SECTION 9.13
PUMPED STORAGE

GEORGE R. RICH
HOWARD A. MAYO, JR.

SIZE OF INSTALLATION

In the typical steam-based utility, it is the function of pumped storage to (1) furnish peaking capacity on the weekly load curve (Figure 1) and (2) generate full pumped storage in an emergency for from 10 to 15 h, as required by the particular system. On the basis of comparative cost estimates, the most economical size of installation is selected.

The principal features of a typical pumped storage project are shown schematically in Figure 2. The overall efficiency is about 2:3; that is, 3 kW of pumping power will yield 2 kW of peak generation. The economy of the process stems from the fact that dump energy for pumping is worth about 3 mills/kW · h, whereas peak energy is worth about 7 milli/kW · h. There is also an operating advantage. Because of the ease and rapidity with which it may be placed on-line and because of its low maintenance charges, pumped storage is ideally suited to peaking operation. On the other hand, for maximum economy, modern high-pressure, high-temperature thermal plants should operate continuously near full load on the base portion of the load curve.

Table 1 shows a typical calculation to determine the reservoir capacity needed for sustaining the weekly load curve. The reservoir capacity to carry full load for a 10- to 15-h emergency is obtained simply by equating the electrical energy in kilowatt-hours in the load to the potential hydraulic energy stored in the upper reservoir.

SELECTION OF UNITS

At this stage of the basic engineering, it is necessary to make (in collaboration with the equipment manufacturers) at least a tentative selection of capacity, diameter, speed, and submergence for the turbomachine. This will be required for refinement of the calculations

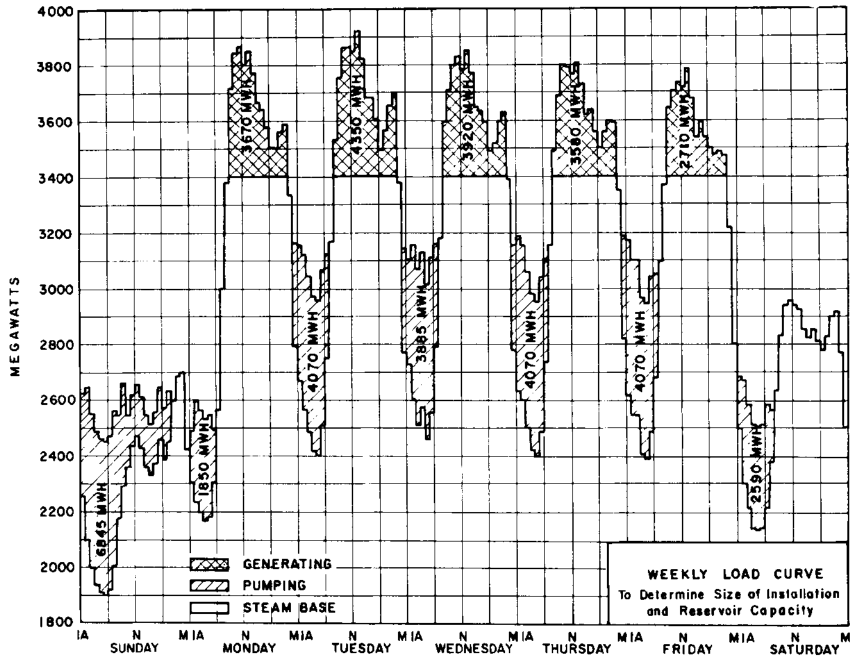

FIGURE 1 Weekly load curve

illustrated in Table 1 and also for the calculation of hydraulic transients that follows. In the head range most attractive for overall project economy, 500 to 1500 ft (152 to 457 m), manufacturers are prepared to offer a single turbomachine that is capable of operating as a pump and whose direction of rotation is opposite that of a turbine. Electrical manufacturers offer a similar machine capable of operating as a synchronous motor for pumping and, in the opposite direction of rotation, as a generator. These machines are designated *pump turbines* and *generator motors.*

For best economy, the speed of the unit should be as high as is practicable without involving an objectionable degree of cavitation of the impeller under the assumed submergence below minimum tailwater. This speed is established (1) by model tests for cavitation at the hydraulic laboratories of the manufacturers and (2) by evaluating experience with similar prototype installations.

Figure 3 is an experience chart showing specific speed $N_s = \text{rpm} \cdot \text{hp}^{1/2}/H^{5/4}$ versus head for the machine acting as a turbine, where H is total head), and Figure 4 shows the specific speed $N_s = \text{rpm}Q^{1/2}/H^{3/4}$ for the machine acting as a pump. These charts presuppose moderate values of submergence because unusually deep settings are uneconomical from the structural standpoint. Three curves are fitted to the installations shown, the equation of the curves being $N_s = K/H^{1/2}$. The depth of submergence may be verified by checking against the value of the cavitation constant $\sigma = (H_a - H_{vp} - H_s)/H$ given by the manufacturer's cavitation model test curves. Here H = total head, H_{vp} = vapor pressure, σ and H_a = atmospheric head. A typical curve of the family is shown in Figure 5.

When the unit has been selected, manufacturers will furnish (in advance of bid invitations) prototype performance curves similar to Figures 6, 7, 8a, and 8b. Figures 8a and 8b are designated four-quadrant synoptic charts and are required for the calculation of hydraulic transients. Figure 8b is for a 5.59-in (142-mm) gate opening. This is the largest gate opening at which the unit will be operating in the pumping cycle. Figure 8a, for an

