act	(PL88-577) 78STAT.890
Water Resources Planning Act	(PL89-80) 79STAT.244
Historic Properties Preservation Act	(PL89-665) 80STAT.915
Wild and Scenic Rivers Act	(PL90-542) 82STAT.906
National Environmental Policy Act	(PL91-190) 83STAT.856
Water Quality Improvement Act	(PL91-224) 86STAT.91
Federal Water Pollution Amendments Act	(PL92-500) 86STAT.816
Endangered Species Act	(PL93-205) 87STAT.884
Coastal Zone Management Act	(PL93-612) 88STAT.1974
Federal Land Policy and Management Act	(PL94-579) 90STAT.2743
Public Utility Regulatory Policies Act	(PL95-617) 92STAT.3117

Familiarity with these statutes is needed to proceed with feasibility analysis. As an example, if one is involved in the study of a river that is reserved under the National Wild and Scenic Rivers system as covered by the Wild and Scenic Rivers Act (PL90-542), it may be futile to try to proceed with development or even try for licensing because of the restraints placed on development by the act.

One should recognize that for study and development by a federal agency, the Federal Energy Regulatory Commission does not have jurisdictional responsibility. This normally falls in the realm of the U.S. Corps of Engineers and the U.S. Bureau of Reclamation. Addresses for contacting these agencies are listed in Appendix B. The procedures to be followed are covered in the Federal Principles and Standards for Planning Water and Land Resources (U.S. Water Resources Council, 1973) and revisions to those standards. In addition, each federal agency has developed internal procedures that cover the various phases of feasibility analysis. More recently, the U.S. Department of Energy has sponsored under the Public Utility Regulatory Policies Act (PURPA) and the White House Rural Development Initiatives a program to make loans for feasibility studies. These are required to make certain assessments of the social, political, and environmental feasibility. In general, those requirements are the same as those that are being administered by the Federal Energy Regulatory Commission.

With all these many requirements it is easy to see why development is sometimes slow and expensive on the front-end study effort. Recently, provisions have been instituted that allow for exemption to obtaining a license for smaller developments. However, this does require filing with FERC for the exemption, as indicated in Fig. 15.4.

One of the challenging problems facing the engineer is responding to these institutional problems and helping to decide whether the development should be made by private entities, private utilities, rural electric cooperatives, public power entities, state governments, or the federal government. In many cases the social, political, and environmental feasibility will depend on which type of entity gains the opportunity to proceed with study, design, and development.

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PROBLEMS

15.1. Prepare a checklist of considerations for evaluating social, political, and environmental feasibility in your area of the country.

15.2. Using Fig. 15.1 or a matrix of your own making, evaluate a proposed hydroelectric site in your vicinity.

15.3. Using the factor profile methodology, develop guidelines for scaling the attribute number on at least five subfactors.

15.4. Report on regulatory requirements that need to be considered in siting a hydroelectric development in your state. Identify appropriate statutes and agency regulations.

15.5. Choose two of the federal acts that are listed in the text as having importance in the study of social, political, and environmental feasibility and report on what restraints are involved. Be sure to document from the law the appropriate parts of the act and the legal reference for finding the information.

APPENDIX A

TURBINE MANUFACTURERS

Allis-Chalmers Corporation
Hydro-Turbine Division
P.O. Box 712
York, PA 17405

Ateliers Bouvier
53 Rue Pierre-Semand
3800 Grenoble
France

Barber Hydraulic Turbines Ltd.
Barber Point, P. O. Box 340
Port Colborne, ON L3K 5N1
Canada

Brown Boveri Corporation
North Brunswick, NJ 08902

Canyon Industries
5342 Mosquito Lake Road
Deming, WA 98224

Dependable Turbines Ltd.
No. 7, 3005 Murray Street
Port Moody, BC V3H 1X3
Canada

Dominion Engineering Works Ltd.
P.O. Box 220
Station A
Montreal, PQ H3C 2S5
Canada

Camille Dumont
Usines DuPont
DeSaint-Uze (Drôme)
France

Escher Wyss
CH 8023 Zurich
Switzerland

Essex Turbine Company
Kettle Cove Industrial Park
Magnolia, MA 01930

Gilbert Gilkes & Gordon Ltd.
Kendal, Cumbria LA9 7BZ
England

Hydroart S.p.A.
Via Stendhal 34
20144 Milan
Italy

Hydro Watt Systems, Inc.
146 Siglun Road
Coos Bay, OR 97420

Independent Power Developers
Rt. 3, Box 285
Sandpoint, ID 83864

KaMeWa
Kristinehamn Works
Box 1010-S
681 01 Kristinehamn
Sweden

The James Leffel & Company
425 East Street
Springfield, OH 45501

Little Spokane Hydroelectric
P.O. Box 82
Chattaroy, WA 99003

McKay Water Power, Inc.
1051 Clinton Street
Buffalo, NY 14206

Moteurs Leroy-Somer
Boulevard Maretlin-Leroy
BP 119
16004 Angoulême CEDEX
France

Neyrpic
Creusot-Loire Group
Rue Général-Mangin, BP 75,
Centre de Tri
38401 Grenoble CEDEX
France

