CHAPTER 3

Classification of Turbines – Main Characteristics

Introduction

This chapter takes into account similarity properties of hydrodynamic machines. That means to describe the performance of a given machine by comparison with the experimentally known performance of another machine. This other machine may be a model with geometrically similar fluid passages, or it may be the same machine with somewhat modified operating conditions such as a change of speed. Such comparisons are simple and reliable, as they are limited to cases in which not only the fluid passages but also the flow inside these passages can be considered geometrically similar.

3.1 Fundamental similarity considerations

3.1.1 Similarity relations

Reduced parameters

Principally every turbine is designed according to the available discharge Q, net head H_n and a chosen optimal rotational speed n. These parameters however, differ over wide ranges from one site to the other.

For this variability it is very useful to have similarity relations at hand for comparison means. In the following it is therefore, introduced some ratio parameters which are designated as reduced quantities[/7/] transferred from corresponding dimensional quantities.

The main turbine Equation (2.24) is rearranged by dividing it through by the net head H_n

$$1 = (1+\varsigma_1)\frac{c_1^2}{2gH_n} - \frac{c_2^2}{2gH_n} + (1+\varsigma_2)\frac{v_2^2}{2gH_n} - \frac{v_1^2}{2gH_n} + \frac{u_1^2}{2gH_n} - \frac{u_2^2}{2gH_n} + (1+\varsigma_3)\frac{c_3^2}{2gH_n} + \frac{E_I^2}{2gH_n} \tag{3.1}$$

In this way all components of the equation have become dimensionless and all energy quantities are expressed as fractions of the total net energy. Since all components on the right hand side of Equation (3.1) are expressions of velocity heads, it is convenient to define

$\underline{c} = \dfrac{c}{\sqrt{2gH_n}}$ as reduced absolute velocity

$\underline{u} = \dfrac{u}{\sqrt{2gH_n}}$ as reduced peripheral velocity

$$\underline{v} = \frac{v}{\sqrt{2gH_n}}$$ as reduced relative velocity

Velocity diagrams based on dimensional values of the velocities are valid for only one single value of the net head H_n. If reduced velocities however, present the corresponding velocity diagrams, these diagrams keep a similar shape. The velocity diagrams based on reduced velocities are therefore beneficial because these diagrams are valid for any value of H_n.

Additional useful reduced quantities are:

$$\underline{h} = \frac{h_p}{H_n}$$ is reduced piezometric head

$$\underline{Q} = \frac{Q}{\sqrt{2gH_n}}$$ is reduced discharge

$$\underline{\omega} = \frac{\omega}{\sqrt{2gH_n}}$$ is reduced angular velocity

Insertion of the reduced parameters in the version (2.23) of the main turbine equation gives

$$\eta = 2(\underline{u}_1\underline{c}_{u1} - \underline{u}_2\underline{c}_{u2}) \tag{3.2}$$

A turbine operating at different heads

Similar flow conditions through a turbine for all values of the net head is prevailing when all reduced velocities are constant. This statement is based on the assumption that the coefficients of losses are independent of the net head H_n. This is valid if the magnitude of H_n does not become extremely large and the pressure in the turbine and the draft tube does not fall below vapour pressure.

Moreover, for similar flow conditions the following relations are fullfilled

- the angular velocity $\dfrac{\omega}{\sqrt{H_n}}$ is constant (3.3)

- the discharge $\dfrac{Q}{\sqrt{H_n}}$ is constant (3.4)

Geometrically similar, but different size of the turbines

Similar flow conditions in geometrically similar turbines create common reduced velocity diagrams for corresponding positions in the turbines.

Taking the ratio between corresponding dimensions may compare the size of geometrically similar turbines. For similar operating conditions the following relations are then valid:

- for the reduced angular velocity the product $\underline{\omega}\,D$ is constant (3.5)

- for the reduced discharge the ratio $\dfrac{\underline{Q}}{D^2}$ is constant (3.6)

By combining these two equations for elimination of D, is obtained

$$\underline{\omega}\sqrt{\underline{Q}} \text{ is constant} \tag{3.7}$$

In these considerations the efficiency of the turbines has been assumed the same. This is an approximation, but deviation between model turbines and the large prototypes are corrected for by upscaling formulas.

3.1.2 Speed number

The reduced discharge $\underline{Q}$ has the dimension of an area, and gives a direct measure of the size of
the turbine. The reduced discharge however, is dependent of the guide vane openings and the angular velocity. Therefore it is reasonable to choose the reduced discharge at the best operating conditions, i.e., maximum efficiency point, of the turbine as its *capacity*/7/. The capacity is designated with a star as $^*\underline{Q}$. To designate the best efficiency point accordingly, other parameters too are marked in the same way so as $^*\omega$, $^*Q$, $^*c$, $^*v$ etc..

The capacity is as stated only a measure of the size of the turbine and does not give any basis for the shape of the axial section of the runner. To get a criterion for the shape the similarity relations must be applied. Further the notation *unit of turbine size* is introduced. This unit means that a capacity equal to 1 is unit area.

A turbine with capacity $^*\underline{Q}$ and diameter D has a geometrically similar unit turbine with a corresponding diameter D_U and capacity 1. By a given set of operating conditions the considered turbine has a reduced discharge $\underline{Q}$ and a reduced angular velocity $\underline{\omega}$. The corresponding unit turbine for the similar conditions has a reduced discharge Q_U and reduced angular velocity $\underline{\Omega}$. According to the similarity relations Equations (3.3 and 3.4):

$$\frac{D}{D_U} = \sqrt{\frac{\underline{Q}}{\underline{Q}_U}} = \frac{\underline{\Omega}}{\underline{\omega}}$$

At the best efficiency point where the capacity of the unit turbine is 1, the unit turbine has the reduced angular velocity

$$^*\underline{\Omega} = {}^*\underline{\omega}\sqrt{^*\underline{Q}} \tag{3.8}$$

which is denoted the *speed number*/7/. The speed number is dimensionless and all geometrically similar turbines have the same speed number.

Admission

The *admission* or capacity ratio is defined as

$$\kappa = \frac{\underline{Q}}{^*\underline{Q}} \text{ at the angular reduced speed } {}^*\underline{\omega} \text{ of the best operating point.} \tag{3.9}$$

The admission is a quantity proportional to the opening of the guide vane cascade of turbines, and it is dimensionless. Its value is $\kappa = 1.0$ for the best operating point, i.e., for $\underline{Q} = {}^*\underline{Q}$ and $\underline{\omega} = {}^*\underline{\omega}$. Some turbines are designed to operate with overload. That means the turbine can operate with a maximum discharge larger than $^*Q$. This maximum discharge is designated $^\bullet Q$. The corresponding value of the admission $^\bullet\kappa = {}^\bullet\underline{Q}/{}^*\underline{Q}$ for $\underline{\omega} = {}^*\underline{\omega}$.

Note

The speed number may also be expressed by the application of the parameters $^*Q$, H_n and $^*n$ directly

$$ ^*\underline{\Omega} = \frac{\pi\, ^*n\sqrt{^*Q}}{30\sqrt[4]{(2gH_n)^3}} \tag{3.10} $$

Another classification parameter than the speed number is also in use. That is the *specific speed* defined in two ways

$$ n_q = \frac{n\sqrt{Q}}{H_n^{3/4}} \quad \text{or} \quad n_s = \frac{n\sqrt{^\bullet P}}{H_n^{5/4}} \tag{3.11} $$

where n is the rotational speed
Q is discharge at the best efficiency point
H_n is the net head at the best efficiency point
$^\bullet P$ is the maximum turbine power

These two expressions of specific speed are not dimensionless, which is the fact for the speed number. The relations between these specific speeds and the speed number are

$$ n_q = 89\, ^*\underline{\Omega} \quad \text{and} \quad n_s = 379\sqrt{^\bullet\eta\, ^\bullet\kappa}\, ^*\underline{\Omega} \tag{3.12} $$

where $^\bullet\eta$ is the efficiency at maximum turbine power
$^\bullet\kappa$ is the admission at maximum turbine power

3.1.3 Classification of turbines

The speed number is a parameter for classification of the turbines. This means that the different types of turbines group themselves in certain ranges of speed numbers. The ordinary turbines cover ranges as specifically marked on a speed number scale as shown in Fig. 3.1.