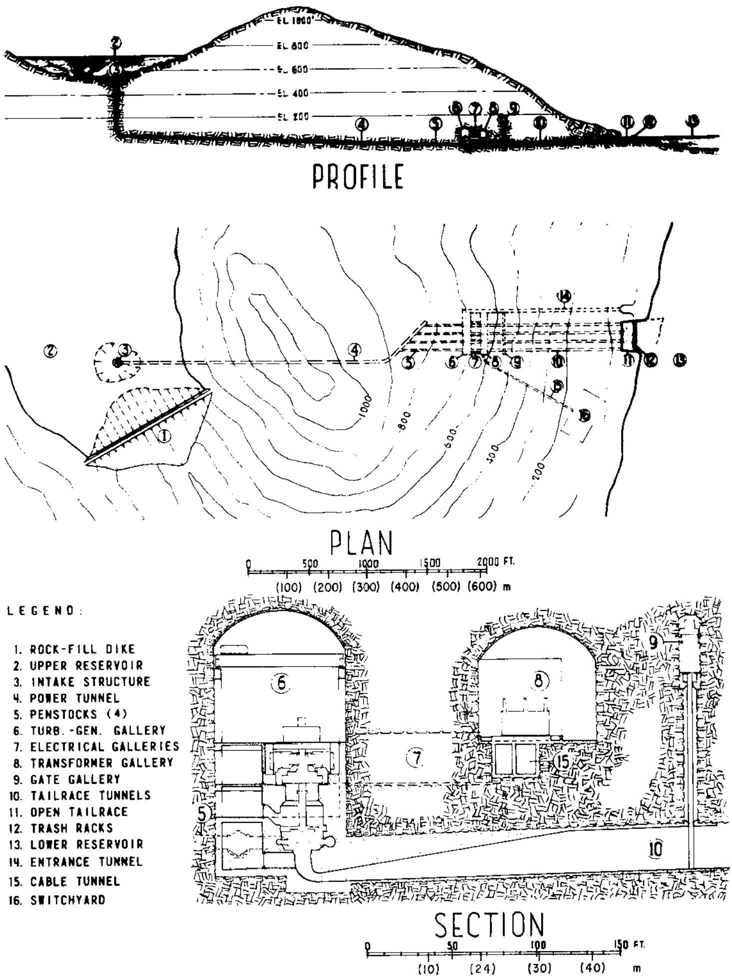

FIGURE 2 Schematic arrangement for a pumped storage project. Elevations are in feet (1 ft = 0.3048 m).

8.94-in (227-mm) gate opening, is for operation on the turbine cycle only. Figure 9 shows a schematic view of a pump-turbine with motor generator and starting motor.

HYDRAULIC TRANSIENTS

In preparing purchase specifications for the generator motor, it is necessary to establish the maximum transient speed and the moment of inertia of the rotor, WR^2 (GD^2). Similarly, for the pump turbine and penstock, it is necessary to determine the maximum water-hammer. This primary calculation, as summarized in Figure 10 and Table 2, is made by the trial-and-error method of arithmetic integration, using various trial values of WR^2 for the condition of full-load rejection on all units during the generating mode with the turbine gates assumed "stuck" in the full-gate position.

TABLE 1 Determination of reservoir capacity

		(1) Generating, MW	(2) Pumping, MW · h	(3) Equivalent generation (col. 2 × $\frac{2}{3}$), MW · h	(4) Net daily change in reservoir, MW · h	(5) Cumulative change in reservoir, MW · h	Remarks
Monday	PM	3670			−3670	−3670	
Tuesday	AM		4070	2710			
	PM	4350			−1640	−5310	
Wednesday	AM		3885	2585			
	PM	3920			−1335	−6645	
Thursday	AM		4070	2710			
	PM	3580			−870	−7515	Reservoir empty
Friday	AM		4070	2710			
	PM	2710			0	−7515	
Saturday			2590	1725	+1725	−5790	
Sunday			6845	4560	+4560	−1230	
Monday	AM		1850	1230	+ 1230	0	Reservoir full

Required reservoir capacity to sustain weekly load curve:

$$\frac{7515 \times 1000 \times 550 \times 3600}{62.4 \times 43{,}560 \times 0.746 \times 900 \times 0.85} = 9600 \text{ acre} \cdot \text{ft}$$

Hours of capacity at full load and 900 ft (274 m) head:

$$\frac{9600 \times 62.4 \times 43{,}560 \times 0.746 \times 900 \times 0.85}{525 \times 1000 \times 550 \times 3600} = \pm 14.3 \text{ h}$$

Pertinent Data
- Generating = rated net head = 900 ft (274 m)
  - Generating capacity at rated head for 3 units = 525 MW
- Pumping = rated net head = 930 ft (283 m)
  - Pumping power at rated head for 3 units = 555 MW
  - Pumping-to-generating ratio = 3 to 2

Upon instantaneous loss of load, the unit builds up overspeed. The increase in speed above normal causes a reduction in turbine discharge, which causes waterhammer, which in turn further increases the power delivered to the rotor. This pyramiding continues until the arrival of negative reflected water hammer from the upper reservoir. The head then decreases. The unit is then so much over speed that it begins to act as a brake, as shown by the four-quadrant synoptic chart (Figure 8). As shown by Figure 10 and Table 2, the process gradually damps down to the steady-state runaway speed and head. Many additional and more refined calculations are made later in the course of the design to establish the optimum governor time and rate of turbine gate closure, as given in detail in the works listed at the end of this section.

STARTING THE UNIT

The procedure for starting the unit is an essential feature of the design. There are three cases to be considered: (1) the pumping mode, (2) the conventional generating mode, and (3) rotating spinning reserve in the generating mode.

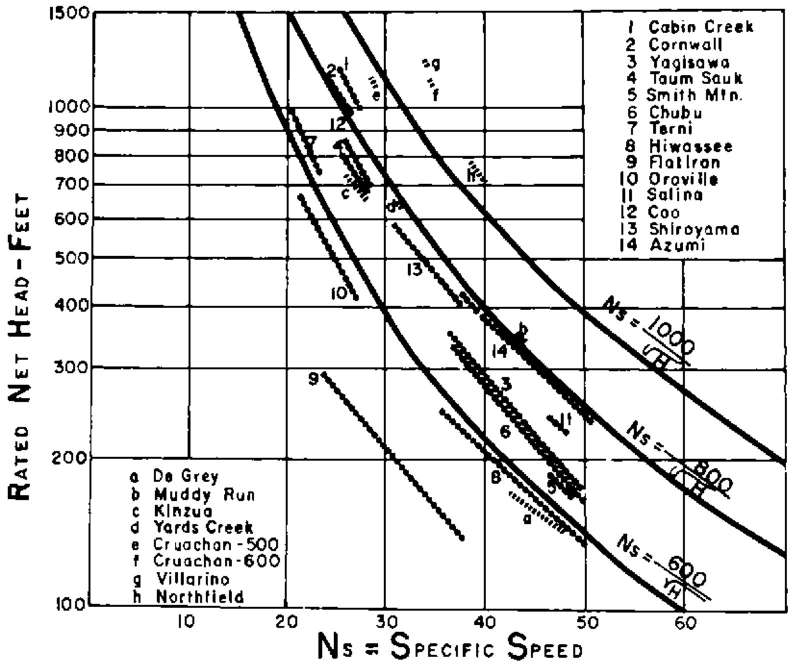

FIGURE 3 Rated net head versus specific speed for a turbine (1 ft = 0.3048 m; $N_s = 0.2622(\mathrm{kW})^{1/2}/\ H^{5/4}$). See Subsection 6.1.4 for the relation of N_s to the universal specific speed Ω_s.

