CHAPTER 2

TURBINE AND PUMP-TURBINE CHARACTERISTICS

2-1. SPECIFIC SPEEDS.

a. The basis for comparison of the characteristics of hydraulic turbines is the specific speed (N_{st}). This is defined as the speed in revolutions per minute (N) at which a turbine of homologous design would operate if the runner was reduced in size to that which would develop one horsepower under one foot of head.

b. The specific speed varies directly as the square root of the horsepower (HP) and inversely with five-quarters power of the head (H) in feet.

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

c. In the metric system (N_{st}) is the speed of a homologous turbine of a size to develop one metric horsepower under one meter head. The metric specific speed is equal to 4.446 times the specific speed in the foot-pounds system.

d. In general, for a given head and horsepower, the higher the specific speed, the higher the speed of the unit and the lower the overall cost of the installation. But there are limits on the specific speed of a runner for a given head and output. Too high a specific speed would reduce the dimensions of the runner to values that would cause excessively high velocities for the water discharge through the throat of the runner and draft tube. Too high a specific speed could reduce the runner structural dimensions and the rotating parts of the generator to such small dimensions that high stresses would make it uneconomical, if not impracticable, to design. Too low a specific speed would unduly increase the size and cost of the generator in order to maintain the WR^2 of the unit. Obviously, there are practical limitations to the range in specific speeds for any head.

e. For Francis turbines, the specific speed is indicative of the type and shape of the runner. A low specific speed runner (high head) has an inlet diameter greater than the discharge diameter while the reverse is true for a high specific speed runner (low head).

f. For propeller turbines, higher specific speeds for higher heads require an increase in the number of blades.

g. Normal N_{st} is defined as the specific speed for best efficiency and rated N_{st} is defined as the specific speed at rated capacity or guaranteed horsepower under the head for which the turbine is designed.

h. Pumping specific speed (N_{sp}) is the speed at which the runner would rotate if reduced geometrically to such a size that it would deliver one U.S. gallon per minute under one foot of head.

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

2-2. PERIPHERAL COEFFICIENTS (ϕ).

a. The peripheral coefficient, a dimensionless number used for convenience in plotting model performance curves, is the ratio of the peripheral velocity of the runner blades to the spouting velocity of the water.

$$\phi = \frac{\text{Peripheral speed of the runner (fps)}}{\text{Spouting velocity of water (fps)}}$$

At the runner throat:

$$\phi_{TH} = \frac{\pi\left(\frac{N}{60}\right)\left(\frac{D_{TH}}{12}\right)}{\sqrt{2g\,He}} = \frac{N\,D_{TH}}{1838\,H^{1/2}}$$

where

ϕ_{TH} = Peripheral coefficient at runner throat

N = Runner speed in revolutions per minute

D_{TH} = throat diameter of the runner in inches.

Note: While D may denote any representative dimension of the runner such as inlet, throat and discharge diameters, it is Corps practice to use throat diameter.

g = Acceleration due to gravity = 32.17 ft./sec^2

H_e = Effective head in feet.

b. The runner speed must be selected to match a synchronous speed for the generator (see Appendix A, Page A-6, "GENERATOR SPEED VS NUMBER OF POLES").

$$N = \frac{120\ Hz}{n}$$

where Hz = frequency in cycles per second and n = number of poles of the generators.

c. While, in general, higher runner speeds for a specified horsepower at a specified head should result in a lower first cost for a turbine, the speed may be limited by the cavitation tendency of the runner, the drop in peak efficiency over the normal range of operation, vibration, and by mechanical design of the turbine or generator. Higher speeds require a lower setting of the runner with respect to tailwater and are accompanied by increased excavation and structural costs. Higher speeds also reduce the head range under which the turbine will operate satisfactorily.

d. Pump-turbines are more subject to cavitation in the pumping mode than in the generating mode. Therefore, the pumping mode determines the setting of the runners.

2-3. SETTING OF TURBINE AND PUMP-TURBINE.

a. The setting of the turbine or pump-turbine is very important. Too low a setting would result in unnecessary excavation and structure costs. Too high a setting could result in excessive cavitation of the runner buckets or blades with a resulting loss in efficiency and increased operating and maintenance costs.

b. The setting of a turbine or pump-turbine can best be determined by the consideration of the Thoma cavitation coefficient Sigma (σ).

$$\sigma = \frac{H_b - H_v - H_s}{H_e}$$

where H_b= Barometric pressure head at elevation of the runner above mean sea level

H_v= Vapor pressure head in feet at water temperature

H_s= For Francis runners is the distance from the lowest point on the runner vanes to tailwater in a vertical shaft unit, and the distance from the highest elevation of the runner band to tailwater for horizontal units.

H_s= For fixed blade and Kaplan runners is the distance from the center line of the blade trunnion to tailwater for vertical units and from the highest elevation of the blades to tailwater for horizontal and inclined units.

H_s= For diagonal flow runners is the distance from the bottom of the gate to tailwater for vertical units and from the highest elevation of the blades to tailwater for other units.

H_e= Net or Effective head on the turbine in feet.

