CHAPTER 5

PUMP TURBINES

5-1. GENERAL.

a. Pump turbines are dual purpose machines. They operate as a pump in one direction and a turbine in the reverse direction.

b. A pump will perform in reverse rotation as a good turbine. However, a turbine does not generally operate in the reverse rotation as a good pump. Consequently, the design of a pump-turbine impeller follows more closely pump design practice than turbine design practice.

c. There is a dependent relationship between the two modes of operation and a compromise can be made to favor one mode of operation over the other.

5-2. BASIC CLASSIFICATIONS.

There are three basic classifications of pump-turbines:

1. Radial flow - Francis type

2. Mixed flow or diagonal type - Fixed blade and adjustable blade (Deriaz)

3. Axial flow or propeller type - Fixed blade and adjustable blade (Kaplan)

5-3. RADIAL FLOW - FRANCIS TYPE.

a. Francis type pump-turbines have been installed for heads of 75 to 1500 feet.

b. The design of the impeller is basically that of a pump impeller rather than that of a turbine runner. The impeller has fewer and longer blades than does a turbine runner with a view to effecting an efficient deceleration of flow in the water passages. The overall diameter of a pump-turbine runner is of the order of 1.4 times larger than the conventional Francis turbine runner. This is due to the requirements for a larger discharge diameter than eye (throat) diameter in the pumping mode. A lower runaway speed results, due to to the choking action of the impeller on the flow at higher speeds. This characteristic affects the cost of the water passages and the cost of the rotating parts of the pump-turbine and motor-generator.

c. As the unit is designed as a pump, it may be preferable to establish the pumping capacity for a specific total head which fixes within narrow limits the turbine capacity. This is particularly true for a combined installation of pump-turbines and turbines. Following the selection of the pumping capacity of the pump-turbine unit to fit the desired program of operation, the turbine capabilities in the generating mode is determined and the rating of the conventional units fixed to give the desired generating capacity for the several specified head conditions.

d. If the installation is strictly a pump-storage scheme, then the selection of the unit would begin with the determination of the required generating capacity at minimum head and the number of units to provide this capacity. Establishing the generating capacity for a given type and specific speed determines within a narrow range the pumping capacity. Establishing the pumping capacity for a specified total head also establishes within narrow limits the generating capacity for the corresponding turbine.

e. It is customary to guarantee the discharge in the pumping mode of a pump-turbine only at rated head or near best efficiency.

f. Only if a suitable runner is available from existing tests is it practicable to specify very closely the requirements for both the pumping and generating cycles.

g. Economics generally favors the higher capacity units unless an excessive number of runner splits is required by machining or shipping limitations. The design of split runners becomes more difficult with higher specific speeds. Runaway speed for higher capacity units is a larger percentage of synchronous speed and the centrifugal force makes the design of the splits more difficult as the additional metal increases the centrifugal force and the stress level.

h. Head losses through the draft tube of a pump-turbine decrease the net available suction head and increase the runner's sensitivity to cavitation. Therefore, pump-turbine runners must be set deeper than turbine runners. In order to reduce the size and cost of pump-turbines, higher specific speeds are utilized for pump-turbines relatively to turbines and consequently require deeper settings. The unit is more subject to cavitation in the pumping mode than in the generating mode. Therefore, the pumping mode determines the setting of the runner.

i. While in general higher specific speeds for pump turbines may appear to be desirable for economic reasons; efficiency, cavitation characteristics, mechanical and hydraulic design must be evaluated to determine the most favorable specific speed. Cavitation increases with higher specific speeds. The use of metals more resistant to cavitation damage may allow the acceptance of higher cavitation levels.

j. Transient Behavior. A transient in hydro-power is the history of what occurs between two states of equilibrium. A study must be made of the transient condition in order to determine the minimum (WK^2) flywheel effect for the rotating parts of the entire unit. The study should include a power failure in the pumping mode and load rejection in the generating mode. The Corps has a computer program which should be utilized when making these studies.

k. Four Quadrant Synoptic Curves.