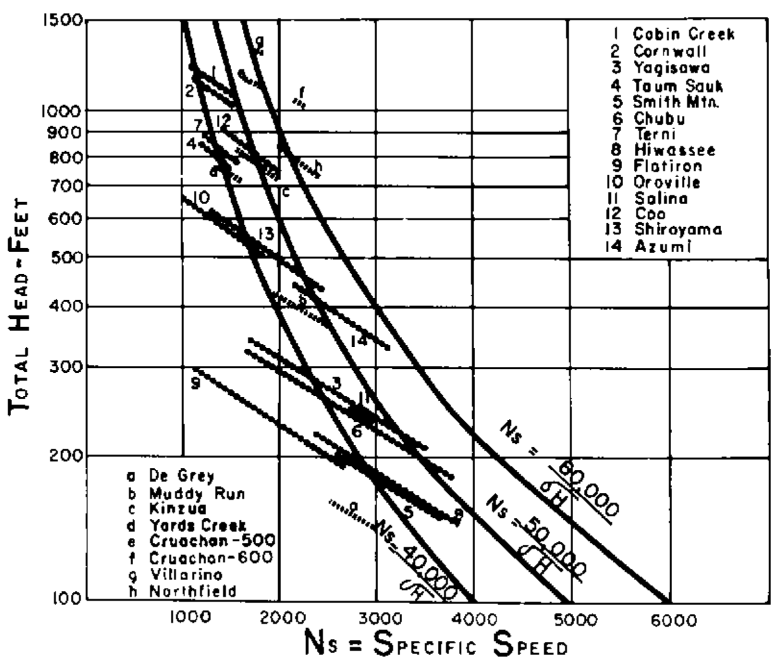

FIGURE 4 Total head versus specific speed for a pump (1 ft = 0.3048 m; $N_s = 51.65(\mathrm{rpm})\ \sqrt{\mathrm{m}^3/\mathrm{s}}/H^{3/4} = 2733 \times \Omega_s$, where the universal specific speed Ω_s is defined in Chapter 1 and Section 2.1).

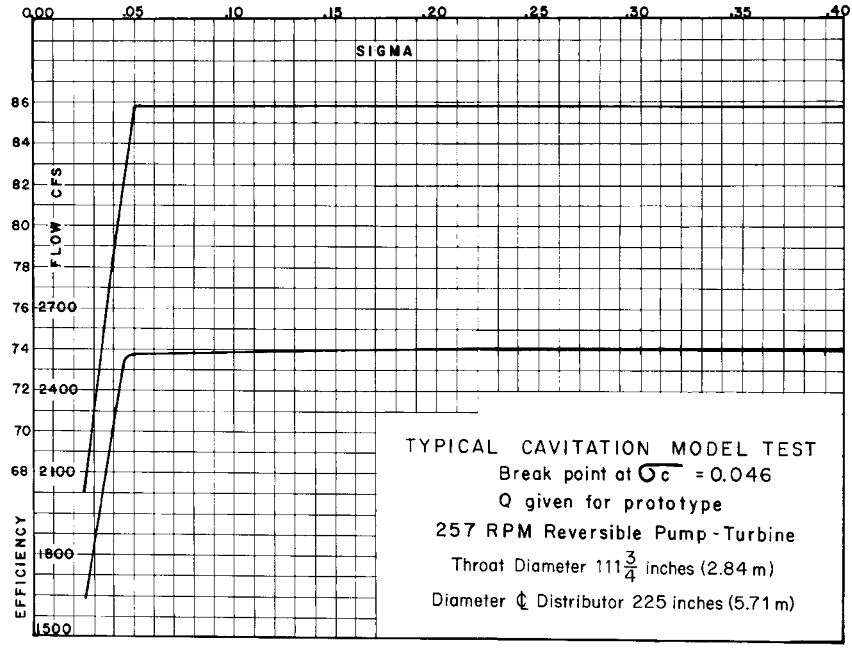

FIGURE 5 Typical cavitation model test (1 ft³/s = 0.0283 m³/s) (Voith Siemens Hydro)

Pumping Mode If we attempted to start the unit as a pump, from rest and with the spiral case and draft tube filled, the power and inrush current would be excessive. Accordingly, the standard type of compressed air system is provided to depress the tailwater elevation to below the bottom of the impeller, with the wicket gates closed. The load to be overcome at starting then consists of the load of the rotating masses, which must be accelerated to synchronous speed, and the load due to windage. Owing to inevitable leakage past the wicket gates, this windage is substantially greater than that due to dry air. The main penstock valves must also be closed during starting, or else the leakage and "wet" windage would be still further increased at the much higher head.

For the larger units generally employed, a separate starting motor, of the induction type with wound rotor, is mounted directly above the main generator motor. It has not been found feasible, in the motor space available, to design an amortisseur winding capable of sustaining the heat from the inrush current resulting from across-the-line starting of the main generator motor, even at reduced voltage. For the smaller units, however, this may be accomplished. In rare instances, where a main unit is always available, this spare may be electrically coupled to the pumping unit and the two started from rest in back-to-back synchronism. The separate starting motor is usually sized to bring the main unit up to synchronous speed in about 10 minutes, as shown by Figure 11.

When the unit has attained full speed, it is synchronized to the line, the compressed air is cut off, the tailwater rises to fill the draft tube, the wicket gates and main penstock valve are opened gradually to prevent shock, and pumping to the upper reservoir begins.

