CHAPTER 3

FRANCIS TURBINES

3-1. GENERAL USE. For many years, Francis turbines were used for low heads. Today they are in general use for heads from 75 feet up to 1600 feet while propeller type turbines have replaced Francis type turbines at the lower heads.

3-2. SPECIFIC SPEEDS.

a. Specific speeds for Francis turbines range from 20 to 90 and is obtained by changing the design proportions of the runner. A general discussion of specific speed is presented in 2-1.

b. A low specific speed Francis runner has a larger entrance diameter than discharge diameter. For a specific speed of approximately 42, the inlet diameter is approximately equal to the throat and discharge diameters. For higher specific speeds, the inlet diameter becomes smaller than the throat and discharge diameters. Also the discharge diameter is larger than the throat diameter.

c. The specific speed (N_s) will remain constant for any other size or head for the same design and the corresponding speed for another homologous runner.

d. Care must be taken when using specific speed values to insure that they are being correctly used. The best efficiency at rated head for a Francis turbine is matched at 85 - 90 percent of the generator KW rating. the normal KW rating of the generator, the horsepower equivalent of which is used in calculating the rated specific speed.

e. In the process of selecting a turbine for a specific installation, the specific speed should also be determined using the lowest head at which the maximum power must be developed (generator KW rating). This will give the highest N_s under which the unit must operate and may dictate the selection of the runner. When the lowest head is appreciably lower than the average operating head and when the power required is exceptionally high in comparison to the requirement under normal head, it may be necessary to install an oversize turbine to meet the low head capacity requirement. In this case the turbine shaft may be sized to meet the generator rating with the provision that the turbine gate openings be restricted when operating at heads where if the gate openings were not restricted, the generator rating would be exceeded. The same head and gate opening restrictions apply to turbines where increase in heads under flood conditions could cause a turbine output in excess of the generator rating.

f. For many years, hydraulic laboratories did not have the facilities for testing the cavitation characteristics of Francis runners. Therefore the cavitation characteristics were estimated on the basis of experience with installations of similar types. During this period a value of 632 was used for K in $N_s = K/H^{1/2}$. In 1951 the manufacturers of hydraulic turbines recommended a K value of 650 to be used in the above formula on the basis that the vertical Francis-type turbines could usually be set with the centerline of the distributor about eight feet above tailwater at sea level. More recent experience indicates an economic advantage for smaller, higher speed units with deeper settings consistent with a K value of approximately 700. This relationship is shown on Figure 1, Appendix C and is recommended for preliminary studies.

3-3. DEVELOPMENT OF PROTOTYPE PERFORMANCE CURVES FROM MODEL TESTS.

a. Model test curves covering a wide range of specific speeds are shown on Figures F1 through F8 in Section I, Appendix D. This method of representation is commonly referred to as "oak tree" or performance hill. The latter designation derives from the fact that the figure is three-dimensional, as each constant efficiency contour represents a coordinate point in the Z-direction perpendicular to the plane of the paper. All data has been reduced to unit values corresponding with D_{TH} = 12 inches (one foot) and head, H = one foot. The ordinate is unit horsepower, HP_1, and the abscissa is peripheral speed coefficient, ϕ_{TH}. All efficiencies are based on D_{TH} = 12 inches. The indicated specific speed is referred to the point of maximum efficiency. Some cavitation characteristics are shown on Figures S1 through S5, Section IV, Appendix D.

b. The significant characteristics of the eight designs are compared on Figure F9 of Appendix C. A number of other designs have been included to aid in the correlation of the data with respect to specific speed. The curves may be used for preliminary selection of the runner throat diameter, speed, design discharge and runner setting. The curve of critical runner sigma is based on a horsepower that is 15 percent greater than the horsepower at best efficiency. The curve must not be used for off-best phi conditions. For studies requiring more complete information regarding the turbine dimensions and performance, a selection must be made from one of the eight designs.

c. Pertinent dimensions of the turbine parts and water passages, expressed as a ratio to D_{TH}, are shown in Tables 1 and 2, and Figure 4 of Appendix C.

d. The following steps are made to select a turbine:

(1) Given the horsepower corresponding to 85 - 90 percent generator rating at rated head, a preliminary value for N_s can be selected from Figure 1, Appendix C. A design is selected from Figures F1 through F8 with a specific speed which most nearly approximates the preliminary value. The design specific speed is used in the ensuing calculations. Speed N is determined from $N = N_s H^{5/4}/HP^{1/2}$, but N must be adjusted to a synchronous speed. It is usually necessary to investigate three synchronous speeds in order to arrive at the most overall economic speed.