(1) The necessary information to analyze transient behavior is provided by model tests made of the model runner and furnished as Four Quadrant Synoptic Curves which show the possible combinations of Unit Discharge (Q_{11}): under one foot of head and one foot eye diameter runner versus Shaft Speed (N_{11}), under one foot of head for one foot eye diameter in both pumping and turbine directions and Torque (T_{11}), under one foot of head for one foot eye diameter versus Discharge (Q_{11}), under one foot head for one foot eye diameter for both pumping and turbine directions.

(2) These curves are prepared from test information obtained from all gate openings in the complete turbine and pump performance curves, plus information from two additional tests. The first one of these tests is identical to a normal pump test except that the sense of rotation of the impeller and of the torque applied to the shaft are opposite to that for a normal pump test with measurements being taken the same as during normal pump tests. The second test involves rotating the impeller in the normal pump direction with water being pumped through the model in the normal turbine direction by service pumps. (During the test the head, discharge, speed, and torque are measured for various gate openings and shaft speeds.)

(3) The Four Quadrant Synoptic Curves may also be supplied showing the relationship of horsepower and discharge to phi (ϕ) for the various gate openings. The curves show the possible combination of head, discharge, torque or power in the following modes of operation:

(a) Pumping operation

(b) Dissipation of energy with rotation in the pumping direction and flow in the generating direction.

(c) Turbine operation

(d) Dissipation of energy with rotation in the generating direction and flow in the pumping direction.

5-4. MIXED FLOW OR DIAGONAL FLOW (DERIAZ).

a. The mixed flow or diagonal flow type pump-turbines are used in the medium head range up to more than 250 feet.

b. While the mixed flow adjustable blade machine (Deriaz) previously manufactured by the English Electric Company are presently manufactured and preferred for use in Japan, the less costly Francis mixed flow types are preferred in the U.S.A.

5-5. AXIAL FLOW - PROPELLER TYPE - FIXED AND ADJUSTABLE BLADE.

a. Alternates for use in the low head range, below 75 feet, are the axial flow machines arranged with the shaft vertically, horizontally or inclined.

b. The Corps of Engineers Harry S. Truman (Kaysinger Bluff) project has the largest capacity slant axis axial flow pump storage machines under construction as of 1 January 1981.

5-6. SPECIFIC SPEEDS - SINGLE STAGE REVERSIBLE PUMP/TURBINES.

a. The turbine specific speed (N_{st}) of a pump turbine in the generating mode is defined as the speed in revolutions per minute (N) at which a pump turbine of homologous design would operate if the runner was reduced in size to that which would develop one horsepower under 1 foot of head.

b. The turbine specific speed is expressed as follows:

$$N_{s_t} = \frac{N\ HP^{1/2}}{H^{5/4}}$$

c. The turbine specific speed is somewhat higher than for a conventional turbine, being in the range of K from 800 to 1250, where K = N_{st} H 1/2. The range of K for a conventional Francis turbine is 700 to 850, based on N_{st} at best efficiency.

d. The pump specific speed (N_{sp}) of a pump turbine in the pumping mode is defined as the speed in revolutions per minute (N) at which the pump turbine runner of homologous design would operate if the runner were reduced geometrically to such a size that it would deliver one U.S. gallon per minute under one foot of head.

e. The pump specific speed is expressed as follows:

$$N_{s_p} = \frac{N\,Q^{1/2}}{H^{3/4}}$$

$$Q = \text{gpm}$$

f. A recommended relationship for selecting pump specific speeds for a range of pumping heads is shown on Figure 2, Appendix C.

g. In selecting the specific speed consideration must be given not only to pump-turbine and generator motor costs, powerhouse and auxiliary equipment costs, but to efficiency in both modes of operation, cavitation characteristics, mechanical, hydraulic design features and to any restrictions imposed by foundation and site conditions.