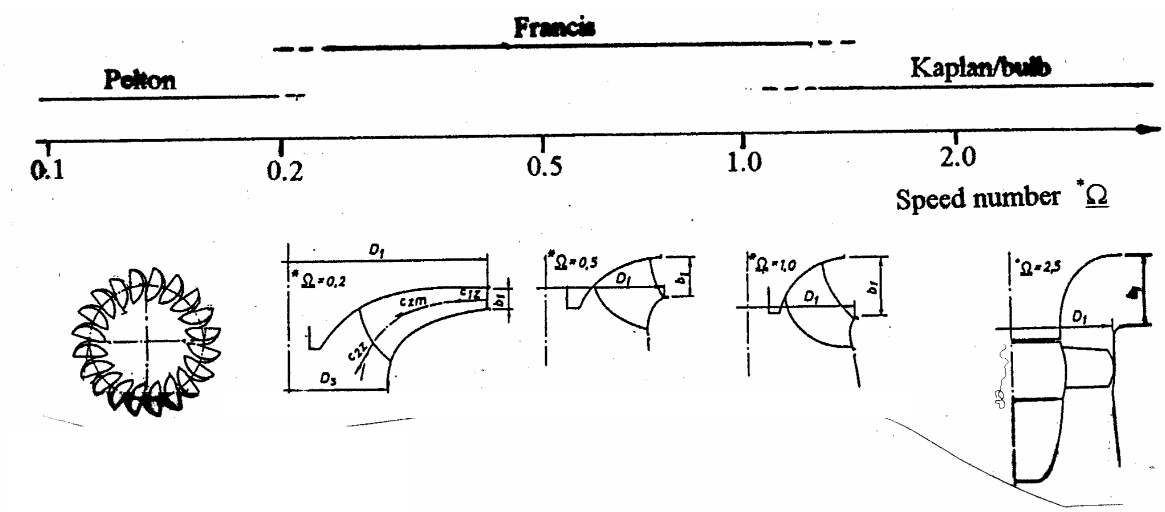

Fig. 3.1 Water turbines classified by speed number $^*\underline{\Omega}$

A main purpose for turbine design is to obtain an optimal adaptation to the plant conditions. When the discharge Q and net head H_n are known, this involve choice of the rotational speed n.

The criterion then is to choose n equal to the maximum synchronous speed for which the turbine-generator unit becomes economically optimal.

According to such guidelines Pelton turbines are built for speed numbers $^*\underline{\Omega} \leq 0.22$, that means for plants with relatively low discharge Q and the largest net heads H_n.

Qualitatively spoken Francis turbines are applied when the discharge Q is larger and the net head H_n lower than suitable for Pelton turbines. The range of Francis turbines $0.2 < {}^*\underline{\Omega} < 1.25$. The lower part of the speed numbers represents the so-called *radial runners*, which have a limit range $2.0 < D_1/D_s < 2.5$.

For increasing discharge Q related to decreasing net head H_n the speed number increases. Then the relation D_1/b_1 between diameter D_1 and the width b_1 decreases and the outlet edge of the blades is positioned farther downstream in the runner.

For speed numbers $*\underline{\Omega} > 1.0$ the discharge Q is as large and the net head H_n as low that the choice has to be Kaplan or Bulb turbines. However, the region $1.0 < {}^*\underline{\Omega} < 1.25$ may be characterised as choosing zone of either Francis or Kaplan turbine.

3.1.4 Performance characteristics

Introduction

By theoretical analysis and available methods for computation it is not possible, even by advanced approaches, to obtain exact results of the real flow state and the performance of turbomachines. The accuracy will be poor or unreliable especially for operating conditions far from the design point.

Therefore experimental research is necessary to carry out in laboratories on models of the prototype turbines. Results from these model tests are valid for the prototype through the similarity relations and certain upscaling formula for the efficiency, provided that the model size is larger than certain minimum international standardised values. A model turbine has to be geometric similar to the prototype in all hydraulic passages from the turbine inlet to the outlet of the draft tube.

A standard for model testing of water turbines is the International Electrotechnical Commission (IEC) Recommendation, Publication 193. In general the code applies to any type of reaction or impulse turbine tested under prescribed laboratory conditions and may accordingly be used for acceptance tests of the prototype turbines as well.

Determination of performance characteristics

Parameters to be measured are those determining the power P_n delivered to the turbine and the power P transferred to the turbine shaft. These powers determine the efficiency, which is expressed in Chapter 2, Equation (2.31):

$$\eta = \frac{P}{P_n}$$

where $P_n = \rho Q g H_n$ is the hydraulic power supplied to the turbine
$P = T\omega$ is the power out of the turbine shaft
Q is the discharge
H_n is the net head
T is the torque

- ω is the angular velocity
- ρ is the density of water
- g is the acceleration of gravity

For the determination of η the hydraulic parameters Q and H_n, and the mechanical parameters T and ω have to be measured in each test point. For the density it is usually sufficient to ascertain the value according to the measured fluid temperature and pressure. The acceleration of gravity g is ascertained according to latitude and altitude.

According to the regulation facilities of the turbines there are also some turbine parameters to be measured. These are the opening of the guide vane canals and in addition the slope of the runner blades on Kaplan and Bulb turbines.

The *testing procedure* is to carry out these efficiency measurements for certain points within an estimated range of operation of the turbine by stepwise changes of the rotational speed and the turbine parameters.

The measured data has to be treated for presentation of the turbine performance curves in a two-dimensional diagram. The efficiency is calculated for each test point directly from the measured data. The measured discharge Q, rotational speed n and power P are recalculated to conditions for net head H_n as constant according to the similarity relations. The resulting data are then ready to be set up in diagrams.

One principle for setting up diagrams is shown in Fig. 3.2. There are two diagrams, with η-curves in the upper one and Q-curves in the lower one, both with the rotational speed n as abscissa. Each of the curves in both diagrams is marked with a corresponding value of the admission $\kappa = Q/{}^{*}Q$. For this illustration of the treating procedure only four κ-values are indicated.

The optimal point on each η-curve is marked. The $\eta_{max.}$-point may be determined by drawing a side projection of the η-optimal-points as shown to the right on Fig.3.2. This point is marked by projection into the Q - n-diagram, and the best operating point of the turbine is then found by the indicated values *Q and *n.

In normal practise the presentation of one diagram for η and one for Q as functions of n is further worked out in one single diagram. This is done by converting the efficiency curves in the upper diagram of Fig. 3.2, into corresponding curves where η is constant in the Q - n-diagram as shown. The result obtained is a hill chart of efficiencies of the turbine, and this presentation is designated performance diagram.

Performance diagrams based on reduced parameters

The results from the efficiency measurements of a model turbine can be converted to the geometric unit turbine on the basis of the similarity relations. First may be determined:

turbine capacity $${}^{*}\underline{Q} = \frac{{}^{*}Q}{\sqrt{2gH_n}} \tag{3.13}$$

reduced angular speed $${}^{*}\underline{\omega} = \frac{{}^{*}\omega}{\sqrt{2gH_n}} \tag{3.14}$$

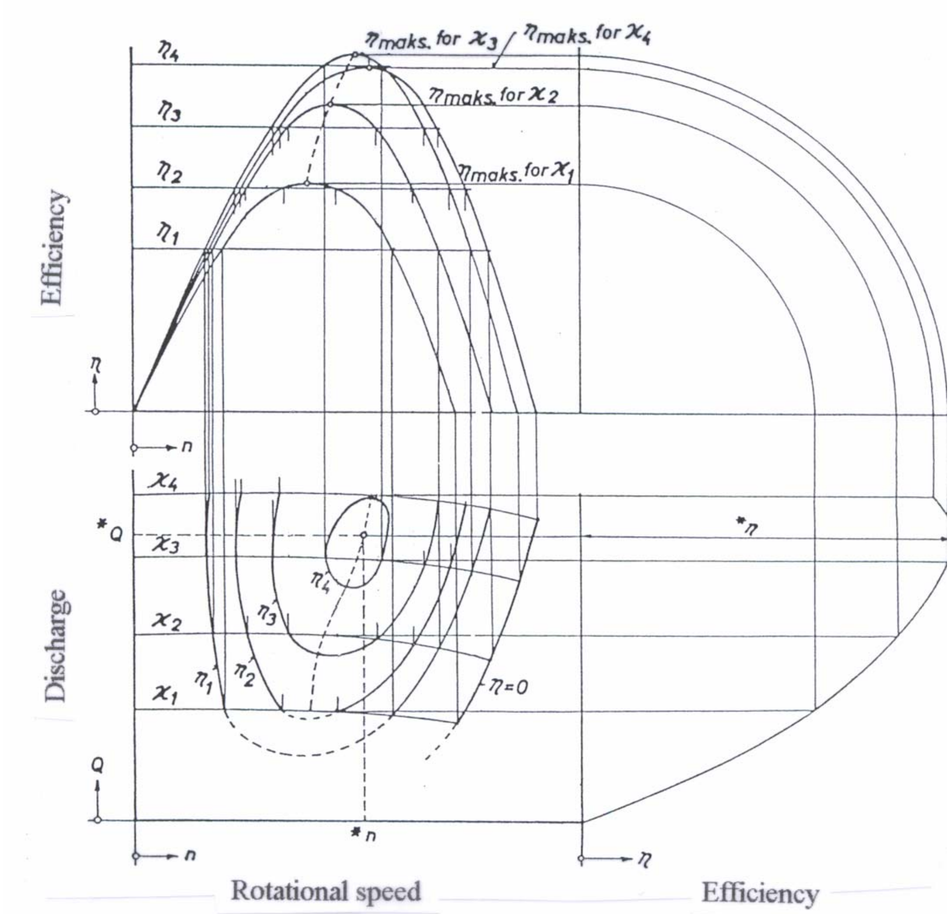

Fig. 3.2 Sketched drawing of the performance diagram of a turbine /5/

speed number $$^*\Omega = {}^*\omega\sqrt{{}^*Q} \qquad (3.15)$$

where $^*\underline{\Omega}$ also represents the angular speed of the geometric similar unit turbine at the best efficiency point.

