HYDROPOWER IN NORWAY

Mechanical Equipment

A survey prepared

by

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Preface

This book was originally aimed to constitute a section called Mechanical Equipment, in a series of books that in total describe the THE DEVELOPMENT OF HYDRO POWER IN NORWAY. In fact much of the literary work during the preparation of this book, was however, evaluated to be a more suitable presentation of the material for a textbook than for the above mentioned series of books. Therefore it was preferred to edit this book separatly with the intention that it may partly serve as a supplementary textbook for students on hydro power machinery.

The subjects being mentioned comprise all main components of a hydro power plant from the upstream end with the basin for water intake to the downstream end of the water flow outlet. Those parts of the plant which are not specifically of the mechanical equipment category, are simply mentioned to inform about their function and position in the system structure.

For the the mechanical equipment it is given basic theory for the hydraulic design of turbines, theory and description of methods for tests of models and prototypes of turbines, theory for the dynamics in water conduits and governing of turbines. Further descriptions are given of the design structure of all actual turbines, valves, governors and auxilliary equipment. In addition a touch on forces transferred to the foundations, causes to damages on the machines, condition control and quality assurance is given.

The preparation of this book has been effected through comparisons and reflections on the material in the references, which are listet at the end of each chapter. However, my colleague Professor Dr.techn. Hermod Brekke, has earlier prepared a manuscript of a book which is containing much of the same subjects as in this case. With allowance from him, I have adopted from his manus quite a lot of the material about the design of the modern turbines and turbine governing. For this important support, I thank Professor Brekke. For the skillful scanning in of the figures, I thank head clerk Rundi Aukan.

Trondheim, December 2001 Arne Kjølle

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CHAPTER 1

Hydropower Machinery

Introduction

Hydropower machine is the designation used for a machine that directly convert the hydraulic power in a water fall to mechanical power on the machine shaft. This power conversion involves losses that arise partly in the machine itself and partly in the water conduits to and from the machine.

The utilization of the power in the waterfall is evaluated by the so-called power *plant efficency* η_a, which is the ratio between the mechanical power output from the machine shaft and the gross hydraulic power of the power plant. The plant efficiency η_a is a variabel quantity that depends on the design of the water conduits to and from the hydropower machine and the operating conditions.

The conduits are normally made with flow cross sections according to optimal design criteria. In practise that means conduit cross section areas are as small as possible to get low investment costs. However, the smaller the cross sections are, the higher the losses become. Similar consequenzies are resulting from increasing lengths of the conduits. Both these effects means a correspondingly lower plant efficiency η_a.

The hydropower machine may be operated with different flow rates Q from time to time according to the variable grid load, the alternating heads and flow discharges in the plant. These circumstances means that the hydropower machines necessarily are equipped with facilities for regulation of the power input and output. In practise this is carried out by regulation of the flow discharge.

The input power to the hydropower machine is however, not efficiently similar utilized at all operating conditions. The fact is that the machine performs the optimal efficiency for only one single combination of flow discharge, water head and rotational speed. This means a characteristic, which is denoted as the efficiency of the machine and generally expressed by

$$\eta = \frac{\text{mechanical output power}}{\text{net input power}} \tag{1.1}$$

where *net input power* means the gross hydraulic power of the power plant minus power losses in the conduits to and from the hydropower machine.

According to the regulation means another quantity is defined as the *admission* κ and expressed as

$$\kappa = \frac{\text{operating discharge}}{\text{discharge at max. efficiency}} \tag{1.2}$$

An example of the efficiency characteristic of a hydropower turbine as a function of the admission, is shown on Fig. 1.1. As being discribed in the following chapters, the efficiency charateristic may be rather different from one turbine type to the other as shown in the example Fig. 1.2.

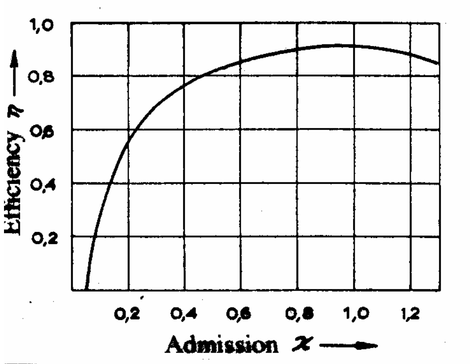


Fig. 1.1 Efficiency of a hydroturbine

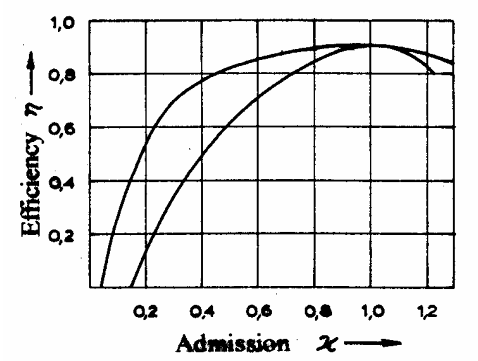


Fig.1.2 Efficiency of two different tubine types

A similar difference may also be found for the same turbine type dependent on the design of the turbine.

By preliminary and approximate evaluations of the output power of hydromachines, the plant efficiency η_a is chosen fairly low, for example $\eta_a = 0.765$ at full load. With discharge Q [m^3/s], water head H [m], water density of $\rho = 1000$ kg/m^3 and the indicated value of η_a the mechanical power output from the machine becomes

$$P = 7.5\ QH \quad [kW] \tag{1.3}$$

1.1 Brief review of hydropower machines

1.1.1 The eldest hydropower machines

The eldest and most primitive type of machines for utilizing water power are the *water wheels*. They were in use in China and Egypt several tousand years ago. The eldest type of these were the *undershot wheels,* Fig. 1.3, which were arranged in the river channel with horizontal shaft. The river water was flowing into the buckets on the underside of the wheel, which in this way was revolved by the velocity energy in the river flow. With this type of water wheels rather low powers were obtained because of low efficiency and the effective flow discharges were small.

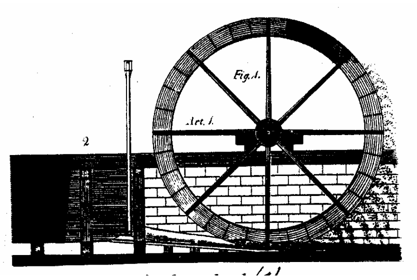
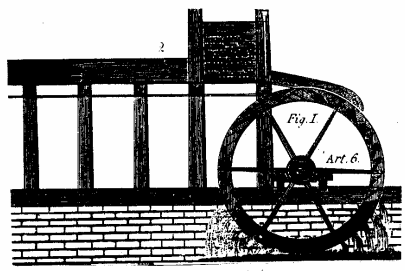


Fig. 1.3 Undershot wheel /1/



F ig.1.4 Overshot wheel /1!

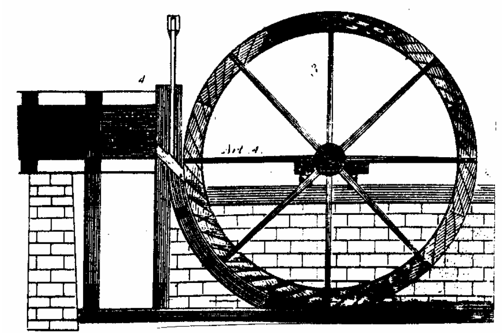


Fig. 1.5 Breast wheel /1/

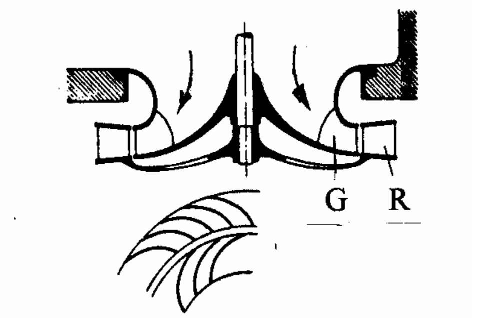
Later on the water wheels were installed in artificial falls that were erected by building a canal to guide the water into the wheel. Depending on the available water head two other wheel types in addition came up to be used. These were the so-called *overshot wheels*, Fig. 1.4, and *breast wheels,* Fig.1.5, which also were arranged with horizontal shaft. On the overshot wheel the water was guided into the buckets on the top side of the wheel, whereas on the breast wheel the water was guided into the buckets somewhere about the height of the wheel shaft. These wheels were revolved by the weight of the water filling in the buckets, and for the best designs of the wheels an efficiency up to 85 % was obtained at water heads higher than three meter.

The water wheels were employed mostly in Middle Europe throughout the Middle Ages. In Norway on the other hand, the so-called *Kvernkallen* (the Old mill-man) was the most widespred hydro machine. This had a vertical shaft to which the runner was fixed. The runner was equipped with radial tilted blades to which the water was flowing along a steep open canal. In this way the Kvernkallen utilized only the velocity energy of the flowing water. And this power transfer went on with relatively great losses especially for the plane blades, and the efficiency might be lower than 50 %..

1.1.2 Turbines

The industrial development throughout the 18. and 19. century the need for more energy was increasing so much that the water wheels no longer got hold of sufficient energy supply. New energy sources like wapour power was adopted, but further development toke place also on utilization of water power. In 1750 the physicist J. A. Segner invented a reaction runner, which has got its name after him. This runner utilized the impulse force from a water jet and was the forerunner of the turbines. Just after this the mathematician Leonard Euler developed the turbine theory, which is valid today too.

Turbine is a designation that was introdused in 1824 in a dissertation of the French engineer Burdin. Next step was made by engineer Fourneyron when he designed and put to operation the first real turbine in 1827. At that time this machine was a kind of a revolution with the unusual great power output of 20 – 30 kW and a runner diameter of 500 mm.



Figur 1.6 Radial turbine of Fournevron /4/

A principle sketch axially through the guide vane cascade G and the runner R of Fourneyron's turbine is shown on Fig. 1.6. Because the water flow has a radial direction through the turbine runner, it may be designated as *radial turbine.*

However, two other turbine designers Henschel and Jonval,developed about 1840, independent of each other, a turbine each with axial water flow through it. In addition they were the first ones to apply draft tube and in that way to utilize the water head between runner outlet and tail water level.

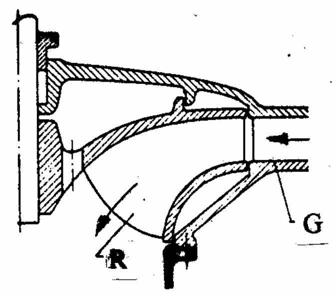
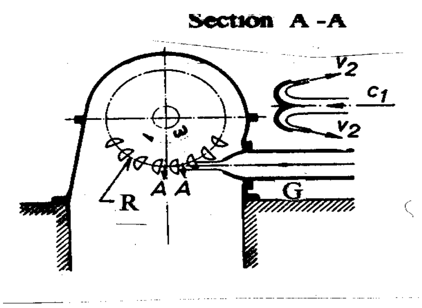


Fig. 1.7 Axial section of Francis turbine /4/

In the time interval until 1850 several designers improved the machines, but most serious was the work by the English engineer Francis. He developed and made a turbin in 1849, which has got its name after him. This turbine looked fairly similar to that one Euler had foreseen. An axial section through the guide vane cascade and the runner of this turbine is shown on Fig. 1.7. The water flows radially through the guide vane cascade G towards the runner R and further out in the axial direction.

In the year 1870 professor Fink introdused an important improvement by making the guide vanes turning on a pivot in order to regulate the flow discharge.

The turbines mentioned above, were operating at relatively low heads compared with turbines nowadays. Addionally the water was flowing into the turbine runner with a certain pressure above the atmospheric through a guide vane cascade comprising the whole periphery of the runner. A turbine design principally different from these, was developed by the American engineer Pelton. He made his first runner in 1890, and Fig. 1.8 shows that the water in this case flows as a jet from a nozzle through atmospheric air into the runner buckets. The buckets are designed to split the jet in two halves, which are deflected almost 180° before they leave the buket. In this way the impulse force from the deflection is transferred to the runner. By the invention of the Pelton turbine a machine type was developed for utilization of the highest water heads in the nature.



Figur. 1.8 Principle of Pelton turbine /2/

In the region of low pressure turbines a great step forward was made when professor Kaplan designed his propeller turbine, which was patented in 1913. This turbine had fixed runner blades, and was developed for utilizing the lowest water heads. Fig. 1.9 shows an axial section through a Kaplan turbine. Not very long after making the propeller turbine, Kaplan further developed his turbine by making the runner blades revolving. This was an important improvement for an economic and efficient governing of the turbine output.

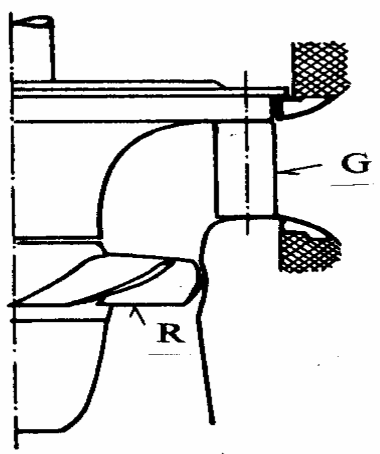


Fig. 1.9 Kaplan turbine /2/

After the brief review above of the hydropower machines, one can summerize that the only machine types that have been developed to be dominant in modern times are Pelton, Francis and Kaplan turbines. The reason is that these three types supplement each other in an excellent way. As a basic rule one can say that Pelton turbines are used for relatively high heads and small water discharges, Kaplan turbines for the lowest heads and the largest water discharges and Francis turbines for the region between the other two turbine types.

1.2 Arrangement of hydropower plants

The discharge operating a water turbine, is conveyed from a river or a water course through an intake and a conduit to the turbine. From the turbine the discharge is conducted through a so-called tail rase canal to a downstream river course.

A brief review of the main details in a hydropower plant is given in the following sections. Fig. 1.10 shows schematically an example of a plant arrangement with indication of the localisation of the details to be mentioned.

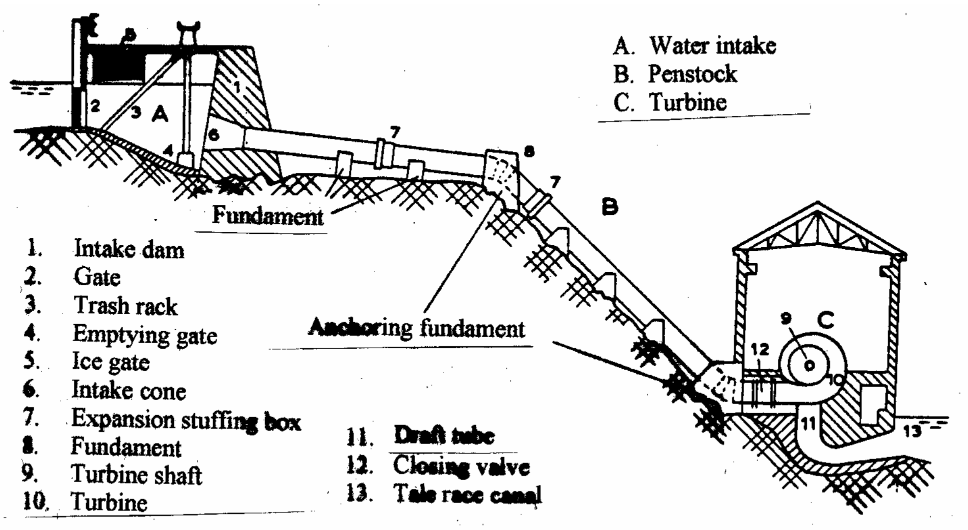


Fig. 1.10 Arrangement of a hydropower plant /2/

Fig.1.11 shows in principle [/3/] the water conduits of a traditional Norwegian power plant with a high head Francis turbine. Downstream from the upstream reservoir the coarse trash rack, intake gate, head race tunnel, surge shaft, sand trap, fine trash rack, penstock isolating valve with air valve, pressure shaft, spherical valve, turbine, draft tube, draft tube gate, outlet surge shaft and tale race tunnel.

Water intake

The water intake is normally constructed in connection with an accumulation dam (1) Fig.1.10, in the river course.

The shallow water intake is equipped with a *coarse trash rack* (3) which prevents trees, branches, debris and stones from entering the conduit system to the turbine. An *intake gate* (2) is arranged to

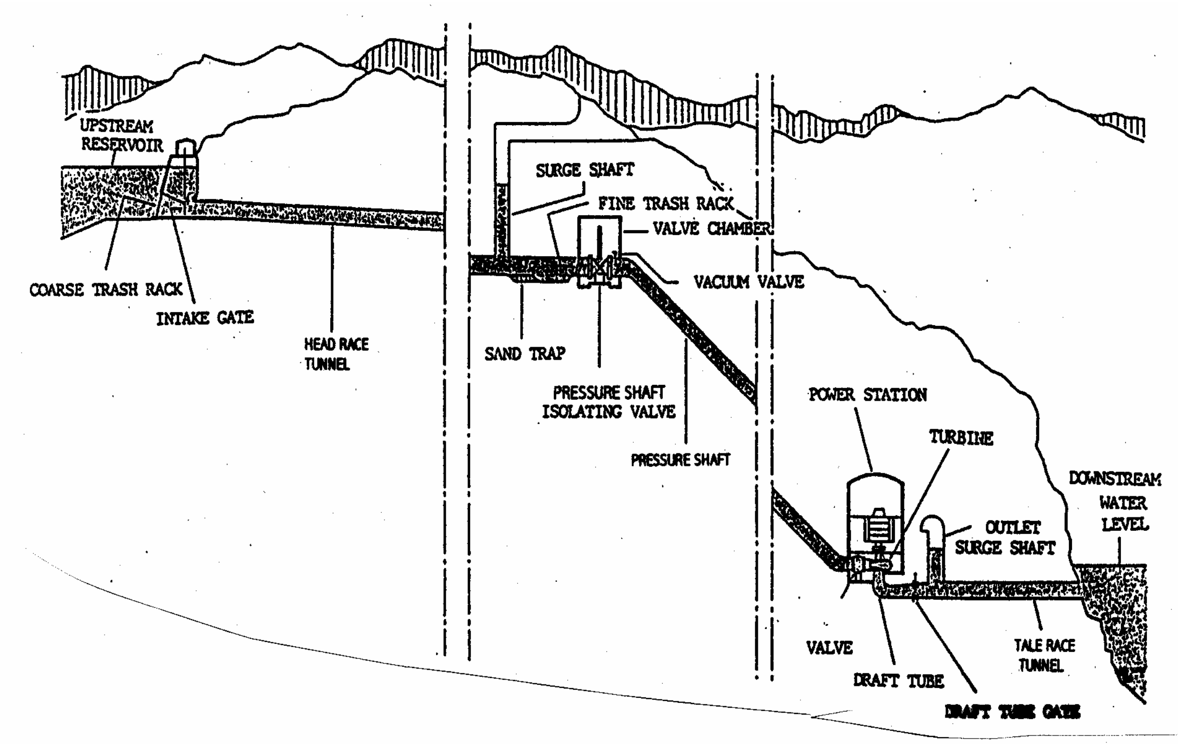


Fig. 1.11 Sketch showing in principle the water conduits of a traditional Norwegian high head power plant /3/

shut off the water delivery when the conduit system has to be emptied. In addition a small gate (4) may be arranged for drainage of the leakage through the main gate.

A deep water intake takes the water directly from the reservoir. It has no trash rack. There is a sump below the intake. Its main function is to collect blasting stones from the piercing of the the head race tunnel into the reservoir. It also traps stones sliding into the reservoir close to the intake. Deep water intakes allows for very strong regulation of the reservoirs. An intake gate is installed with the same function as described for the shallow water intake.

Conduit system

From the water intake to the turbine it is a conduit system constructed as open canal, tunnel, penstock or pressure shaft or a combination of these.

Open canals are usually digged in the ground, blasted in rock or built up as a chute of wood or concrete.

In high head power plants it is normally a so-called *head race tunnel* between the water intake and the pressure shaft. It may either be drilled and blasted or bored with a tunnel boring machine (TBM). The latter method leaves a much smoother wall surface than the first one, and consequently the head loss is significantly smaller for the same cross section. At the end of the head race tunnel there is a *sand trap*. Beside the sump in the tunnel floor the cross section of the tunnel is gradually increased to reduce the water velocity and allow for a better sedimentation of suspended particles.

At the downstream end of the head race tunnel there is also a *surge chamber system*. The function of the surge chamber is briefly to reduce water hammer pressure variations and keep the mass oscillations, caused by load changes, within acceptable limits and decrease the oscillations to stable operation as soon as possible. At the end of long head race tunnels it is also normally istalled a gate. This makes it possible to empty the pressure shaft and penstock upstream of the turbine, for inspection and maintenance without emptying the head race tunnel. Before the water enters the pressure shaft it passes a *fine trash rack*. It is the last protection of the valve and the turbine against floating debris or smaller stones if the sand trap is full or omitted.

The pressure shaft may either be lined or unlined. Where the rock is of sufficiently high quality the shafts are normally unlined. The excavation of the rock masses may be done either by drilling and blasting or by boring with TBM-machines. Shafts in lower quality rock is being lined either by concrete or by steel plate lining embedded in concrete. Lining of the shafts reduces the losses but increases the costs.

A steel penstock connects the shaft with the valve in the machine hall. Inside the rock the penstock is embedded in a concrete plug. Penstocks are normally welded pipe constructions of steel plates. A flange connects the penstock with the valve.

Penstocks above ground are mounted on foundation concrete blocks where the penstock may slide according to thermal expansion. In certain positions the penstocks are fixed in reinforced concrete anchoring blocks (8) on Fig. 1.10. Between these anchoring blocks the penstocks are equipped with expansion stuffing boxes (7) in Fig. 1.10.

At the upstream end of a penstock an automatic isolating valve is normally installed. This valve closes automatically if a pipe rupture should occur.

Turbine

The main parts of the turbine, with reference to Fig. 1.10, are:

- the guide vane cascade, usually adjustable, gives the water flow the velocity and the direction required for the inlet to
- the runner where the hydraulic power is transferred to mechanical power on
- the turbine shaft (9) to which the runner is fixed. The turbine shaft is guided in a
- radial bearing and an
- axial bearing that is loaded with the axial force from the runner, caused by the water pressure and impulses from the flow, and the weight of the rotating parts.
- The scroll case (10) conducts the water flow to the guide vane cascade.
- The draft tube (11) conducts the water flow from the turbine outlet into the tale race canal.

Closing valve

Upstream of the turbine a closing device (12) on Fig. 1.10, is installed. Depending on water head and capasity it may be a gate, butterfly valve, gate valve or a spherical valve. By submerged turbines a closing device, normally a gate, is installed also at the outlet from the draft tube.

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CHAPTER 2

Energy Conversion

Introduction

A water flow from an upper level to a lower level represents a hydraulic power potential. This power flow can be utilised in a water power plant by conversion to mechanical power on the shafts of turbines. However, some fractions of the power potential are lost partly in the plant's conduits and partly in the turbines.

In this chapter a brief description is given of the power conversion in the turbines that are common in hydro power stations.

2.1 Fundamentals and definitions

Specific energy

The specific energy of a hydro power plant is the quantity of potential and kinetic energy which 1 kilogram of the water delivers when passing through the plant from an upper to a lower reservoir. The expression of the specific energy is Nm/kg or J/kg and is designated as [m^2/s^2].

In a hydro power plant as outlined on Fig. 2.1, the difference between the level of the upper reservoir z_{res} and the level of the tail water z_{tw} is defined as the *gross head*

$$H_{gr} = z_{res} - z_{tw} \tag{2.1}$$

The corresponding *gross specific hydraulic energy*

$$E_{gr} = gH_{gr} \tag{2.2}$$

where g is the acceleration of gravity.

When a water discharge Q [m^3/s] passes through the plant, the delivered power is

$$P_{gr} = \rho Q g H_{gr} \tag{2.3}$$

where P_{gr} is the *gross power* of the plant
ρ is the density of the water
Q is the discharge

To look further on the hydropower system in Fig. 2.1 the specific hydraulic energy between the Sections (1) and (3) is available for the turbine. This specific energy is defined as *net specific energy* and is expressed by

$$E_n = gH_n \tag{2.4}$$

and the *net head* of the turbine $H_n = \dfrac{E_n}{g}$ (2.5)

As shown on Fig. 2.1 there are two ways of expressing the evaluation of the net head. The one way

$$H_n = h_p + c^2/2g$$

And the other way

$$H_n = H_{gr} - \frac{E_L}{g} = H_{gr} - H_L$$

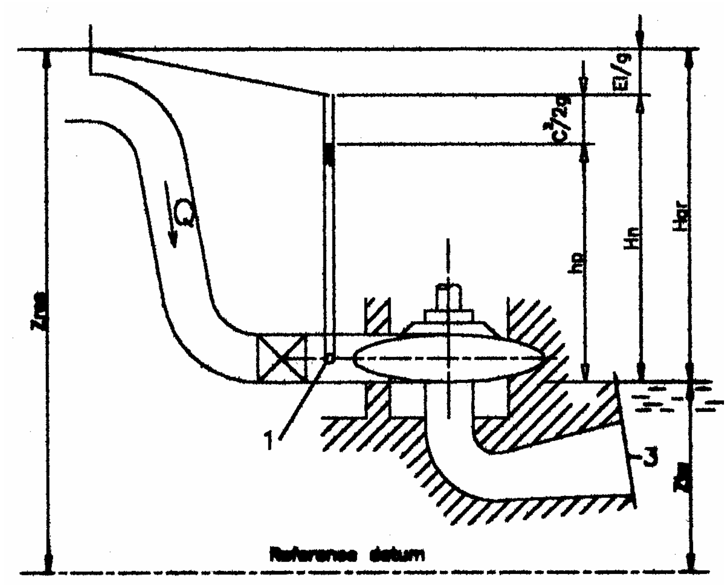

Fig. 2.1 Hydro power plant. Definition of gross head H_{gr} and net head H_n

where h_p is the piezometric head above tailwater level measured in section (1), $c^2/2g$ is the velocity head in section (1) and E_L/g is specific hydraulic energy loss between reservoir and section (1) converted to head loss H_L.

Note.

Some comments to the specific energy definitions according to Equations (2.2) and (2.4) should be mentioned. For efficiency tests of hydro turbines a relatively high exactness of the determination of the specific energy is required. Therefore an international standard exists for the measurements and evaluations of such tests. The name of it is **International Standard IEC 41.**

In addition to the specifications of relevant levels this standard take into account the influences of: Compressibility and temperature effects of the water; the weight of the air column difference between the reservoir and the tail water; the difference of specific kinetic energy between defined sections of the system and at last that the acceleration of gravity depends on the altitude and latitude.

The specific energy expressed by putting H_{gr} corresponding to Fig. 2.1, in Equation (2.2) and H_n in Equation (2.4) respectively, is consequently approximations according to this standard. However, the mentioned influences are relatively small, i.e., totally of the order 1% in extreme cases. This means that these influences are essentially smaller than the tolerance accuracy of the hydraulic dimensioning of the turbomachines. Therefore the hydraulic considerations of calculation and design in the following sections are based on constant values of the acceleration of gravity and the density of water, no influence of temperature and the weight of the air column.

2.2 Transforming hydraulic energy into mechanical energy

2.2.1 General considerations

Ordinary turbines.

The discharge and the net head for turbines differ in wide ranges from one power plant to another. This indicates that not only different types of turbines but also a very large register of sizes of turbines are needed.

The ordinary turbine types in water power plants are distinguished in two main groups:

- impulse turbines
- reaction turbines

This distinction is based on the difference between the two cases of energy conversion in these turbines. Briefly these two ways of energy conversion may be pronounced as follows.

- Basically the flow energy to the *impulse turbines* is completely converted to kinetic energy before transformation in the runner. This means that the flow passes the runner buckets with no pressure difference between inlet and outlet. Therefore only the impulse forces being transferred by the direction changes of the flow velocity vectors when passing the buckets create the energy converted to mechanical energy on the turbine shaft. The flow enters the runner at nearly atmospheric pressure in the form of one or more jets regularly spaced around the rim of the runners. This means that each jet hits momentarily only a fraction or part of the circumference of the runner. For that reason the impulse turbines are also denoted *partial turbines.*

- In the *reaction turbines* two effects cause the energy transfer from the flow to mechanical energy on the turbine shaft. Firstly it follows from a drop in pressure from inlet to outlet of the runner. This is denoted the *reaction part* of the energy conversion. Secondly changes in the directions of the velocity vectors of the flow through the canals between the runner blades transfer impulse forces. This is denoted the *impulse part* of the energy conversion. The pressure drop from inlet to outlet of the runners is obtained because the runners are completely filled with water. Therefore this group of turbines also have been denoted as *full turbines.*

The most commonly used turbines today are:

- impulse type: *Pelton turbines*
- reaction type: *Francis turbines*
  Kaplan turbines
  Bulb turbines

In the following sections the hydraulic energy transfer in Pelton turbines as well as the reaction turbines is described in more detail [/1/,/3/,/4/].