H_s may be positive or negative, depending on whether or not the referenced point on the runner is above or below tailwater. When the referenced point is below tailwater, it is negative. If "a" is the distance from the center line of the turbine distributor to the lowest point on the runner vanes for Francis units and to the centerline of the blades for propeller-type runners, then the distance from the centerline of the distributor to tailwater for H_s positive is $(a + H_s)$, and for H_s negative is $(a - H_s)$.

c. It is customary for manufacturers to add a safety allowance to the cavitation coefficient Sigma (σ).

$$\sigma = \frac{H_b - H_v - H_s - \text{Safety}}{He}$$

2-4. CRITICAL SIGMA (σ_c).

a. Over the years, since facilities for making cavitation tests have been available, there have been several methods proposed and used for determining critical sigma from model tests. There has been no fixed agreement on a standard method of determining critical sigma and in using manufacturers' critical sigma curves. It is important that the method used in determining critical sigma for a particular model be clearly established as a manufacturer may have used a different method of determining critical sigma, depending on the method in use at the time of the test.

b. In some model tests, the cavitation limits were considered to be at the points where power drops off and the discharge increases, thus decreasing the efficiency.

c. In other model tests, critical sigma was considered to be the value obtained at the point of intersection of the constant horizontal HP or Q (pump) curve with the slope of the line under cavitating conditions.

d. The International Code for Model Acceptance Tests of Hydraulic Turbines, IEEE Publication 193, gives the following three definitions of sigma:

(1) σ_0, the lowest value of sigma for which the efficiency remains unchanged as compared to non-cavitating conditions,

(2) σ_1, the lowest value of sigma for which a drop of one percent in efficiency is attained compared to a non-cavitating condition, and

(3) σ_s, Standard Sigma, the value sigma at the intersection of the constant efficiency line (non-cavitating) with the strongly dropping straight line along which measuring points align themselves for a high cavitating degree.

e. The Corps of Engineers Specifications defines "the Critical Sigma of the turbine or pump turbine for such desired turbine output or pump capacity and head shall be the sigma corresponding to the tailwater level of such tests which results in a one percent decrease in efficiency or turbine output, or pump power input which ever occurs first." (See CE 2201.01, .02 and .03, paragraph MT-4.5).

f. Because of the shape of some model sigma curves, considerable judgment is necessary in determining critical sigma.

g. Prototype experience is necessary to determine the factor of safety to include with H_s and also how much prewelding of the runner blades can be used as a trade-off against deeper submergence.

2-5. PERFORMANCE CURVES.

a. Turbine prototype performance curves are plots of efficiency and discharge versus horsepower for various heads and gate openings and are based on laboratory test data of a model homologous to the prototype with regards to runner and water passages.

b. The power is stepped up from the model by the formula:

$$HP_p = HP_m \left(\frac{D_p}{D_m}\right)^2 \left(\frac{H_p}{H_m}\right)^{3/2}$$

c. The turbine discharge, neglecting any step-up in efficiency, may be calculated by inserting the value of horsepower calculated from the formula under (b) above into the formula for turbine horsepower $HP = WQHE_m/550$. H is the net or effective head and E_m is the model efficiency.

d. The expected efficiency of the prototype turbine is the model efficiency plus not more than 2/3 of the step-up in efficiency ($E_p - E_m$) as determined by the Moody formula where E_m is the maximum model turbine efficiency at best speed or phi (Ø). The allowable step-up in efficiency is added to all efficiency points to obtain the expected prototype corrected efficiencies, (E_c).

$$E_p = 100 - (100 - E_m)\left(\frac{D_m}{D_p}\right)^{1/5}$$

$$E_c = E_m + \left(\frac{2}{3}\right)(E_p - E_m)$$

e. No step-up in power is permitted by the guide specifications however the corrected efficiency is used in calculating prototype discharges.

$$Q = \frac{550\ HP}{w \cdot H_e\ E_c}$$

f. Pumping performance curves are plots of efficiency, head and horsepower versus discharge at various gate openings and are based on laboratory test data for a model homologous to the prototype with regards to runner and water passages. For pumping, unless otherwise stated, the head is the total head from the suction pool to the discharge of the spiral case. The prototype head and discharge capacity values are stepped up from the model by the affinity laws and the capacity values so determined should not be less than the guaranteed values. The pump corrected efficiencies are obtained by adding the allowable step-up in efficiency to all efficiencies points. The expected head-capacity curves are developed using corrected capacity values which are the values stepped up from the model multiplied by the ratio E_c/E_m. The expected efficiency capacity curves are developed using the corrected efficiency and capacity values and the horsepower values used in developing the expected horsepower-capacity curves are computed using the formula HP = $wQH/550E_c$, where Q, H, and E_c are taken directly from the expected performance curves. The maximum pump input horsepower determined from the curves should not exceed the maximum pump horsepower permitted by the specifications.

2-6. MODEL-PROTOTYPE RELATIONSHIPS. Affinity laws and model to prototype relationships for turbines and pump turbines are included in Appendix A.

2-7. GUARANTEES.

a. When available, previous model tests can be used as the basis for guarantees, the guaranteed efficiency values should be set 1/4 percent less than the indicated model efficiencies. See also Paragraph 1-11.

b. Likewise, horsepower guarantees should be set two percent less than the values shown on the expected Horsepower vs. Efficiency curves for the prototype.