Considering now an arbitrary point of operation of the test turbine, the reduced discharge through the unit turbine corresponding to this point is

$$\underline{Q}_U = \frac{\underline{Q}}{^*\underline{Q}} \qquad (3.16)$$

For the performance diagram it is reasonable to introduce $\underline{Q}/{}^*\underline{Q}$ instead of Q for the ordinate. For the abscissa it is adequate to introduce the ratio between the unit turbine angular speed $\underline{\Omega}$ and the speed number $^*\underline{\Omega}$ so that

$$\frac{\underline{\Omega}}{^*\underline{\Omega}} = \frac{\underline{\omega}}{^*\underline{\omega}} \qquad (3.17)$$

By this presentation of the performance diagram is obtained an efficiency hill chart which is *valid for all turbines geometric similar to the unit turbine*.

3.1.5 Cavitation and suction head

Cavitation

When the pressure in a liquid is lowered down to vapour pressure, cavities are created in the liquid, i.e., bubbles filled with air and vapour. This may occur in the low-pressure regions of the turbines, especially at the outlet of runners and inlet of the draft tube in reaction turbines.

However, the cavity bubbles will again collapse when coming into regions of higher pressure. These collapses produce a strong characteristic noise, and bubbles collapsing on surfaces of runner blades, runner discs, draft tube wall and so on, may damage the surfaces by a more or less terrific erodation. The whole range of turbine operation should therefore be free of cavitation. In practise that means to estimate maximum suction head.

Suction head

The conception of suction head is described in the IEC Publication 41.This publication presents definitions and specifications of *the net positive suction energy NPSE and the net positive suction head NPSH*

For normal calculations it is sufficient to evaluate the suction head as schematically shown in the Fig. 3.3 for different arrangements of the turbine axis. The suction head h_s is defined as positive when measured upwards and as negative downwards from the tail water level. This suction head however, does not alone determine the pressure and the cavitation conditions. But by application of the Bernoulli's equation the pressure head system can be expressed as follows

$$H_A = h_v + h_s + (1 - \varsigma_s)\frac{c_s^2}{2g} + \Delta h \qquad (3.18)$$

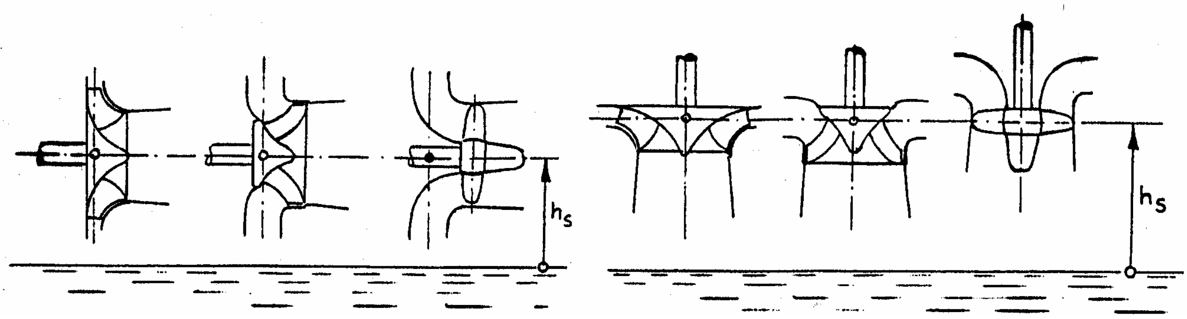

Fig. 3.3 Specification of suction head h_s of reaction turbines /5/

where h_v is the vapour pressure head
H_A is the atmospheric pressure head
h_s is the suction head
c_s is the mean velocity in the narrowest cross section area of the draft tube
ς_s is the coefficient of the total head loss in the draft tube
Δh is the resulting pressure head above the head of the vapour pressure

The pressure head Δh is just the pressure which is defined as *net positive suction head NPSH.* According to Equation (3.18) this is

$$NPSH = H_A - h_v - h_s - (1 - \varsigma_s)\frac{c_s^2}{2g} \tag{3.19}$$

Cavitation limits

The performance of a turbine under operating conditions with cavitation, can be experimentally investigated in several ways. Depending on the physical phenomenon as basis for the investigation, the following methods of detecting cavitation may be mentioned:

- By the change of the hydraulic performance of the machine as expressed by head, power, capacity or efficieny.
- By visual or photographic observation of the vapour pockets or bubbles on the runner blades.
- By observation and, if possible, measurements of the noise and vibration accompanying the operation of the machine.

Of these methods only the first one has so far given proper reliable results of practical value. On the other hand, it is also recognised that a change in the hydraulic performance is not a sufficient reliable indication of cavitation. The reason is that appreciable noise and other indications of cavitation have at times been observed without accompanying changes in any of the performance characteristics.

Testing of cavitation in a turbine has to be carried out on a model in a laboratory. The NPSH of the turbine is gradually reduced under otherwise constant operating conditions. As long as the efficiency η, the discharge Q and the power P of the turbine remain at constant values, no cavitation occurs. Conditions with cavitation occur as soon as η, Q and P start decreasing and the limit value of NPSH is then reached.

Similarity relations to include cavitation

Provided that similar hydraulic cavitating flow remain unchanged relative to the flow canals, the relations of hydraulic similar flow, i.e. Equations (3.3) and (3.4), are valid also for flow including cavitation.

The net head H_n in the Equations (3.3) and (3.4) may then be replaced by NPSH. The law of Thoma is introduced

$$\sigma = \frac{NPSH}{H} \quad \text{is constant} \tag{3.20}$$

which is valid for similar cavitating conditions in turbines and pumps.

3.2 Pelton turbines

3.2.1 Main hydraulic dimensions

Pelton turbines represent the lowest region of speed numbers and may extend a tiny overlap in the lower end of the Francis turbine range. The reduced angular velocity of a Pelton turbine is $\underline{\omega} = 2\underline{u}_1/D_1$. If the turbine has one single jet with a reduced velocity $\underline{c}_1$ and a jet diameter d, then the reduced discharge becomes $\underline{Q} = (\pi/4)d^2\underline{c}_1$. Correspondingly the speed number with one jet

$$^*\underline{\Omega} = {}^*\underline{\omega}\sqrt{^*\underline{Q}} = \frac{2\,^*\underline{u}_1}{D}\sqrt{\frac{\pi}{4}\,^*d^2\,\underline{c}_1} \tag{3.21}$$

Intended values of the reduced velocities are

$\underline{c}_1 = 0.98$ and $^*\underline{u}_1 = 0.485$

The largest speed number with one jet is in practise limited to $^*\underline{\Omega} = 0.1$. This high value is not recommended because maximum jet diameter $^\bullet d$ has to be kept $^\bullet d/D < 0.1$. An example shows that at maximum admission $^\bullet\kappa = 1.4$ and $^\bullet d/D = 0.1$, one jet makes a speed number $^\bullet\underline{\Omega} = 0.085$. The maximum number of jets is considered as six. In this case the speed number becomes

$^\bullet\underline{\Omega} = 0.085\cdot\sqrt{6} = 0.21$

Pelton turbines can be applied up to heads around 2000 m, however with decreased speed number because that is favourable in regard to avoid high stresses and fatigue problems and cavitation damage.

3.2.2 Pelton bucket dimensions

To indicate intended values as a guide to the bucket dimensions a specifying concept/7/ is given on Fig. 3.4. According to the specifications on this figure the following values should be acceptable.