The maximum transient load on the generator motor thrust bearing occurs just as pumping begins. Prior to the advent of pumped storage, thrust bearings were designed to carry the weight of the rotating parts plus the hydraulic thrust at the steady-state condition. Now a greatly increased thrust of short duration must also be accommodated. Because of the short duration of this transient excess load, it may usually be carried safely by the bearing as designed for the steady-state requirement, depending on the detailed

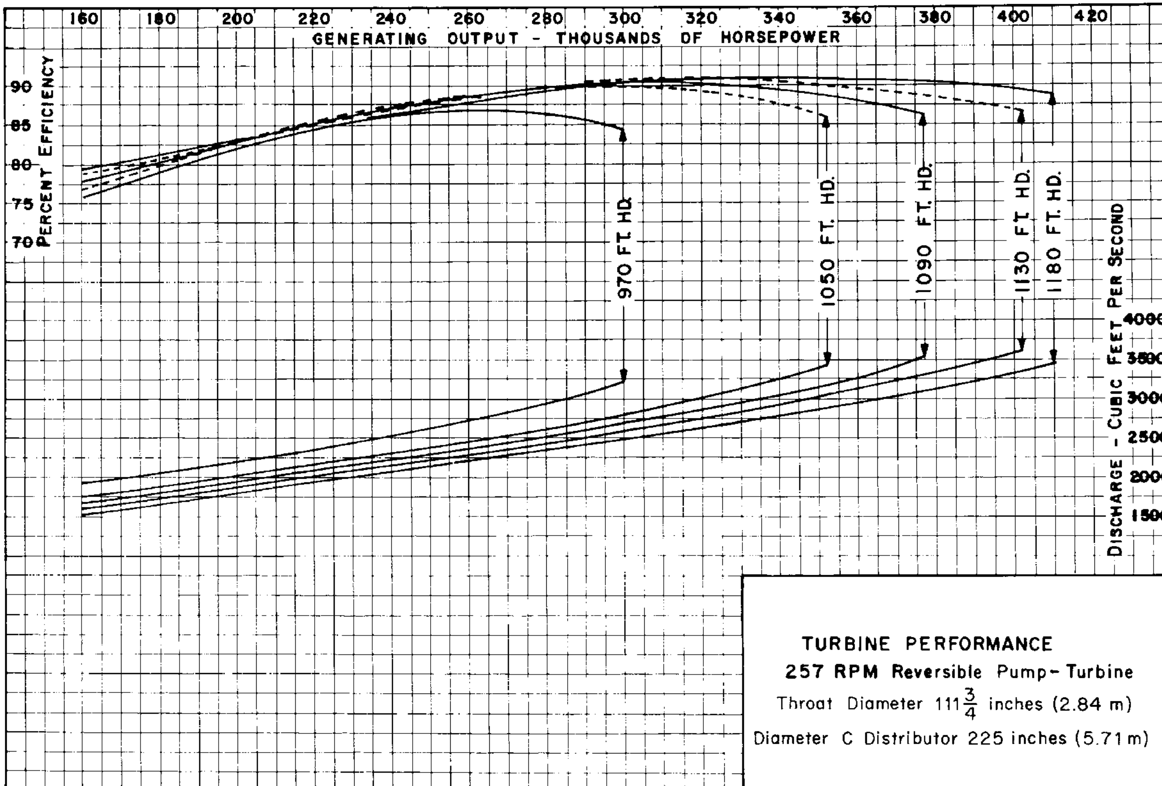

FIGURE 6 Turbine performance (1 ft^3/s 0.0283 m^3/s; 1 ft = 0.3048 m; 1 hp 0.746 kW) (Voith Siemens Hydro)

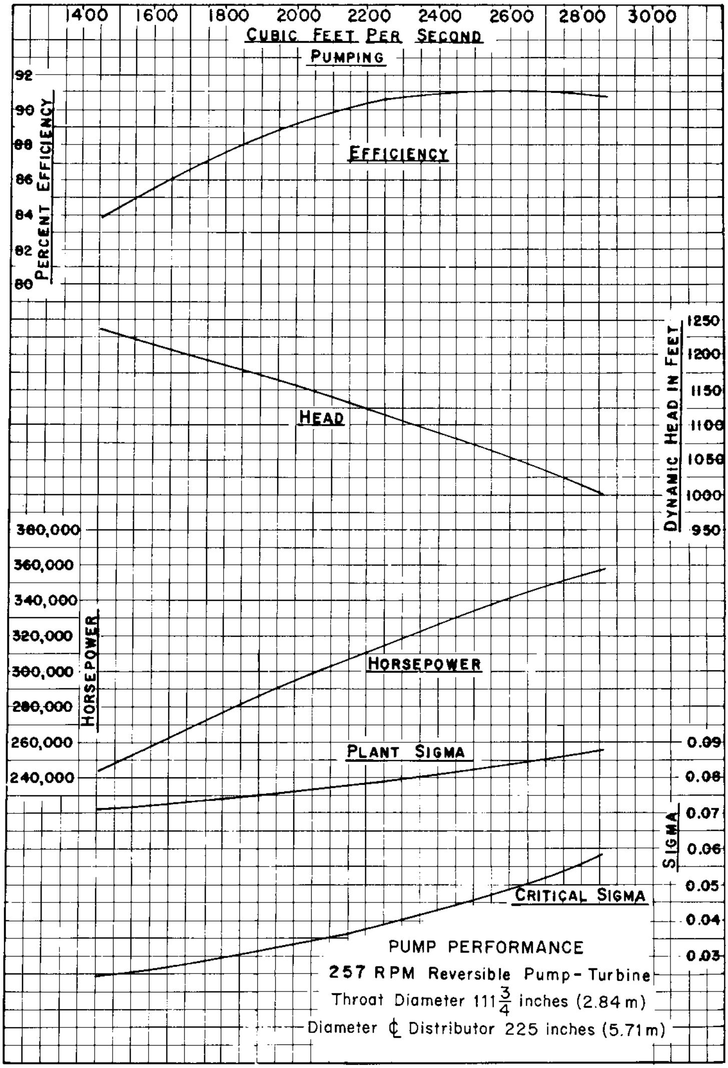

FIGURE 7 Pump performance (1 hp = 0.746 kW; 1 ft = 0.3048 m; 1 ft^3/s = 0.0283 m^3/s) (Voith Siemens Hydro)

design of the bearing. It is now standard practice to provide a high-pressure oil pumping system to ensure that there will be a film of oil between the bearing surfaces before the unit starts rotating.

Conventional Generation For generation in the conventional manner, the unit may be started from rest under its own power without assistance from the starting motor.