2.2.2 Impulse turbine - Pelton

A section through a Pelton runner is shown in Fig. 2.2. The water jet from the nozzle hit the buckets that are spaced equidistant around the runner disc. In the left lower corner of the figure a look into a bucket in the same direction as the incoming jet is shown. To the right of the figure a section through two neighbouring buckets are shown for the indicated section line A - A. In this section is shown how the jet flow is split symmetrically at the bucket edge and passing over the bucket. The jet deflection is nearly up to 180°, but limited to a little smaller angle because the leaving jets have to run clear of the subsequent bucket behind.

For a net head H_n the theoretical velocity of the water jet out of the nozzle is found according to Bernoulli´s equation

$$c_1 = \sqrt{2gH_n} \tag{2.6}$$

However, an energy loss occurs in the nozzle. This influence is corrected for by a friction coefficient φ, and the velocity then becomes

$$c_1 = \varphi\sqrt{2gH_n} \tag{2.7}$$

A value of the friction coefficient based on experience may be φ = 0.98.

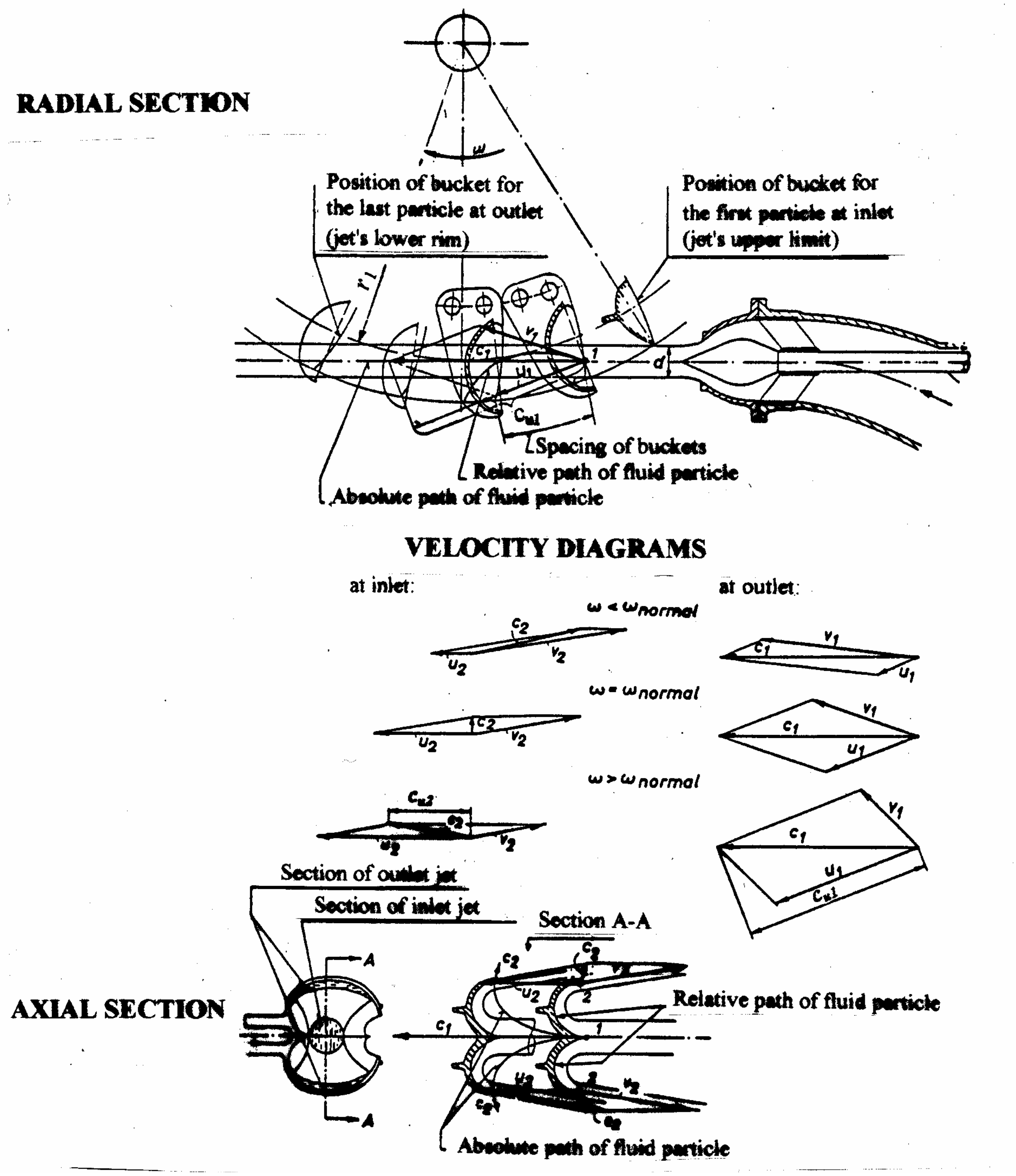

Fig. 2.2 The water jet flow into a Pelton runner. Velocity diagrams at inlet and outlet of the buckets /3/

The runner is assumed to rotate with a constant angular speed ω.

One water particle is considered, for example the particle in the centre of the jet just at the splitting edge of the bucket and marked as position (1) on the Fig. 2.2. This bucket is drawn for a position corresponding just to the moment that the full jet is entering the bucket. The *absolute* velocity c_1 is known from Equation (2.7). The *peripheral* velocity of the runner $u_1 = r_1\omega$ corresponds to radius r_1 in position (1). The direction of this velocity vector is the same as the tangent in position (1) of the circle with radius r_1.

When the absolute velocity c_1 of the water particle and the peripheral velocity u_1 in position (1) are known, the velocity v_1 of the water particle *relative* to the bucket in position (1) can be found. The absolute velocity c_1 is the *geometric sum* of the peripheral velocity u_1 and the relative velocity v_1. Therefore one has to draw the three velocity vectors from point (1) so that c_1 is the diagonal in the parallelogram with u_1 and v_1 as sides. This parallelogram is called the *velocity diagram* of the water particle at the inlet of the bucket.

The water particle moves over the bucket and changes its direction gradually until it leaves the bucket at position (2) as shown in section A - A on Fig. 2.2. During this movement the particle transfers its impulse force corresponding to the change from the direction of the relative velocity vector v_1 to the relative velocity vector v_2. The magnitude of v_2 depends on energy losses during the passage of the bucket. These losses can be expressed as $\zeta_2 \frac{v_2^2}{2}$ where ζ_2 is defined as loss coefficient. From experience an estimated value of $\zeta_2 = 0.06$. The relation between v_1 and v_2 is found according to Bernoulli's equation:

$$h_1 + \frac{v_1^2}{2g} = h_2 + \frac{v_2^2}{2g} + \varsigma_2 \frac{v_2^2}{2g} \tag{2.8}$$

In this case $h_1 = h_2$, and therefore

$$(1 + \varsigma_2)\frac{v_2^2}{2g} = \frac{v_1^2}{2g} \tag{2.9}$$

and

$$v_2 = \frac{v_1}{\sqrt{1 + \varsigma_2}} \tag{2.10}$$

The magnitude of velocity v_2 is nearly as great as v_1 and has a direction as shown in position (2), section A - A on Fig. 2.2. The peripheral velocity u_2 is assumed the same as u_1 because in an approximate approach it is supposed that the water particle enters and leaves the runner bucket at the same radius of the runner. Therefore the velocity diagram may be drawn from position (2) with u_2 and v_2 as the parallelogram sides and the absolute velocity c_2 as the diagonal. This velocity diagram shows that the magnitude of c_2 is much smaller than of c_1, which is just the main intention to obtain, because $c_2^2/2$ is a direct measure for the loss at outlet of the turbine.

The passage of the water particle from position (1) to position (2) lasts a certain time interval, and during this time the runner rotates a corresponding angle. If the corresponding positions of the bucket and the absolute flow path under this passage over the bucket are drawn, one get an absolute flow path which ends up with the velocity vector c_2 as shown in section A - A on Fig. 2.2. The absolute velocity vector is therefore tangent to this path all the way of the passage.

The description of the movement over the bucket of one water particle is valid also for all the other particles of the full jet. These particles will however trace different paths, but even so, in practise it is assumed that the impulse force and the corresponding torque transfer to the runner is the same from all water particles in the jet.

When the runner is rotating and the bucket starts splitting the jet, it gradually takes up more and more of the full jet area until it cut the jet completely. But immediately after this cut the succeeding bucket starts splitting the jet and repeats the same behaviour as the preceding bucket. In the time interval between the first water particle enters and the last one leaves a bucket at a distance as indicated on Fig. 2.2, is covered.

The total deflection of the jet may be made greater for a relatively large spacing between the buckets compared with a smaller one. Therefore the spacing between the buckets is made as big as possible, but not larger than to secure that all water particles will hit the bucket.

With the same torque transfer to the runner from all water particles in the jet, the velocity diagrams for the inlet and outlet respectively for one of the water particles is valid for all the water particles of the jet. The general expression for the transferred power P_R is therefore

$$P_R = \rho Q(u_1 c_{u1} - u_2 c_{u2}) \tag{2.11}$$

where Q is discharge
u_1 is the peripheral velocity of the runner where the jet hit the bucket
u_2 is the peripheral velocity of the runner at jet outlet of the runner
c_{u1} is the component of the absolute velocity c_1 in the direction of u_1
c_{u2} is the component of the absolute velocity c_2 in the direction of u_2

As previously mentioned the peripheral velocity $u_1 = u_2$ for the Pelton turbine. Moreover it is suggested to insert the peripheral velocity corresponding to the runner diameter to which the centre line of the jet is tangent. The power equation then becomes:

$$P_R = \rho Q u_1 (c_{u1} - c_{u2}) \tag{2.12}$$

How this power varies if the rotational speed is changed, is interpreted as follows. As basis is ascertained that the absolute velocity c_1 is constant and therefore c_{u1} is constant. The discharge Q is constant while the angular velocity ω is varied. The rotating speed $u_1 = r_1\omega$. From Fig. 2.2 it is found that c_{u2} varies when ω is varied. In the case $u_1 = 0$, i.e., the runner is at standstill, the power $P_R = 0$ and $c_{u2} \approx - c_{u1}$. When the runner is rotating, the velocity component c_{u2} decreases as the rotational speed increases and it approximates to zero when u_1 increases towards $c_1/2$. At the same time it is observed that the power P_R increases when u_1 increases from zero.

If u_1 increases towards c_{u1}, c_{u2} also increases and approaches to c_{u1} so that $(c_{u1} - c_{u2})$ advances towards zero. In this case the power again approaches towards zero. This case corresponds to the run away speed.

A closer examination shows that the transferred power P_R to the runner has its maximum value when c_{u2} is close to zero and accordingly u_1 approximately like $c_1/2$.

Regulation of the power means to regulate the discharge Q by adjustment of the needle position to larger or smaller openings of the nozzles. For constant angular speed the regulations really cause minor or no changes of the velocity diagrams.

2.2.3 Reaction turbines

Francis, Kaplan and Bulb turbines are the reaction turbines normally applied. The transfer of hydraulic energy into mechanical energy is principally similar in these turbines. However, the hydraulic design of Francis turbines differ so much from that of Kaplan and Bulb turbines that an interpretation of the energy transfer will be given for each of these two groups.

Francis turbines

Fig. 2.3 shows an axial section through a Francis turbine with the guide vane cascade (G) and the runner (R). The runner is fastened to the turbine shaft (S).

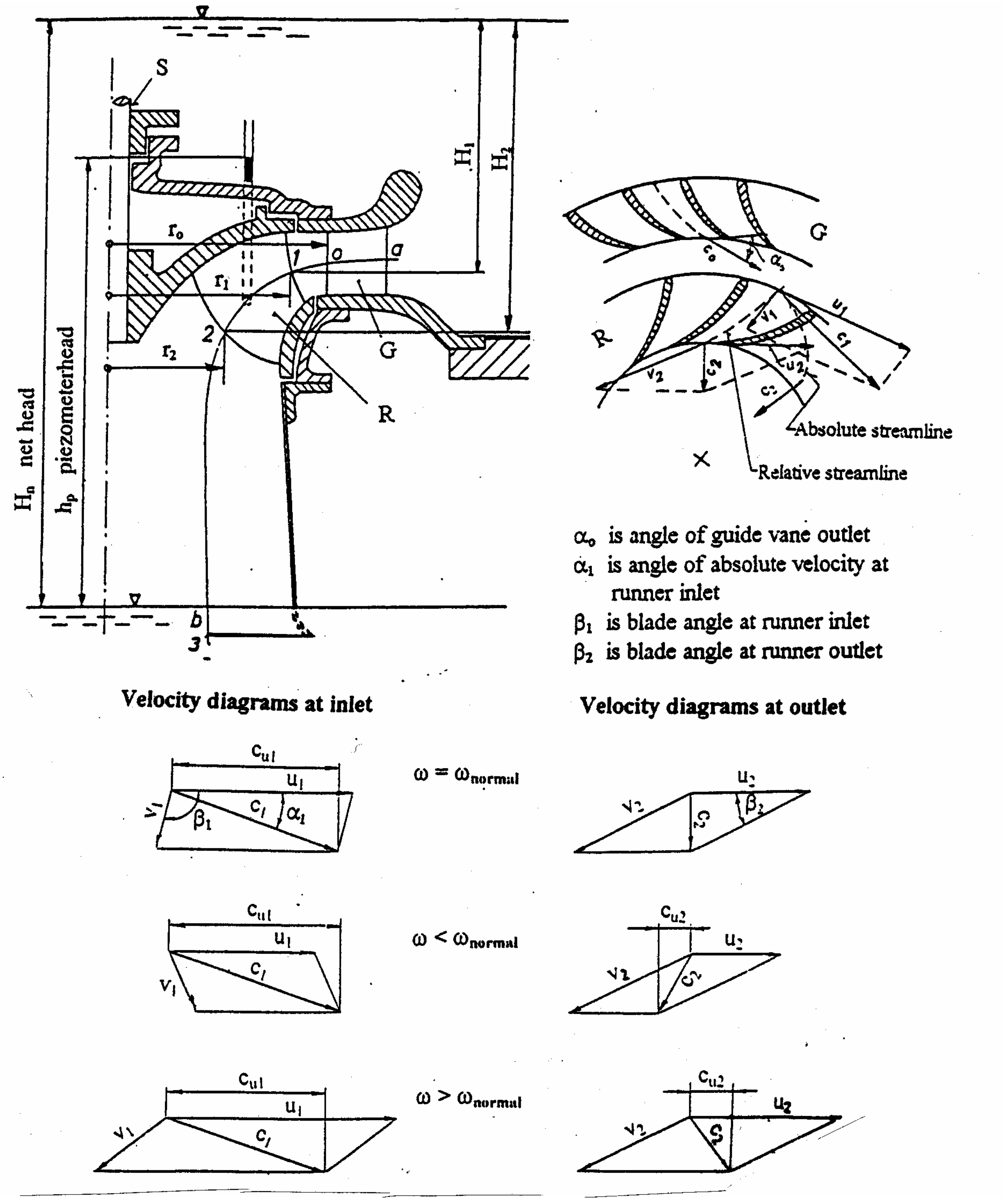

Fig. 2.3 Axial section through a Francis turbine. Velocity diagrams at inlet and outlet of the runner /3/

The energy transfer will be considered on the basis of the movement of one water particle along the coaxial stream surface of revolution having a contour (a - b), in the axial section through the turbine. To observe the movement of this particle through the turbine, a section across the guide vanes and runner blades perpendicular to the paper plane in Fig. 2.3 is needed. This should be made along the contour (a - b), i.e., along (a - o) through the guide vane cascade and along (1 – 2) through the runner cascade.

However, the stream surface (a - b) in Fig.2.3 has a double curvature. Therefore the surface cannot be unfolded unless by a transformation from the real surface to a single curved surface. In practise this can be done by using conform transformation method because the angles of geometry then are kept unchanged from the real surface to the transformed surface.

The section through the runner cascade shown to the right on Fig. 2.3, is assumed to represent a conform transformation onto a conical surface tangential to the contour (1 - 2).

It is assumed that the turbine is installed at the bottom of an open reservoir filled with water up to a certain level above the guide vane cascade. The guide vane direction angle α_o is assumed constant and the runner is rotating with a certain angular speed ω and the water is filling all the runner canals completely.

The consideration is assumed to start with a water particle at the inlet edge of the guide vane cascade. Through the guide vane canal the water particle is assumed to follow the streamline in the middle of the canal width as shown on the figure. The guide vanes are designed so that the movement of the particle is changed from the radial direction at inlet to leave the outlet edge of the canal with a rather large velocity component in the peripheral direction. The outlet edge of the guide vanes is indicated with the index (o), and the *absolute* velocity of the water particle at this edge is accordingly denoted c_o. The direction of c_o is supposed to coincide with the direction of the vanes at the outlet of the guide vane cascade.

It is assumed that the water particle passes without friction through the ring chamber between the guide vane outlet and the inlet of the runner. Therefore it will keep an unchanged vortex momentum. That means rc_u is constant, and the relation between the rotational components c_{uo} and c_{u1} of the absolute velocities c_o and c_1 respectively become

$$c_{u1} = c_{uo}\frac{r_o}{r_1} \tag{2.13}$$

where r_o is the radius to the guide vane outlet marked (o)
r_1 is the radius to the inlet of the runner marked (1)

The peripheral velocity of the runner corresponding to radius r_1 is found by $u_1 = r_1\omega$.

Now the absolute velocity c_1 and the peripheral velocity u_1 are determined. The relative velocity v_1 is then found as one side in the parallelogram where the peripheral velocity vector u_1 is the other side and the absolute velocity vector c_1 is the diagonal. These three velocity vectors are drawn in Fig. 2.3 and form the velocity diagram of the water particle at the inlet of the runner canal.

During the movement through the runner canal the particle changes its direction again as shown on the figure. By this deflection an impulse force is transferred by giving the runner a torque in the rotational direction.

The relative streamline through the runner canal is also drawn in the Fig.2.3. At the outlet edge of the canal on this streamline the relative velocity is denoted v_2. The figure shows that the relative velocity v_2 has got a rather large peripheral component in the opposite direction of the rotation. The magnitude of v_2 can be found by means of the continuity equation

$$v_2a_2 = v_1a_1 \tag{2.14}$$

where a_1 and a_2 are the areas of the runner blade canal taken perpendicular to the streamline at inlet and outlet of the canal respectively. The direction of v_2 is the same as the outlet direction of the runner blades. The peripheral velocity $u_2 = r_2\omega$ where r_2 is the radius to the marked point (2) at the runner outlet.

The velocity diagram for the water particle at the outlet edge of the runner canal can be determined by drawing the parallelogram with the sides u_2 and v_2 from point (2) and thereafter the diagonal which is the resultant c_2 drawn from the same point.

The passage of the water particle from point (1) to point (2) in the runner canal, needs a certain time interval for this movement, and simultaneously the runner rotates a certain angle. By drawing corresponding positions of the runner canal in the rotational direction and the position of the particle in the canal for some intermediate time intervals, the absolute path of the water particle is found. A such path is drawn on Fig. 2.3, and the absolute velocity vector is everywhere tangent to the absolute flow path.

The absolute as well as the relative movement of all particles in the water flow through the turbine will behave in the same way as described for the considered water particle. In a corresponding way the same impulse and torque is supposed to be transferred to the runner from all water particles.

The power transfer P_R to the runner from the water flow is then

$$P_R = \rho Q(u_1 c_{u1} - u_2 c_{u2}) \tag{2.15}$$

As mentioned for the impulse turbine, $c_2^2/2$ represents the energy at outlet. However, during the passage of the draft tube a fraction of this energy is recovered by retardation of the flow velocity, but a flow friction loss occurs also in the draft tube which again means a slight reduction of the the recovered energy.

A discussion of Equation (2.15) may be carried out in the same way as done for the Pelton turbine. Examples are drawn in Fig. 2.3 of the velocity diagrams at inlet and outlet of the runner respectively for three angular velocities, $\omega = \omega_{normal}$, $\omega < \omega_{normal}$ and $\omega > \omega_{normal}$. A difference should however, be remarked that when the power is near maximum, u_1 and c_{u1} nearly have the same magnitude, while u_1 is approximately half of c_{u1} for the Pelton turbine.

For regulating the discharge Q of the turbine the width of the guide vane canals must be varied. An increase of Q means to adjust the guide vanes to a larger angle α_o and a decrease of Q means an adjustment in the opposite direction. This regulation causes corresponding changes in the direction of the absolute velocity c_1. Accordingly the velocity diagrams will change.

Both the variation of the angular velocity ω and the regulation of the discharge Q involve changes in the direction and magnitude of the relative velocity v_1. The relative velocity v_2 varies accordingly in magnitude with the regulation of Q. Moreover the difference $(u_1c_{u1} - u_2c_2)$ and thereby also the power transfer, is entirely dependent of these changes.

The most efficient power transfer however, is obtained for the operating condition when the relative velocity v_1 coincide with blade angle β_1 at the runner inlet and simultaneously the rotational component $c_{u2} \approx 0$. Therefore the hydraulic lay out of all reaction turbine runners are based on the data of rotational speed n, discharge Q and net head H_n for which the optimal efficiency is wanted.

Kaplan and Bulb turbines

The hydraulic design of Kaplan and Bulb turbine runners is quite similar while the flow direction through the guide vane cascades is radial in Kaplan turbines and approximately axial in Bulb turbines. This means no significant difference for an interpretation of the flow through these turbines. Therefore an illustration of the flow through a Kaplan is valid also for a Bulb turbine.

Fig. 2.4 shows an axial section through a Kaplan turbine with the guide vane cascade (G), runner (R) and the shaft (S).

In the same way as for Francis turbines the consideration of the flow through the Kaplan turbine is based on the movement of one water particle along the flow surface with the contour (a - b) as shown on Fig. 2.4. This particle is given a flow direction in the guide vane canal before it enters the runner canal. The particle movement through the guide vane canal and runner canal is shown to the right in the figure. The canal section across guide vane cascade is radial (perpendicular to the plane) along contour (a - o) and across the runner blades the section is cylindrical along the contour (1 - 2).

The considerations are further based on a constant guide vane direction angle α_o and constant angular velocity ω. The particle flow through the turbine is quite analogous to that described for the Francis turbine. Therefore the description is focused mainly on the velocity diagrams.

The fluid flow in the axisymmetrical hollow space between outlet of the guide vane canal marked (o), and the inlet of the runner marked (1), is denoted as free vortex. The flow is again assumed free of losses along the flow path. The relation between the rotational component c_{uo} of the absolute velocity c_o and the rotational component c_{u1} of the absolute velocity c_1 then becomes

$$c_{u1} = c_{uo}\frac{r_o}{r_1} \quad \text{or} \quad c_{u1} = c_{uo}\frac{u_o}{u_1} \tag{2.16}$$

The peripheral velocities are $u_1 = r_1\omega$ and $u_2 = r_2\omega$. By the given angle α_o at the outlet of the guide vane canal and the angle β_2 at the outlet of the runner canal all data are now prepared for drawing of the velocity diagrams at inlet and outlet of the runner. The drawing of these diagrams is as described for Francis turbines, and in Fig. 2.4 examples are shown for three different angular speeds, $\omega = \omega_{normal}$, $\omega < \omega_{normal}$ and $\omega > \omega_{normal}$.

The designation $\omega = \omega_{normal}$ means again the rotational speed for which the turbine obtain the lowest energy loss at outlet represented mainly by $c_2^2/2$. This is also the operating condition for which the turbine obtain the highest hydraulic efficiency for the given angle α_o of the guide vane canal.

The movement that is considered for a single water particle through the turbine, is an illustration also for all the other particles in the total water flow. As mentioned for Francis turbines that is an assumption for Kaplan and Bulb turbines as well that all particles transfer the same impulse and torque to the runner. The power transfer from the flow is therefore generally expressed by Equation (2.15)

$$P_R = \rho Q(u_1c_{u1} - u_2c_{u2})$$

A look at Fig. 2.4 indicate also that the peripheral velocity $u_2 = u_1$. The power equation then becomes

$$P_R = \rho Q u_1(c_{u1} - c_{u2}) \tag{2.17}$$

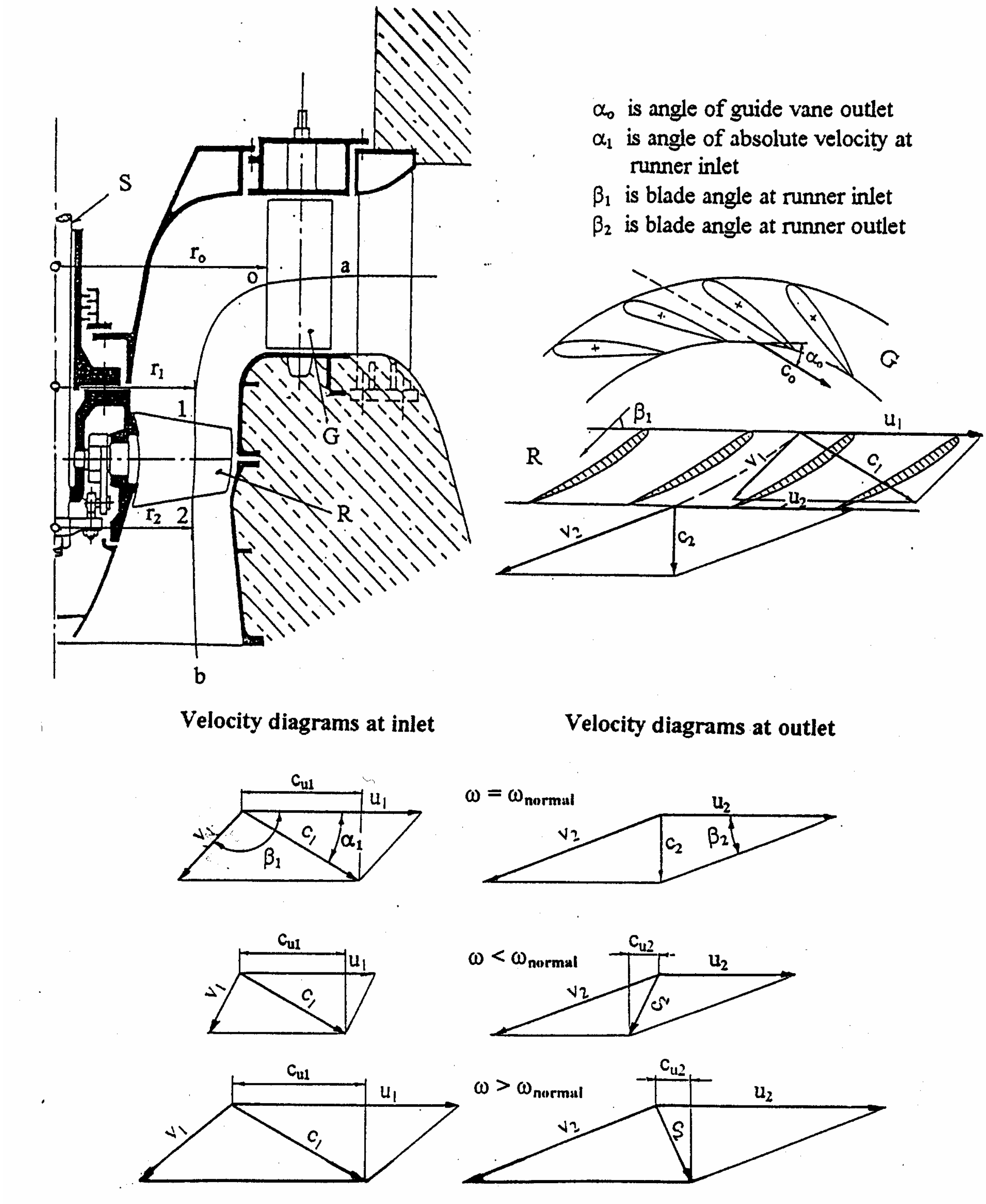

Fig. 2.4 Axial section through a Kaplan turbine. Velocity diagrams at inlet and outlet /3/

An interpretation of power regulation by changing the discharge Q with adjustment of the openings of the guide vane cascade follows the same reasoning as for Francis turbines. However, in Kaplan and Bulb turbines the direction of the runner blades can also be varied. This property will be further interpreted in Chapter 3.

2.2.4 The main equation of turbines

Losses along a flow path

The flow through turbines is exposed to energy losses. In reaction turbines the flow friction and change of flow directions in the guide vane cascade, the runner and the draft tube mainly cause these losses. According to the turbulent flow conditions these energy losses are defined [/4/] by

$\varsigma_1 \dfrac{c_1^2}{2}$ is the energy losses in the guide vane cascade

$\varsigma_2 \dfrac{v_2^2}{2}$ is the energy losses in the runner

$(1+\varsigma_3)\dfrac{c_3^2}{2}$ is the energy losses in the draft tube

The coefficients ζ_1, ζ_2 and ζ_3 are assumed as constants [/4/], but their values depend on the operating conditions of the turbine. In general this dependence behaves so that ζ_1 and ζ_2 have a smaller variation than ζ_3 when operating conditions changes. Values of general validity of these coefficients cannot be given. However, at favourable operating conditions for a reaction turbine one can estimate values of ζ_1 and ζ_2 in the region 0.06 - 0.15 and of ζ_3 = 0.1 - 0. 3.