5-7. PRELIMINARY DATA FOR FRANCIS PUMP-TURBINES.

a. Model performance data for pump-turbines is shown in Appendix D. This data is based on model tests covering low, intermediate and high specific speed designs. The original test data has been reduced to the more convenient form shown on Figures PT1, PT2 and PT3, which are based on D_{TH} = 12" and H = 1 foot. The performance for the generating mode is presented in the same format as that adopted in the other Sections covering the conventional Francis and propeller turbine designs. The discharge, efficiency and critical sigma curves for the pumping mode represent the envelope performance for best efficiency from a number of fixed wicket gate tests. The curve of pumping specific speeds is derived from the other data. Pertinent dimensions of the pump-turbine and associated water passages, expressed as ratios to D_{TH}, are shown in Table 3 and Figure 4 of Appendix C.

b. Information established by the Project Planning and Field Survey Studies is listed in paragraph 1-5. For the pumping cycle this information includes the pumping requirements and maximum, minimum and average heads.

c. The results of planning studies will generally dictate the rated pumping conditions. This information should include the rated gpm discharge and the associated rated dynamic head. An appropriate specific speed for the rated head is obtained from Figure 2, Appendix C. A preliminary value for speed (N), is then calculated from the formula:

$$N = \frac{N_{s_p} H^{3/4}}{Q^{1/2}}$$

The calculated value is rounded to the nearest synchronous speed.

d. If the user is chiefly interested in the size of the prototype unit, the following empirical relationship may be used to approximate a value for ϕ_{TH}.

$$\phi_{TH} = 0.0015\ N_s^{0.785}$$

e. The runner throat diameter is calculated as follows:

$$D_{TH} = \frac{1838\ \phi_{TH}\ H^{1/2}}{N}$$

f. The following procedure is used to calculate prototype pumping performance from the model performance curves:

(1) From Figure 2 pick off the value of N_{sp} corresponding with the rated dynamic head.

(2) Inspect the model performance curves, noting the specific speed range of each, and select the design that best suits the value of N_{sp}.

(3) From the selected curves, note the value of Q_1 for maximum pumping efficiency. Also note that this value is in cfs units. The prototype runner throat diameter (eye diameter for pump impeller) is calculated from the following relationship, where the subscript 1 refers to the model and subscript 2 refers to the prototype:

$$\frac{Q_2}{Q_1} = \left(\frac{D_2}{D_1}\right)^2 \left(\frac{H_2}{H_1}\right)^{1/2}$$

D_1 = 12 inches and H_1 = 1 foot.

(4) At this same point note the value of ϕ_{TH}. The pump speed is calculated from the relationship:

$$N = \frac{1838\ \phi_{TH}\ H^{1/2}}{D_{TH}}$$

Round to the nearest synchronous speed.

(5) Readjust ϕ_{TH} for synchronous speed, pick off a new Q1 from the model curves and repeat steps (3) and (4), if required. Continue this process until the re-adjusted value of ϕ_{TH} and corresponding Q1 from the model curves produce a value of D_{TH} that results in a synchronous speed in step (4).

(6) With fixed values for D_{TH} and N, the ϕ_{TH} corresponding to other pumping heads is calculated from the following relationship:

$$\phi_{TH} = \frac{N\ D_{TH}}{1838\ H^{1/2}} = \frac{Kn}{H^{1/2}}$$

(7) For given head and corresponding value of ϕ_{TH} extract Q_1 from the model curves and calculate the uncorrected value of prototype discharge from the following relationship:

$$Q_2 = Q_1 \left(\frac{D_{TH}}{12}\right)^2 H^{1/2}$$

(8) The prototype efficiency of the pump-turbine is determined by the Moody formula in accordance with MT-4.5 of Guide Specifications CE-2201.02 HYDRAULIC PUMP-TURBINES - FRANCIS TYPE Paragraph MT-4.5. The efficiencies are increased by the same amount in both the pumping and generating modes of operation. The following formula is applied to the peak efficiency value:

$$E_2 = 100-(100-E_1)\left(\frac{D_m}{D_p}\right)^{0.2}$$

$E_2 - E_1$ = step-up in efficiency

Using two-thirds of the step-up, the model efficiencies (E_1) are increased by 2/3 ($E_2 - E_1$) to give the expected or prototype efficiencies, E_2.

(9) The values of discharge in step (7) are increased by the ratio (E_2/E_1) to account for the increased efficiency of the prototype. The increased values are used in plotting the expected head-cfs curve of the prototype.