Rotating Spinning Reserve In considering the requirements for starting the unit for rotating spinning reserve, it will be assumed that the utility is a participant in a grid sys-

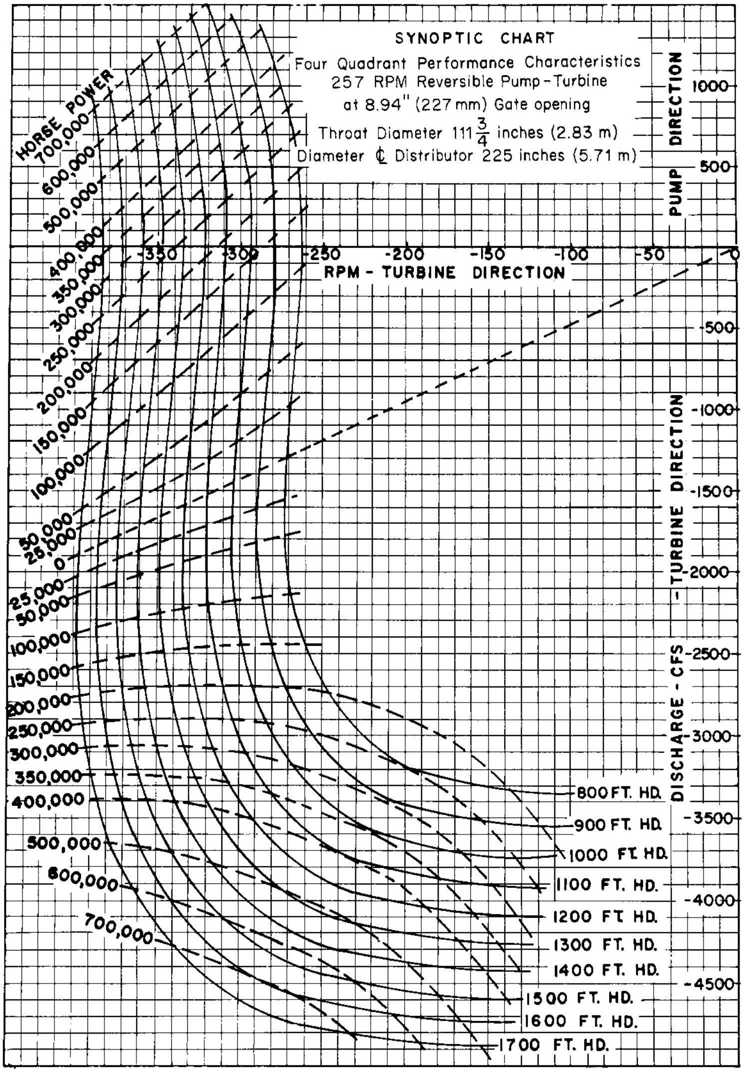

FIGURE 8A Synoptic chart for an 8.94-in (227-mm) opening (1 hp = 0.746 kW; 1 ft = 0.3048m; 1 ft^3/s = 0.0283 m^3/s) (Voith Siemens Hydro)

tem of interconnection by means of extra-high-voltage (EHV) transmission lines of high capability. This means that, immediately upon loss of generation by the local utility, the EHV connection will carry the necessary load for the short time required (about 30 s) for the local pumped storage units to absorb full load. In readiness for just such an emergency, these local pump turbines will be motoring on the line in the generating direction, with the wicket gates and main penstock valves closed and with the tailwater depressed. The loads and procedure for bringing the units up to speed for synchronizing to the line for rotating spinning reserve are identical with those given for operation in the pumping mode except that rotation is in the generating direction.

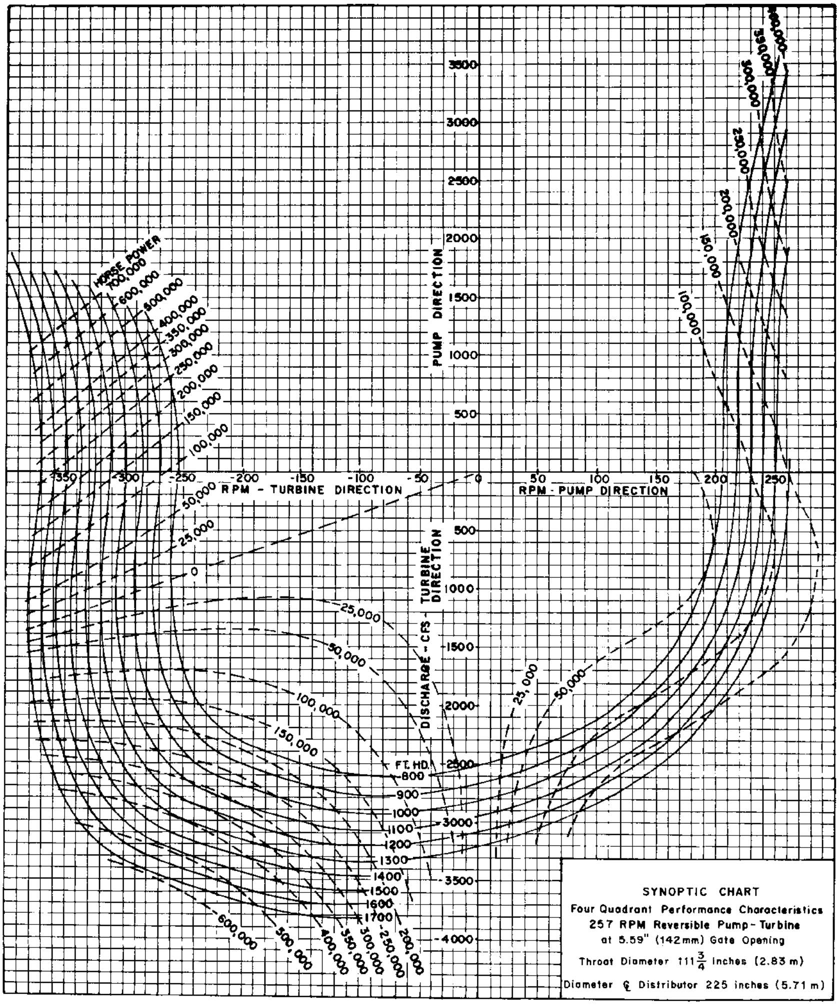

FIGURE 8B Synoptic chart for a 5.59-in (142-mm) opening (1 hp = 0.746 kW; 1 ft = 0.3048 m 1 ft^3/s; = 0.0283 m^3/s) (Voith Siemens Hydro)

GOVERNOR TIME AND SURGE TANKS

In pumped storage plants, the time specified for the governor servomotors to move the wicket gates through a complete stroke is generally not less than 30 s. The reasons for this are that (1) about 30 s is the minimum practicable time for penstock valve operation if the valve operating machinery is not to be made unduly complicated by the incorporation of dashpots and accessories and (2) one of the primary purposes of massive EHV connection is to carry loads from emergency outages for 30 to 60 s or more until the local pumped storage units take over. In the lengths of waterways that are permissible economically, penstock and tunnel velocities may usually be accelerated to full load in a much shorter time