In addition to the losses by friction and changed directions, so-called impact losses can occur at the inlet of the runner. These losses which are designated by E_I^2, are introduced when the relative velocity vector of the flow enters the runner with another direction than the inlet direction of the blade.

The total sum of the losses along the flow path is

$$h_L = \frac{1}{2}\left[\varsigma_1 c_1^2 + \varsigma_2 v_2^2 + (1+\varsigma_3)c_3^2 + E_I^2\right] \tag{2.18}$$

The available net head for the turbine is designated H_n. The specific energy head transferred to the runner is then

$$h_R = H_n - h_L \tag{2.19}$$

Note

When the operating conditions by regulations deviate quite a lot from the design operating conditions, the same assumptions of the magnitude of the loss coefficients along the flow paths do not hold completely.

The main equation of turbines

The total available power of a plant is

$$P_n = \rho Q g H_n \tag{2.20}$$

The net head H_n is defined at the inlet of the turbine referred to the level of the tail water of

reaction turbines or the outlet of the nozzle of a jet turbine.

The power transfer from the fluid to the turbine runner is

$$P_R = \rho Q(u_1 c_{u1} - u_2 c_{u2}) \tag{2.21}$$

The ratio between these powers

$$\eta_h = \frac{P_R}{P_n} = \frac{1}{gH_n}(u_1 c_{u1} - u_2 c_{u2}) \tag{2.22}$$

which is defined as *hydraulic efficiency.*

The following rearrangement of Equation (2.22) is called the *main turbine equation*

$$\eta_h H_n = \frac{1}{g}(u_1 c_{u1} - u_2 c_{u2}) \tag{2.23}$$

Another version of the main turbine equation is obtained by substituting for u_1c_{u1} and u_2c_{u2} from the velocity triangles respectively

$$H_n = (1+\varsigma_1)\frac{c_1^2}{2g} - \frac{c_2^2}{2g} + (1+\varsigma_2)\frac{v_2^2}{2g} - \frac{v_1^2}{2g} + \frac{u_1^2}{2g} - \frac{u_2^2}{2g} + (1+\varsigma_3)\frac{c_3^2}{2g} + \frac{E_1^2}{2g} \tag{2.24}$$

This last version expresses the total sum of the energy transfer in the runner and the head losses in the guide vane cascade, the runner and the draft tube as equal to the net head H_n. For Pelton turbines however, it should be remarked that the two last terms in equation (2.24) are omitted, i.e., these terms do not occur in the energy conversion analysis for these turbines..

2.3 A brief outline of the hydraulic design of turbines

The axial flow out of reaction turbines

By the considerations of the flow along the stream surface of revolution having a contour given in Fig. 2.3, the main obtained result was the *main turbine equation* which is expressed in several equations (2.23 - 2.24). For the conditions being the design basis of a turbine, the major objective of the design is to obtain the same hydraulic efficiency for any flow surface of similar properties as the surface with the contour a - b, through the turbine over the whole width of the flow space.

To deal with this design process in detail is not the aim in this text. Therefore only a brief description of it will be given. Principally this will be about the hydraulic designing.

One of the first steps in the design is to form the contours of the axial section of the turbine. It may start with a trial of the form and thereafter a computational verification of the distribution of the meridional flow velocity through the axial section. The determination of the meridional velocity profile is based on the law of irrotational flow

$$\frac{dc_z}{dn} = -\frac{c_z}{R} \tag{2.25}$$

where c_z is the meridional component of the absolute velocity
n is the length parameter of the width perpendicular to the flow surface
R is the radius of the curvature of the flow path contour

When the velocity profile is determined according to equation (2.25) a control of how far this is correct, has to be done by use of the continuity

$$dQ = 2\pi r c_z dn \qquad (2.26)$$

equation where r is the radius to the actual position on the flow surface. The examination of a correct flow distribution may be done in the following way. The axial flow section is thought to

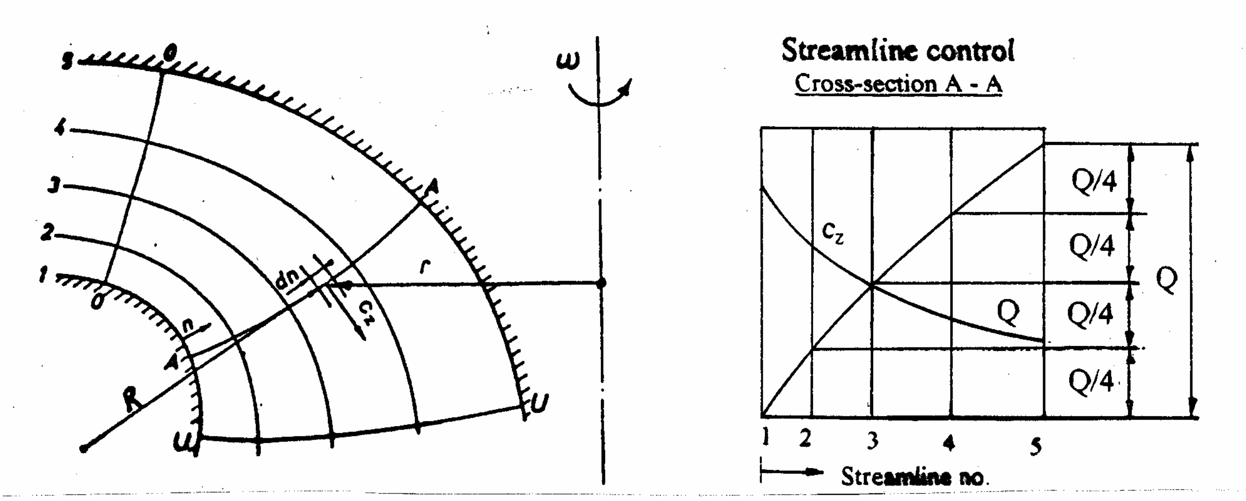

Fig. 2.5 Axial section of a Francis runner divided in four subsections by five numbered paths /3/

be divided in a certain number of coaxial subsections through the turbine as shown on Fig. 2.5, where the number of subsections is four. Each of these subsections shall represent a fraction of the total discharge Q, that means one quart of Q. To obtain this the velocity profiles of c_z from equation (2.25) are tested and corrected by use of equation (2.26) until the discharge is the same in all the subsections. Further it is important to check that the shape of the axial section performs an even or smooth meridional velocity c_z so that it does not increase or decrease disorderly in magnitude along the flow path.

Runner canals and blade forms

The shaping of the blades and the runner canals is a topic task for skilled designers. This process is again a combination of theoretical computations, experiences, certain criteria and rules.

As one of the first steps an estimation of the number of blades has to be done. The turbine manufacturers have normally specific criteria of their own for this choice. But there are also some general requirements that have to be satisfied. For example the smallest openings between blades must be large enough to let expected firm contamination in the flow run freely through. Further the designers have to give the blades a shape based on control procedures so that the energy conversion in the runner for the whole flow path is satisfied.

To have an idea of some details in this type of design work, it may be beneficial to have a look at a section through a few pairs of blades and the canals they constitute. At first a glance back on Figs. 2.3 or 2.4, gives examples of blade forms and canals. In some of these canals a streamline is drawn. This streamline served as basis for the considerations being carried out on the kinematics of the hydraulic energy transfer in reaction turbines. Furthermore this streamline was assumed to represent the average velocity in the corresponding cross-sections along the canals. Now a look in more detail on the flow conditions through the runner canals will show velocity distributions over the cross-sections as indicated on Fig. 2.6. To determine the velocity profile in the respective cross-sections of the canals, the following differential equation may be used

$$\frac{dv}{dn} = -\frac{v}{R} - 2\omega\cos\delta \tag{2.27}$$

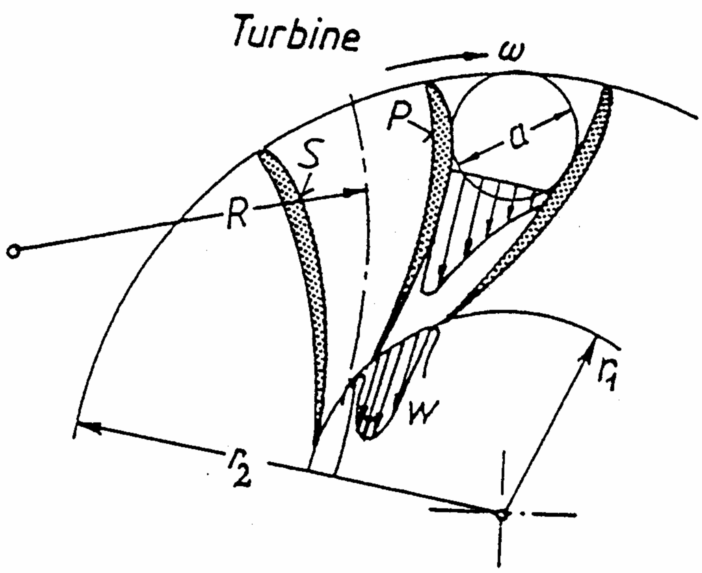

Fig. 2.6 Section through runner blades. An example of the skew velocity profile

where

v is the relative velocity in the runner canal
R is the curvature radius of the relative streamline
ω is the angular speed of the turbine
δ is the angle between the turbine axis and the perpendicular to the cone surface in which the section through the runner blade canal are depicted
n is the length parameter of the width of the canal perpendicular to the streamlines.

The estimations of the velocity profiles according to equation (2.27) have again to be examined by means of the equation of continuity.

In general one can say that the blades should never be formed with particular sharp curvature. It is considered as ideal to give the blades a smooth form between the inlet and the outlet of the canals. This may often hit upon opposing requirements that lead to judgements and compromises between blades of short length and sharp curvature on one side and long blades with a smooth curvature on the other. Long canals are exposed to larger viscous friction losses than the short ones, while short canals are exposed to larger impulse and impact losses. The best solution is to be found somewhere in between these extreme limits. In such cases skilled designers are of course the right experts to do the job.

Pelton buckets

The design of buckets of Pelton runners [2] involves preliminary drafts of the hydraulic form of the buckets followed by computational and experimental examination and adjustment of the draft [2].

To have an idea of how the hydraulic modelling of the buckets may be done, Fig. 2.7 shows a calculated flow picture over a Pelton bucket at a certain moment. This analysis is based on the fact that the accelerations of the fluid particles must be perpendicular to the surface of the water flow in the bucket at any moment. The traces of fluid particles can then be found as shown on the figure. The following equations are used to find the accelerations and traces:

$$a_x = \frac{d^2x}{dt^2} + \omega^2 R\cos\phi + 2\omega u_y \tag{2.28}$$

where $\cos\phi = x/R$ and R = runner radius

$$a_y = \frac{d^2y}{dt^2} + \omega^2 R\sin\phi + 2\omega u_x \tag{2.29}$$

$$a_z = \frac{d^2z}{dt^2} \tag{2.30}$$

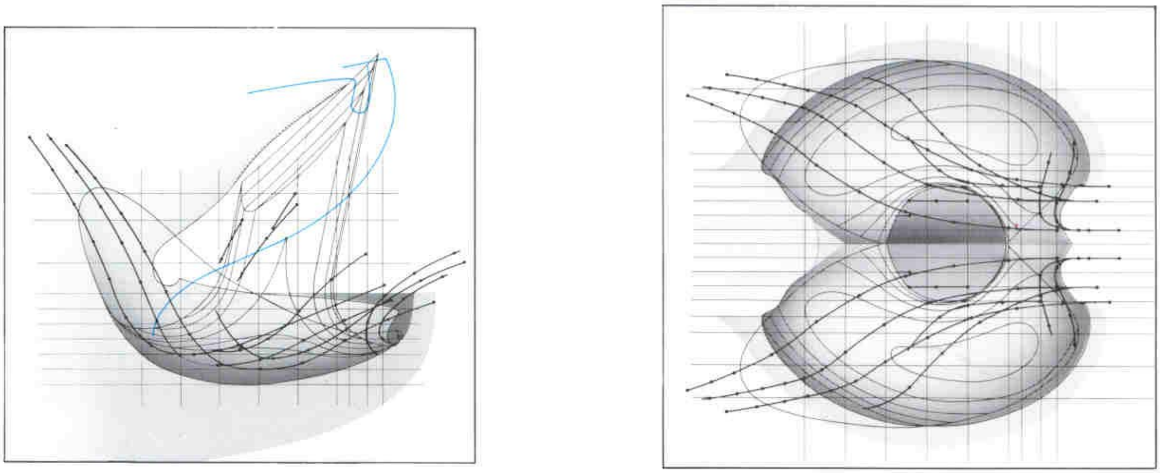

Fig. 2.7 Flow traces over a Pelton bucket at a certain moment (calculated) /2/

2.4 Efficiency

The equation of hydraulic efficiency, Eq. (2.22), expresses

$$\eta_h = \frac{P_R}{P_n}$$

The power transfer to the runner is further exposed to additional losses before the resulting power P is transferred to the generator shaft. These losses are composed of mechanical friction in the bearings and stuffing boxes, viscous friction from the fluid between the outside of the runner and the covers of the reaction turbines and ventilation or air friction losses of the runner in impulse turbines.

Through the space between the covers and the outside of the runner a leakage flow also passes according to the clearances of the labyrinth seals, from the inlet rim to the suction side of the runner. Some energy is also required for operation of the turbine governor, tapping water for sealing boxes, ejectors and cooling of bearings and the governor oil.

On account of all these losses the turbine efficiency is always lower than the hydraulic efficiency. Therefore, at the discharge Q and the power P transferred from the turbine shaft to generator shaft, the turbine efficiency is

$$\eta = \frac{P}{P_n} = \frac{P}{\rho Q g H_n} \tag{2.31}$$

Usually the maximum efficiency point which is represented by the best operating conditions, is reaching values of say $\eta = 0.93$ to 0.95 of the larger and best reaction turbines. Corresponding values estimated for the hydraulic efficiency $\eta_h = 0.95$ to 0.97. For the best Pelton turbines $\eta_{\max}$ reaches values about 0.92

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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

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

$\underline{u} = \frac{u}{\sqrt{2gH_n}}$ 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 $\frac{\omega}{\sqrt{H_n}}$ is constant (3.3)

- the discharge $\frac{Q}{\sqrt{H_n}}$ 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 $\frac{\underline{Q}}{D^2}$ 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 $\underline{Q}_U$ 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.

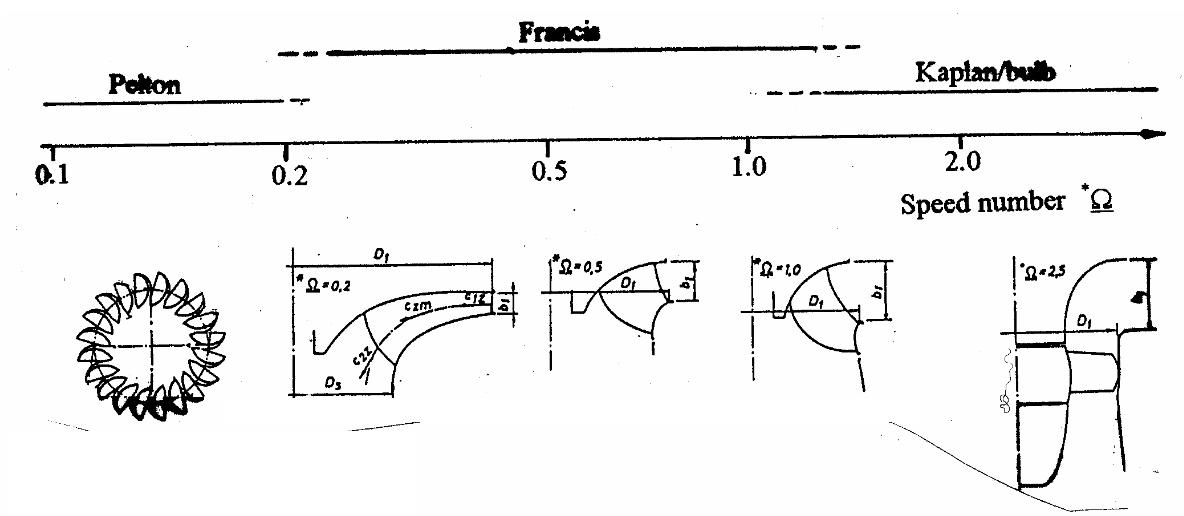

Fig. 3.1 Water turbines classified by speed number $^*\underline{\Omega}$

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

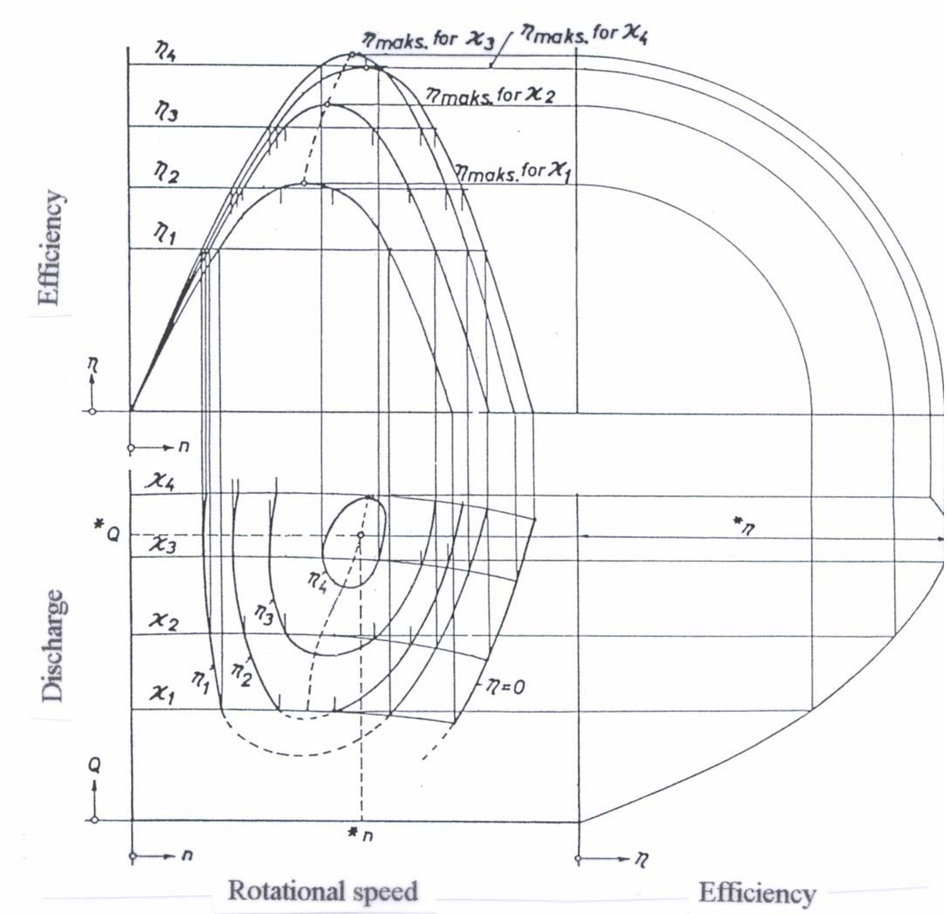

Fig. 3.2 Sketched drawing of the performance diagram of a turbine /5/

speed number $${}^*\Omega={}^*\omega\sqrt{{}^*Q} \tag{3.15}$$

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

$$\underline{Q}_U = \frac{\underline{Q}}{{}^*\underline{Q}} \tag{3.16}$$

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

$$\frac{\underline{\Omega}}{{}^*\underline{\Omega}} = \frac{\underline{\omega}}{{}^*\underline{\omega}} \tag{3.17}$$

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

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

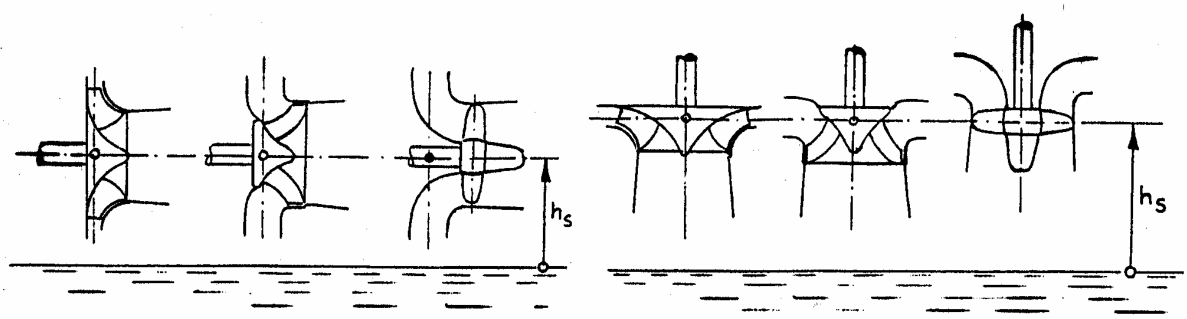

Fig. 3.3 Specification of suction head h_s of reaction turbines /5/

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

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

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

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

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

$$^{*}\underline{\Omega} = {}^{*}\underline{\omega}\sqrt{{}^{*}\underline{Q}} = \frac{2\,{}^{*}\underline{u}_{1}}{D}\sqrt{\frac{\pi}{4}\,{}^{*}d^{2}\,\underline{c}_{1}} \qquad (3.21)$$

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

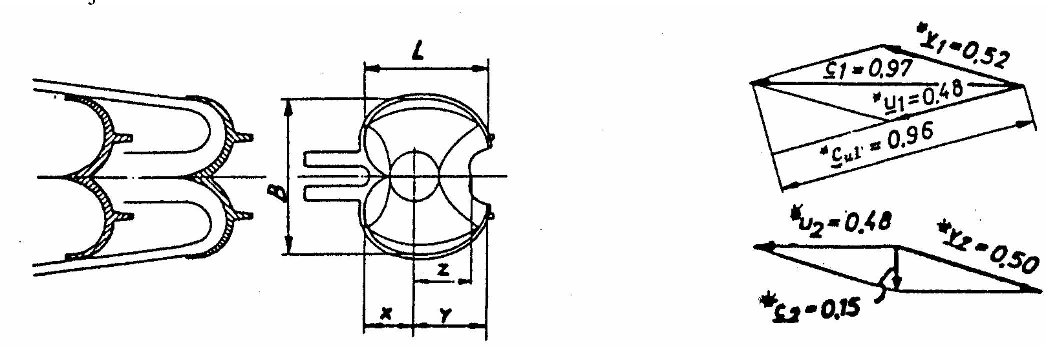

Fig. 3.4 Pelton turbine, indication of main dimensions and the bucket form /5/

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 $0.5{}^*Q < Q < 1.5{}^*Q$.

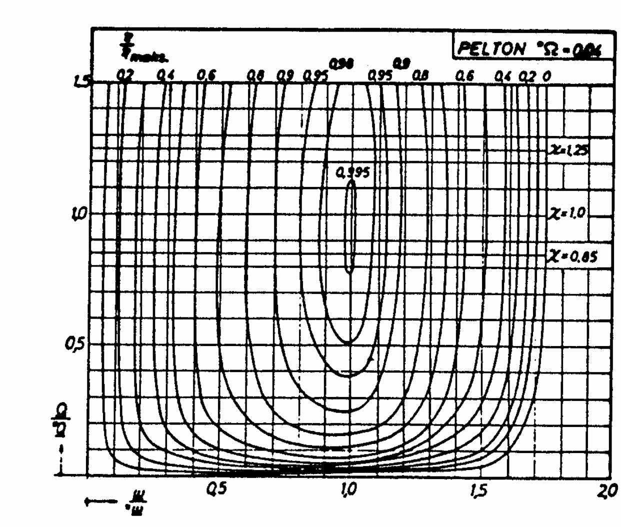

Fig. 3.5 Performance diagram[5] *of a Pelton turbine, $^*\underline{\Omega}$ = 0.04*

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,

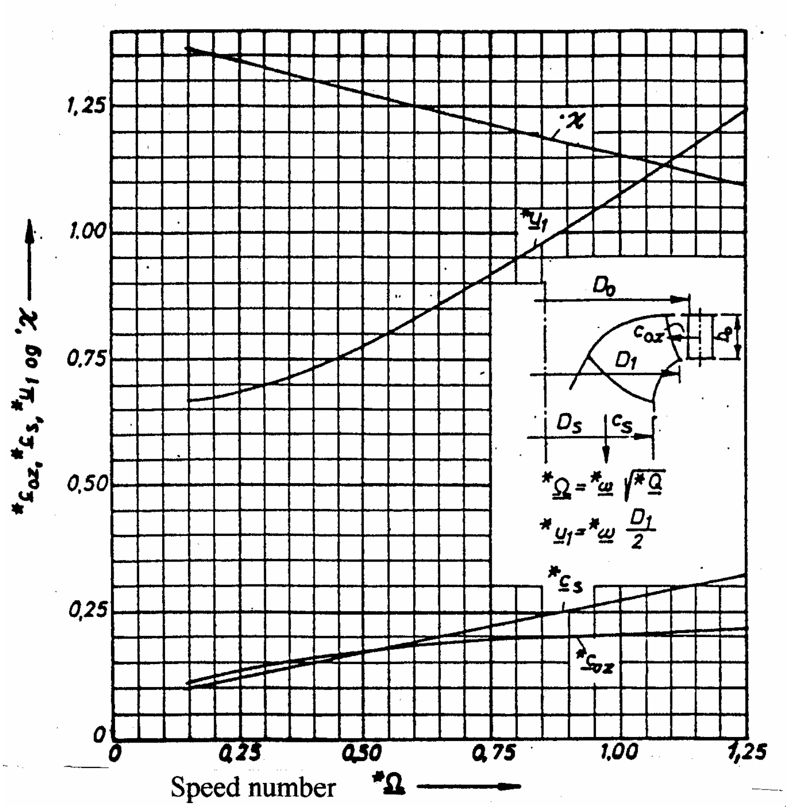

Fig. 3.6 Hydraulic design data for Francis turbines (intended values as a guide) /4/

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.

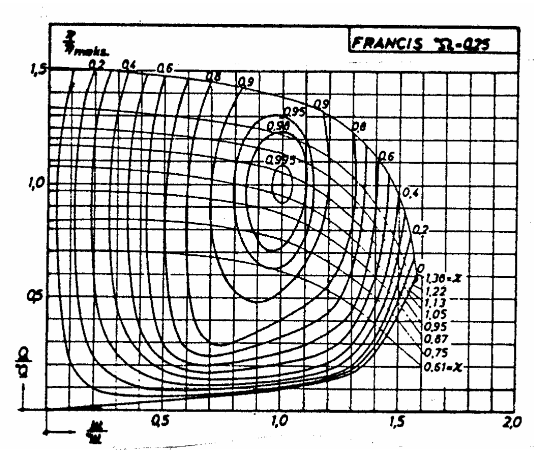

Fig. 3.7 Performance diagram/5/ of a Francis turbine $^*\underline{\Omega}$ = 0.25

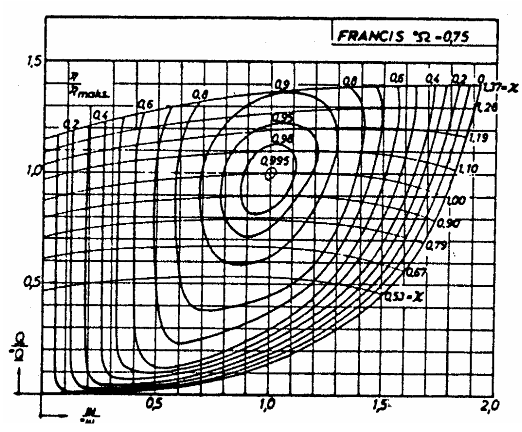

Fig.3.8 Performance diagram/5/ of a Francis turbine $^*\underline{\Omega}$ = 0.75

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

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

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

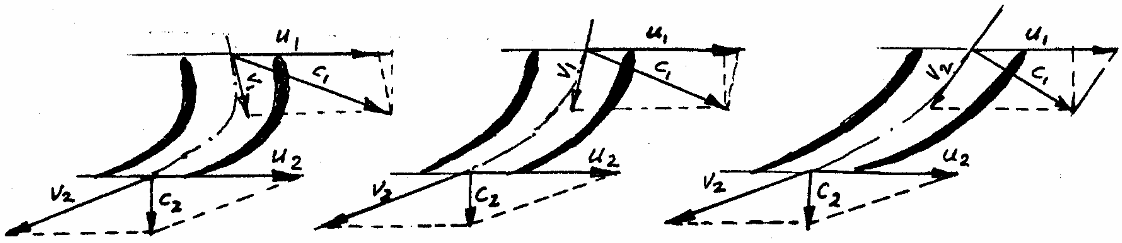

Fig. 3.9 Blade cascades for different reaction ratios with $\underline{c}_{u2} = 0$; /2/ .

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.