(10) With Q, H and E being taken directly from the performance curves, the horsepower-cfs curve is completed using the formula:

$$HP = \frac{Q\,H\,w}{550\,E}$$

g. The following procedure is used to calculate prototype performance in the generating mode from the associated hill curves shown at the bottom of the model curves in Appendix D.

(1) ϕ_{TH} values for the range in net operating heads are calculated from the formula established in (6) above.

(2) Prototype horsepowers for each head are calculated by intercepting the fixed gate curves at the associated value of ϕ_{TH}, noting the corresponding HP_1 and substituting known values in the following formula:

$$HP_2 = HP_1 \left(\frac{D_2}{D_1}\right)^2 \left(\frac{H_2}{H_1}\right)^{3/2}$$

$$HP_2 = HP_1 \left(\frac{D_{TH}}{12}\right)^2 \left(\frac{H_2}{1}\right)^{3/2}$$

(3) The expected prototype efficiency (E_2) is calculated by adding the step-up determined in f(8) above to the associated model efficiency (E_1), at each fixed gate point.

(4) With HP, H and E being taken directly from the expected performance curves, the expected prototype discharge is calculated from the following formula:

$$Q = \frac{550\ HP}{w\ H\ E}$$

5-8. SETTING OF PUMP-TURBINE RUNNER - FRANCIS TYPE.

a. The setting, which is generally referred to the elevation of the distributor centerline, is determined by critical sigma in the pumping mode. The appropriate sigma value for each pumping head can be picked off directly at each corresponding ϕ_{TH} on the model curves, then substituted in the following formula to compute the submergence, H_s:

$$\sigma_c = \frac{H_b - H_v - H_s - \text{safety}}{H}$$

(1) Appropriate values for H_b and H_v are shown on Figure 6, Appendix C.

(2) A recommended safety margin is calculated by substituting in the following formula:

Safety margin = $0.2\ D_i + 0.4\ H^{1/2}$

where D_i is the prototype runner inlet diameter in feet. This diameter is expressed as a dimensionless ratio, D_1 in Table 3 and Figure 4. The actual diameter is the product of the ratio times D_{TH}, in feet.

b. The values of critical sigma from the model tests are referred to the bottom of the runner. Likewise, the submergence (H_s) computed above is referred to this same point. The distance, a, from the distributor centerline to the bottom of the runner is expressed as a dimensionless ratio, d, in Table 3 and Figure 4. The distance is the product of the ratio times D_{TH}, in feet. The setting is calculated as follows:

Elevation distributor centerline = Tailwater Elev. + H_s + a

5-9. <u>SPIRAL CASING AND DRAFT TUBE - FRANCIS TYPE.</u>

a. The turbine manufacturer is responsible for the design of the water passages from the upstream end of the turbine casing inlet to the discharge of the draft tube. Deviations from the model dimensions can affect performance.

b. Pumping considerations dictate the design of the spiral casing and draft tube, except for the number and width of draft tube piers which depend on structural requirements.

5-10. <u>DRAFT TUBE LINERS.</u> Draft tube liners for pump-turbines should extend to the pier noses.

5-11. RUNAWAY SPEEDS.

a. Runaway speed tests for Francis type or fixed-blade propeller type pump-turbines are conducted the same as those for the Francis type turbines.

b. The runaway speed tests for adjustable blade propeller type pump turbines are conducted the same as for the adjustable blade turbine.

c. The calculation of prototype maximum runaway speed is the same as that shown in paragraph 4-8c.

d. It is sometimes necessary to limit the gate opening of the prototype Francis pump-turbine to limit overspeed because of the difficulty of designing a generator to withstand the higher overspeed of high gate openings.

e. The effect of sigma on a Francis type pump-turbine can be neglected, however, sigma must be considered for propeller type pump-turbines.

5-12. AIR ADMISSION. An air and check valve (or valves) should be installed for Francis-type pump turbines to permit operating at gate openings below 50 percent.

5-13. RUNNER SEAL CHAMBER DRAINS. The seal chamber pipe drain header should discharge in the vertical leg of the draft tube on the side furthest away from the draft tube exit.

5-14. SAMPLE CALCULATIONS. A typical calculation for a pump turbine installation is included in Appendix E.