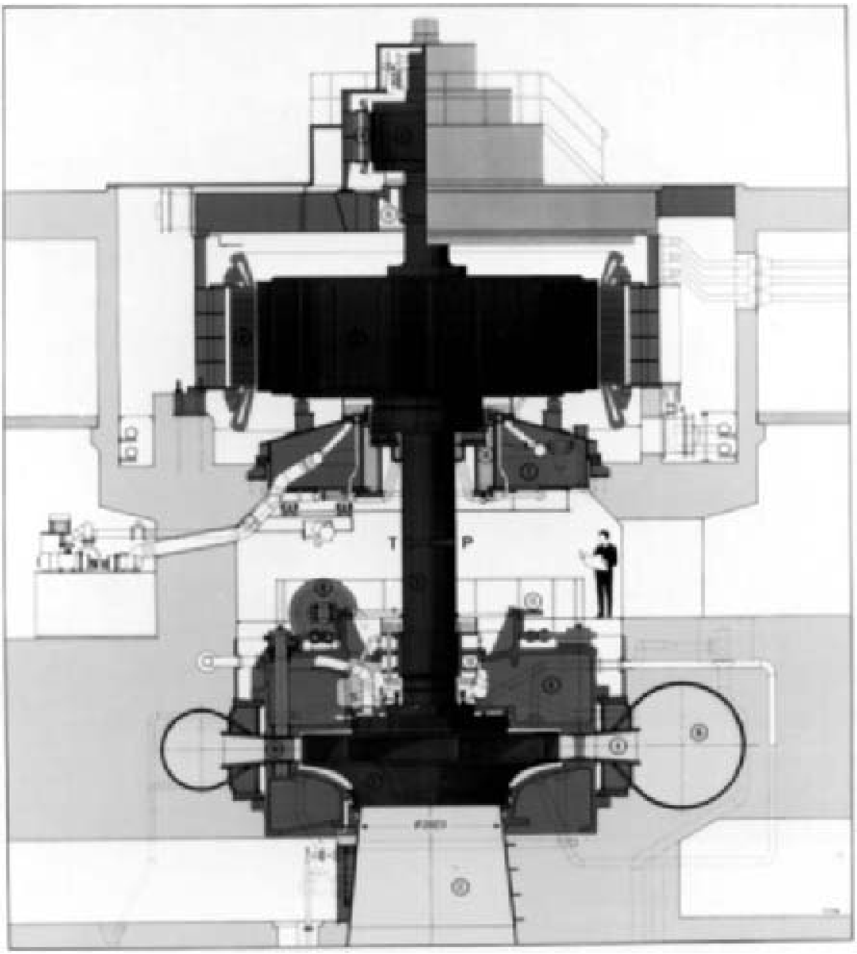

FIGURE 9 Pump-turbine with motor generator. Starting motor is located on top, above the motor-generator. The pump-turbine runner—surrounded by adjustable vanes, stay vanes, and spiral case—is located above the conical draft tube that has a minimum throat diameter at the runner eye of 2823 mm (111.1 in). In the turbine mode, flow is downward and out of the draft tube, and power generated is 210 MW at a head of 250 m (820 ft). In the pumping mode, the flow direction is reversed, and the motor input power is 198.1 MW at a head of 260 m (853 ft). Speed is 272.7 rpm in both modes (Voith Siemens Hydro)

without excessive positive or negative waterhammer, so surge tanks are not necessary. However, in the exceptional case of a long tailrace tunnel flowing as a closed conduit, under the relatively low head of tailwater, even a 30-s closure could be sufficient to produce negative waterhammer great enough to cause separation of the water column and damage to the unit. For such cases, a surge tank[1] at the downstream face of the power station is required. In a dual-purpose project for municipal water supply and by-product power, the length of the tunnel is dictated by water supply economics and may be as great as 40 to 50 miles (64 to 80.5 km). In such cases, a surge tank on the upstream side of the power station may be indicated.

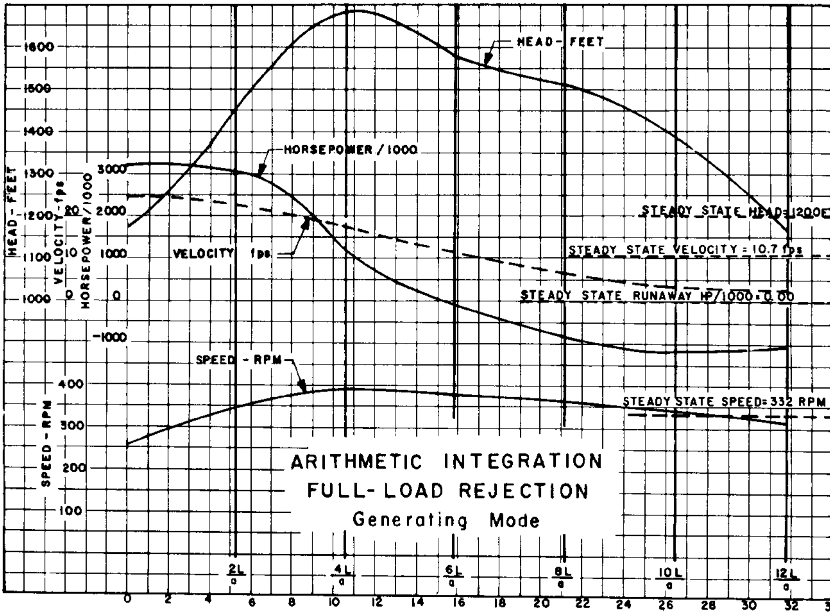

FIGURE 10 Arithmetic integration, full load rejection (1 hp = 0.746 kW; 1 ft = 0.3048 m; 1 ft/s = 0.3048 m/s; 1 ft^3/s = 0.0283 m^3/s) versus time in seconds.

ECONOMIC DESIGN

A proposed pumped storage project for peaking service must be able to show a liberal margin of economic superiority over competing thermal types. This requires that the undertaking be designed in accordance with the so-called lean project concept, in which all elements are closely tailored to the specific purpose of pumped storage and many features considered standard in the conventional hydroelectric plant are found superfluous.