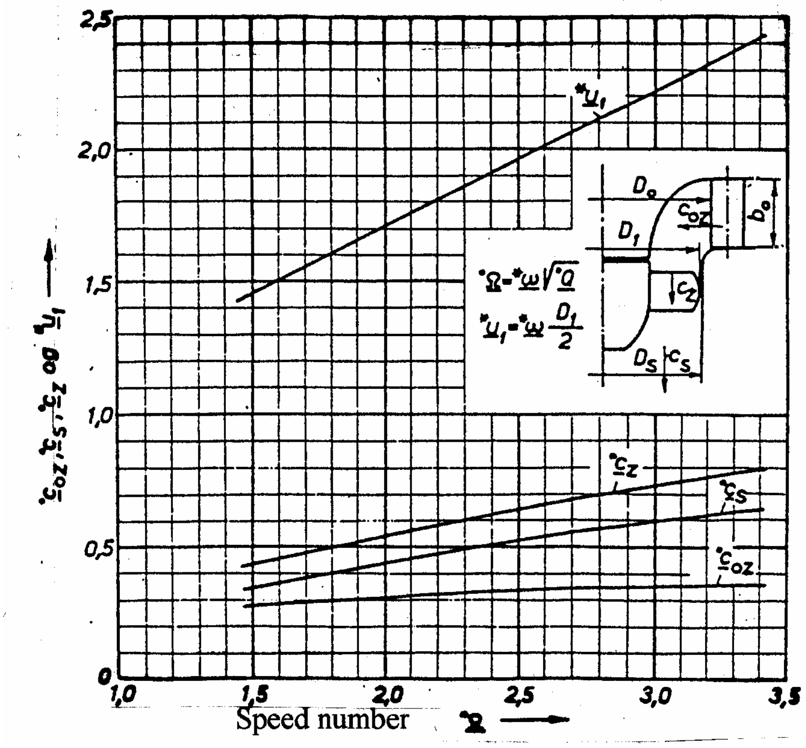

Fig. 3.10 Hydraulic design data for Kaplan turbines (intended values as a guide) /4/

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.

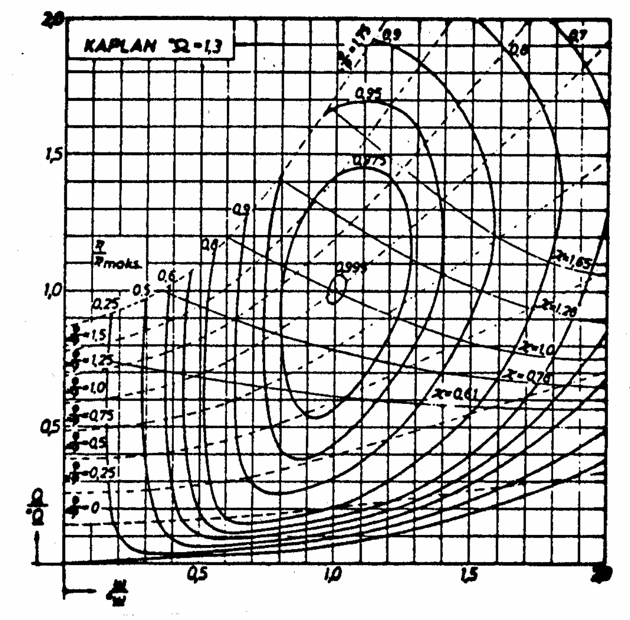

Fig. 3.11 Performance diagram[5] of a Kaplan turbine, $^*\underline{\Omega}=1.3$

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.

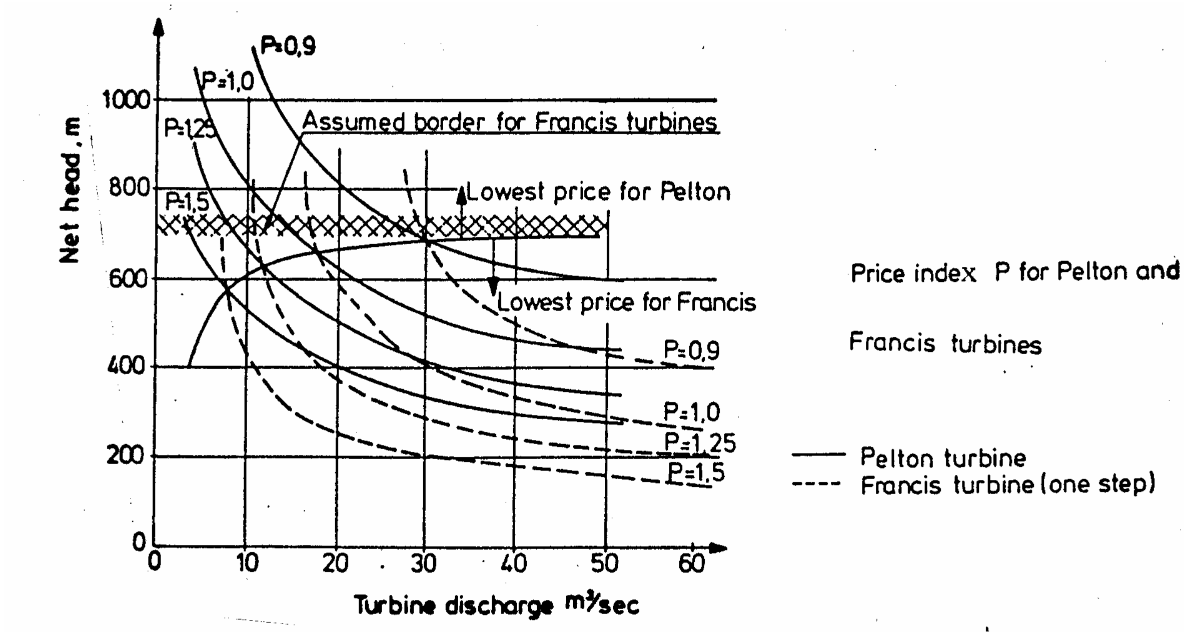

Fig. 3.12 Comparison of price index between Pelton and Francis turbines, /1/.

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.

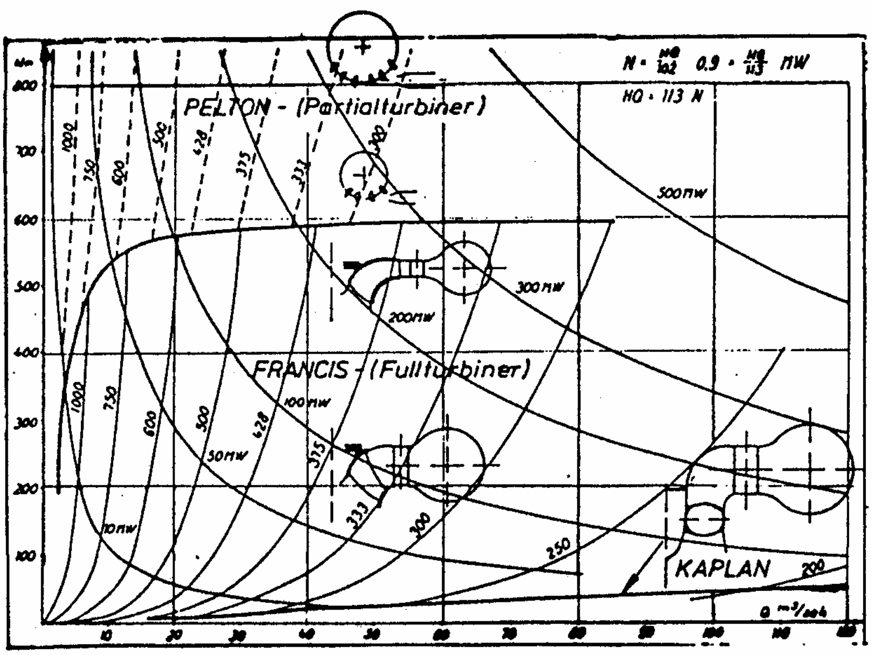

Fig. 3.13 Diagram indicating decreasing prices versus increasing speed number of turbines, /1/,/2/.

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.

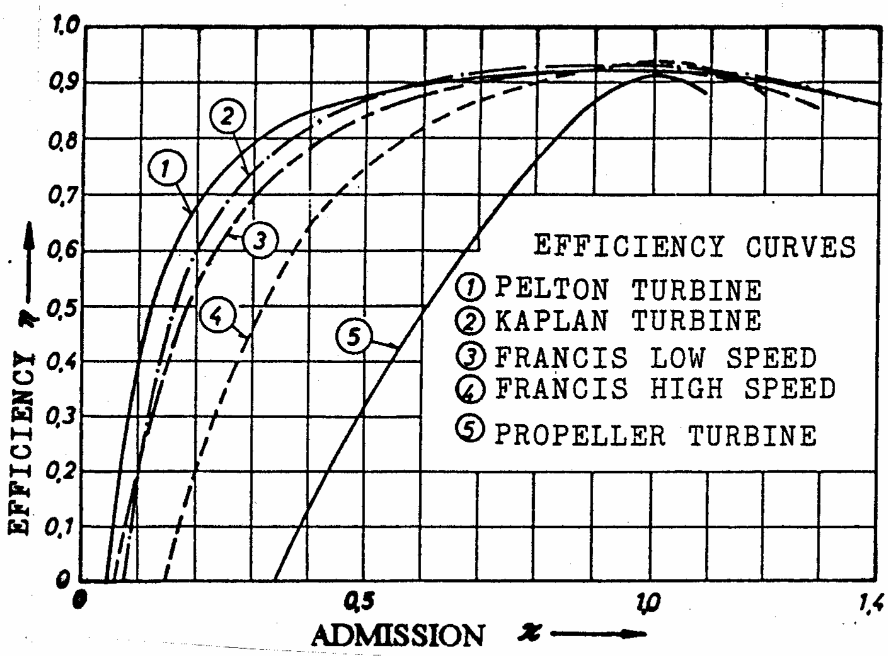

Fig. 3.14 Efficiency characteristics of different types of turbines as functions of capacity ratio /4/

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.

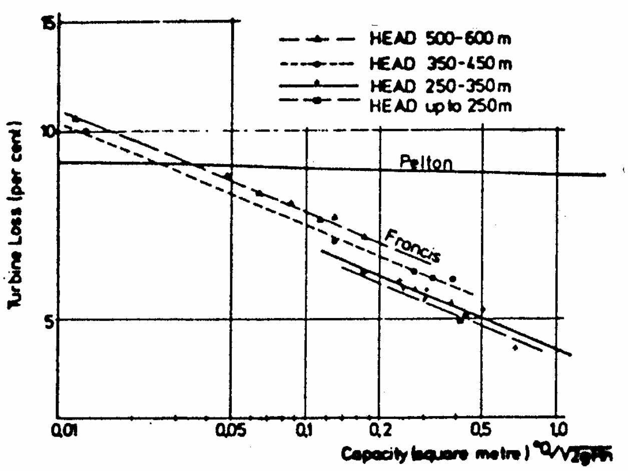

Fig. 3.15 Losses in Pelton and Francis turbines as function of capacity and , /1/,/2/.

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

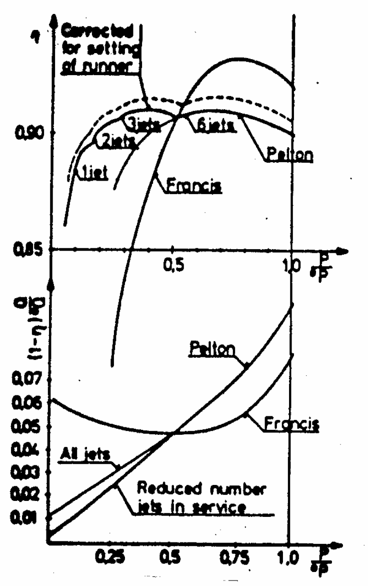

Fig. 3.16 Comparison of losses in a F rancis turbine and Pelton turbine of 50 MW at H = 470 m

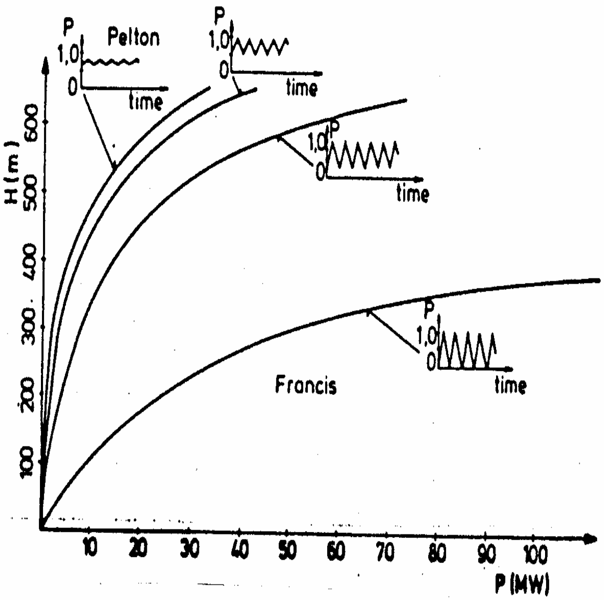

Fig. 3.17 Boundary curves between Pelton and Francis turbines based on different kinds of operation

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.

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CHAPTER 4

Governing Principles

Introduction

Turbine governors are equipment for the control and adjustment of the turbine power output and evening out deviations between power and the grid load as fast as possible.

The turbine governors[/3/,/4/,/5/,/6/] have to comply with two major purposes:

1. To keep the rotational speed stable and constant of the turbine-generator unit at any grid load and prevailing conditions in the water conduit.
2. At load rejections or emergency stops the turbine admission have to be closed down according to acceptable limits of the rotational speed rise of the unit and the pressure rise in the water conduit.

Alterations of the grid load cause deviations between turbine power output and the load. For a load decrease the excess power accelerates the rotating masses of the unit according to a higher rotational speed. The following governor reduction of the turbine admission means deceleration of the water masses in the conduit and a corresponding pressure rise.

To keep the rise of the rotational speed below a prescribed limit at load rejections, the admission-closing rate must be equal to or higher than a certain value. For the pressure rise in the water conduit the condition is opposite, e.g., the closing rate of the admission must be equal to or lower than a certain value to keep the pressure rise as low as prescribed.

For power plants where these two demands are not fulfilled by one single control, the governors are provided with dual control functions, one for controlling the rotational speed rise and the other for controlling the pressure rise. This is normal for governors of high head Pelton and Francis turbines.

For Pelton turbines the principle is:

- To set the closing rate of the needle control of the nozzles to a value which satisfies the prescribed pressure rise
- To bend the jet flow temporarily away from the runner by a deflector so the speed rise does not exceed the accepted level.

For Francis turbines the principle is:

- To set the closing rate of the guide vane opening to a value, which satisfies the rotational speed, rise limits

- To divert as much of the discharge through a controlled by-pass valve that the pressure rise in the conduit is kept below the prescribed level.

4.1 Feedback control system

The governor function for a turbine with water conduit is shown in the block diagram on Fig. 4.1.

The input reference signal is compared with the speed feedback signal. By a momentary change in the load a deviation between the generator power output and the load occurs. This deviation causes the unit inertia masses either to accelerate or to decelerate. The output of this process is the speed, which again is compared with the reference.

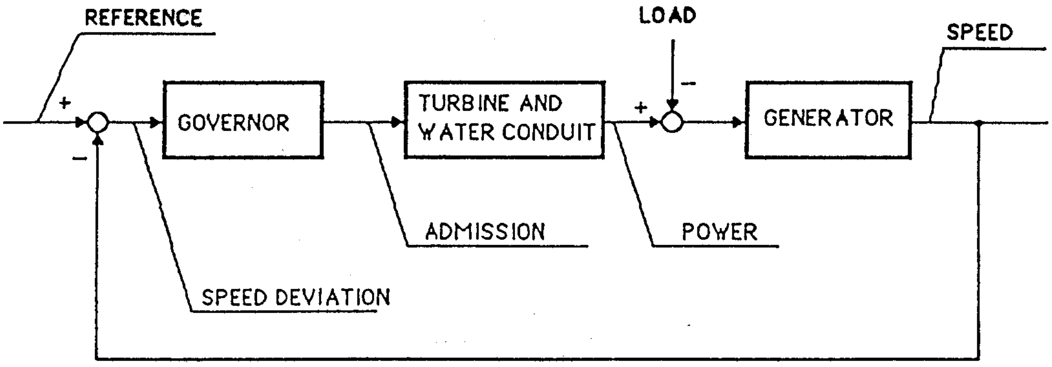

Fig. 4.1 Block diagram of a turbine closed loop system /6/

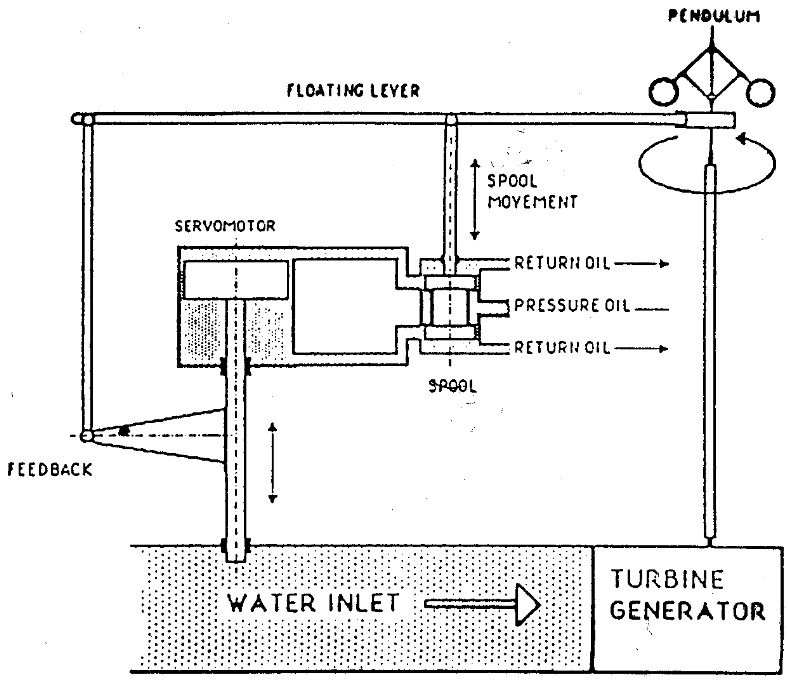

Fig. 4.2 A hydraulic governor with a direct acting pendulum /6/

A simple but classic example of a turbine governor is shown schematically in Fig. 4.2. This is a governor with a belt driven centrifugal pendulum. For explaining the governor actions it is chosen to start at a moment of stable equilibrium between power and load. In this condition the control valve is closed by the spool, which is in the neutral position.

When a decrease in the grid load occurs, the rotational speed starts increasing and the pendulum sleeve and the connected end of the floating lever moves upwards. The lever moves the spool accordingly upward out of the neutral position and opens the hydraulic conduits to the servomotor.

High-pressure oil flows to the piston topside. The piston moves downwards and reduces the gate opening and the turbine power. At the moment when the power is equal to the load, the rotational speed culminates as indicated on Fig. 4.3.

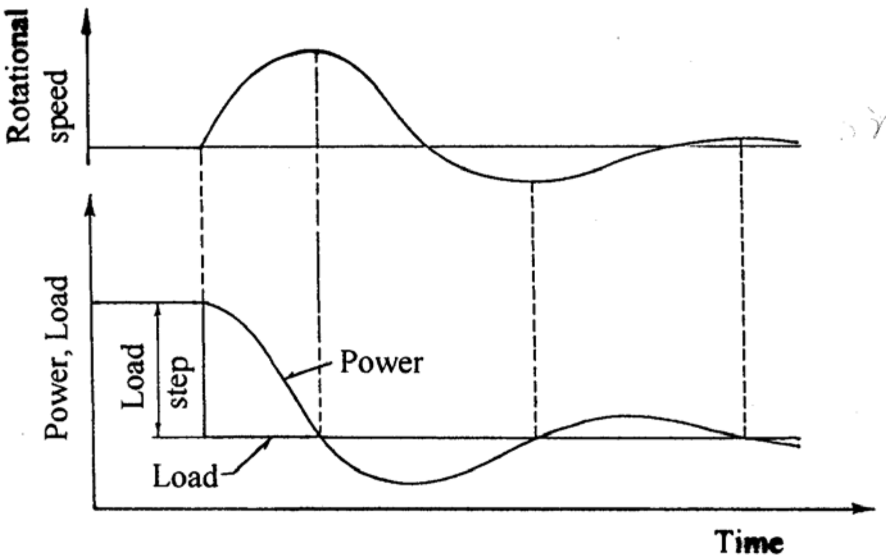

Fig.4.3 Time response of power output and rotational speed after a load reduction step

At this moment however, the spool valve is still open. The piston movement continues and the power output decreases even more. Consequently the speed decreases and the pendulum sleeve and the spool are moving downwards again.

During this movement the spool valve passes the neutral position and opens then for high-pressure oil flow to the opposite side of the piston. The piston movement is thereby returned and the power output increasing. Next time the rotational speed culminates the power again is equal to the load and therefore a succeeding swing in the speed and power output take place as previously described.

As Fig. 4.3 indicates, the swings are strongly damped because of the feedback system. This feed-back is arranged by a linkage connection between the servomotor rod and the end of the floating lever that is opposite to the sleeve as shown on the Fig. 4.2. When the piston moves in the closing direction, the floating lever moves the spool accordingly in the same direction towards the neutral position. In this way a stable control process is obtained.

A governor of a type as shown in Fig. 4.2 has a one-stage amplifier. Water turbine governors have normally several stages of amplifiers. The governors in use are of various designs, e.g., as mechanical-hydraulic and electro-hydraulic products. Design of governors and governor systems are described in Chapter 10.

4.2 Governor adjustment facilities

4.2.1 Proportional-integral-derivative functions (PID)

In governor systems as shown in Fig. 4.2 the rotational speed is dependent on the load for power/load

equilibrium conditions. That means higher rotational speed at zero loads than at max. load and this dependency are linear.

This type of governor function is designated as *proportional control* and denoted by the label P.

For units delivering the power to a grid system the frequency has to be constant at any load. Governors for these units therefore have adjustment means also for automatically recovering of the speed according to the grid frequency during regulation processes.

This type of governor function is designated as *integral control* and denoted by the label I.

Groups of governors are provided also with a derivative function in addition to the above functions. It may be utilised for improvement of the phase angle of the frequency response for the governor system.

This type of governor function is designated as *derivative control* and denoted by the label D.

Mechanical hydraulic governors are provided with PI-functions only while electro-hydraulic governors are designed with PID functions.

Governors with PID-functions have large ranges for adjustment of each of the PID function parameters.

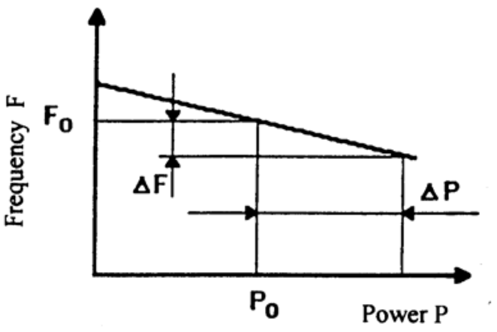

Fig.4.4 Permanent droop and strength of regulation

4.2.2 Permanent speed droop

A special and major property of a turbine governor is the permanent droop. It is defined as the percentage change in the frequency for a 100 % change of the power output from the unit.

For example a unit with a set value of the permanent droop at 5 % will respond with 40 % power increase for a change in frequency from 50 Hz to 49 Hz.

The permanent droop is adjustable and in practice it is chosen in the range 0 - 6 %.

In connection with the permanent droop another notion also plays a certain role. That is the *strength of regulation*[4]*, which* is defined as

$$\text{Strength of Regulation} = -\frac{\Delta P}{\Delta F}$$

where ΔP is power change

ΔF is frequency change

as illustrated in Fig. 4.4.

The load distribution between turbine-generator units connected to the same grid is dependent on the permanent droop setting of these units. In Fig. 4.5 two units with different permanent droop are shown where the load distribution for a given frequency change is indicated.

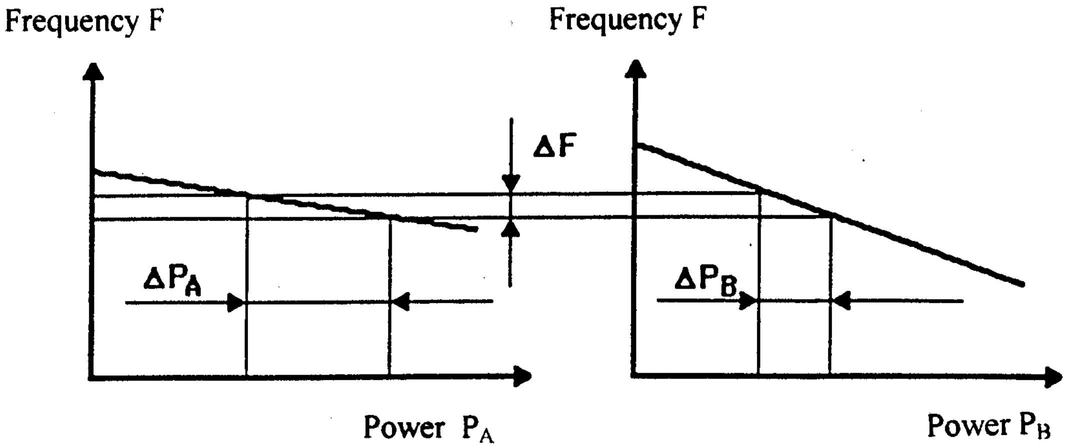

Fig. 4.5 Load distribution for different permanent droop of two units connected to the same grid /4/

4.3 Turbine governing demands

4.3.1 Frequency and load regulation

The governor shall be able to maintain stability of the generating unit when running on an isolated grid. Generally the units are designed for stable operation up to full load. In this mode of operation the governor shall keep the frequency within certain limits of deviation.

Load regulation on a rigid system is the most common operation mode. Each unit has little influence on the grid system frequency. The governor controls the load to the desired value. The variation of the load as function of the change in frequency is dependent on the permanent droop setting.

A special mode of operation is the manual mode where the guide vane openings are controlled manually by means of a mechanical hydraulic load limiter. In this mode only the load can be controlled.

4.3.2 Start and stop sequence control

During the period of start the unit shall be run up to nominal speed as quickly and smoothly as possible. A start can be carried out both manually and automatically. The admission must be opened only when permitted by all overriding start conditions.

In shut down mode the admission shall be closed as quickly as possible but limited by the magnitude of the pressure rise in the tunnel and pressure shaft system. Due to safety reasons, the shut down signal will be given simultaneously to different stages in the governor, e.g. closing of the load limiter or the emergency operated shut down valve. The shut down valve is also functioning if the ordinary voltage supply has failed. The stop command can be given both manually and automatically.

4.3.3 Disconnection, load rejection

Disconnection means to open the generator main circuit switch. The generator is thereby separated from grid and the turbine power output results in a speed rise of the unit. The function of the governor is then to shut down the turbine not faster than the caused pressure rise is kept below the guaranteed level.

4.3.4 Load limiting

Load limiting must be possible according to external conditions. The load limiter device may be operated both manually and automatically.

4.4 Regulation requirements of water power plants

The governing of water turbines requires limitations of speed rise and pressure rise as well as fulfilment of regulating stability demands. To guarantee these requirements the adaptation of the dynamic behaviour of the conduit system and the generator inertia mass have to be examined carefully. Basic theory and principles for these examinations are surveyed in the following.

An example of the layout of a traditional high head power plant is shown in Fig. 4.6.

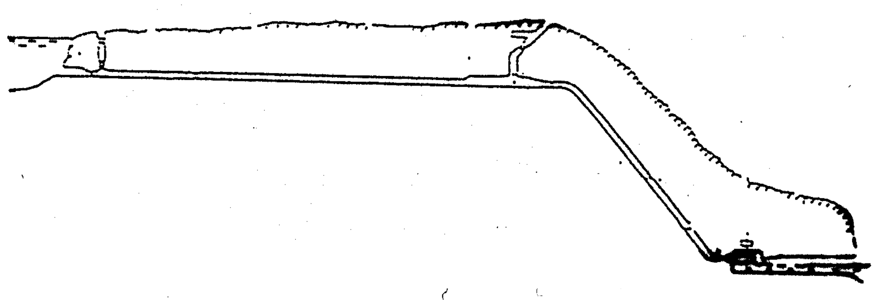

Fig. 4.6 The layout of a high head Hydro Electric Power Plant /6/

In such a power plant the pressure rise and regulating stability may be divided in two parts:

1. The mass oscillation problem for the tunnels and surge shafts both on the pressure side and the suction side.
2. The water hammer problem of the conduit between the turbine and the water level in the surge shafts both on pressure side and suction side.

4.4.1 Mass oscillations

The mass oscillation stability problem is treated according to the criterion for the minimum critical Thoma cross section area of the surge shaft, ref. /9/, for obtaining stable mass oscillations.