(2) Having selected a speed, a preliminary runner diameter may be determined using the selected design and the relationship of $\phi_{TH} = N D_{TH}/1838 H^{1/2}$ and D_{TH} may be adjusted to get ϕ_{TH} at rated head to be at or near best gate. Also, it may be necessary to change the runner diameter so the HP_1 picked off from the performance hill for the phi values corresponding to other head conditions will give the required prototype horsepower. This may necessitate a change in speed with a corresponding shift away from best phi at rated head conditions. This includes calculating ϕ_{TH} for the minimum head conditions at which the capacity value of the unit or units is based, and the horsepower output at this head. It may be necessary as stated above to readjust N and D_{TH} to get the necessary HP_1 when stepped up to give the required prototype horsepower at the minimum head condition. See also comments under 3-2(e) regarding the necessity of installing an oversize turbine to meet the low head capacity requirements.

(3) Performance curves may now be developed from the model tests using the appropriate model - prototype relationships included in Appendix A.

e. As previously pointed out, hydraulic turbines are not off-the-shelf items of equipment. Model tests previously made and prototypes of models test are indicative of the performance that can be expected and the turbine manufacturer can alter or design a model based on experience to meet the requirements specified for a particular procurement. This accounts for some of the scatter of points shown on Figure F9 of Appendix D.

f. For specific speeds N_s = 33 and below, increasing the vent (i.e. opening between buckets) will permit increasing the power and shifting the point of best efficiency to the right. Decreasing the vent will reduce the power and shift the point of best efficiency to the left. Increases or decrease in vent opening as much as 15 to 20 percent may be made. A small percentage increase in the inlet diameter may also be possible as a means of shifting the point of best efficiency and slightly reducing the power. Increasing the vent openings too much may result in a loss of efficiency.

g. For specific speeds greater than 33, the model may be changed by increasing or decreasing the vent opening up to approximately 10 percent. Small percentage increase and decrease in inlet diameter may be possible for runners with specific speeds at best gate up to N_s of approximately 60. For specific speeds above 60, a small increase in inlet diameter is usually permissible. These are the means by which models can be adjusted to give desired project performance.

h. Increasing the vent openings increases the power of the runner but may result in a drop in efficiency. Too large an increase in vent opening could cause the power and efficiency to drop off too sharply.

i. It should be noted that N_s for best gate is approximately 7-1/2 to 11 percent smaller than N_s for full gate at rated head.

j. The setting for the three synchronous speed runners can now be determined.

3-4. SETTING OF RUNNER.

a. Overall plant efficiency is dependent on the design of the water passages from forebay through the tailrace. However, the turbine manufacturer is only responsible for the design between the turbine casing inlet and the discharge of the draft tube. Therefore the following dimensions are necessary for inclusion in the turbine specifications as limiting dimensions:

(1) Elevation of center-line of distributor.

(2) Maximum elevation of low point of draft tube floor.

(3) Horizontal distance from center-line of unit to the end of the draft tube.

b. The setting of the runner can be established by calculating H_s (the distance from the lowest point on the runner buckets to tailwater corresponding to the Q for the prototype horsepower) and substituting the value of sigma obtained from the model tests curves for HP_1 corresponding to the prototype horsepower in the formula:

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

Depending on the value of sigma, H_s may be positive or negative. The setting of the bottom of the runner blades may be above or below the elevation of the tailwater corresponding to the discharge Q for the prototype horsepower. The distance ratio from the centerline of the distributor to the bottom of the runner is listed in Tables 1 and 2, Appendix C. This ratio multiplied by the prototype runner diameter, D_{TH}, gives the dimension which when added to or subtracted from H_s yields the setting for the centerline of the distributor of the prototype.

c. In determining the setting of the runner, the possibility of lower future tailwater levels due to degradation of the river channel below the dam must be considered. Also, the time required for build-up of tailwater under low load factor operation conditions, if applicable, must be considered. Both factors dictate a lower setting. Foundation conditions at the site may make it economically desirable to set the unit higher by using a lower specific speed runner or to set the unit lower by using a higher specific speed runner. There is also the possibility of an economic "trade-off" between the maximum output of the runner at the lower heads, cost of excavation, a draft tube with a shorter vertical leg, and more stainless steel pre-welding of the runner to reduce pitting of the runner due to the higher setting.