$B = 2.8$ to $3.4\cdot{}^\bullet d$
$L = 2.3$ to $2.7\cdot{}^\bullet d$
$x = 1.0$ to $1.1\cdot{}^\bullet d$
$y = 1.3$ to $1.6\cdot{}^\bullet d$
$z = 1.1$ to $1.2\cdot{}^\bullet d$

Bucket section longitudinal of the jet — View perpendicular to the jet — Velocity triangles at the inlet and outlet of the bucket

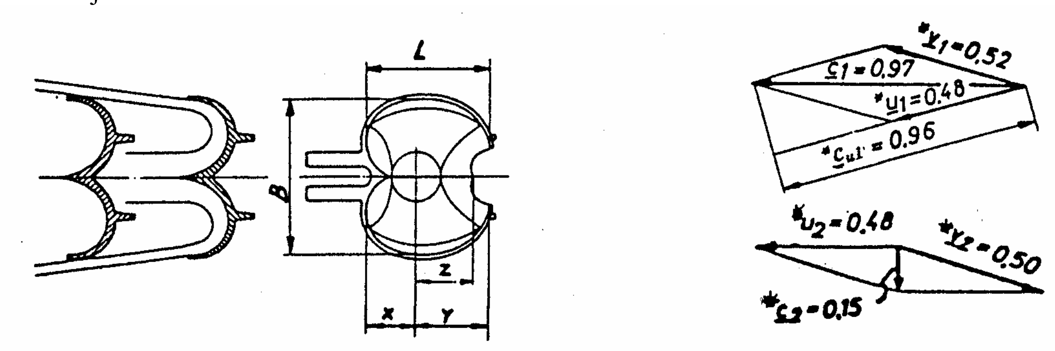

Fig. 3.4 Pelton turbine, indication of main dimensions and the bucket form /5/

3.2.3 Performance diagram

Fig. 3.5 shows an example of a performance diagram for a unit Pelton turbine with speed number $^*\underline{\Omega} = 0.04$. That means a one jet turbine and this diagram is valid for all geometric similar Pelton turbines with the same speed number. The efficiency is represented by relative efficiency curves

$\underline{\eta} = \eta/\eta_{max}$.

In the performance diagram it is of major interest to examine the efficiency η as a function of the discharge Q along the ordinate for $\underline{\omega}/^*\underline{\omega}$ =1.0. It is then observed a rather steep progress from a relative efficiency $\underline{\eta} = 0$ at $Q \approx 0$ up to $\underline{\eta} = 0.9$ at $Q \approx 0.25{}^*Q$. Further it increases to $\underline{\eta} = 0.95$ at $Q \approx 0.5{}^*Q$, and $0.95 < \underline{\eta} < 1.0$ for the interval $0.5{}^*Q < Q < 1.5{}^*Q$.

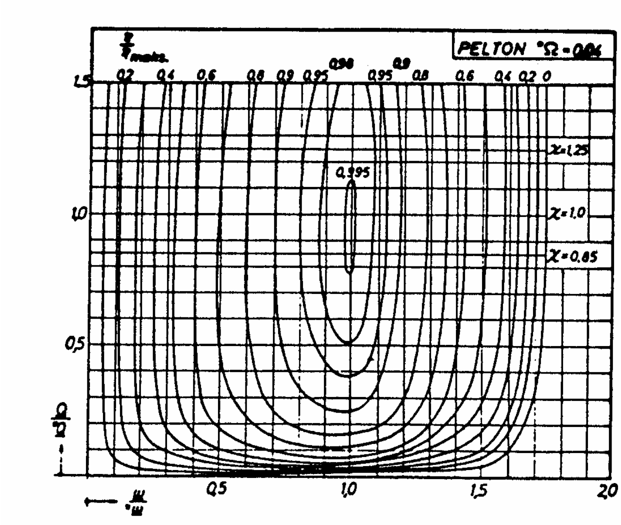

Fig. 3.5 Performance diagram[5] *of a Pelton turbine,* $^*\underline{\Omega} = 0.04$

Another important characteristic is the *runaway speed.* This can be read in the diagram for $\underline{\eta} = 0$ at $\underline{Q}/^*\underline{Q} = 1.0$. The read value $\underline{\omega}/^*\underline{\omega} \approx 1.75$, which means a runaway speed $n = 1.75{}^*n$.

The admission κ can be read along the ordinate of $\underline{\omega}/^*\underline{\omega} = 1.0$. In diagram Fig. 3.5 there are indicated three κ-curves, and these curves are lines parallel to the abscissa axis and therefore independent of the runner speed. This is always the case in Pelton turbines because the jet passes through the free open air from the nozzle to the bucket.

The shape of the hill chart is naturally dependent on the speed number. This dependency however, does appear only with relatively small changes.

3.3 Francis turbines

3.3.1 Main hydraulic dimensions

Francis turbines are located in the region of speed numbers $0.2 < {}^*\underline{\Omega} < 1{,}25$. Hydraulic design of these turbines is based on reduced quantities. Intended values, which may be applied as a guide of the reduced velocities, are given as functions of the speed number in the diagram Fig. 3.6. Reduced peripheral velocity ${}^*\underline{u}_1$ corresponds to the largest diameter D_1 of the runner cascade. Reduced meridional velocity ${}^*\underline{c}_s$ corresponds to the smallest cross section of the runner outlet and reduced meridional velocity ${}^*\underline{c}_{oz}$ corresponds to the diameter of the outlet edge of the guide vanes. In the diagram it is shown a curve of the admission ${}^\bullet\kappa$, which indicate values of the opening of the guide vane cascade for the admission at maximum discharge and maximum turbine output power. As shown in the diagram, this parameter decreases as the speed number increases. The reason is that runners of low speed numbers perform a more even efficiency curve than runners of high-speed numbers. For the extreme high-speed numbers of Francis turbines ${}^\bullet\kappa$ approaches values between 1.1 and 1.2.

The range of speed numbers for Francis turbines indicates that they may be applied for heads within a wide range. Turbines of the lowest speed numbers are today built for heads up to 700 m, and turbines of the highest speed numbers may be built for any low head of some few meters.

3.3.2 Performance diagrams

The Figures 3.7 and 3.8 show two examples of performance diagrams of unit Francis turbines,

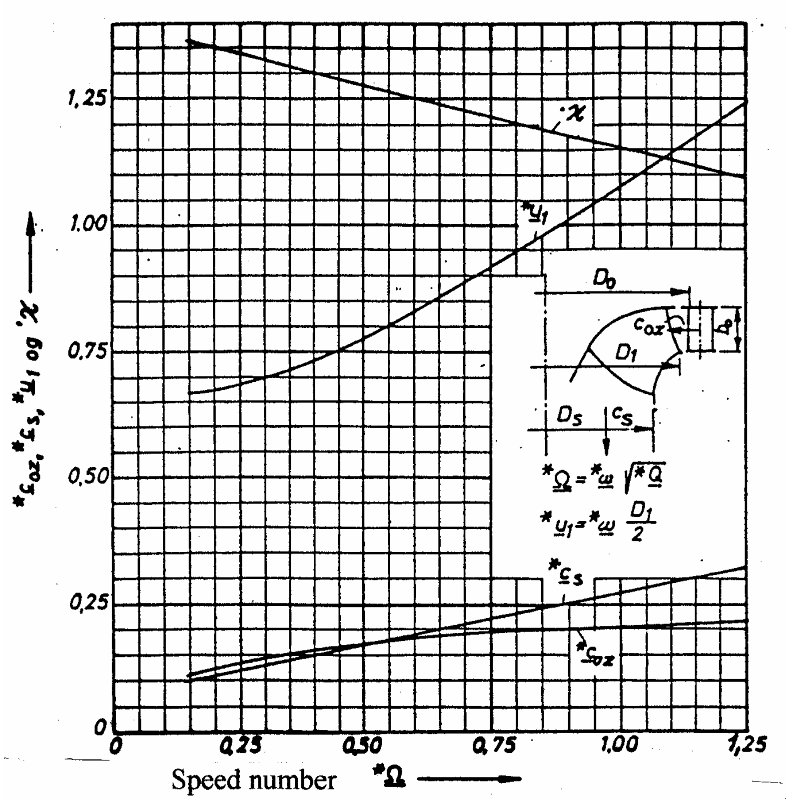

Fig. 3.6 Hydraulic design data for Francis turbines (intended values as a guide) /4/

the one on Fig. 3.7 with a speed number $^*\underline{\Omega}$ = 0.25 and the other on Fig. 3.8 with a speed number$^*\underline{\Omega}$ = 0.75. The shapes of the hill chart of these two diagrams indicate considerable differences. Both the "copiousness" and "width" on the top of the hill chart decrease all the way from the low speed Pelton turbines to the most high-speed Francis turbines.