This criterion which is based on the equation of continuity, the equilibrium of forces and the assumption of an ideal turbine governor, is well described in the literature.

An ideal turbine governor is assumed to be able to keep the product of discharge and pressure as constant. However, this statement is not fulfilled if the tunnel is short with short surging time because the turbine governor then will be too slow to obtain constant output of the turbine. For short tunnels the margin to the critical Thoma cross section area of the surge shaft must be increased accordingly.

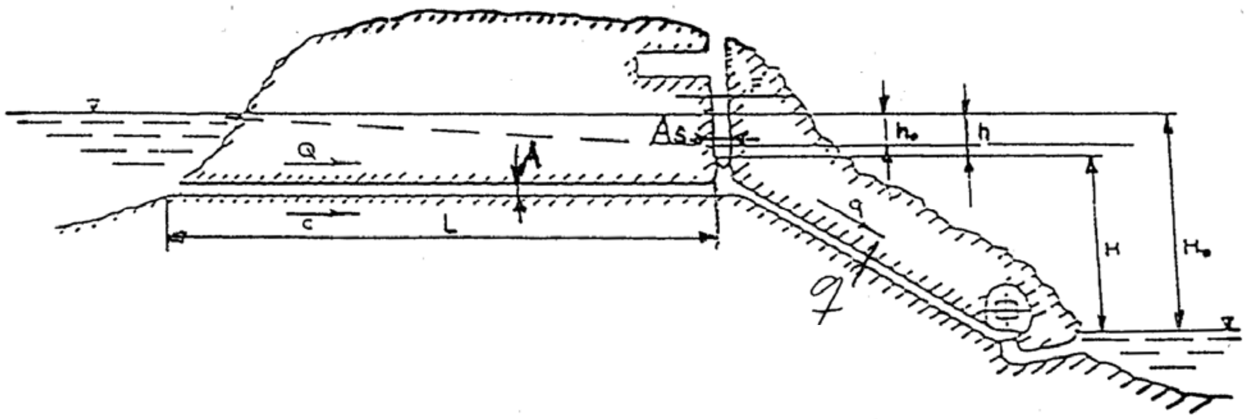

Fig. 4.7 Tunnel and surge shaft with values used in the simple Thoma criterion /6/

With reference to Fig. 4.7 the equation of continuity is

$$A_s \frac{dh}{dt} = q - Q \tag{4.1}$$

where A_s is the cross section area of the surge shaft
$h = H_o - H$ is the head difference between reservoir level and surge shaft level
q is the turbine flow
Q is the flow in the tunnel

The equilibrium of forces

$$\frac{L}{A}\frac{dQ}{dt} = g\left[h - h_o\left(\frac{Q}{Q_o}\right)^2\right] \tag{4.2}$$

where A is the cross section area of the tunnel
g is the acceleration of gravity
$h_o = (f/R_h)LQ_o^2/(2g\,A^2)$ is the head loss in the tunnel
f is the Darcy-Weissback friction factor
L is the length of the tunnel
Q_o is the steady state flow before disturbance or mean flow
R_h is the hydraulic radius of the tunnel cross section area

The equation of discharge and pressure control of the turbine (with an idealised turbine governor)

$$q(H_o - h) = Q_o(H_o - h_o) \tag{4.3}$$

According to ref./8/ the three equations give the following critical cross section area

$$A_{scr} = \frac{LQ_o^2}{2gh_o(H_o - h_o)A} \qquad \text{for } h_o < \frac{1}{3}H_o \tag{4.4}$$

When substituting for head loss h_o and H_o - h_o = H_n which is the net head,

$$A_{scr} = \frac{AR_h}{fH_n} \tag{4.5}$$

For a tunnel with an air cushion surge chamber the cross section area of the surge shaft A_s could be substituted for an equivalent cross section area $A_{eq.}$ By the equation of polytrophic changes of the air volume, the area A_{eq} is calculated for the compression of the air and surging of the level. According to ref./2/ the equation for A_{eq} is

$$A_{eq} = \frac{1}{\frac{1}{A_w} + \kappa\frac{H_o'}{V_o}} \tag{4.6}$$

where A_w is the water level area in the air cushion surge chamber
$H_o' = H_o + 10$ is the absolute air pressure head [m] in the air cushion surge chamber
V_o is the air volume in the air cushion surge chamber
κ is the polytropic exponent normally $\kappa \approx 1.3$

Normally $\frac{1}{A_w} << \kappa\frac{H_o'}{V_o}$ so that the formula may be simplified to

$$A_{eq} = \frac{V_o}{\kappa H_o'} \tag{4.7}$$

Then the critical air volume can be found by the simplified equation

$$V_{ocr} \approx \kappa H_o' A_{scr} \tag{4.8}$$

Normally a safety factor to the critical cross section area should be 1.3 - 1.5. According to ref./2/ however, it has been proven that for smooth full profile driven tunnels a larger margin should be used because of a smaller difference between steady state flow friction and friction of flow oscillations. For short tunnels an even larger margin is required than for long tunnels because a real turbine governor cannot satisfy Equation (4.3) for short tunnels on account of the shorter surging time of the pressure.

4.4.2 Water hammer pressure rise versus closure time and speed

The water hammer problems may be divided in to main areas:

1. The water hammer transient problems causing pressure rises that affects the stresses in penstocks and stress carrying parts of the turbine.
2. The governing stability problems caused by pressure oscillations in the conduit system.

The treatment of water hammer problems are based on ref. /1/. General solutions of governing stability problems and transients are based on ref. /8/. Special stability theory for power plants with influence of the frictional damping of oscillations in rough and smooth tunnels as well as the influence from the turbine characteristics is presented in ref. /2/.

The general equations

With reference to Fig. 4.8 parameters additional to those being used previously are:

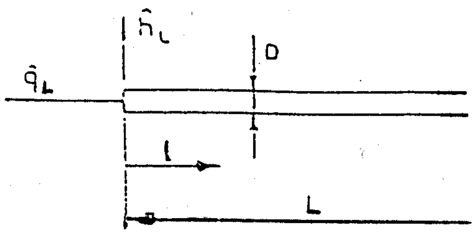

Fig. 4.8 Serction of pipe with diameter D

A is the cross section area of the pipe
D is the diameter of the pipe
l is the parameter of the pipe length from the pipe outlet
L is the length of the pipe

The most commonly used dynamic equations of water hammer problems yields:

The equation of continuity

$$\frac{\partial q_L}{\partial l} = \frac{AH_o g}{a^2 Q_o}\frac{\partial h_L}{\partial t} \tag{4.9}$$

Equilibrium of for

$$\frac{\partial h_L}{\partial l} = \frac{Q_o}{H_o gA}\left(\frac{\partial q_L}{\partial t} + Kq_L\right) \tag{4.10}$$

where $h_L = \frac{\Delta H}{H_o}$ is relative pressure variation

$\Delta H = H - H_n$ is the pressure head variation
H is the instantaneous pressure at the turbine inlet
H_n is the net head
$K = \frac{\pi D\tau}{\rho Q_o q_L}$ is the head loss friction factor, ref. /2/ (4.11)
$q_L = \frac{\Delta Q}{Q_o}$ is the relative flow variation
ΔQ is the change of the flow
ρ is the density of the fluid
τ is the friction shear force per unit length of the pipe

These basic Equations (4.9), (4.10) and (4.11) may be solved according to methods described in refs. /2/ and /7/.

Important parameters

The stability of the governing process of a hydro turbine-generator unit is dependent on the velocity c of the water in the conduit, the inertia of the rotating masses of the unit and the length L of the water conduit. The length L is defined as the distance between the turbine and the surge chamber or the distance between the turbine and the reservoir if there is no surge chamber.

According to the dependency of these parameters, some important compound parameters are created:

$T_a = \frac{\pi^2 n^2 I}{(30)^2 P_n}$ is the acceleration time of the rotating masses of the unit [sec]

$T_w = \frac{\sum Lc}{gH_n}$ is the inertia time constant of the water masses in the conduit [sec] (4.12)

$h_w = \frac{ac_m}{2gH_n} = \frac{T_w}{T_a}$ is the water hammer number (4.13)

where a is the propagation speed of the water hammer wave
g is the acceleration of gravity
H_n is the nominal head
$c = \frac{Q_n}{A}$ is the water velocity in the conduit
A is the cross section area of the water conduit
Q_n is the nominal flow
$c_m = \frac{\sum Lc}{\sum L}$ is the mean water velocity
n is the rotational speed
I is the inertia of the rotating masses
P_n is the nominal power of the unit
$T_r = \frac{2\sum L}{a}$ is the penstock reflection time

Simplified analysis of pressure rise from water hammer

During shut down of the unit a fast closure of the guide vanes (or needles of Pelton turbines) is required to avoid high speed rise. A fast closure however, causes a high pressure rise in the penstock and the speed rise has influence on the flow through reaction turbines.

For Pelton turbines the speed rise problem is normally solved by jet deflectors which deflect the jets quickly away from the runner and thus allowing for a slow closure of the needles. The corresponding solution of the speed rise problem of Francis turbines is to bypass the discharge through an energy dissipater with a valve controlled by the governor, see Chapt. 10.

For keeping speed rise and pressure rise within specified limits, some simple formulas are used for determination of closing and opening times of a gate at the downstream end of a penstock. These formulas[/3/] which are presented in the following, are valid for linear closure with constant speed. Frictional damping is neglected and the relative pressure $z = H/H_n$ is introduced.

- Closing time T_{CL}:

For elastic water and a long elastic penstock $2h_w < (1 + \sqrt{z})$, then the closing time for linear closing from 100% opening

$$T_{CL} = T_w \frac{2}{1-z} \tag{4.14}$$

Note: T_{CL} is negative for $\Delta H > 0$

- Opening time T_O:

For elastic water and elastic penstock

$$T_O = T_w \frac{2\sqrt{z}}{1-z} \tag{4.15}$$

Note: T_O is positive and $|\Delta H| < 1.0$

- For inelastic water and inelastic penstock:

Inelastic water and inelastic penstock means a short penstock $2h_w < 1 + \sqrt{z}$.

Closing time

$$T_{CL} = T_w \frac{\sqrt{z}}{1-z}$$

Opening time (4.16)

$$T_O = T_w \frac{\sqrt{z}}{1-z}$$

In this case maximum pressure occurs from large openings.

Note

The maximum value z is obtained with the maximum closing speed; that means 100% closure during the time T_{CL}. If the closure from a certain opening to zero takes the time 2L/a, then the reduction in velocity will be $\Delta c = c_o(2L/a)/T_{CL}$. By substituting for $z = 1 + \Delta H/H_o$, $T_w = L\Delta c/(gH_o)$ and $T_{CL} = 2L/a$ in formula (9.14), the maximum pressure rise becomes

$$\Delta H = \frac{a\Delta c}{g} \tag{4.17}$$

The simplified formulas may be used also for complex high pressure systems. In such cases the length of the penstock should be regarded as the length from the turbine to the nearest water level in the surge shaft. If an air cushion surge chamber is used, the nearest water level will be in the air cushion chamber where full reflection of pressure waves occurs. The air cushion surge chamber will be located within a relatively short distance from the turbine, and thereby excellent regulating conditions will be obtained.

Speed rise of turbine and generator

The speed rise of the turbine may also be calculated in a simplified way. According to the 2. law of Newton the maximum rotational speed n is found by the formula:

$$\frac{n}{n_n} d\left(\frac{n}{n_n}\right) = \frac{P - P_G}{T_a} dt$$

where n is rotational speed of the turbine
n_n is the nominal rotational speed of the turbine
P_G is the generator load
P is the turbine power output
P_n is the nominal turbine power output
$T_a = \frac{\pi n I}{30 P_n}$ is the acceleration time of the rotating masses of the unit
I is the moment of inertia of the rotating masses of the unit
t is the time

By introducing the speed rise parameter ε‘ and integrating one get the area A_n as indicated in Fig. 4.9.

$$\varepsilon' = \frac{1}{2}\left(\frac{n^2 - n_n^2}{n_n^2}\right) = \int_0^{T_{CL}} \left(\frac{P - P_G}{T_a}\right) dt = \frac{A_n}{T_a} \qquad (4.18)$$

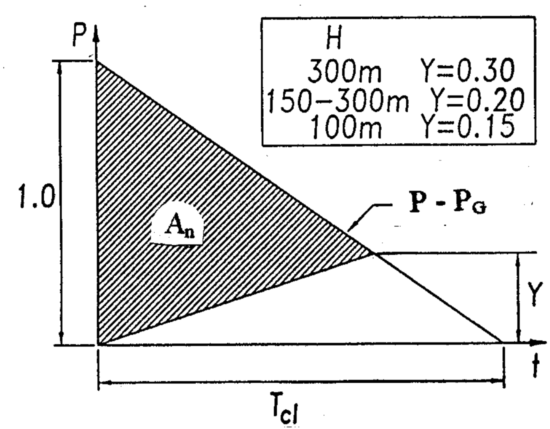

Fig. 4.9 Illustration of the integration of the accelerating torque versus time /3/

The presentation of the integrated torque as like the shaded area A_n in fig. 4.9, is based on a simplified consideration of the effect of the zero efficiency point. The zero efficiency point occurs just before closure of the guide vane cascade. The effect of this may be accounted for by simply to evaluate ε‘ in equation (4.18) satisfactorily by the following equation

$$\varepsilon' = \frac{1}{2} T_{CL} \frac{(1 - Y)(P - P_G)|_{t=0}}{T_a} = \frac{A_n}{T_a} \qquad (4.19)$$

where

T_{CL} is the closing time
$(P - P_G)|_{t=0}$ is the difference between turbine power and generator power at t = 0
Y is a numerical value dependent on the net head H_n

Values of Y are:

High head turbines $H_n > 300$ m — Y = 0.30
Medium head turbines 150 m $< H_n <$ 300 m — Y = 0.20
Low head turbines $H_n < 150$ m — Y = 0.15

The pressure rise may also be accounted for by multiplying the expression of ε‘ by $z^{3/2}$, where $z = H/H_n$ is the relative pressure. Then the speed rise becomes

$$\varepsilon' = \frac{A_n}{T_a} z^{\frac{3}{2}} \qquad (4.20)$$

Further the maximum speed can be calculated according to Equation (4.20)

$$2\varepsilon' = \frac{n^2 - n_n^2}{n_n^2} = (\frac{n}{n_n})^2 - 1$$

From this

$$\frac{n}{n_n} = \sqrt{1+2\varepsilon'}\,, \quad \text{and} \quad n = n_n + \Delta n$$

then

$$\frac{\Delta n}{n} = \sqrt{1+2\varepsilon'} - 1 \qquad (4.21$$

4.5 Governing stability

4.5.1 Modes of operation

Regarding governing conditions there are three different modes of operation on isolated loads:

1. Steady state operations when the unit is operating at constant load, head and command input.
2. The total system is subject to small changes caused by fluctuations in load or command input. In this mode none of the governor elements will reach the limit of closing or opening speed. The stability guarantees are always referred to this mode.
3. The total system is subject to changes, which is resulting in speed limits, closing and opening movements of parts of the governor system. This is the situation during load rejections when the main servomotors are operating at maximum closing speed.

When the generator is connected to a large grid with constant frequency, the function of the governor is only to change the generator output. If the generator is being disconnected, a load rejection will occur, which means that the system is in mode 3.

For stability analysis of operation on isolated grid it will often be sufficient to consider the system on the basis of the three compound parameters T_a, T_w and h_w[35/]. However, some power plants have a rather complex tunnel and conduit system which makes it necessary to work out stability analysis of the complete water way system.

As an example a conventional high or medium high head system with inlet tunnel, surge chamber, penstock with a 45^o slope and a tailrace tunnel, is sketched in Fig. 4.10. Concerning stability analysis the constants T_w and h_w are calculated from the length of the penstock between the water level in the surge chamber and the nearest free surface level in the tail water.

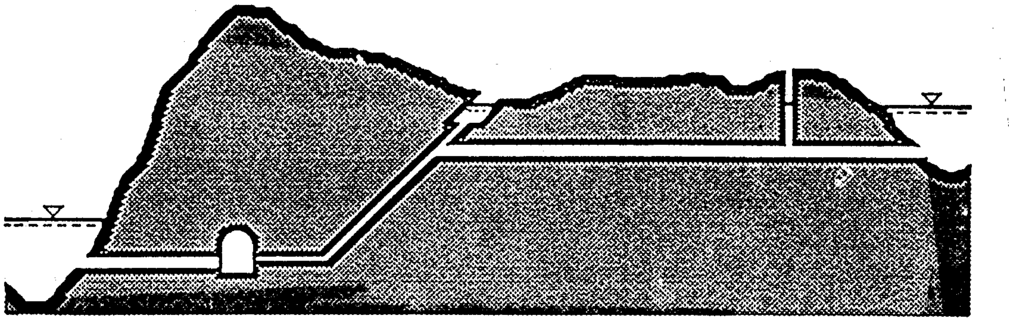

Fig.4.10 System with surge chamber /6/

In Norway high pressure tunnels have been excavated for increasing pressure levels. As a consequence the surge shaft length has increased. Therefore a relatively large part of travelling water

hammer waves in the conduit passes the surge shaft because of the increased inertia mass of water in the surge shaft. These waves are more or less reflected from every free water level in all branches of the tunnel and conduit system. In turn this leads to unstable turbine governing.

An improvement of the conditions has been made by the introduction of an air cushion surge chamber schematically shown in Fig.4.11. By means of such surge chambers the distance from the turbine to the nearest water level becomes short as well, and an excellent governing stability is thereby obtained.

In addition a short closing time of the turbine governor can be obtained with low speed rise without a high-pressure rise.

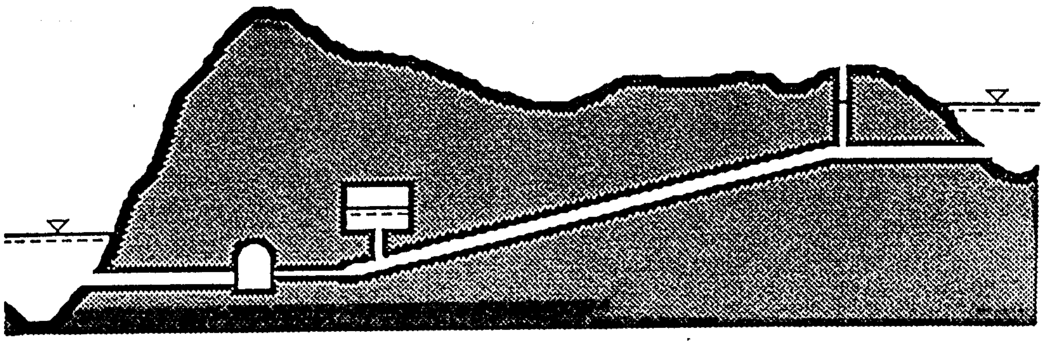

Fig.4.11 System with air cushion surge chamber /6/

For the stability analysis the Equations (4.9) and (4.10) have been Laplace transformed and the frictional shear force has been found to be a function of the flow oscillations and the frequency of the oscillations, ref. /2/. It should be emphasised that the influence from the turbine characteristics must be included to find the correct stability margins.

4.5.2 Rules of thumb

Some rules of thumb[/3/] may be established to make a brief judgement of the governing stability based on the water hammer number h_w and the relation between the time constants T_a and T_w. These rules are:

$2 < h_w$	$T_a/Tw \geq 2.5$
$1 < h_w < 2$	$T_a/T_w \geq 3.0$
$0.7 < h_w < 1$	$T_a/T_w \geq 3.5$
$0.5 < h_w < 0.7$	$T_a/T_w \geq 4.0$
$0.3 < h_w < 0.5$	$T_a/T_w \geq 4.5$

An efficient test for proving the governing stability is the frequency response test. Because the frictional damping of oscillatory flow is of great importance to know, it is most correctly estimated by this test.

Note

Modern governors with pressure feed back signal systems have additionally improved the stability of the governing systems.

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CHAPTER 5

Performance Tests

Introduction

Performance tests of hydro turbines are part of the commissioning and acceptance of the delivery of electro-hydraulic equipment.

The tests of hydro turbines should be made in accordance with the IEC recommendations. This is a claim by model tests as well if the prototype is a low head turbine and performance tests on the prototype is difficult to carry out with acceptable tolerances.

A turbine must in general meet a guarantee of the efficiency within a certain range of output and head variation. The turbine power output must fulfil the guarantee as a function of the net head as well. The operation in the given range of head shall also be without damage such as cavitation pittings and fatigue problems, on the turbine during the guarantee period.

Shuts down tests of inlet valves are also often a part of the acceptance tests when required.

In the course of the running time wear occur on vital parts of the turbine and the efficiency decreases accordingly. Therefore efficiency tests of a turbine may come into question several times.

Detailed guidelines for field acceptance tests are given in the International standard IEC 41[/9/] and for model tests in IEC 193[/10/].

5.1 Tests on prototype

5.1.1 Principles for test

The efficiency of hydro turbines may be determined according to two different principles:

1. Measuring the output P and the available power P_n at the turbine inlet and calculate the efficiency

$$\eta = \frac{P}{P_n} \tag{5.1}$$

2. Measurement of the total power losses ΔP_{loss} in the turbine and the suction pipe and determine the efficiency

$$\eta = 1 - \frac{\Delta P_{loss}}{P_n} \tag{5.2}$$

5.1.2 Measurement of the turbine power

Power output P

For a prototype turbine the electrical power output from the generator P_G is measured. Through the knowledge of the efficiency η_G of the electrical generator the turbine power output P is calculated.

The generator efficiency as function of cosϕ and the load can always be obtained from the generator manufacturer or measured on site as part of the generator performance tests.

The available power P_n

$$P_n = \rho Q g H_n \tag{5.3}$$

The discharge Q and the net head H_n are to be measured. The density ρ = 1000 kg/m^3 is generally used in the calculations if the contract of the measurements do not involve requirements of an exact check. For the acceleration of gravity g = 9.82 m/sec^2.

The net head H_n is evaluated above tail water level just at the outlet of the suction pipe of a full turbine, and above average level of the inlet of the jets in the buckets of a Pelton turbine. Net head H_n is composed of the hydraulic pressure head and the velocity head in the pipe cross section area A just in front of the turbine. For the velocity head it is usual to evaluate this directly as the average velocity head $c_m^2/(2g) = Q^2/(2gA^2)$.

The main methods for the determination of the discharge in hydro power plants are described in the following section.

5.1.3 Methods for determination of discharge

Measuring methods being used for the determination of the discharge in water power plants are:

1. Current meter method
2. Pitot tube gauging
3. Pressure-time method (Gibson method)
4. Tracer methods
5. Ultrasonic method
6. Weirs
7. Standardised differential pressure devices
8. Volumetric gauging method
9. Relative discharge measurement

Weirs, venturimeters, nozzles and volumetric meters are used especially for the measurements of discharge of smaller turbines.

Not all of these methods are generally applicable, and which of them are to be used in the respective power plants is a matter of choice based on main aspects as compatibility, economy and accuracy.

5.1.3.1 Current meter method

The current meter method requires a number of propeller-type current meters. These are located at specified points in a suitable cross section of an open channel or closed conduit. Simultaneous measurements of local mean velocity with the meters are integrated over the gauging section to estimate the discharge.

Current meters are instruments designed as propellers with 2 or 3 blades. Fig. 5.1 shows an example of a current meter design with a two-blade propeller.

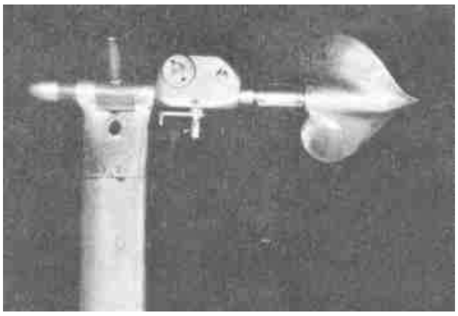

Fig. 5.1 Current meter /3/

The current meter is put in the flow with the propeller axis parallel to the flow direction and the propeller peak against the flow. The rotational speed n of the propeller is a linear function of the flow velocity c in the measuring point.

$$c = kn + b \qquad (5.4)$$

where k and b are constants of the respective current meter and has to be determined by calibration tests.

The rotational speed of the current meter is detected by an electric contact giving a pulse frequency signal proportional to rotational speed.

The flow velocity is recorded in the centre of gravity of each grid element of the cross section area as shown on Fig. 5.2. By considering an arbitrary element with area A_i where the recorded velocity is c_i, the discharge $\Delta Q_i = c_i A_i$ through this element. If the flow cross section is divided in n elements, the total flow discharge is found by

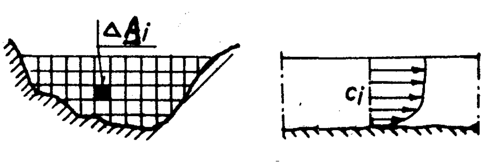

Fig. 5.2 Flow cross section /3/

$$Q = \sum_{i=1}^{n} c_i A_i \qquad (5.5)$$

Depending on the conditions on site the arrangement of current meter measurements may be carried out by a number of current meters installed in a kind of structure being built across the flow section.

Current meter measurements may be applied in open drains, channels, rivers as well as in closed pipes. To achieve accurate results, it is however important that the flow through the cross section of the measurements is regular and as rectilinear as possible.

The accuracy by current meter gauging of the discharge depends essentially on factors related to the flow, the quality of measurements, a careful reflection of the gauge point distribution and the method of discharge calculation. With good measuring techniques and flow conditions, the estimated uncertainties[/9/] should be about:

- in closed conduits $\pm$ 1 to $\pm$ 1.5 %
- in open channels with rectangular section $\pm$ 1.2 to $\pm$ 2 %
- in open channels with trapezoidal section $\pm$ 1.4 to $\pm$ 2.3 %

5.1.3.2 Pitot tube gauging

The pitot tube gauging means to measure the stagnation pressure of the flow velocity directed into the tube end opening. Pitot tubes are found in a great variety of designs. Fig. 5.3 shows an example of a frequently applied type called Prandtl tube.

Standardised pitot tubes are reported in ISO 3966[/13/], which covers the design, installation and use of these tubes. This standard also gives guidelines for the selection and installation of pitot

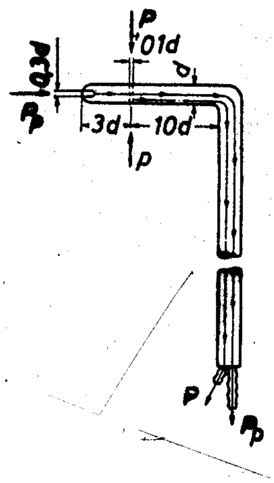

Fig. 5.3 Prandtl tube /3/

static tubes, choice of measuring section and the computation of the discharge and its uncertainty. ISO 3966 shall be used only with the standardised Pitot tubes that are described therein and equipped with a single total pressure tap and one or more static pressure taps. Such tubes may be used uncalibrated and the flow coefficient assumed to be unity.

The local velocity v_i is given by:

$$v_i = \sqrt{2\Delta p_i / \rho} \tag{5.6}$$

where Δp_i is the difference between the total stagnation pressure and the static pressure measured with the pitot tube located at point "i".

Pitot tubes are applied in the same way as described for current meter measurements, for gauging the flow velocity in chosen points of a flow cross section. The total discharge Q is also determined analogous to the scheme described for current meters.

Pitot tubes are not well fit for velocity measurements in liquids when the flow velocity is lower than 1 m/s.

With good measuring techniques and flow conditions the estimated uncertainty should be about 1.5 to 2.5 %.

5.1.3.3 The pressure-time method (Gibson method)

The pressure-time method[/1/] for discharge determination is based on the pressure rise when a flow regulating device in a closed conduit reduces the water flow.

The pressure rise on the upstream side of the regulation device depends on the closing speed,

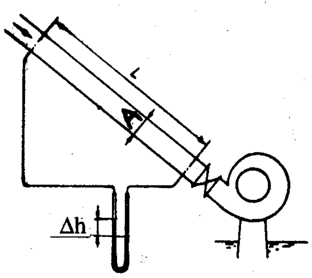

Fig. 5.4 Measurement of differential pressure in a pipeline /3/

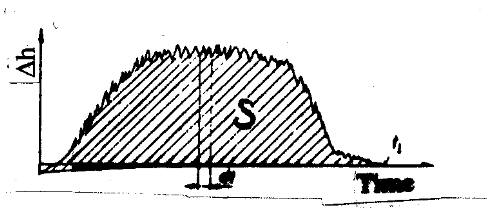

Fig. 5.5 Pressure differential-time-diagram /3/

the conduit length, the net head and the flow velocity in the conduit at the start of the closing operation. On Fig. 5.4 is schematically shown a turbine connected with a pipeline. A differential manometer is connected to the pipe through pressure tappings in the pipe wall, and the distance between the upstream and downstream tappings is called L.

When the turbine admission has a closing movement, a pressure head difference Δh as function of time may be recorded on the differential manometer, as shown on the diagram Fig. 5.5.

This pressure head differential-time-diagram is a measure of the total flow.