3-5. SPIRAL CASE AND DRAFT TUBE.

a. While the turbine manufacturer is responsible for the design of the water passages from the turbine casing inlet to the discharge of the draft tube, there are limitations which are prudent to impose such as the velocity at inlet to the spiral case, the number and width of draft tube piers, the velocity at discharge of the draft tube and the elevation of the lowest point of the draft tube that will be permitted.

b. The diameter of the inlet to the spiral case may be the same as, or preferably less than, that of the penstock but the velocity at the inlet should not exceed 22 percent of $\sqrt{2gH}$. If the velocity is higher, a loss in efficiency and power result. There may be instances where it is desired to install, in an existing plant, a larger unit than the structure was designed to accommodate. In this case the increased head loss (H_L) between the net head measurement section and the runner is approximately 2/3 of the increase in velocity head. The reduced efficiency E and power HP can be calculated by the following:

$$E_r = \frac{E\,(H - H_L)}{H}$$

$$HP_r = HP \left(\frac{H - H_L}{H} \right)^{3/2}$$

$$H_L = 2/3 \left(\frac{V_2^2}{2g} - \frac{V_1^2}{2g} \right) = \frac{Q^2}{3g} \left(\frac{1}{A_2^2} - \frac{1}{A_1^2} \right)$$

V_1 = Velocity at the inlet of the normal casing.
V_2 = Velocity of the smaller casing.
A_1 = Area of inlet of the normal casing.
A_2 = Area of the smaller casing.

c. Deviations from strictly homologous water passages may also affect runaway speed, thrust, critical sigma as well as design of moving parts.

d. While procedures based on model laws and model and prototype tests are necessary to the study and selection of equipment, they need to be augmented by skills and judgment acquired by experience.

3-6. RUNAWAY SPEED.

a. The runaway speed of the prototype turbine is determined from model tests by running the model at various gate opening for the full range of model RPM (N_1) or phi (ϕ) to maximum RPM or ϕ at minimum values of efficiency and power and extending the curves to zero. The corresponding value, $\phi_{max.}$, is shown on Figures F1 through F8 of Appendix D. Prototype maximum runaway speed is given by the following:

$$N_{max} = \phi_{max} \left(\frac{12}{D_{TH}} \right) \left(\frac{60}{\pi} \right) (2g\,H)^{1/2}$$

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

b. It is difficult to design a generator to withstand the highest overspeed conditions. Therefore, it is sometimes necessary to limit the maximum gate opening of the prototype turbine in order to limit the overspeed.

c. While runaway speed is affected by sigma, for all practicable purposes, its effect, on a Francis turbine can be neglected.

d. With medium head Francis turbines the maximum overspeed occurs at full gate but for higher heads where the inlet diameter of the runner is somewhat greater than the discharge diameter, the maximum runaway speed may occur at less than full gate

3-7. DRAFT TUBE LINERS. Draft tube liners should extend a distance equal to at least one discharge diameter of the runner below the point of attachment to the bottom ring.

3-8. AIR ADMISSION.

a. When Francis units are operating at part gate, air must be admitted to the center of the runner cone or hub. An air valve, mechanically connected to the wicket gate mechanism controls the admission of air. If the tailwater can be higher than the elevation of the valve and also, if a tailwater depression system is used, a check valve must be installed. Depending on the specific speed of the turbine and its required submergence, it may be necessary for the runner to have alternate passages to admit air through the runner relief holes and to use a compressed air supply for air admission.

b. For a required horsepower at a given head, higher specific speeds will require deeper settings and increased air admission at part gate opening for stable operation. It will also be necessary in some cases to provide fins in the draft tube to reduce power swings to an acceptable level.

3-9. RUNNER SEAL CHAMBER DRAINS. When runner seal drains are required, the seal chamber pipe drain header should discharge in the vertical leg of the draft tube at a location furthermost away from the draft tube exit.

3-10. SAMPLE CALCULATIONS. The basic calculations for a typical installation are included in Section 1, Appendix E.