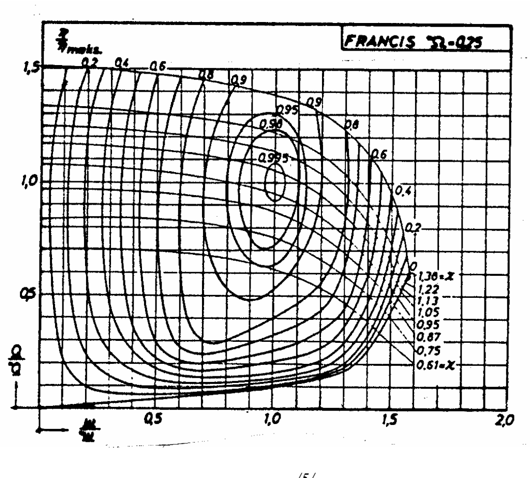

Fig. 3.7 Performance diagram/5/ of a Francis turbine
$^*\underline{\Omega}$ = 0.25

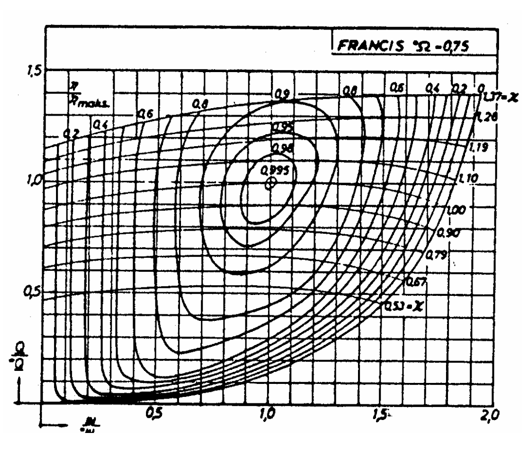

Fig.3.8 Performance diagram/5/ of a Francis turbine
$^*\underline{\Omega}$ = 0.75

The relative efficiency $\underline{\eta}$ as a function of reduced discharge $\underline{Q}$ along the ordinate through $\underline{\omega}/{}^*\underline{\omega} = 1.0$, is not so steep for Francis turbines as for the Pelton turbine on Fig. 3.5. For the speed number ${}^*\underline{\Omega} = 0.25$ the efficiency $\underline{\eta}$ increases from $\underline{\eta} = 0$ at $\underline{Q}/{}^*\underline{Q} = 0.1$ up to $\underline{\eta} = 0.9$ about $\underline{Q}/{}^*\underline{Q} = 0.55$ and further to $\underline{\eta} = 0.95$ about $\underline{Q}/{}^*\underline{Q} = 0.62$.

For the speed number ${}^*\underline{\Omega} = 0.75$ the efficiency increases from $\underline{\eta} = 0$ at $\underline{Q}/{}^*\underline{Q} = 0.125$ up to $\underline{\eta} = 0.9$ about $\underline{Q}/{}^*\underline{Q} = 0.62$ and further to $\underline{\eta} = 0.95$ about $\underline{Q}/{}^*\underline{Q} = 0.82$.

As a conclusion it may be pronounced that the efficiency characteristics of hydro turbines become more and more pointed with increasing speed number.

It is important also to notice that the idle running discharge increases with an increasing speed number. As seen from the diagrams the reduced idle running discharge has the value $\underline{Q}/{}^*\underline{Q} = 0.1$ for ${}^*\underline{\Omega} = 0.25$ and $\underline{Q}/{}^*\underline{Q} = 0.125$ for ${}^*\underline{\Omega} = 0.75$.

The runaway speed increases also with increasing speed number. For ${}^*\underline{\Omega} = 0.25$ and $\kappa = 1.0$ the runaway speed is about $n = 1.5{}^*n$, and for ${}^*\underline{\Omega} = 0.75$ and $\kappa = 1.0$ the running speed is about $n = 1.8{}^*n$.

Another quality to notice is the different courses of the admission κ-curves in the performance diagrams on the Figures 3.7 and 3.8 respectively.

3.3.3 Cavitation, suction head and reaction ratio

In Section 3.1.5 cavitation and suction head was considered and Thoma coefficient σ introduced. This is here supplemented by a method being used by a turbine manufacturer, to determine the NPSH.

The peripheral velocity u_2 at the outlet of the runner is a significant parameter for estimation of the tendency of cavitation and finally the setting of the turbine [/2/]. Therefore the speed number and the NPSH are converted to expressions with a dependency of the velocity u_2.

The speed number expressed by the reduced velocity $\underline{u}_2$ and blade angle β_2 at the runner outlet is

$$ {}^*\underline{\Omega} = \underline{u}_2^{3/2} \sqrt{\pi \tan(\pi - \beta_2)} \tag{3.22} $$

An equation for NPSH is established as follows

$$ \mathrm{NPSH} = H_n \, \underline{u}_2^2 \left[a{}^{\bullet}\kappa^2 \tan^2(\pi - \beta_2) + b \right] = \frac{u_2^2}{2g} \left[a{}^{\bullet}\kappa^2 \tan^2(\pi - \beta_2) + b \right] \tag{3.23} $$

where $1 < a < 1.2$ and $0.05 < b < 0.5$ depending on the speed number, blade number and blade geometry. ${}^{\bullet}\kappa$ is the admission of maximum discharge.

According to Equation (3.23) a given setting of a Francis turbine (and also for a Kaplan turbine) appoints a limit both for the speed number ${}^*\underline{\Omega}$ and the peripheral velocity u_2 at the rim of the outlet of the runner due to tendency of cavitation.

For the blade loading and cavitation occurrence the *reaction ratio* is also a very important parameter. The reaction ratio is describing the pressure drop from the runner inlet to the outlet at best efficiency flow, i.e. $c_{u2} = 0$ in Equation (2.22) or (2.23). If the losses in Equation (2.24) are neglected, this equation becomes

$$ 2(\underline{u}_1\underline{c}_{u1} - \underline{u}_2\underline{c}_{u2}) = \underline{h}_1 + \underline{c}_1^2 - \underline{h}_2 - \underline{c}_2 \tag{3.24} $$

Now $\underline{c}_1^2 = \underline{c}_{u1}^2 + \underline{c}_{1z}^2$ and $\underline{c}_2^2 = \underline{c}_{u2}^2 + \underline{c}_{2z}^2$. Further is assumed $\underline{c}_{1z} = \underline{c}_{2z}$. By introducing these transformations into Equation (3.24), the reaction ratio is

$$\underline{h}_1 - \underline{h}_2 = 2\underline{u}_1\underline{c}_{u1} - \underline{c}_{u2}^2 = \underline{c}_{u1}(2\underline{u}_1\underline{c}_{u1} - \underline{c}_{u2}) \tag{3.25}$$

Decreased <----------- Reaction ratio ------------> Increased

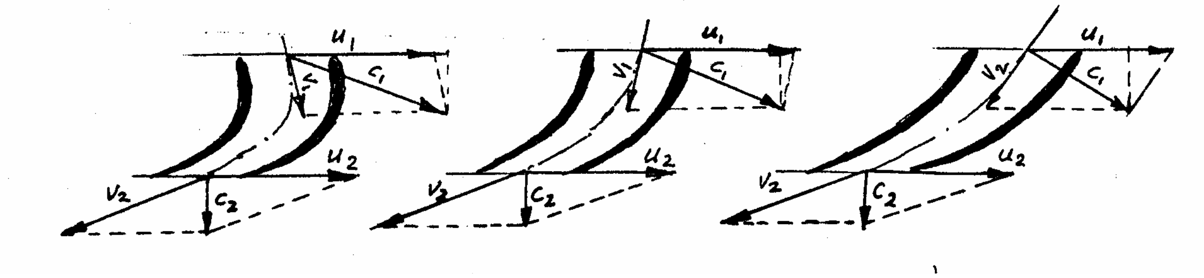

Fig. 3.9 Blade cascades for different reaction ratios with $\underline{c}_{u2} = 0$; /2/.

Fig. 3.9 is illustrating qualitatively different magnitudes of $\underline{c}_{u1}$ depending on the different inlet directions of the blades, and consequently different reaction ratios.

The reaction ratios for different Francis turbines may be clarified by an example. For the best efficiency point of two turbines a hydraulic efficiency $\eta_h = 0.97$ is assumed. From the diagram Fig. 3.6 intended values of the peripheral velocity $^*\underline{u}_1$ may be read.