The equilibrium of the retarded water flow mass for an element of the cross section in the considered pipe length L may be expressed by:

$$\rho g \Delta h dA = -\rho L dA \frac{dc}{dt} \tag{5.7}$$

If t_l is the time during which the velocity changes, and if Δh_{loss} is the pressure head loss due to friction between the two pipe sections, then:

$$A \int_{t=0}^{t=t_l} dc = -\frac{gA}{L} \int_{t=0}^{t=t_l} (\Delta h + \Delta h_{loss}) dt \tag{5.8}$$

The discharge Q before the closure operation begins, is then:

$$Q = Ac_o = \frac{gA}{L} \int_{t=0}^{t=t_o} (\Delta h + \Delta h_{loss}) dt + Ac_{t_l} \tag{5.9}$$

The discharge q = Ac_{t1} is the leakage flow past the gate after shut off. This leakage must be determined separately with the machine running.

The pressure-time method is applicable on flow in closed pipes only. Moreover the measuring length L must be 9 meter or two times the pipe diameter if this product is greater than 9 meter.

The pressure-time method requires especially good instrumentation[/1/] and a highly qualified staff of specialists to carry out the tests. Under favourable conditions an overall uncertainty of about $\pm$ 1.5 % to $\pm$ 2 % may be expected.

Indications are that applying the pressure-time method in conduits less than 1 meter in diameter leads to overestimating the discharge.

5.2.3.4 Tracer methods

Allen salt velocity method

A salt in water dilution increases the electric conductivity of the water. By injecting a salt dilution dose in a water flow conduit, the transit time of this dose between two electrodes in the conduit can be traced electrically. The conduit volume V between these two electrodes divided by the average transit time t_{mean} of the passage of this salt dose, gives the true value of the discharge:

$$Q = \frac{V}{t_{mean}} \tag{5.10}$$

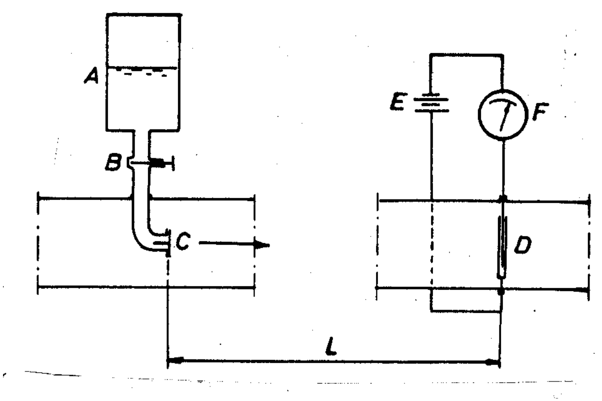

Fig. 5.6 Allen salt velocity method /6/

Fig. 5.6 shows schematically an arrangement[/6/] for the application of this method.

A dilution of common salt is kept in the container A. This dilution is pressurised to a certain level. As soon as the rapid operating valve B opens, an adequate dose of salt dilution is forced into the pipeline through the spring loaded valve C. The injected dose is transported along the pipeline with the same velocity as the main flow. However, it will be rapidly diluted and its extension in the flow direction is durably increasing due to the larger flow velocity in the central part than in the neighbourhood of the wall of the pipe cross section.

In cross sections of the chosen lengths of the pipe, electrodes are installed as shown schematically on the figure. When an electric voltage E and an electric recorder F is connected in series with the electrodes, the recorder will record an electric current dependent on the conductivity of the water analogous to the diagram shown on Fig. 5.7.

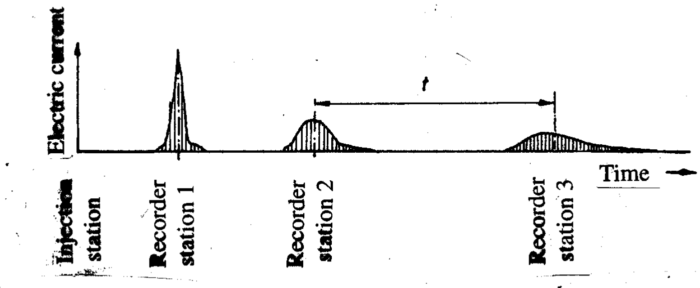

Fig. 5.7 Record of a salt cloud passing the electrode stations in a pipeline /6/

Fig. 5.7 Record of a salt cloud passing the electrode stations in a pipeline /6/

For the application of this method expert knowledge of a number of details in the experimental equipment is needed.

For the evaluation of the discharge, several methods have been applied for the calculation of the time interval of the passage of the salt cloud between the respective electrode stations. However, these methods do not differ from each other so much that not any of them is recommended as the most preferable. Therefore this is a matter of choice for the experts.

With good measuring techniques and flow conditions, it is generally accepted that the discharge may be determined to an accuracy around $\pm$ 1 % to $\pm$ 1.5 % by the use of the Allen salt velocity method.

Salt dilution method

This method is apt to be called the chemical method. It incorporates Mohr's procedure for titration of chlorides by means of silver nitrate.

The method has been used for the measurements of the discharge in mountain rivers[/5/]. The results being obtained has shown a development of accuracy that has led to application of this method for efficiency determination of water turbines.

The salt dilution method is principally different from the Allen salt velocity method. In a flow through pipes or open channels as shown on Fig. 5.8, two cross sections (I) and (II) with a certain mutual distance, are chosen. In cross section (I) a steady continuous flow of a homogeneous and relatively strong concentrated solution of sodium dichromate is injected into the main water flow in points evenly distributed over the cross section. The flow downstream in the channel becomes a dilution with a concentration depending on the relation between the magnitude of the main water flow and the magnitude of the injected flow of salt solution. In cross section (II), which is a distance far enough

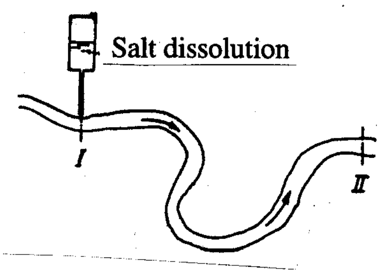

Fig. 5.8 Dilution method /5/

downstream to ensure thorough mixing, samples are taken out from several positions in the cross section. By means of Mohr's titration method and application of silver nitrate and potassium chromate an accurate gauging of the concentration of the dilution may be obtained in these samples.

It is not necessary to know the geometric characteristics of the pipe, but it is essential to ensure that reverse or side currents do not exist which could abort some of the injected solution. Also the concentration of salt in the natural water must be constant and not exceed 15 % of the concentration at the sampling point during injection of the salt solution.

The discharge Q can be determined from:

$$Q = q\frac{C_1 - C_2}{C_2 - C_o} \tag{5.11}$$

where Q is the discharge to be measured
q is the discharge of the salt solution injected
C_o is the initial concentration of the salt in natural water
C_1 is the concentration of salt in the injected salt solution
C_2 is the concentration of the salt dilution at the sampling station

For the application of this method the flow must be perfectly turbulent, otherwise the mixture of the salt solution and the main water flow will be uneven. Moreover, the salt solution must be injected in points positioned relatively close to each other over the cross section. The samples in the cross section downstream as well must be taken in points correspondingly close distributed.

The concentration of the injected salt solution may be one part by weight of salt to four parts by weight of water.

Instead of the salt solution being described above, other radioactive and non-radioactive tracers can be used, provided the recommendations and procedures described in Parts 1, 6 and 7 of ISO 2975[/12/] are applied.

With good measuring techniques and flow conditions, the obtained accuracy of the discharge determined by the dilution method should be about $\pm$ 1 % to $\pm$ 1.5 %.

5.1.3.5 Ultrasonic method

Small-magnitude pressure disturbances are propagated through a fluid at velocity which is the sound velocity *relative* to the fluid. If the fluid also has a velocity, the *absolute* velocity of the pressure disturbance propagation is the algebraic sum of the two. Since the discharge is related to fluid velocity, this effect may be used in several ways as the operating principle of ultrasonic flow metering.

The term ultrasonic refers to the fact that the pressure disturbances usually are short bursts of sine waves whose frequency is above the range audible to human hearing.

The various methods[/2/,/4/] of application of the above phenomenon all depend on the existence of transmitters and receivers of acoustic energy. A common approach is to utilise piezoelectric crystal transducers for both functions. In a transmitter electrical energy in the form of a short burst of high-frequency voltage, is applied to a crystal and causing it to vibrate. If the crystal is in contact with the fluid, the vibration will be communicated to the fluid and propagated through it. The receiver crystal is exposed to these pressure fluctuations and responds by vibrating. The vibratory motion produces an electric current signal in proportion according to the action of piezoelectric displacement transducers.

Fig. 5.9 shows the most direct application[2] of these principles. With zero flow velocity the transit time t_o of pulses from the transmitter to the receivers is given by

$$t_o = \frac{L}{a} \tag{5.12}$$

where L is the distance between transmitter and receiver.

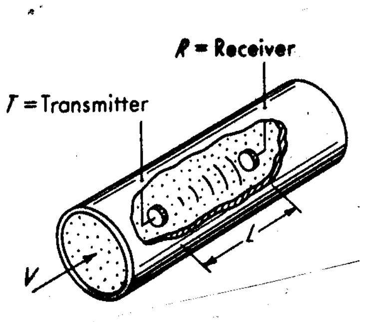

Fig. 5.9 Principle of ultrasonic signalling /2/

The velocity

a is the acoustic (sound) velocity in the fluid

If the fluid is moving at a velocity c, the transit time becomes

$$t = \frac{L}{a+c} = L\left(\frac{1}{a} - \frac{c}{a^2} + \frac{c^2}{a^3} - .....\right) \approx \frac{L}{a}\left(1 - \frac{c}{a}\right) \tag{5.13}$$

and defining $\Delta t = t_o - t$, then

$$\Delta t \approx \frac{Lc}{a^2} \tag{5.14}$$

Thus, if a and L are known, measurement of Δt allows calculation of c. However, while L may be taken as constant, a varies both with temperature and pressure and may cause significant error because of its appearance as a^2. Also, Δt is quite small since c is a small fraction of a. Since it is not directly provided for measurement of t_o in this arrangement, the modification of Fig. 5.10a may be preferable. If t_1 is the transit time with the flow and t_2 is the transit time against the flow, then it is obtained

$$\Delta t = t_2 - t_1 = \frac{2Lc}{a^2 - c^2} \approx \frac{2Lc}{a^2} \tag{5.15}$$

This Δt is twice as large as before and is also a time increment that may be directly measured. However, the dependence on a^2 is still a drawback.

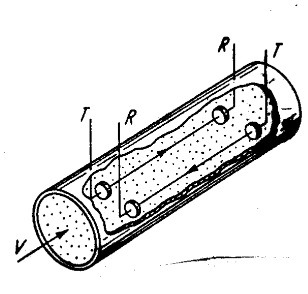

Fig. 5.10a Sound signal sent with and against the flow direction /2/

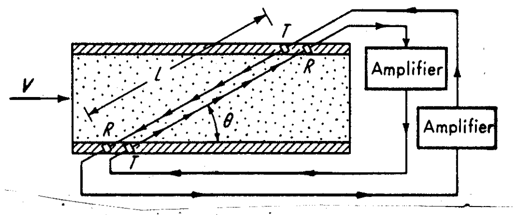

Fig. 5.10b Two self-excited oscillated systems /2/

In Fig. 5.10b two self-excited oscillating systems are created by using the received pulses to trigger the transmitted pulses in a feedback arrangement. The pulse repetition frequency in the forward propagating loop is $1/t_1$ while in the backward loop is $1/t_2$. The frequency difference $\Delta f = 1/t_1 - 1/t_2$ can be measured by multiplying the two signals together to get a beat frequency. Since $t_1 = L/(a + c \cos\theta)$ and $t_2 = L/(a - c \cos\theta)$, then

$$\Delta f = \frac{2c\cos\theta}{L} \tag{5.16}$$

which is independent of a and thus not subject to errors due to changes in a.

The above analysis assumes a square velocity profile which does not occur in practice. For actual profiles c can be replaced by c_{mean} as long as the profiles are symmetrical about the pipe centre line. If certain mathematical conditions such as continuity and differentiability are met by the velocity distribution, the discharge can be obtained from the equation for a circular section:

$$Q = k\frac{D}{2}\sum_{i=1}^{n} W_i c_{meani} D \sin\alpha_i \tag{5.17}$$

where D is the diameter of the pipe in the intersecting acoustic plane
W_i are weighting coefficients depending on the number of paths and the applied integration technique
n is the number of acoustic paths in one plane
k is correction coefficient which accounts for the error introduced by the integration technique
α_i defines the angular location of the end path relative to D

Experience with the acoustic methods of discharge measurement is limited and obtained accuracy is about ± 2 %.

5.1.3.6 Weirs

The measurement principle is to measure the discharge by interposing a thin plate weir in a free surface flow and observe the head over the weir. A unique functional relationship between the discharge and the head over the weir is employed. In order to have the best known relationship, only rectangular weirs without side contraction sharp crested, with complete crest contraction and free overflow shall be used.

The basic formula for calculating the discharge is due to Poleni and can be written as[/9/]:

$$Q = \frac{2}{3}Cb\sqrt{2gh^3} \tag{5.18}$$

where Q is the discharge
C is the discharge coefficient
b is the length of the weir crest (perpendicular to the flow)
g is the acceleration of gravity
h is the measured upstream head over the weir

The weir plate shall be smooth and plain, particularly on the upstream face, and shall remain unaltered for the whole duration of measurements. It shall preferably be made of metal which can resist erosion and corrosion. It shall be rigid, watertight and perpendicular to the walls and to the bottom of the channel.

Fig. 5.11 shows a sketch of a rectangular weir. The surface of the weir crest shall be a horizontal, flat and smooth surface perpendicular to the upstream face of the plate. Its intersection with the upstream face shall be straight and form sharp edges free from burrs or scratches. The width e of the edge, perpendicular to the upstream face, shall be 1 to 2 mm. If the weir plate is thicker than the allowable crest width, the downstream edge shall be chamfered at a 45° angle.

Aeration of the free efflux from the weir shall be secured with a ventilation sufficient to keep the air underneath the free efflux at approximately atmos-pheric pressure. The weir is commonly located on the low pressure side of the turbine, and care shall be taken to ensure that smooth flow exists in the approach channel. With this location it shall moreover, be far enough from the turbine or the discharge conduit outlet to enable the water to release the air bubbles before reaching the weir.

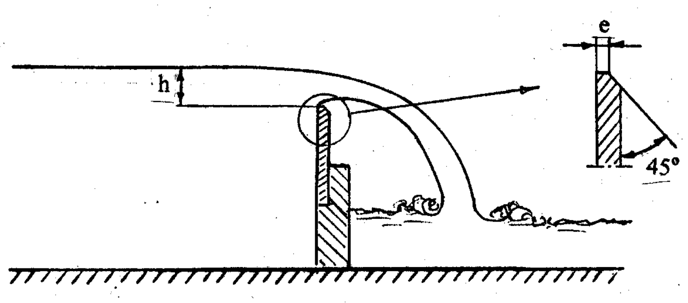

Fig. 5.11 Sketch of a sharp-crested rectangular weir

The approach channel shall be straight and of a uniform cross section and with smooth walls for a length of at least 10 times the length of the weir crest b. Along this length the bottom slope must be very small (< 0.005).

The sides of the channel above the level of the crest of the weir shall extend without discontinuity at least $0.3h_{max}$ downstream of the plane of the weir.

With good measuring techniques and flow conditions, estimated obtainable accuracy should be about ± 1.7 % to ± 3 %.

5.1.3.7 Standardised differential pressure devices

Discharge determination by pressure differential is based on installing a device creating a constricted cross section in the conduit and gauging the pressure difference generated by this constriction. Such devices are orifice plates, nozzles and venturi tubes.

The method of discharge measurement by differential pressure devices is the subject of ISO 5167[/14/] supplemented by ISO 2186[/11/], concerning pressure signal transmission.

These standards give all the necessary directions concerning the design and the setting of the primary element, the choice of the section of measurement, the value of the flow coefficient, the computation of discharge and its uncertainty. These standards apply only in the range of the pipe diameter D and Reynolds number R_{eD} specified in ISO 5167.

Whenever possible to satisfy the requirements of the ISO standards, it is unnecessary to calibrate the apparatus as the flow coefficients indicated in the standards may be used provided their resulting accuracy is considered sufficient. All data necessary to estimate the total uncertainty in discharge measurement are given in ISO 5167.

With good measuring techniques and flow conditions, obtainable accuracy is estimated to ± 1% to ± 1.5 % for orifice plate, nozzle and venturi tube.

5.1.3.8 Volumetric gauging method

The conventional volumetric gauging method is confined to low discharges, because of the limitation caused by the size of the tanks or reservoirs required. Therefore, it is unlikely to be applied to discharge measurement in the field.

Nevertheless, a variant of this method can be adopted for large scale discharge measurements[/9/]. It consists in determining the variation of the water volume stored in the headwater or tailwater pond on the basis of the variation of the water level. If necessary, provision shall be made for isolating the pond to ensure that there shall be no inflow to or no outflow from it during the measuring time.

Artificial ponds best suited for volumetric measurements are concrete basins with vertical walls. with increasing size the ponds are generally provided with inclined concrete walls. These ponds are particularly suitable for volumetric measurements if the slope of the walls remains constant over the whole measuring range. The shape of a basin and the slope of the walls should be considered carefully in the planning stage of the plant if the basin is to be used for volumetric measurements.

Approximate values of the uncertainty of the volume determination of concrete ponds with vertical walls should be $\pm$ 0.5 % to $\pm$ 0.8 %, and for concrete pond with sloping banks $\pm$ 0.7 % to $\pm$ 1.0 %.

5.1.3.9 Relative discharge measurement

Relative discharge measurement[/9/] can be done by measurement of the pressure difference between suitably taps on the scroll case of a turbine as shown on Fig. 5.11.

This is the Winter-Kennedy method and the discharge is usually well represented by

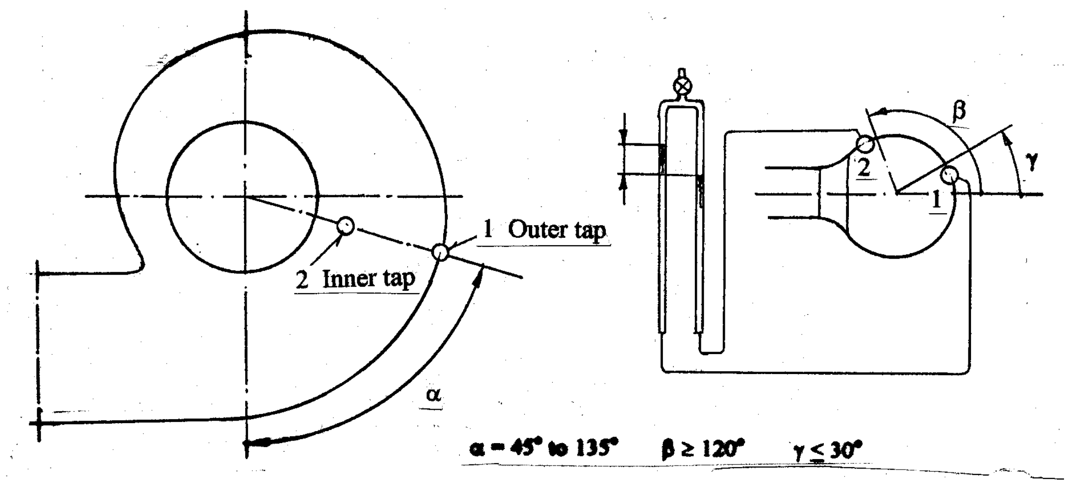

Fig. 5.12 Winter-Kennedy measurement

$$Q = kh^n \tag{5.19}$$

where h is the reading of a differential manometer connected between the taps
n is theoretically equal to 0.5

As a general rule this method is applicable to turbines only. In installations with a steel scroll case it requires taps[/9/] located in the same radial section of the scroll case. The outer tap 1 is located at the outer side of the scroll case. The inner tap 2 shall be located outside the stay vanes on a flow line passing midway between the two adjacent stay vanes. It is recommended that a second pair of taps be located in another radial section.

5.1.4 Thermodynamic measurement of flow losses

5.1.4.1 Measurement of power losses

The energy flow losses through a hydro turbine, is converted to heat energy in the water flow. Thus the temperature in the discharge increases through the passage. On this basis *the thermodynamic method*[/17/] for determination of turbine efficiency is established.

However, the relatively small temperature changes in the water flow cause applicability limits for the method. For example, a turbine with net head H_n = 427 m and an efficiency η = 90 %, the energy loss in the turbine corresponds to 10 % of H_n, which in this example means a temperature increase about 0.1 ^oC.

In practice, by lack of uniformity in measured values, limitations of measuring equipment and relatively high magnitude of corrective terms, the range of application of this method is therefore limited to heads above 100 meter.

Measuring equipment

The instrument for the application of the thermodynamic method consists of: elements for temperature measurements, calorimeter through which water is drawn off from the turbine, and precision manometer for pressure measurements.

An usual principle for measurements of temperatures is to connect two platina resistance thermometers S_1 and S_2 in a Wheatsones bridge together with two constant resisters R_1 and R_2 as schematically shown on Fig. 5.13.

The nominal resistance of the thermometers S_1 and S_2 is about 100 Ω. These thermometers have relatively high temperature sensitivity and linear temperature dependence.

Drawing off water through a calorimeter at the turbine inlet is in principle shown on Fig. 5.13. A hollow probe is mounted radially through a bore in the wall of the conduit. The drawn off flow is conducted through the probe via the regulating valve R into the chamber M for measurement of temperature and pressure.

Precision measurement of pressure is shown to the right on Fig.5.13. The pressure in chamber M is regulated to any desired level by the valve R, and this pressure may be measured through a branched off pipe from the chamber.

5.1.4.2 Efficiency and specific energies

The hydraulic efficiency of a turbine is, as expressed in Equation 2.22:

$$\eta_h = \frac{P_R}{P_n} = \frac{E_R}{E_n} \tag{5.20}$$

where E_R is the specific mechanical energy at the runner
E_n is the available specific hydraulic energy at the turbine inlet

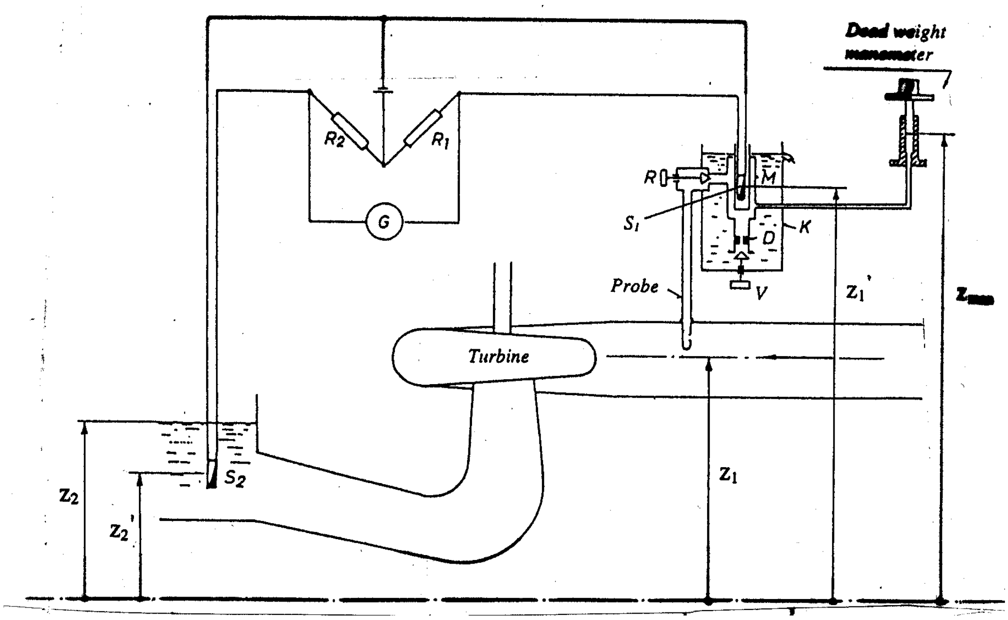

Fig. 5.13 Arrangement of thermodynamic measurements on a turbine /3/

These specific energies are now to be expressed from calculations based on the first and second laws of thermodynamics together with empirical data for some physical and thermodynamic properties of the fluid in use. The properties include compressibility, cubic expansion or Joule-Thomsen coefficient, and specific heat at constant pressure.

On this basis and with reference to Fig. 5.13:

The specific available hydraulic energy of the turbine may be determined by

$$E_n = \frac{p_{abs1} - p_{abs2}}{\rho} + \frac{c_1^2 - c_2^2}{2} + g(z_1 - z_2) \tag{5.21}$$

where
p_{abs1} is the absolute static pressure at the turbine inlet, level z_1
p_{abs2} is the absolute at turbine outlet, level z_2
c_1 is the average flow velocity at the turbine inlet, pos. 1
c_2 is the average flow velocity at the turbine outlet, pos. 2

g is the acceleration of gravity
z_1 is the level of the turbine inlet, pos. 1
z_2 is the level of the tail race water, pos. 2

The specific mechanical energy at the runner

$$E_R = a(p_{abs1} - p_{abs2}) + c_p(\theta_1 - \theta_2) + \frac{c_1^2 - c_2^2}{2} + g(z_1 - z_2) \tag{5.22}$$

where
a is the isothermal factor of water [m^3/kg]
c_p is the specific heat capacity of water [J/kg/K]
θ_1 is the temperature of the water at the turbine inlet [K (Kelvin degrees)]
θ_2 is the temperature of the tail race water, pos. 2 [K]

From first and second laws of thermodynamics and differentiation of the enthalpy h, is known:

$$dh = dq + \frac{dp}{\rho} = \theta ds + \frac{dp}{\rho}$$

or

$$dh = \theta\left(\frac{\partial s}{\partial \theta}\right)_p d\theta + \theta\left(\frac{\partial s}{\partial p}\right)_\theta dp + \frac{dp}{\rho} \tag{5.23}$$

where h is the enthalpy
q is the heat flow
s is the entropy
ρ is the density of the water

From the thermodynamics

$$\left(\frac{\partial s}{\partial p}\right)_\theta = -\left(\frac{\partial(1/\rho)}{\partial \theta}\right)_p$$

and the specific heat at constant pressure

$$c_p = \theta\left(\frac{\partial s}{\partial \theta}\right)_p \tag{5.24}$$

Therefore the differential of the enthalpy can be converted in

$$dh = c_p d\theta + \left[\frac{1}{\rho} - \theta\left(\frac{\partial(1/\rho)}{\partial \theta}\right)_p\right] dp$$

and the isothermal factor of water is then:

$$a = \frac{1}{\rho} - \theta\left(\frac{\partial(1/\rho)}{\partial \theta}\right)_p \tag{5.25}$$

In the International Standard IEC 41[/9/] tables are given of the properties of water for:

- the isothermal factor a [m^3/kg] with total range from
  $a = 1.0184 \cdot 10^{-3}$ at temperature 0 ^oC and absolute pressure p = 1 bar, to
  $a = 0.8790 \cdot 10^{-3}$ at temperature 40 ^oC and absolute pressure p = 150 bar
- the specific heat c_p [J/kg/K] with total range from
  c_p = 4207 at temperature 0 ^oC and absolute pressure p = 1 bar, to
  c_p = 4145 at temperature 40 ^oC and absolute pressure p = 150 bar

5.1.4.3 Measuring technique

The measurements to be carried out with the equipment described in Section 5.2.4.1, are to determine the quantities in the Equations (5.21) and (5.22), which means two measuring procedures for each point to be tested of the turbine efficiency.

To illustrate the measuring technique an ordinary arrangement of the thermodynamic measurements on a Francis turbine as shown on Fig. 5.13, is used.

Determination of the specific available hydraulic energy E_n

The probe is a kind of a pitot tube. With the opening directed against the flow velocity in the conduit, the total pressure that is the sum of the static pressure and the stagnation pressure in the conduit, can be measured. By closing the valve V downstream of the orifice D, the drawn off flow is shut off, and the total pressure $p_{abs.man} = p_{abs1} + c_1^2/2$ is measured by the dead weight manometer at the reference level z_{man}.

The pressure p_{abs2} is the barometric at the tail water level z_2. The velocity c_2 is a relatively small quantity and $c_2^2/2$ may be neglected as a first approximation. But after the net head $H_n = E_n/g$, the hydraulic efficiency η_h and the generator output P_G are determined, the velocity c_2 can be calculated as $c_2 = P_G/(\eta_G\eta_h A_2)$ where A_2 is the cross section area of the outlet of the suction pipe.

On the base of these measurements and observations of z_{man} and z_2 the specific available hydraulic energy is

$$E_n = \frac{p_{abs.man} - p_{abs2}}{\rho} - \frac{c_2^2}{2} + g(z_{man} - z_2)$$

and the net head $H_n = E_n/g$.