Obermeyer Hydraulic Turbines Ltd.
10 Front Street
Collinsville, CT 06022

Ossberger Turbinan Fabrik
Iveissenberg
Postfach 425
D-8832 Bayern
West Germany

Skandia Axial Flow Turbines
Hydro Engineering
P.O. Box 10667
Winslow, WA 98110

Small Hydroelectric Systems & Equipment
5141 Wickersham
Acme, WA 98220

Sorumsand Verstad A/S
Kvaener Group
N-1920 Sorumsand
Norway

Vevey Engineering Works, Ltd.
CH-1800 Vevey
Switzerland

Voest-Alpine AG
Postfach 2
A-4010 Linz
Austria

J. M. Voith GmbH
Postfach 1940
D7920 Heidenheim
West Germany

APPENDIX B

PRINCIPAL U.S. GOVERNMENT AGENCIES CONCERNED WITH HYDROPOWER

Federal Energy Regulatory Commission
Office of Electric Power Regulation
825 North Capitol Street N.E.
Washington, D.C.

Regional Offices (address Regional Engineer, Federal Energy Regulatory Commission)

730 Peachtree Station
Atlanta, GA 30308

230 South Dearborn Street
Chicago, IL 60604

819 Taylor Street
Fort Worth, TX 76102

26 Federal Plaza
New York, NY 10278

333 Market Street
San Francisco, CA 94111

U.S. Army Corps of Engineers
Office of the Chief of Engineers
Washington, D.C.

U.S. Army Corps of Engineers
Hydrologic Engineering Center
609 Second Street
Davis, CA 95616

U.S. Army Corps of Engineers
The Institute for Water Resources
Kingman Building
Ft. Belvoir, VA 22060

U.S. Army Corps of Engineers
District offices (address the District Engineer, U.S. Army Engineering District)

Alaska
P.O. Box, 7002
Anchorage, AK 99510

Albuquerque
P.O. Box 1580
Albuquerque, NM 87103

Baltimore
P.O. Box 1715
Baltimore, MD 21203

Southwest Region
Commerce Building, Suite 201
714 South Tyler Street
Amarillo, TX 79101

Upper Missouri Region
Federal Office Building
316 North 26th Street
P.O. Box 2553
Billings, MT 59103

Lower Missouri Region
Building 20
Denver Federal Center
Denver, CO 80225

Engineering and Research Center
Building 67
Denver Federal Center
Denver, CO 80225

U.S. Environmental Protection Agency
Administrator
Waterside Mall,
4th and M Streets S.W.
Washington, D.C. 20460

Regional Offices (address Regional Administrator, E.P.A.)

Region I
2203 Kennedy Federal Building
Boston, MA 02203

Region II
26 Federal Plaza, Room 1009
New York, NY 10007

Region III
6th and Walnut Streets
Philadelphia, PA 19106

Region IV
345 Courtland Street, NE
Atlanta, GA 30365

Region V
230 South Dearborn Street
Chicago, IL 60604

Region VI
First International Building
1201 Elm Street
Dallas, TX 75270

Region VII
324 East 11th Street
Kansas City, MO 64106

Region VIII
1860 Lincoln Street, S
Denver, CO 80203

Region IX
215 Fremont Street

Corps of Engineers
logic Engineering Center
cond Street
CA 95616

Corps of Engineers
istitute for Water Resources
ian Building
lvoir, VA 22060

Corps of Engineers
:t offices (address the
strict Engineer, U.S. Army
igineering District)

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. Box, 7002
horage, AK 99510

uerque
. Box 1580
uquerque, NM 87103

iore
. Box 1715
imore, MD 21203

Buffalo
1776 Niagara Street
Buffalo, NY 14207

Charleston
P.O. Box 919
Charleston, SC 29402

Chicago
219 South Dearborn Street
Chicago, IL 60604

Detroit
P.O. Box 1027
Detroit, MI 48231

Ft. Worth
P.O. Box 17300
Ft. Worth, TX 76102

Galveston
P.O. Box 1229
Galveston, TX 77553

Honolulu
Building 230
Ft. Shaffer, Honolulu, HI

Huntington
P.O. Box 2127
Huntington, WV 25721

Jacksonville
P.O. Box 4970
Jacksonville, FL 32201

Kansas City
700 Federal Building
601 East 12th Street
Kansas City, MO 64106

Little Rock
P.O. Box 867
Little Rock, AR 72203

Los Angeles
P.O. Box 2711
Los Angeles, CA 90053

Louisville
P.O. Box 59
Louisville, KY 40201

Memphis
668 Federal Office Building
Memphis, TN 38103

Region VIII
1860 Lincoln Street, Suite 900
Denver, CO 80203

Region IX
215 Fremont Street
San Francisco, CA 94105

Region X
1200 6th Avenue
Seattle, WA 98101

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