Francis turbine: $^*\underline{\Omega} = 0.35$, $^*\underline{u}_1 = 0.72$ (from Fig. 3.6) amounts a reaction ratio $\underline{h}_1 - \underline{h}_2 = 0.516$
Francis turbine: $^*\underline{\Omega} = 1.0$, $^*\underline{u}_1 = 1.075$ « « amounts a reaction ratio $\underline{h}_1 - \underline{h}_2 = 0.766$

The reaction ratio is consequently increasing with increasing speed number. For the lower end of the speed number range the reaction ratio is approaching that of the Pelton turbine when $\underline{u}_1/\underline{c}_{1x}$ --> 0.5 and ($\underline{h}_1 - \underline{h}_2$) --> 0. For increasing speed numbers, the runners approach those of Kaplan/Bulb turbines with the highest reaction ratios.

Cavitation at the inlet of the blade cascade in low head turbines, i.e. Francis and Kaplan/Bulb turbines, is significantly linked with blade loading.

3.4 Kaplan turbines

3.4.1 Main hydraulic dimensions

Kaplan and Bulb turbines belong to the same grope of turbines. They represent the extension of speed numbers above the range of Francis turbines, however with a minor overlap in the Francis range, and consequently expressed by

$$^{\bullet}\underline{\Omega} = {^{\bullet}\underline{\omega}} \sqrt{^{\bullet}\underline{Q}}$$

indicating maximum discharge and maximum output power from the turbine. According to this convention the Kaplan and Bulb turbines are located in the region of speed numbers $1.5 \leq {^{\bullet}\underline{\Omega}} < 3.5$.

In the diagram Fig. 3.10, curves are given for average values of the reduced velocities $^*\underline{u}_1$, $^{\bullet}\underline{c}_s$ and $^{\bullet}\underline{c}_{oz}$ corresponding to the locations given in the diagram as functions of the speed number.

The speed number $^{\bullet}\underline{\Omega}$ does not represent the best efficiency point of the turbine operating conditions. Because of the excellent regulating possibilities of the Kaplan and Bulb turbines they perform a relatively even efficiency curve. In practise therefore it seems favourable to design the turbine for the best operating conditions about a capacity $^{*}\underline{Q} = 0.65\cdot{}^{\bullet}\underline{Q}$ to $0.70\cdot{}^{\bullet}\underline{Q}$.

3.4.2 Performance diagram

An example of the performance diagram of a unit Kaplan/Bulb turbine is shown on Fig. 3.11. This diagram is rather different from those shown for the Pelton turbine Fig. 3.5 and the Francis turbines in Figs. 3.7 and 3.8. The reason is the regulation parameter for the slope of the runner blade angle φ. The regulation of the opening of the guide vane cascade and the regulation of the runner blade angle is interconnected in such a way that the turbine operates with optimal efficiency in every case of operation.

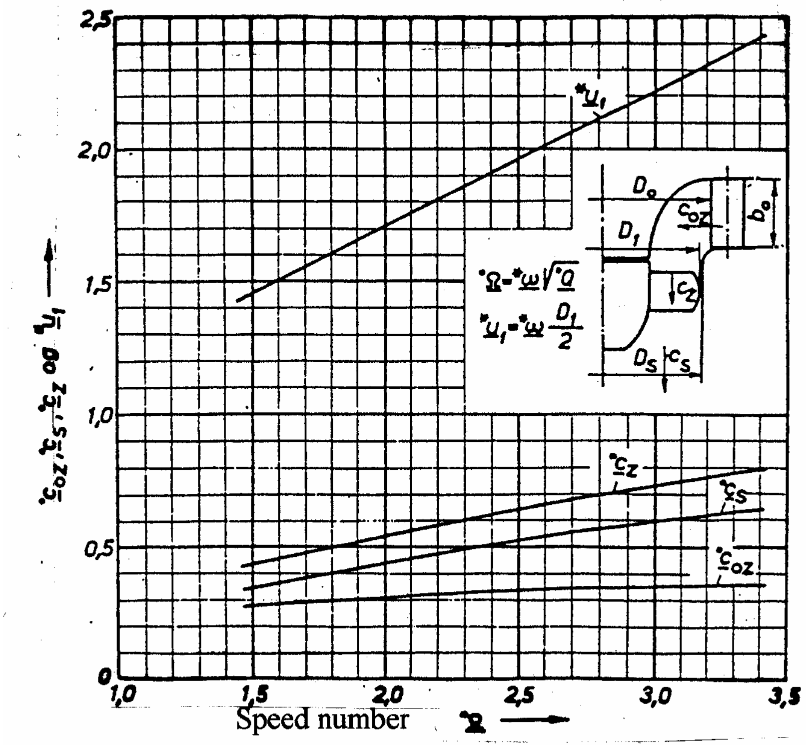

Fig. 3.10 Hydraulic design data for Kaplan turbines (intended values as a guide) /4/

To determine the overall optimal operating conditions of a Kaplan/Bulb turbine, proper model performance tests have to be carried out. The basis for the test procedure is three free variable parameters:

n = the rotational speed [RPM]
α = direction angle defining opening of the guide vane cascade[^{o}]
φ = angle defining the direction of the runner blades [^{o}]

With a chosen blade angle φ, efficiency η and discharge Q are measured for chosen values of the guide vane angle α as functions of the rotational speed. Then the blade angle φ is changed to another position and a new series of η-curves and Q-curves are measured in the same way as in the first case. Analogous series of measurements are further carried out for as many φ-values as being prescribed.

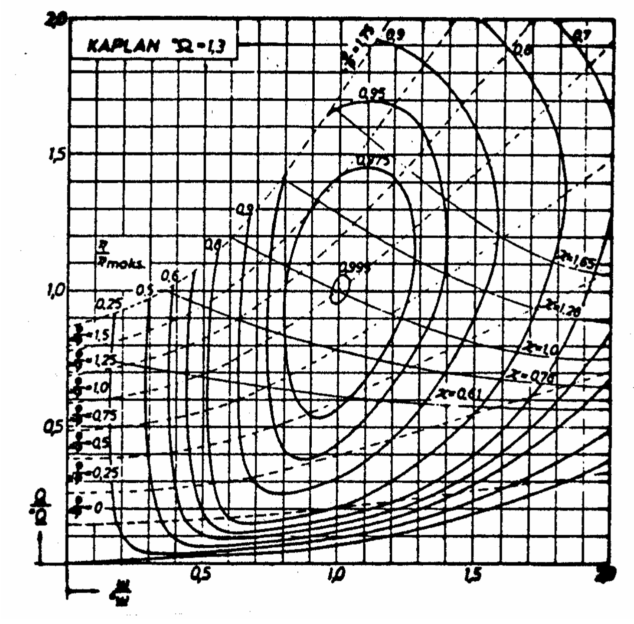

Fig. 3.11 Performance diagram/5/ of a Kaplan turbine, ${}^*\underline{\Omega}=1.3$

The succeeding treatment of the data is to work out diagrams from which the combinations of parameter values can be determined corresponding to optimal performance for any point of operation. Finally the results of this working procedure can be presented in a performance diagram as in Fig. 3.11.

In this performance diagram the relative efficiency $\underline{\eta}$ as a function of reduced discharge $\underline{Q}$ along the ordinate through $\underline{\omega}/{}^*\underline{\omega}$ = 1.0 is steeper than of the Francis turbines Figs. 3.9 and 3.10, but not so steep as of the Pelton turbine Fig. 3.7.

The runaway speed of a Kaplan/Bulb turbine is strongly dependent of the combination of the guide vane angle α and the blade angle φ and may differ from say 1.5*n to 3.0*n.

3.4.3 Cavitation, suction head and reaction ratio

The remarks about cavitation, suction head and reaction ratio in section 3.3 on Francis turbines, are valid also for Kaplan/Bulb turbines.

3.5 Choice of turbine

The four types of turbines Pelton, Francis and Kaplan and Bulb turbines mainly have each their own region of speed numbers. In that way they are ideally supplementing each other. Therefore, when the actual speed number is estimated, the determination of the turbine type is normally done as well. On the other hand there are many questions to deal with before relevant values of speed numbers are estimated. Main problems are connected with evaluation of the costs of the plant and the trends in the development of the design of turbines. Furthermore the operating conditions play a principal role.

3.5.1 Choice between Pelton and Francis turbines

The *costs* of Pelton and Francis turbines are compared in the diagram Fig. 3.12, where the abscissa divides the diagram in one upper field where the Pelton are the cheapest turbines and one lower field where the Francis are the cheapest ones.

The prices of the turbines of the same type become cheaper the higher the rotational speed. That means a decreasing price with increasing speed number. Fig. 3.13 again shows the limit price curve between Pelton and Francis fields in a (Q - H)-diagram, which also has curves of constant power output and constant rotational speed for Pelton and Francis turbines respectively.