Determination of the specific mechanical energy E_R

The drawn off flow is exposed to heat exchange with the surroundings by the flow through the apparatus. Therefore the measured temperature has to be corrected for the corresponding temperature change. This is done by measuring the temperature and pressure for a series of different magnitudes of the drawn off flow by regulating the valve R.

On the base of these temperature and pressure observations the pressure and temperature corresponding to zero heat exchange can be determined. The total pressure in this case at the level z_1' of the temperature sensor S_1 is $p_{abs.z_1'} = p_{abs.man} + \rho g(z_{man} - z_1')$ where $p_{abs.man}$ include stagnation pressure $\rho c_1^2/2$, and the temperature is θ_1.

The temperature θ_2 is measured by sensor S_2 in the position of level z_2' in the outlet of the suction pipe. The corresponding pressure $p_{abs.z_2'}$ is the sum of the barometric pressure p_{abs2} at level z_2 and the difference of gravity $\rho g(z_2 - z_2')$.

The velocity c_2 is the same as in the determination of E_n.

The isothermal factor a and the specific heat capacity c_p are to be taken from tables (f.ex. IEC 41) and evaluated for the average pressure $(p_{abs.z_1'} + p_{abs.z_2'})/2$ and the average temperature $(\theta_1+\theta_2)/2$.

By these measurements and observations the specific mechanical energy becomes

$$E_R = a(p_{abs.z_1'} - p_{abs.z_2'}) + c_p(\theta_1 - \theta_2) - \frac{c_2^2}{2} + g(z_1' - z_2') =$$

$$= a\left[p_{abs.man} + \rho g(z_{man} - z_2)\right] + g(z'_1 - z'_2)(1 - a\rho) + c_p(\theta_1 - \theta_2) - \frac{c_2^2}{2}$$

and the corresponding head utilised by the runner is $H_R = E_R/g$.

The measuring technique as illustrated for the measurements of a Francis turbine, is valid also for the measurements of a Pelton turbine. However, the temperature sensor S_2 has to be positioned in the tail water downstream of the turbine runner outlet at a distance which is longer or equal to a minimum length defined in the Standard IEC 41. Moreover, the level of the tail water is usually lower than the average level of the nozzles, and this level difference has to be corrected for in the temperature measurements.

The practise of measurements described above, is not the only method the different practitioners of the thermodynamic method are applying. In addition a great many of them instead of evaluating the specific energy quantities, prefer to convert these quantities into heights. Further information about these details are given in the Standard IEC 41.

5.1.4.4 Corrections for leakage and friction

The hydraulic efficiency $\eta_h = E_R/E_n$ is determined on the basis of the measurements described in Subsection 5.1.4.2.

In addition to the energy losses by the flow through the turbine there are some other losses also to be taken into account. In practise that is mainly losses in the bearing, the leakage flow and its friction losses outside the runner shrouds. With certain adaptation and arrangement of measuring equipment these losses too are measured. The corresponding power is denoted P_L and the mechanical efficiency

$$\eta_m = 1 - \frac{P_L}{P_R} = 1 - \frac{P_L}{\rho Q E_R} \tag{5.26}$$

where the turbine discharge is

$$Q = \frac{P_G}{\eta_G \rho E_n}$$

Finally the turbine efficiency becomes

$$\eta = \eta_m \eta_h \tag{5.27}$$

With good measuring techniques and flow conditions, the efficiency of a turbine determined by the thermodynamic method, is in general obtained with an accuracy around $\pm$ 1.0% for the higher heads and $\pm$ 1.5 % for the lower heads.

5.1.5 Dynamic properties of the turbines

During comissioning, shaft vibrations and vibrations of the guide vanes, top and bottom cover and the draft tube may be recorded if the dynamic properties are regarded to be unfavourable. Noise level is often recorded, but a mutual agreement must be made to include noise level in the guarantee.

For shaft vibrations a new IEC code somewhat similar to the ISO norms for pumps, is under progress.

5.1.6 Cavitation behaviour of prototype

During commissioning cavitation may be observed by abnormal noise. In the future hydrophones seem to be a tool for indicating severe cavitation at an early stage.

Cavitation damage deeper than 0.5 mm is normally defined as eroded surface. The eroded volume is than calculated to be 0.5 x deepest point multiplied by the eroded surface deeper than 0.5 mm according to norms. However, for satisfactory operation no material pitting should occur and this is the goal for the owner and producer of a turbine.

5.1.7 Governor test – Rejection tests

Rejection of a turbine generator unit is actually a governor test[/15/]. But the turbine characteristic has also an influence on speed and pressure and this test is therefore normally combined with the turbine performance tests.

The governor system and turbine performance however, will always be tested in a shut down test of each unit normally at 25 %, 50 %, 75 % and 100 % load. If overload has been guaranteed, a rejection test must also be made at the guaranteed overload.

It should be emphasised that for a high head power plant with more than one unit, simultaneous part load rejection with all units normally gives higher pressure rise than full load rejection.

During the rejection test the maximum pressure should be recorded by fast pressure transducers with short connecting pipes in order to record the real pressure peaks with minimum damping.

Besides the maximum pressure peak, the maximum speed of the unit should be recorded.

During the shut down test also the minimum pressure in the draft tube should be recorded, and special attention should be paid to this if the draft tube is relatively long.

The surging of the water level in the draft tube surge shaft should be carefully observed because in some cases a quick unloading after a full rejection followed by a new repeated rejection may cause overflow and drowning of a cavern power station.

5.2 Model tests and scale effect of efficiency from model to prototype

Performance tests of prototype turbines by means of model tests may be used to prove guarantees given by the manufacturer. Model tests may also be used to compare models from several manufacturers. Such tests have to be carried out in neutral laboratories.

5.2.1 Laboratory qualifications

The laboratories qualified for performance tests of turbines by means of model tests, have technical data around the following values[/7/]:

Pump capacity:	H_{max} = 160 [m] and Q_{max} = 1.5 [m^3/sec]
Dynamometer:	P = 350 [kW] and n_{max} = 1500 [RPM]
Traceable calibration range:	H: 0 - 155 [m] and Q: 0.05 - 1.5 [m^3/sec]
Water reservoir:	V = 650 [m^3]

The flow system in which the model test turbines are installed, must have an upstream inflow and a downstream outflow system with dimensions and designs satisfying certain standard rules. The supply pump must be continuously adjustable in head and capacity within the calibration range, and the control system must be prepared to keep any operation point in this range constant within required limits.

The parameters to measure on the models are: pressure p and/or head H, discharge Q, torque T, angular velocity ω and density ρ of the water. The laboratory must be equipped with facilities for calibration and checking control of the metering devices to required accuracies for all these parameters. A device for metering and control of the air content in the water is also needed.

Normally the indications from the metering devices (transducers) of the respective parameters, are converted to electric signals which are transmitted to recording devices in a central control room. The recorded data may further be fed into a computer, which is programmed for evaluation of the resulting efficiency and other relevant data.

In addition to the facilities for calibration and measurements of the parameters of the model tests, some other instruments as a precision barometer, thermometer and a hygrometer, are necessary for control of the environmental conditions in the laboratory.

5.2.2 Model tests

Required size and surface roughness of the models

A major point by the model tests is the minimal size of a model, for which reliable test results must be obtained for the succeeding evaluation of the corresponding prototype turbine. This will generally depend on Reynolds number of the tests and the roughness of the surfaces being in contact with the flow. However, provided that the models are manufactured with hydraulic smooth surfaces, the Reynolds number creates the criterion for the sizing of the models.

In practise this means to establish a lower limit of the Reynolds number. The basis for the evaluation of this limit is the distribution of the laminar and turbulent flow layers on the runner vanes, which is representing the skin friction losses. In addition this distribution is also influencing the stability conditions and the flow direction out of propeller runners .

Tests have however shown that synonymous critical values of the Reynolds number may be established. In the Standard IEC 193[/10/] the values in the following table are adopted for the minimum Reynolds number $R_{e\ min}$ of the models of Kaplan/propeller, Francis and Pelton turbines.

The definitions of the Reynolds numbers are:

- for Francis and Kaplan/propeller turbines $R_e = D_s \frac{\sqrt{2gH}}{\upsilon}$
- for Pelton turbines $R_e = B \frac{\sqrt{2gH}}{\upsilon}$

where D_s is the diameter of the runner at the outlet of Francis and Kaplan turbines
B is the width of a Pelton bucket
H is the head of the model turbine
ν is the kinematic viscosity of the water

	Kaplan model turbine	Francis model turbine	Pelton model turbine
$R_{e\ min}$	$2 \cdot 10^6$	$2.5 \cdot 10^6$	$3.5 \cdot 10^6$
$D_{s\ min}$	250 mm	250 mm	B_{min} 80 mm
H_{min}	1 m	2 m	40 m

Besides the lower limits of the model size, it is required that all hydraulic details from inlet to outlet of the model turbine are geometric similar to the prototype. This means inclusion of the scroll case and the suction pipe in the similarity requirements as well. Likewise such components as bends, branch pipes and valves at the turbine inlet may be included in these requirements.

The flow leading surfaces of the model shall have the same relative smoothness as the prototype.

Testing procedure

The testing procedure for determination of performance characteristics of the model turbine[/8/], is reported in Subsection 3.1.4.

Accuracy of model tests

The probable errors of the measurements of an operating point on the turbine[/8/], may be estimated statistically by introducing the error of each of the measured quantities. These errors may be summed up, as an example in the following way

$$\Delta\eta = \sqrt{\Delta\eta_H^2 + \Delta\eta_Q^2 + \Delta\eta_T^2 + \Delta\eta_\omega^2 + \Delta\eta_\rho^2} \tag{5.28}$$

where $\Delta\eta_H$ is the error of the efficiency caused by the error of the head
$\Delta\eta_Q$ is the error of the efficiency caused by the error of the discharge
$\Delta\eta_T$ is the error of the efficiency caused by the error of the torque
$\Delta\eta_\omega$ is the error of the efficiency caused by the error of the angular speed
$\Delta\eta_\rho$ is the error of the efficiency caused by the error of the density of the water

It may be required in model tests that the probable error of the efficiency shall be $\Delta\eta \leq 0.25$ %.

In practise it is difficult however, even with the best measuring methods, to obtain lower errors of the measured quantities than the following values:

$$\Delta\eta_H = \pm\, 0.1\ \%,\quad \Delta\eta_Q = \pm\, 0.2\ \%,\quad \Delta\eta_T = \pm\, 0.1\ \%,\quad \Delta\eta_\omega = \pm\, 0.05\ \%,\quad \Delta\eta_\rho = \pm\, 0.05\ \%$$

With these values the probable error of the efficiency becomes $\Delta\eta = 0.255$ %.

Cavitation tests on model

Cavitation, suction head and similarity relations including cavitation are dealt with in Subsection 3.1.5. Methods for testing and determination of cavitation limits on models are indicated there as well.

According to these outlines the Thoma parameter for the model:

$$\sigma = \left(\frac{\mathrm{NPSH}}{\mathrm{H}}\right)_{\mathrm{model}} \tag{5.29}$$

can be calculated from the measured pressure referred to the reference height according to Standard IEC 41 for NPSH. On the base of the measured σ value the critical setting H_s of the prototype can be calculated.

However, the content of nucleis and air in the water has a great influence on the value of σ measured in a model. The lowest value of σ will be obtained in the laboratory by degassed water.

Further the content of silt in the water at site for the prototype also has some influence.

For safety reasons it has been recommended to use degassed water and inject nucleis (micro bubbles of air) until the highest σ value is obtained. However, so far the experts on cavitation have not agreed upon a standard which includes nuclei injection, and the majority of laboratories have no nuclei injection systems. So far natural water saturated with air has been regarded to be the best alternative by many laboratories for model tests, because degassed water gives a lower value of σ and a lesser margin for the prototype if plant σ is based on model test.

Runaway test

It is a question of safety for the dimensioning and design of the rotating parts of the turbine and generator unit to know the runaway speed.

In the discussions of the performance diagrams of Pelton, Francis and Kaplan turbines in the Sections 3.2, 3.3 and 3.4 respectively, the runaway speed of these turbines has been mentioned. With reference to these diagrams, the runaway speed differs essentially from one type of turbines to another.

On the base of these facts the runaway test with zero torque should be carried out during the normal model performance tests. It should be noted however, that the runaway speed normally increases with low values of the Thoma parameter σ. For this reason the runaway tests should be run at the plant σ. Runaway test on prototypes should be avoided at site due to the consequences if the generator cannot withstand the centrifugal forces.

5.2.3 Scale effect on efficiency from model to prototype

The efficiency of a prototype will normally be higher than for the model. The reason is less friction due to higher Reynolds number $R_e = U_2D_2/\nu$ where U_2 and D_2 is the circumferential speed and the diameter respectively of the runner at outlet and ν is the water viscosity.

The formula for upscaling the efficiency from the model to the prototype for Francis turbines according to IEC code[/10/] yields:

$$\Delta\eta = (1-\eta_m)V\left(1-\left(\frac{R_{em}}{R_{ep}}\right)^{\alpha}\right) \tag{5.30}$$

where $\Delta\eta$ is the difference in efficiency of the prototype and the model
η_m is the efficiency of the model turbine
R_{em} is the Reynolds number of the model
R_{ep} is the Reynolds number of the prototype
V is the scaleable part of the losses. $V \approx 0.7$ according to IEC code[/10/]. However, it is proven that $V = f({}^*\underline{\Omega})$
α is exponent, estimated $\alpha = 0.16$

For Kaplan turbines a similar formula as for Francis turbines is established, but with a different value of V. For Pelton turbines a minor scale up effect at part load has been proven. At best efficiency point the increase in efficiency is normally very small and at full load a decreased efficiency is normally observed for multinozzle turbines.

A scale effect of the Thoma cavitation parameter σ has not yet been proved. However, the σ value for the prototype is regarded to be the same as for the model.

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CHAPTER 6

Pelton Turbines

Introduction

The specified data of water flow rate, head and rotational speed determine the Pelton turbine designs.The number of jets depends on the speed number and stepping up values of the speed number means to step up in number of jets. For a certain plant that means often a matter of choise between numbers of jets for the turbine. If for example the design should be either four or five jets of the turbine, then the cost of the generator for a five jet turbine will be correspondingly lower than the alternative of a four jet turbine.

However, it is a serious demand that the jets enter the buckets so far from each other that no mutual disturbances can occur. In practise this means that six jets represent the upper limit of their number, and an evaluation of the speed number according to Equation (3.8) will show that speed numbers of Pelton turbines ranges below $^{*}\underline{\Omega} = 0.2$.

Pelton turbines can be arranged either by a horizontal or a vertical shaft. In general a horizontal arrangement is found only in the medium and smaller sized turbines with one or two jets. Some horizontal Pelton turbines have however, been built with four jets as well.

6.1 Horizontal Pelton turbine arrangement

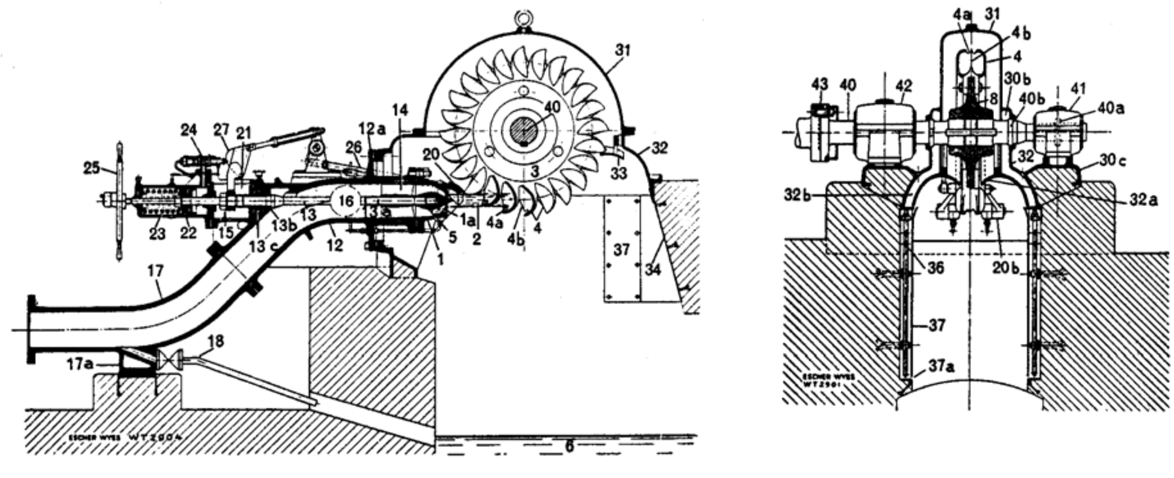


Fig.6.1 Horizontal Pelton turbine with one jet (To the left: Radiell section; to the right: Longitudinal section) (Courtesy of Escher Wyss)

Main details:

1. *Nozzle*
2. *Jet*
3. *Runner*
4. *Runner bucket*
5. *Needle head*
6. *Tail water level*
8. *Runner disc*
12. *Inlet bend*
13. *Needle rod*
17. *Lower bend*
18. *Emptying pipe*
20. *Deflector*
21. *Feed back rod*
22. *Servomotor piston for needle movement*
23. *Closing spring for needle*
24. *Solenoid valve for the control of needle servomotor*
25. *Steering wheel for needle movement*

31 and 32. Turbine housing

32a. Screen for guiding water from runner outlet to tail water

32b. Drain canals for for guiding spray water away from shaft

34. Steel lining

40. Turbine shaft

40a. Cam for axial bearing

42. The outer turbine bearing

43. Coupling

The flow passes through the inlet bend (12), the nozzle outlet (1) where it flows out as a compact jet through atmospheric air into the wheel buckets (4). From the outlet of the buckets the water falls through the pit down into the tail water canal (6).

Further comments to some of the details are given in Section 6.3.

Fig. 6.2 shows another example of a horizontal Pelton turbine with two runners and two jets on each runner.

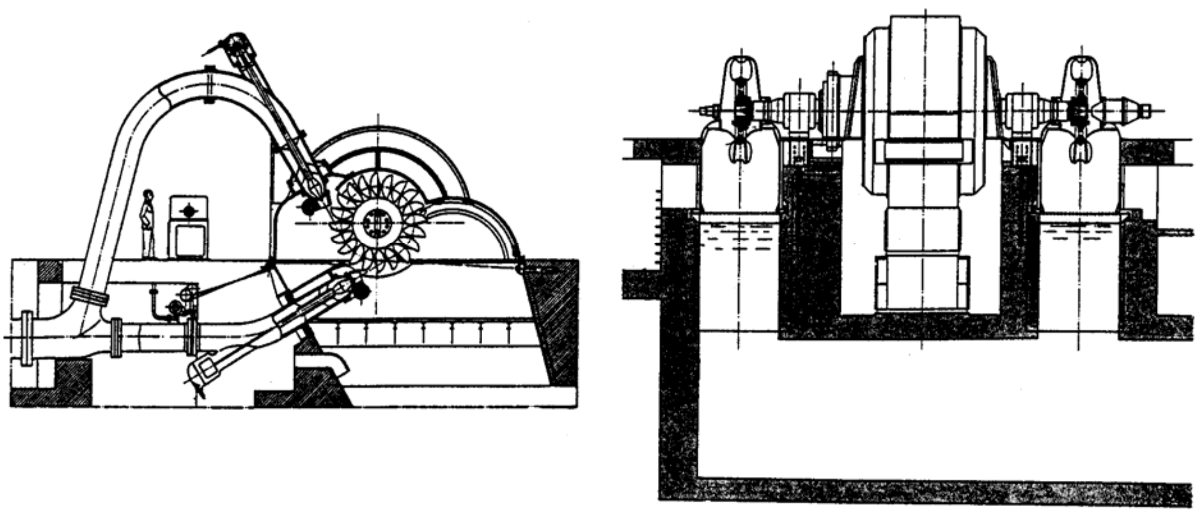


Fig. 6.2 Cross sections through a horizontal Pelton turbine with two runners and two jets to each runner. (Courtesy of Atelier des Charmilles)

6.2 Vertical Pelton turbine arrangement

Large Pelton turbines with many jets are normally arranged with vertical shaft. The jets are symmetrically distributed around the runner to balance the jet forces. The Fig. 6.2 and Fig. 6.3 show as an example of the vertical and horizontal section respectively of the arrangement of a six jet vertical Pelton turbine.

1. Pipes for efficiency tests
2. Distributor pipe
3. Deflector mechanism
4. Turbine shaft
5. Deflector servomotor
6. Wheel pit cover
7. Guide bearing
8. Turbine housing
9. Main injector with needle servomotor
10. Runner
11. Runner cart rails
12. Runner cart
13. Inspection platform

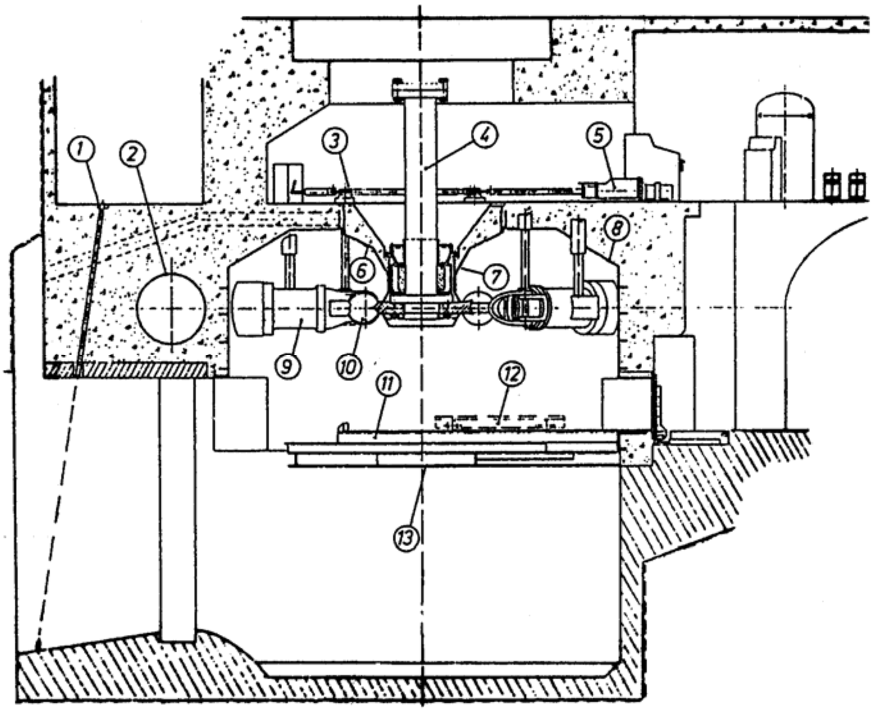


Fig. 6.3a Vertical multinozzle Pelton turbine, vertical section (Courtesy of Kværner Brug)

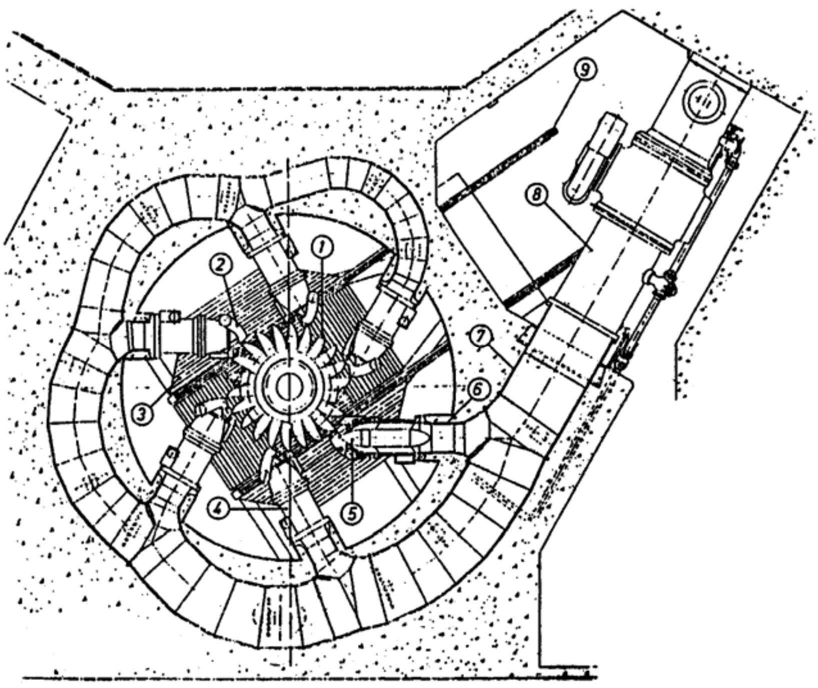


Fig. 6.3b Vertical multinozzle Pelton turbine, horizontal section (Courtesy of Kværner Brug)

Numbered details in Fig. 6.3b:

1. *Runner*
2. *Deflector*
3. *Inspection platform*
4. *Main injector with needle servomotor*
5. *Needle*
6. *Bifurcation*
7. *Distributor*
8. *Expansion/dismantling joint*
9. *Runner cart rail*

6.3Main components and their functions

The examples of Pelton turbines shown in the Figures 6.1, 6.2, 6.3a 6.3b, indicate that the turbine constructions consist of a rather great number of components. Naturally some of these do not exist in every manufactured Pelton turbine. This matter of fact imply that the constructions appear different from one manufacturer to the other or from a small to a bigger size of the turbine.

For these reasons our considerations will mainly according to ref. /*1*/, be dedicated to main components which are vital in all turbines. Of these components further attention will be paid to the folloving parts referred to Figs.6.1, 6.3a and 6.3b:

- runner
- shaft and bearings
- oil reservoir
- guide bearing
- bend and distributor
- straight flow injector
- deflector
- turbine housing

6.3.1 Runner

The Pelton runners may be designed either for casting of the disc and buckets in one piece, i.e. monocast, or the disc and each of the buckets are casted in separate pieces. The method first mentioned is preferred and common for the Pelton turbines in modern power plants where the the turbine units are of the high power and bigger sizes. Fig. 6.4 shows a photograph of a runner of monocast type, and Fig. 6.5 shows a runner with buckets having ears through which they are bolted to the disc.

Some details of such buckets on Fig. 6.5 show how some manufacturers in addition to bolts (9) also use pins (10) or wedges (11) between the buckets. The reason for this is that, while the runner is rotating, the buckets run in and out of the jets with a frequency according to the rotating speed and the number of jets. During this process they are exposed to shocks occurring with the same frequency. In the same way the bolts (9) are strained. To take care of these bolts conical pins (10) may be fitted into corresponding holes and driven in to a certain degree of prestressing between the buckets and the disc. If such pins are too weak, wedges (11) can be used as shown to the right on Fig. 6.5.

The material of the runner and buckets are chosen according to the head, stresses, content of sand in the water and other strain factors, and may be cast iron, cast steel or casted of high quality alloy steel. For the large high head turbines the main strain factors are cavitation, sand erosion and cycle fatigue. Runners working under such conditions are normally casted of 13%Cr4%Ni alloy steel. In addition to the strength qualities the material must be adequately weldable.

Fig. 6.4 A monocast Pelton runner

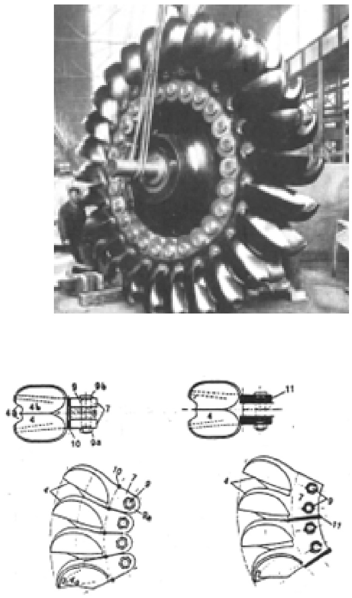


Fig. 6.5 Pelton runner with bolted buckets

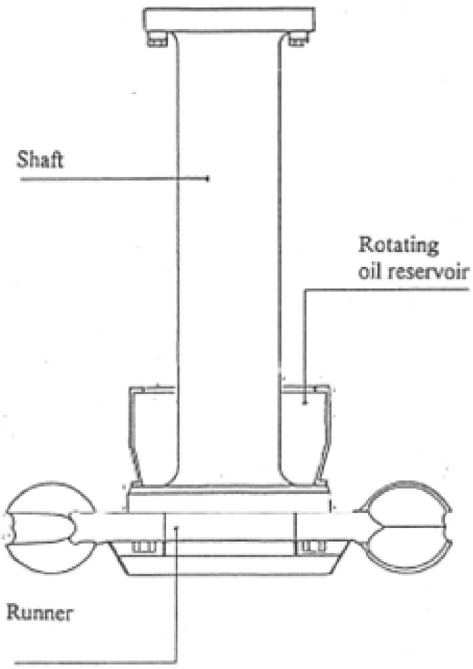


Fig. 6.6 Rotating parts of a vertical Pelton turbine /1/

The shape of the buckets is decisive for the efficiency of the turbines. For keeping and improving high efficiency characteristics great efforts of theoretical analysis and model tests follow the runner design. Limitations however, in these works are that bucket shape always will be a compromise between a hydraulically ideal and a structural optimum design.