Selection of the proper site is of paramount importance. The rock should be strong and tight so no more than local grouting is needed to inhibit leakage and no drainage system for the concrete lining of the tunnels is required. The upper storage reservoir should preferably be located in a natural basin in elevated highlands to provide the requisite storage capacity by use of low rimdikes. It is a great advantage if the lower reservoir can be located on tidewater. The head, probably the most important natural feature, should be high, in the range from 500 to 1500 ft (152 to 457 m). High head means smaller water quantities and consequently smaller sizes for a given power output. High head also permits higher turbine speeds, lower torque, and a smaller generator. The intake need be only a bell mouth at the end of the supply tunnel; consequently, headgates, cranes, and accessories can be eliminated. The individual units should be of large capacity to ensure minimum equipment costs.

Table 3 shows a typical form for the economic comparison of pumped storage and competitive thermal peaking, and Table 4 is a typical form for the project cost estimate with its overheads.

Table 5 shows the range of machine requirements for a deep upper reservoir in which the head variation due to daily drawdown is relatively large. Note that the maximum electric motor load to develop the full hydraulic capability of the pump, occurring at the minimum head of 970 ft (296 m), is 297 MVA. However, this loading is of comparatively short duration and may be sustained at 80°C temperature rise, which corresponds to 115% of normal rating. The normal rating at 60°C temperature rise would then be $\frac{297}{1.15}$, or 258 MVA,

TABLE 2 Arithmetic integration at full-load rejection (generating mode; wicket gates at 92% opening)

Interval $\frac{2L}{a}$	Time, s	Trial V (ΔV), ft/s	ΔH, ft	$\sum\Delta H$, ft	H_f, ft	$H_o + H_f + \sum\Delta H$ = total H, ft	Trial speed, rpm	Q (8 units), ft³/s	Check V, ft/s	Hp, 8 units (000)	Average hp, 8 units (000)	N^2, rpm²	N_2^2 2 N_1^2 rpm²	Check T, s
0	0	24.5	—	—	−36.8	1173.2	257	28,100 (3,510)[a]	24.7	3200	—	66,000		
1	5.3	−1.9 22.6	274	274	−28.8	1455.2	342	25,800 (3220)	22.6	3040	3120	117,000	51,000	5.30
2	10.6	−5.3 17.3	766	492	−16.9	1685.1	390	19,700 (2460)	17.3	1180	2110	152,000	35,000	5.30
3	15.9	−6.0 11.3	867	375	−7.2	1577.8	378	12,900 (1610)	11.3	−104	538	142,900	9100	5.30
4	21.2	−4.7 6.6	680	305	−2.5	1512.5	368	7520 (940)	6.6	−784	−444	135,500	−7400	5.30
5	26.5	−3.25 3.35	470	165	−0.6	1374.4	345	3819 (477)	3.35	−1200	−992	119,000	−16,500	5.30
6	31.8	−0.85 2.50	123	−42	−0.4	1167.0	316	2850 (256)	2.5	−1080	−1140	100,000	−19,000	5.30

L = 12,322 ft, A = 1140 ft², a = 4650 ft/s, $H_f = 0.0565V^2$, $WR^2 = 1023 \times 10^6$ (8 units), $2L/a$ = 5.3 s, H_o = 1210 ft, $\Delta H = a\,\Delta V/g = (4650 \times \Delta V)/32.2 = 144.5\,\Delta V$

$$\Delta T = \frac{4\pi WR^2(N_2^2 - N_1^2)}{2g \times \text{av. hp} \times 550 \times 3600} = \frac{4\pi^2 \times 1023 \times 10^6(N_2^2 - N_1^2)}{64.4\ \text{av. hp} \times 550 \times 3600} = \frac{320(N_2^2 - N_1^2)}{\text{av. hp}}$$

SI conversions: m/s = 0.3048 × ft/s; m = 0.3048 × ft; m³/s = 0.0283 × ft³/s; kW = 0.746 × hp.

[a]Values in parentheses are per-unit values.