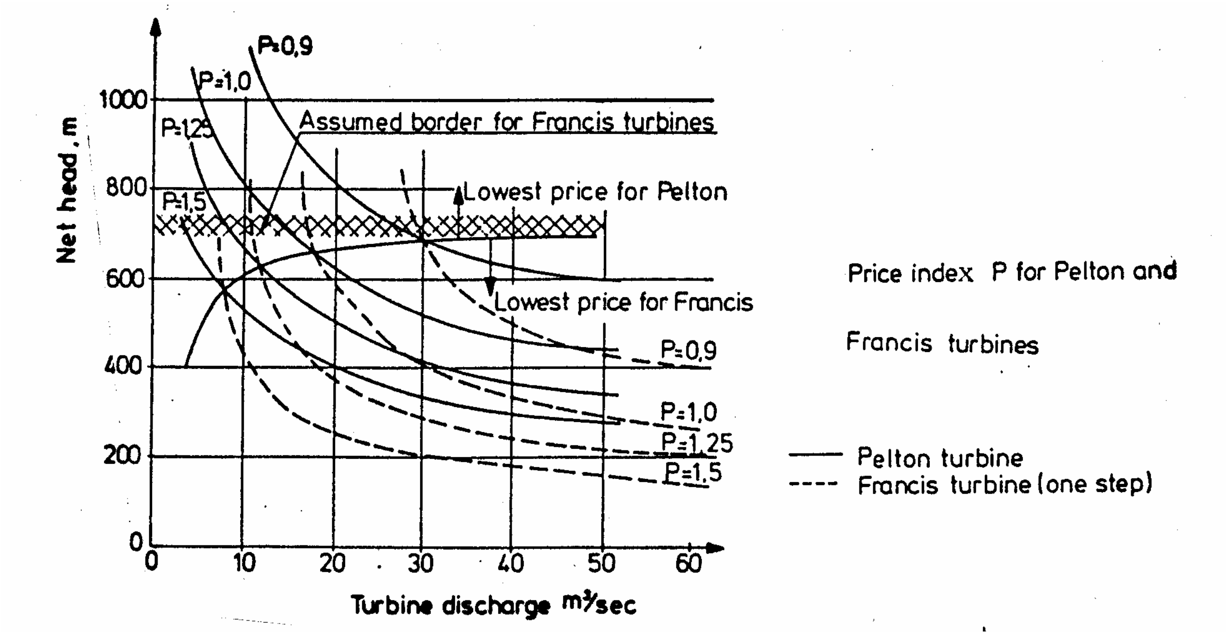

Fig. 3.12 Comparison of price index between Pelton and Francis turbines, /1/.

The Figures 3.12 and 3.13 show that Francis turbines can be the cheapest choice for heads up to 700 meters and even higher for large units. But for higher heads than this limit the Pelton turbines are ruling the domain.

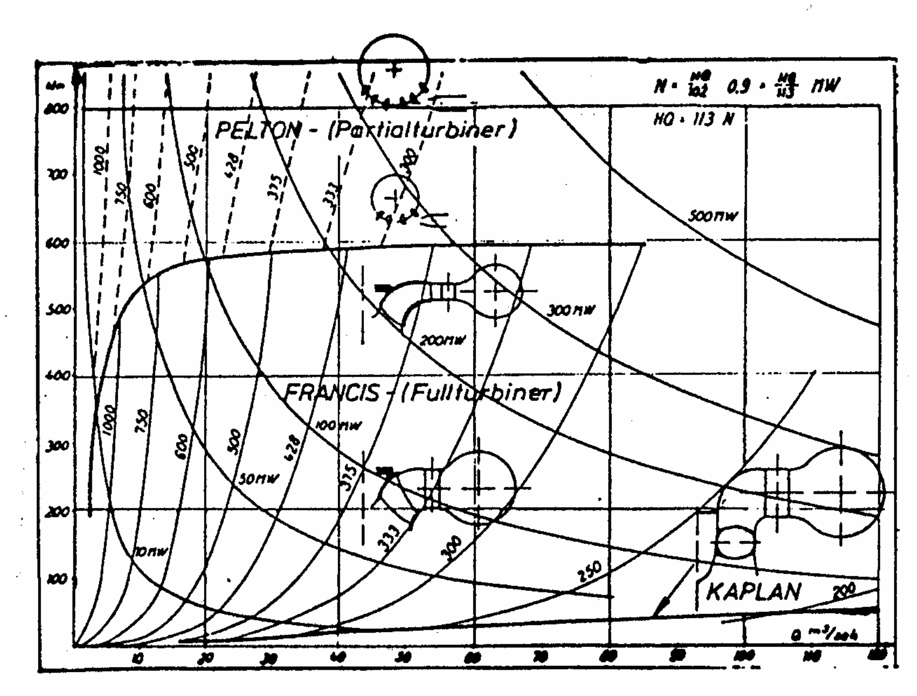

Fig. 3.13 Diagram indicating decreasing prices versus increasing speed number of turbines, /1/,/2/.

Efficiency characteristics

However, to compare the *economy* of the turbines the *efficiency characteristics* must be compared as well. Fig. 3.14 shows average qualitative efficiency curves of four types of turbines as functions of capacity ratio. Generally the Pelton and Kaplan turbines perform the wellrounded efficiency curves, while the Francis turbines and also the propeller without any adjusting control of the runner blades, perform more pointed efficiency curves the higher the speed number.

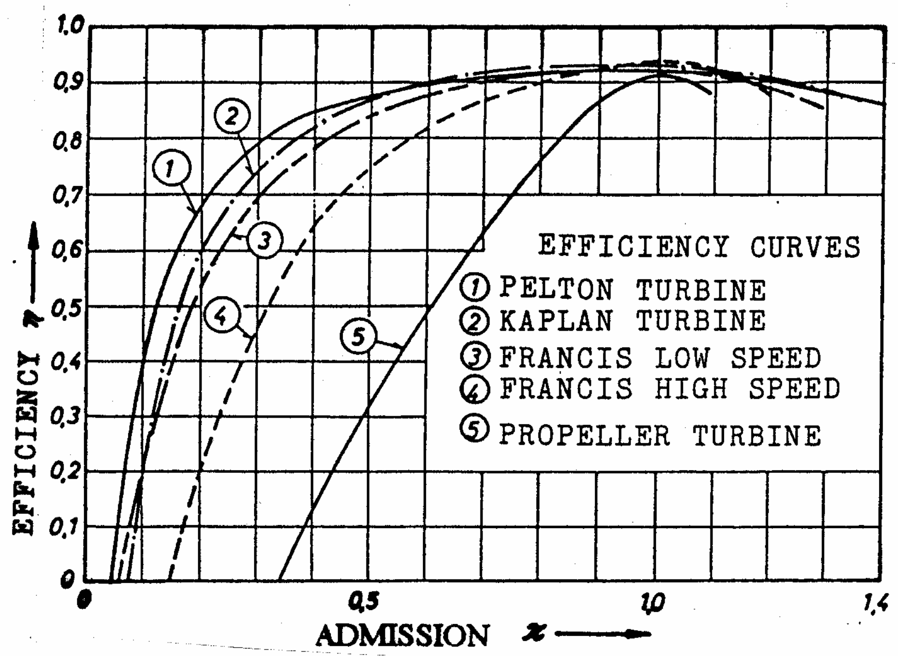

Fig. 3.14 Efficiency characteristics of different types of turbines as functions of capacity ratio /4/

For the best efficiency point it can be noticed a slight increase in the optimal efficiency value of Francis turbines with increased speed number. This is due to a reduction of frictional losses. But the character of this tendency depends also on the design and the finish of the guide vanes and the runner.

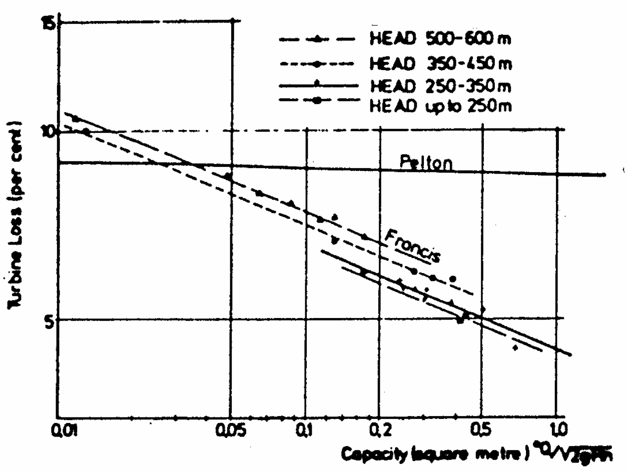

Fig. 3.15 Losses in Pelton and Francis turbines as function of capacity and , /1/,/2/.