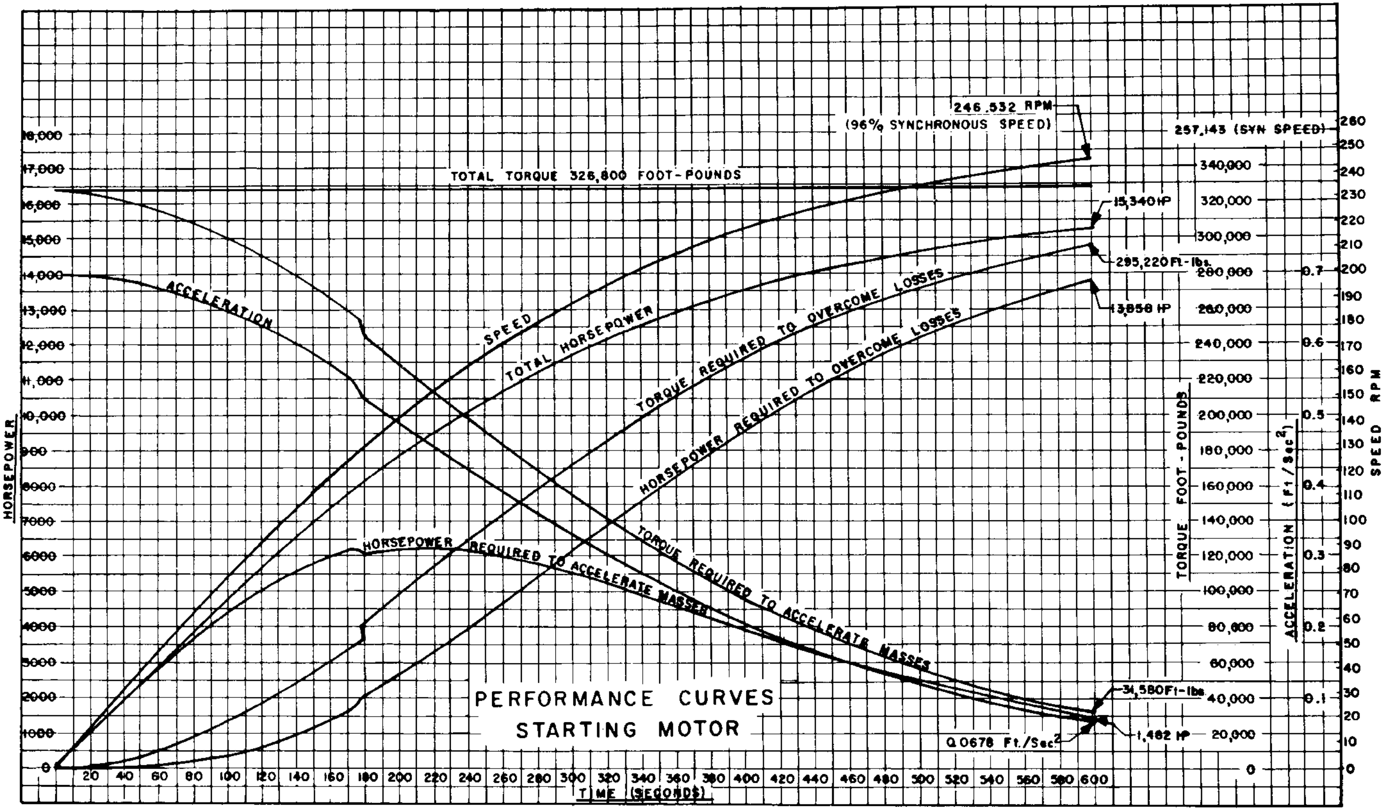

FIGURE 11 Performance curves, starting motor (1 hp = 0.746 kW; 1 ft · lb = 0.138 kg · m)

TABLE 3 Economic evaluation of hypothetical pumped storage plant versus conventional steam reheat plant

	Reheat		Pumped storage	
Installed capacity MW		3,600		3,600
Number of units		4		16
Plant investment (000 omitted)				
Generation		$450,000		$260,000
Transmission		—		47,100
Total		$450,000		$307,100
Total per kilowatt		$ 125		$ 85
Annual generation (10^6 kW · h)				
Peak load		1,300		1,300
Base load		20,000		
Total		21,300		1,300
Annual capacity factor, %		67.5		4.1
Annual fixed charge rates, %				
Generating plant		13.35		10.90
Transmission plant		—		11.35
Annual costs	Total (000)	Per kW · yr	Total (000)	Per kW · yr
Fixed charges				
Generating plant	$ 60,076	$16.69	$28,340	$ 7.87
Transmission plant	—	—	5,347	1.49
Total	$ 60,076	$16.69	$33,686	$ 9.36
Fuel	59,044	16.40	5,018[a]	1.39
Operation and maintenance	9,900	2.75	3,492	.97
Total	$129,020	$35.84	$42,196	$11.72
Credit to reheat unit:				
Replacement for base				
Load generation	68,480	19.02		
Net cost of 3600-MW capacity				
and peak-load generation	$ 60,540	$16.82	$42,196	$11.72
Differential annual cost in				
favor of hypothetical plant			$ 9,172	$ 5.10

[a]Fuel cost for pumping energy.

TABLE 4 Hypothetical pumped storage plant cost estimate-summary

Account no.	Item	Cost
330	Land and land rights	$ 4,311,000
331	Structures and improvements	
−1	Powerhouse substructure	20,791,000
−2	Service bay substructure	2,023,000
−3	Powerhouse superstructure	553,200
−4	Cofferdam	600,000
332	Reservoirs, dams, and waterways	
−2	Reservoir clearing	123,000
−4	Dikes and embankments	25,162,000
−8	Intakes	1,203,000
−12	Power tunnels and waterways	76,526,500
	Tailrace	4,940,000

Account no.	Item	Cost
333	Waterwheels, turbines, and generators	
−2	Fire protection system	86,000
−5	Motor generators	31,776,800
−8	Spherical valves	7,680,000
−9	Turbines and accessories	23,915,520
−11	Piezometer system	24,000
−13	Spiral case and draft tube unwatering systems	136,000
	Air depression system	208,000
	Draft tube racks, piers, and guides	2,071,600
334	Accessory electrical equipment	6,496,000
335	Miscellaneous power plant equipment	2,046,000
335	Roads, railroads, and bridges	2,393,600
352	Transmission plant—structures and improvements	2,226,500
353	Station equipment	12,119,400
		$222,412,120
	Contingencies, overhead, and engineering	32,587,880
	Total estimated cost of project	$260,000,000

TABLE 5 Operating range of hypothetical project

	Reservoir elevation, ft			
	1000	1060	1120	1160
Tailwater elevation, ft	0	0	0	0
Gross head, ft	1000	1060	1120	1160
Generating cycle				
Friction head loss, ft	30	30	30	30
Net head, ft	970	1030	1090	1130
Best gate turbine output, hp (000)	—	310		
Full-gate turbine output, hp (000)	310	—	380	400
Turbine output blocked at, hp (000)	—	—	350	350
Turbine efficiency, %	85	88.3	87.5	88.5
Turbine discharge, ft³/s	3300	3000	3200	3100
Generator output at 98% efficiency, MW	227	227	256	256
Generator MVA (0.90 PF)	252	252	284	284
Pumping cycle				
Frictional head loss, ft	20	20	20	20
Net head, ft	1020	1080	1140	1180
Maximum possible discharge, ft³/s	2700	2350	2000	1730
Pump power, hp (000)	350	324	298	277
Pump efficiency, %	89.5	89	87	84
Motor power, at 98% efficiency, MW	262	242	223	206
Motor MIVA (0.90 PF)	297	275	252	234

SI conversions: m = 0.3048 × ft; kW = 0.746 × hp; $m^3/s = 0.0283 \times ft^3/s$.

which is shown by the tabulation to be adequate to carry pumping and generating loadings of more protracted duration. This utilization of overload rating for Class B insulation affords substantial economies in electric machine cost.

REFERENCE

1. Rich, G. R. *Hydraulic Transients,* Dover, New York, 1963.

FURTHER READING

American Society of Mechanical Engineers. *Waterhammer in Pumped Storage Projects.* ASME, New York 1965.

Chapin, W. S. "The Niagara Power Project." *Civ. Eng.,* April 1961, p. 36.

Davis, C. V., and Sorensen, K. E. *Handbook of Applied Hydraulics.* McGraw-Hill, New York, 1969.

McCormack, W. J. "Taum Sauk Pumped-Storage Project as a Peaking Plant." *Water Power,* June 1962.

Parmakian, J. *Waterhammer Analysis,* Dover, New York, 1963.

Rich, G. R., and Fisk, W. B. "The Lean Project Concept in the Economic Design of Pumped-Storage Hydroelectric Plants." Paper presented at World Power Conference, 60, September 1964.

Rudulph, E. A. "Taum Sauk Pumped Storage Power Project." *Civ. Eng.,* January 1963.