The losses in Francis and Pelton turbines at best efficiency point versus capacity and the head as parameter are compared in the diagram Fig. 3.15 which confirms the statement mentioned above.

The losses in power production due to the variation of turbine efficiency versus power output is compared in Fig. 3.16 for a Francis turbine and a Pelton turbine, each of 50 MW at 470 m head. At best efficiency point the Francis turbine is the best choice for operation from 50 % load up to full load. However, if a typical peaking operation is wanted at very low load, the Pelton turbine will be the best choice especially if an automatic selection of operating jets is chosen.

A valuation by integration of the output loss over the whole output range of the turbines makes clear that the Pelton turbine is the best choice if the time in operation is equally distributed from zero load to full load. This is illustrated in Fig. 3.16. However, if any operation below 25 % load is avoided or the loads are higher than 25%, the Francis turbine will be the best choise.

A similar comparison between Pelton and Francis turbines may be made for different sizes and different heads of the turbines. On Fig. 3.17 boundary selection curves may be drawn as shown for different operations. The zigzag illustrates the different kinds of operations as follows: Zigzag for peaking operation from 0 to 100 % load (the lowest curve), for 25 % - 100 % load variation, for 50 % - 100 % load variation and finally best efficiency point operation ± 10 % load variation (the uppermost curve).

Time out of operation

The time out of operation for repair is a significant criterion for the choice between Francis and Pelton turbines[1]. The cases of repair are normally related to sand erosion. With sand laden water

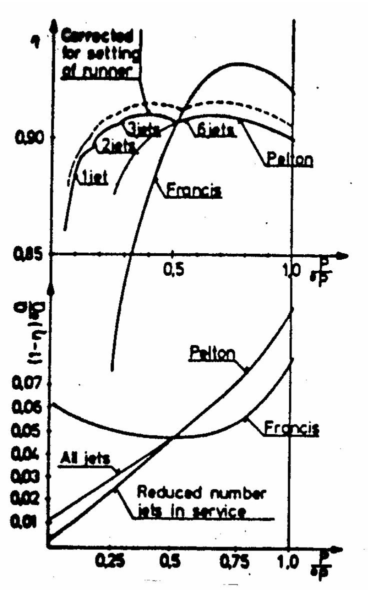

Fig. 3.16 Comparison of losses in a F rancis turbine and Pelton turbine of 50 MW at H = 470 m

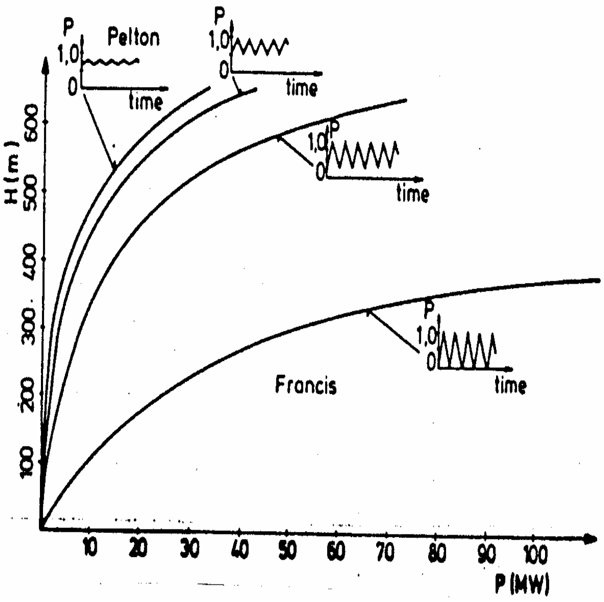

Fig. 3.17 Boundary curves between Pelton and Francis turbines based on different kinds of operation

repair may be necessary every year to keep the loss of power production at minimum. If the water is clean a well-designed turbine will be in operation at least 10 years without any repair.

The parts most exposed to sand erosion are those with the surfaces where the water velocity and the acceleration is high. Such parts are:

In Pelton turbines:
- Nozzles
- Needles
- Runner buckets

In Francis Turbines:
- Guide vanes and guide vane facing plates
- Runner inlet and outlet
- Labyrinth seals

The behaviour of the erosion from sand laden water in high head Francis turbines occurs slightly different and depends on the design of runners. This appears by a lower erosion in runners with splitter blades than in ordinary runners, because the double number of blades in the low velocity region at the runner inlet make the flow more uniform. At the outlet of the runner where the velocity is high, the number of blades is lower than in an ordinary runner, but even so the velocity distribution is fairly uniform in this region.

Dismantling and assembly of the sand eroded parts takes shorter time for Pelton than for Francis turbines. Therefore Pelton turbines will normally be preferred where much sand erosion is expected. However, this again depends rather strongly on the plant's operation schedule. If one or more turbines are stopped for a long period of for example two months a year, Francis turbines may be chosen even if the water has a high sand content because there will be enough time for an annual repair.

3.5.2 Choice between Francis and Kaplan turbines

The choice between Francis and Kaplan turbines may be made in a similar way as for Francis and Pelton turbines. But the parameters are somewhat different.

In general the Kaplan turbines are chosen for heads below 30 m. But Kaplan turbines have been built for higher heads as well even up to 60 m. The reason is to a large extent given by the more wellrounded efficiency curve compared with a low head Francis turbine (see Fig. 3.14). A Kaplan turbine offers also an advantage with its smaller dimensions for a certain capacity than the corresponding Francis turbine. Especially for large machines where capacities of 200 - 500 m^3/sec are wanted the Kaplan turbine is chosen. In this case one big high-speed unit allowing for a cheaper powerhouse than the alternative with more than one Francis turbine or one big Francis turbine which can handle such capacity.

The upper economic and practical limit for the Kaplan turbine head is in the range of 60 m, though extreme cases of 70 - 75 m have been planned for this turbine type as well. The head limit is caused by mechanical strength problems in hub and blades.

For low heads Bulb turbines will be an alternative to the Kaplan turbines. The Bulb turbine offers more favourable inlet flow conditions to the runner than a Kaplan turbine. These favourable flow conditions have the effect that the runner diameter of a Bulb turbine may be made 15 % smaller than for a Kaplan turbine under otherwise equal conditions. The flow conditions will also reduce the cavitation risk for the Bulb turbine, which means a less submergence is needed than for the Kaplan turbine.

The Bulb turbine is still more favourable if only one unit shall be built because the scroll casing of a Kaplan turbine makes the power station much wider. The Bulb turbine will however, reach an upper limit design head because of the concentrated hydraulic load on the concrete foundation through the ribs. Thus the pressure will be limited to 15 - 20 m head for this turbine type.

References

1. Brekke, H.: A Discussion of Pelton Turbines versus Francis Turbines for high Head Plants. Joint Symposium on Design and Operation of Fluid Machinery, Colorado State University Fort Collins, Colorado, USA, June 12. – 14. 1978.
2. Brekke, H.: The Layout of Power Plants and Choices of Turbine Types for Electricity Production on Isolated Load. CONFERENCIA LATINOAMERCANA DE ELECTRIFICATION RURAL. Lima, Peru, November 26. – 30. 1979.

3. Christie, H.: Features from the Design of Francis Turbines in Norway (in Norwegian). Teknisk Ukeblad no. 10, 1064, Oslo, Norway.

4. Kjølle, A.: WATER TURBINES: Hydraulic Design Base and Turbine Design and Manufacture. Lecture 2 and 3 at Central-South Institute of Mining and Metallurgy, Changsha , Hunan, China, 03.11. – 04.11.1983.

5. Kjølle, A.: Water Power Machines (in Norwegian), Universitetsforlaget, Oslo, Norway 1980.

6. Kjølle, A.: Hydraulic Measurements (in Norwegian), lectures at NTNU, 1971, Trondheim, Norway.

7. Sundby, G: WaterPower Machines, Notes from Lectures at NTH 1938 (in Norwegian). Water Power Laboratory, NTNU, Trondheim, Norway.

Bibliography

1. Nechleba, M.: Hydraulic Turbines. Artia-Prague. Constable & Co. Ltd., London, England 1957.

2. Raabe, J.: Hydraulische Maschinen und Anlagen. Zweite Auflage der Teile 1 bis 4 in einem Band. VDI-Verlag GmbH 1989.

3. Wislicenus, G. F.: Fluid Mechanics of Turbomachinery, Volume 1 and 2, Dover Publication, New York, USA, 1965.

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