The surfaces, over which the water jet flows are milled, ground and polished to the correct shape. The correctness is checked with templates. Surfaces close to jet flow out of the downstream neighbour bucket, and places where fatigue cracks may occur, are ground and polished as well.

The runner disc is fastened to the shaft by bolts and nuts[/1/] as shown on Fig. 6.6. The bolts are screwn in threaded holes in the shaft flange and protrude through holes in the disc and have tightening nuts screwn on the other end. This connection establishes a pure frictional joint by a corresponding prestressing of the bolts by means of heat or a special oil hydraulic tool. This type of joint enables a simple dismantling and replacement of the runner. A shield to reduce windage losses caused by air and droplet circulation protects the nuts.

6.2.2 The turbine shaft

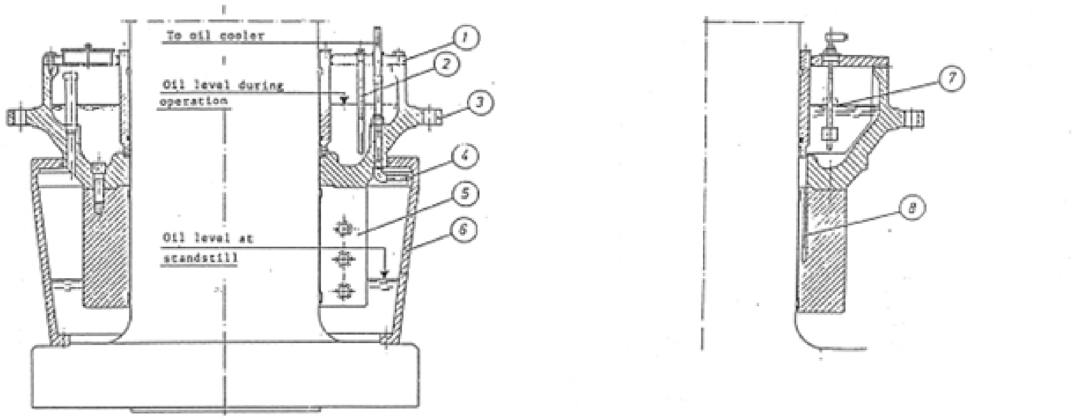
The turbine shaft of vertical Pelton turbines is made of forged Siemens-Martin steel with an integral flange at both ends as shown on Fig. 6.6. A hole is drilled centrally through the whole length of the shaft. The surface of the shaft is machined in a lathe to a final appropriate tolerance and even surface.

An oil reservoir is a rotating member bolted to the shaft flange[1] as shown on Fig. 6.6.

6.2.3 Turbine radial bearing

The details of a turbine bearing shown on fig. 6.7, is a radial guide bearing[1]. The bearing house (3) consists of two halves bolted together and mounted to a flange on the runner pit cover.

Two semicircular segments bolted together are



1. Bearing cover
2. Oil temperature sensorl
3. Bearing house.
4. Scoop
5. Bearing shell
6. Lower rotating oil reservoir
7. Level switch
8. Temperature sensor, bearing shell

Fig. 6.7 Turbine guide bearing (Courtesy of Kværner Brug)

forming the bearing shell (5) as a rigid cylinder, which is mounted to the lower side of the bearing house (3). These segments are provided with four fixed babbit bearing pads. They ensure a proper centring of the turbine shaft. Oil pockets between the four fixed bearing pads create the entrance of cooled lubricating oil to the bearing pads. At standstill the total oil quantity is staying in the rotating reservoir (6).

The rotating oil reservoir encloses the bearing shell. The oil rotates together with the reservoir as a layer on the wall. A stationary scoop (4) is installed with one end against the rotational speed of the oil in reservoir (6). The scoop is fastened to the bearing house (3) with a pipe connected to an external oil cooler. From the cooler a pipe returns to the upper oil reservoir of (3). The stagnation velocity pressure at the scoop opening causes an oil flow from the lower reservoir through the oil cooler to the upper reservoir.

To keep a certain oil quantity in the rotating reservoir the scoop is set at a predetermined distance from the wall. For large turbines however, two scoops are normally installed, and one of these with a larger distance from the wall than the other. This is advantageous under start while the oil layer in the rotating reservoir is thick enough for both scoops to deliver oil from the lower to the upper reservoir. Thus the time needed to establish the stationary oil circulation is decreased.

During the stationary circulation the oil flow exists as a continuous thin film along the wall of the rotating reservoir. An air stream outside the wall of the rotating reservoir mainly cools the oil layer. The slightly conical shape of the wall of the rotating reservoir is ideal for this cooling. Air-cooling may however be insufficient especially if the shaft rotational speed relative to the guide bearings is in the high-speed range. Therefore an external cooler is normally provided.

The cooled oil flows downwards along the shaft as it is distributed to the four pockets between the bearing pads. The oil film follows the shaft rotation, enters the bearing segments and establishes the load carrying coolant film in the bearing pads.
The bearing shell (5) is provided with a throttling edge at the lower end. The throttling edge controls the oil circulation in the bearing and ensures that the oil pockets in the shell are always filled with oil during operation.

6.2.4 Bend and distributor

The distributor pipe (7) for a multijet Pelton turbine[/1/] is shown on Fig. 6.3b. From this a bifurcation (6) is made to each of the injectors (4). The distributor is designed to provoke an acceleration of the water flow through the bifurcation towards each of the main injectors. This design is advantages by contributing to keep a uniform velocity profile of the flow.

The distributor pipe is a welded plate design manufactured completely from fine grain high tensile steel. The maximum main stress for this material must be limited to about 200 MPa. The bifurcation is reinforced with external and internal ribs. Because of the steel quality, welding of the distributor must be performed with a specified heat treatment.

The distributor is completely embedded in concrete when installed in the power house. However, to transfer the large axial forces to the power house an extension must be welded to the inlet flange. This is done for reducing the specific pressure on the concrete to avoid crushing and cracking.

The distributor is provided with a manhole with cover as access for internal inspection and maintenance. A manually operated drain valve is installed underneath close to the inlet flange of the distributor. This valve should be operated only when the main spherical valve upstream of the distributor is closed.

The distributor is joined to the main spherical valve via a joint, which is installed for dismantling purposes. This is furnished with a telescope flange connection to the distributor entrance. The main injectors are joined to the bifurcation by means of rigid flange connections.

An automatic relief valve is normally installed on the top of either the distributor entrance section or the dismantling joint. This valve closes automatically when most of the air in the distributor is let out during water filling, and remains closed as long as the distributor is pressurised.

For emergency stop a water jet braking system is provided to obtain a fast reduction of the rotational speed of the runner after the nozzles are closed. This system consists of one automatically operated needle valve connected to one or two brake jet nozzles, which are fed via pipes directly from the penstock. The braking valve is controlled by a solenoid valve operated by an emergency control system actuated by the water pressure from the penstock.

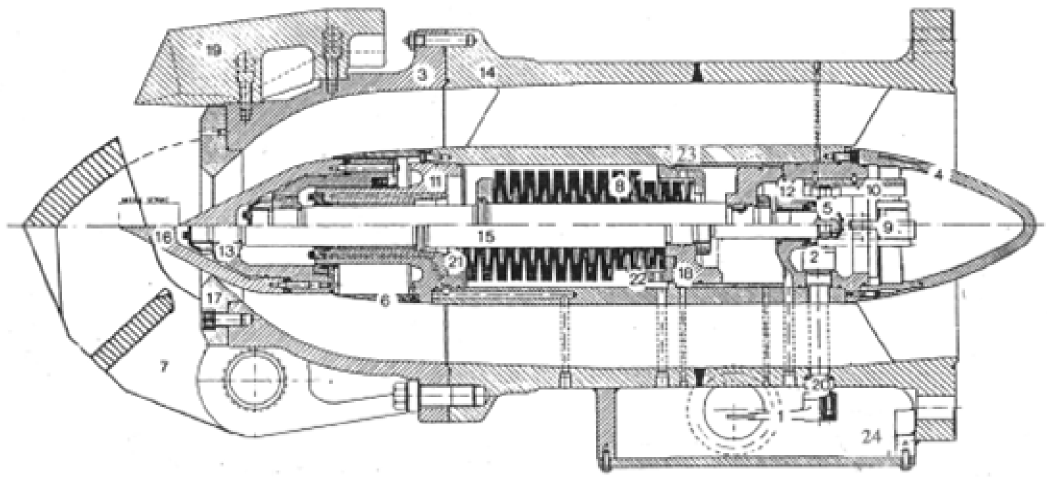
For emptying the penstock upstream of the spherical valve a draining system is provided. This consists of five parts: a pipe installed just upstream of the spherical valve, a manually operated gate valve with a bypass valve, one manually operated needle valve in series downstream of the gate valve and an outflow pipe. The outflow pipe has its outlet above the water surface in the turbine pit to prevent cavitation during drainage of the penstock.

6.2.5 Straight flow injector

The straight flow injector with needle servomotor is shown on Fig. 6.8.

The injector[/1/] consists of three outer parts bolted together, the main body (14), the beak (3) and the nozzle ring (17). The beak is provided with two external brackets with self-lubricating bushings for the deflector (7) support.

Inside the injector two fins to the main body fix the inner cylindrical body (23). The inner body contains: the needle, servomotor with needle rod (15), disc spring column (8) and feedback mechanism.



1. Arm
2. Arm with split boss
3. Beak
4. Cap
5. Carrier
6. Cover
7. Deflector
8. Disk spring
9. Elbow lever
10. Guide
11. Guiding piece
12. Lid
13. Needle head
14. Main injector
15. Piston rod
16. Needle tip
17. Nozzle
18. Piston
19. Shield for beak
20. Spindle
21. Spring plate
22. Spring retainer
23. Inner injector body
24. Oil box

Fig. 6.8 Straight flow injector (Courtesy of Kværner Brug)

The needle consists of two parts, the head (13) and the tip (16). The guidance for the needle is the guidance piece (11). A U-seal ring in the needle head seals against the water pressure outside the guiding piece. The seal ring is protected against sand by a synthetic rubber scraper ring. Another U-seal ring in the guidance piece seals against the oil pressure in the servomotor.

The needle servomotor is double acting and operated with oil pressure from the governor oil system through a control valve. Disc spring elements and the water pressure from the penstock balance the needle. The disc spring column (8) will function satisfactorily also when one of the discs is broken.

With this spring design the servomotor will work even though the oil pressure should fall to about 25% of the normal level. If the oil pressure should be decreasing towards zero the water pressure will move the needle servomotor towards closed position and the turbine brought to stop if the generator is disconnected from the grid.

When the servomotor is out of operation and no pressure either from oil or water exists, the needle will be in the middle position and the spring discs relieved.

The governor feedback mechanism is located inside the upstream cap (4). The feedback shaft (20) is passing through one of the structure ribs into the external oil box (24). At the end of the shaft the arm (1) is fastened and further connected via a mechanical linkage system to a control cubicle on the turbine floor.

During operation with load alteration and corresponding regulation the feedback system will move the control valve in the control cubicle back to neutral position when the needle has reached correct position.

Due to the high exposure of sand erosion and cavitation the nozzle ring (17), the needle head (13), the guiding piece (11) and piston rod (15) with the nut are made of corrosion resistant steel normally 13% Cr 4% Ni. The beak is also lined with stainless steel.

The piston (18) is made of cast iron. The servomotor cylinder is lined inside with a cast iron bushing which cover the stroke length of the piston. On the upstream side the piston rod is supported in the servomotor cover with a bushing and a U-seal ring against the oil pressure.

The chamber bounded by the piston rod (15), the needle head (13) and the end of the guiding piece (11) has a volume varying with the piston movement. This chamber is vented to the atmosphere through a hole drilled along the guiding piece (11) and corresponding to a drilled hole in the wall of the inner injector body (23) and finally through the structure rib.

A replaceable cast steel shield (19) mounted on the beak protects the jet against exit water from the runner during normal operation.

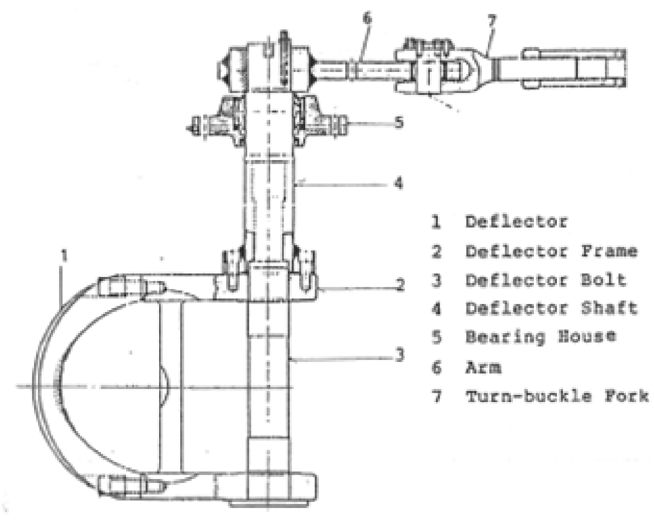


Fig. 6.9 Deflector mechanism (Courtesy of Kværner Brug)

6.2.6 Deflector mechanism

The deflector[1] has the function to bend the jet away from the runner at load rejections to avoid too high speed increase. Moreover it protects the jet against exit water spray from the runner.

The deflector mechanism is shown on Fig. 6.9. The deflector arc (1) is bolted to the deflector support structure frame (2). The deflector support frame is connected to the deflector shaft (4). The deflector pivot (3) is supported in the beak on water lubricated bearings. The deflector shaft (4) is supported in a self-lubricating spherical bearing located in the bearing housing (5) on the top plate floor. A seal ring around the deflector shaft bearing housing prevents water and moisture from penetrating into the bearing.

The deflector arc (1) is made of stainless corrosion resistant cast steel 13% Cr 4% Ni.

The governor deflector servomotor controls the deflectors through rods, turnbuckle forks and lever (6) which is fastened to the deflector shaft (4). In addition a dead band-eliminating servomotor is mounted in the deflector mechanism to prevent play in the governor lever system.

The control valve of the needle servomotors is joined by a link connection to feed forward mechanism connected to the deflector servomotor.

6.2.7 Turbine housing

The housing of a vertical Pelton turbine[/1/] is shown in Fig. 6.3a, pos. (8). This consists of an upper turbine housing and a lower pit lining. Both parts are made of steel plates welded together at site during construction. Moreover, cylindrical connections for the distributor inlet flanges are welded to the pit lining.

The pit lining is cylindrical and the upper turbine housing is externally reinforced by ribs and anchor bolts. The entire housing is embedded in reinforced concrete. Together these parts form a rigid unit with passages for needle servomotor piping and feedback mechanisms and the deflector shafts through the pit cover up to the turbine floor.

The wheel pit cover pos. (6) on Fig. 6.3a is conical. A plane top plate is provided with a flange for support of the turbine guide bearing, pos. (7). The shape of the wetted side of the wheel pit cover is important for leading the exit water effectively away from the runner.

The wheel pit cover is filled with concrete except for the canals of embedded pipes leading air from the tail race tunnel to the runner disc. The wheel pit cover is welded to the upper and lower plate on the turbine housing and forms a rigid support for the turbine guide bearing.

An inspection platform below the runner, pos. (13) on Fig 6.3, is designed to carry the weight of the runner and the main injectors during assembly and dismantling. Transport rails are mounted on the inspection platform, and a runner cart is provided for the transport of heavy parts in and out of the turbine pit.6.3 Condition control.

6.3 Condition control

6.3.1 Turbine guide bearing

The first oil change should be done after 3 - 6 months of operation[/1/]. The later oil changes have to be done as required by evaluating oil sample tests.

To empty the bearing for oil it has to be done at standstill by pumping through the oil level pipe in the bearing housing.

If babbit metal particles are found in oil samples, the bearing should immediately be dismantled for inspection.

6.3.2 Runner

The runner should be regularly inspected[/1/] to record possible damages from foreign objects in the water. The time interval between each inspection is dependent on sand content in the water.

The runner inspection is done visually by means of magnaflux and/or dyes penetrant. Particular attention should be paid to the area between the buckets.

If minor cracks or defects have been formed, these should be removed by grinding and polishing according to advises from the manufacturer.

The special shape of the runner buckets makes it difficult to detect material defects just below the surface. These defects may penetrate up to the surface during the first operation time period. To prevent an extensive crack propagation they must be rectified as soon as possible.

6.3.3 Main injector with needle servo motor

The needle tip and the nozzle should be inspected[/1/] with respect to cavitation damage and damage from foreign objects. If the water contains fine silt or sand, the needle tips may loose their original shape.

It is of great importance that the nozzle is inspected from the inside.

The needle servomotor should be run to neutral mid position and the oil pressure shut off for safety reasons. The leakage indicators should be regularly inspected.

6.3.4 Seal ring in deflector bearing

Leakage in this seal does not require immediate replacement of the seal ring, but replacement should be done as soon as possible to avoid bearing corrosion. The seal ring must be provided with a spring of stainless material.

6.3.5 Filter

Filter for breaking system control water supply should be checked and cleaned if necessary. It may be cleaned after closing of the valves for this water supply system from the penstock.

6.4 Monitoring instruments

The turbines are normally equipped with the following instruments:

- A contact manometer for reading of penstock pressure just upstream of the spherical valve. The manometer transmits signals either to the control room or to the operation control centre if the pressure exceeds or falls below certain pre-set values.
- A manometer for reading the pressure in the distributor.
- A contact thermometer for measuring the temperature of the turbine bearing. The reading instrument is mounted on the wall close to the governor control desk. The temperature sensor is located in one of the pads of babbit metal.

  The thermometer is provided with a contact that gives an alarm signal to the control room at a temperature which is normally 5 ^{o}C above the highest stable set temperature of the bearing.

If the temperature should rise a further 3 ^{o}C, another contact will give automatic emergency shut down. The stable set temperature will vary from one bearing to another. By symmetric operation (i.e. balanced jet forces on the runner) the temperature will however, be approximately 75 ^{o}C. By unsymmetrical operation with a reduced number of jets in action, the unbalanced jet force must be carried by the turbine bearing. In this case an increase in the temperature of 5 - 7 ^{o}C will occur. Another sensor for direct transmission of the temperature reading to the control room is normally also installed in one of the bearing pads.

- A resistance thermometer is mounted in the upper oil reservoir with remote display in the control room for reading of the bearing oil temperature.
- Two oil level switches are installed to give warning signals at high and low oil levels in the bearing. Low oil level may be caused by oil leakage in the oil reservoir, or defects in the oil scoops that may lead to spraying oil so that an oil evaporation occurs.

  Cooling water leakage into the bearing normally causes a too high oil level. In this case an overflow may occur with oil pollution in the environment. This is also a warning against filling too much oil in the bearing.

6.5 Assembly and dismantling

The assembly operation is of great importance to the completed turbine quality. At first this has to do with the reliability of operation, the further maintenance and the hydraulic efficiency. Furthermore the progress of the assembly should be carefully planned, as it is essential for ensuring commissioning at the right time. The civil works, the turbine and generator assembly is often directly interdependent and must therefore be carefully co-ordinated. The use of the crane in the machine hall as well must be co-ordinated between the parties involved.

The turbine must in general be ready for embedding in concrete prior to the commencement of the formwork. Moreover the generator assembly cannot start before the turbine shaft alignment is completed. This is again dependent on the completed embedding of the turbine and the completion of cleaning operations.

The great advantage of the Pelton turbine compared with the Francis turbine is its simple maintenance and dismantling. The dismantling is almost without exception performed in the opposite sequence of assembly.

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CHAPTER 7

Francis Turbines

Introduction

In Chapter 3 the hydro turbines are classified by speed numbers, and the Francis turbines are in the range $0.2 < {}^*\underline{\Omega} < 1.5$. This wide range imply that the hydraulic design of the runner in these turbines differ rather much from the lowest to the highest speed numbers.

In general the Francis turbines have a guide vane cascade encompassing the whole circumference of the runner. Adjustable vanes in the cascade create the canals which are equal in shape and size, for regulation of the discharge and flow direction before entering the runner. The water flow fills up all canals completely in the guide vane cascade and the runner respectively. Therefore the water is under pressure when it enters the runner.

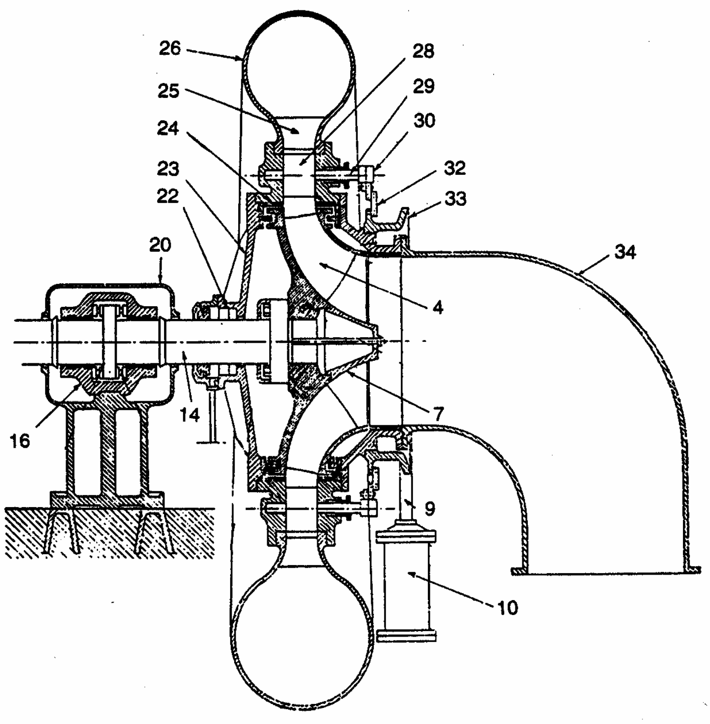


Fig. 7.1 Horizontal Francis Turbine

The Francis turbines may be divided in two groups, the one group with horizontal and the other with vertical shaft. In practise it is normal that turbines with comparatively small dimensions are arranged with horizontal shaft, while larger turbines have vertical shaft. The vertical arrangement is normally used also for small dimensions if the tail race water level is above the turbine centre.

7.1 Horizontal Francis turbine

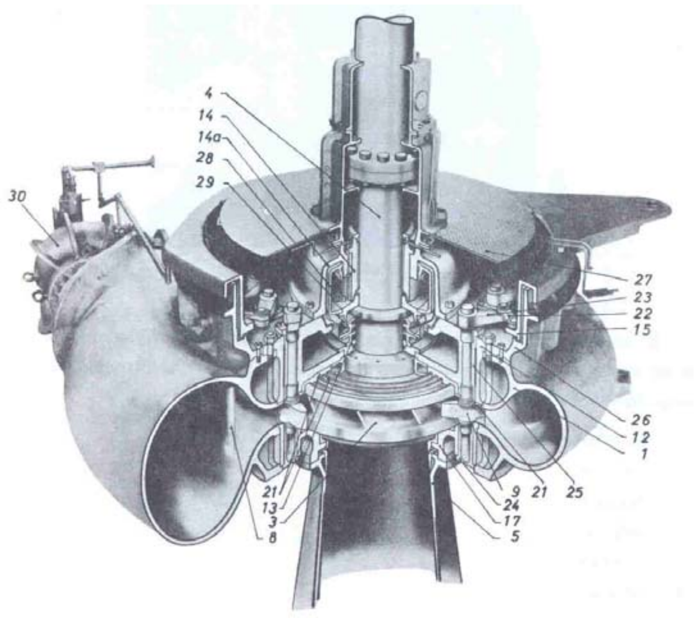
A design example of a horizontal Francis turbine is shown in Fig. 7.1, which is an axial section through the turbine. The water from the supply penstock flows through the scroll casing (26), the guide vane cascade (28), the runner (4), the draft tube (34) and into the tail race canal. To obtain an even, quite undisturbed water flow without thrust losses through the turbine, it is of great importance that any sharp edges or any sharp bends in the flow path should never exist.

The numbered details on Fig. 7.1 are:

4. Runner
7. The runner cone
9. Servomotor rod
10. Servomotor
14. Turbine shaft
16. Bearing pad
20. Bearing cover
22. Shaft sealing box
23. Turbine cover
24. Runner seal ring
25. Stay vane
26. Scroll case
28. Guide vane
29. Guide vane stem
30. Guide vane lever
32. Link
33. Regulating ring
34. Draft tube

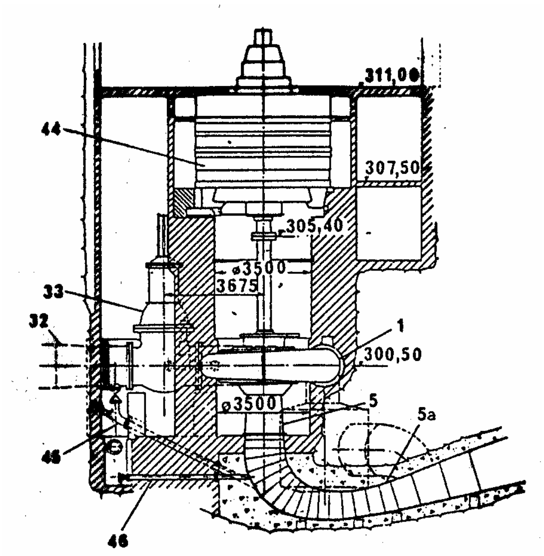
7.2 Vertical Francis turbine

An illustration of a vertical Francis turbine is shown in Fig. 7.2. This figure represents a



1. The scroll casing
3. Runner
4. Shaft
5. Draft tube cone
8. Stay vanes
9. Guide vanes
12. Upper cover
13. Sealing box
14. Guide bearing
14a. Bracket for the bearing(14)
15. Regulating ring
17. Lower cover
21. Replaceable wear and labyrinth rings
22. Link
23. Lever
24. Lower bearing for guide vane
25. Upper bearing for guide vane
26. Bearing for the regulating ring
27. Floor
28. Rotating oil cylinder
29. Oil scoop fastened to (14a) and (14) with the opening against the rotating oil in rotating oil cylinder (28)

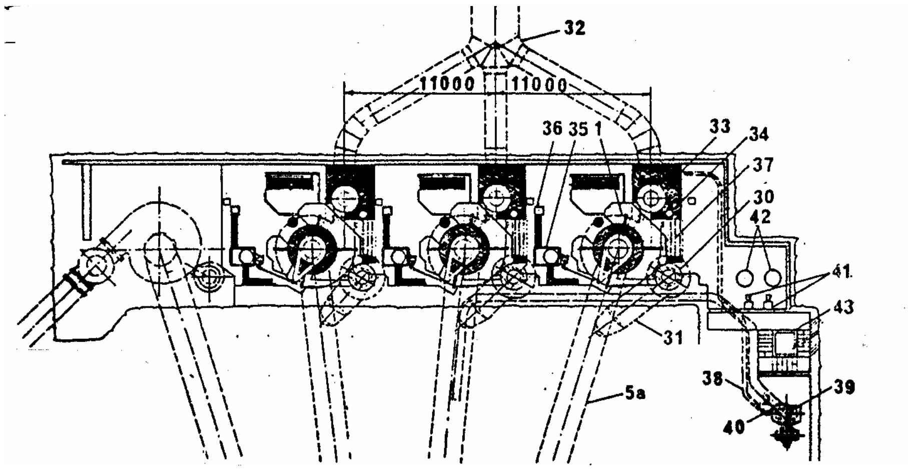
Fig. 7.2 Perspective of an axial section through a Francis turbine perspective of an axial section through the turbine. The position numbers point to details that are named in the list underneath.



Vertical section

An illustration of a vertical Francis turbine arrangement is given in Fig. 7.3, where the units are built up in an underground cavern. A vertical section through a unit is shown to the left and a horizontal section across all units in the cavern is shown to the right at the bottom of the figure.

The water flow enters the turbine through the pipe (32), the gate valve (33), scroll casing (1) and further through the guide apparatus and runner, draft tube cone (5), bend (5a) out in the tail race tunnel. The other position numbers are listed below.



Horizontal section

30. Relief valve and energy dissipation chamber
31. Outlet from energy dissipation chamber
34. Valve in the bypass pipe for pressurising the scroll casing before opening the gate valve
35. Governor housing with servomotor and accumulator
36. Pilot control of the governor
37. Ejector pipes
38. Penstock for auxiliary turbine
39. Auxiliary turbine
40. Tail race pipe from the auxiliary turbine
41. Vertical drain pumps
42. Motors for water cooling pumps
43. Staircase and conveying pit
44. Generator (above highest level of the tail water)
45. Emptying the penstock
46. Emptying pipe from the draft tube

Fig 7.3 Power plant with vertical Francis turbines

7.3 Main components and their functions

The examples of the building up of Francis turbines in Figs. 7.1 to 7.4, show that these turbine constructions consist of a great number of components. Many of these components are tailor made, and not all of them are found in every turbine. The constructions are to some extent time dependent and the different manufacturers construct some details different. Moreover the turbine constructions depend on the turbine size.

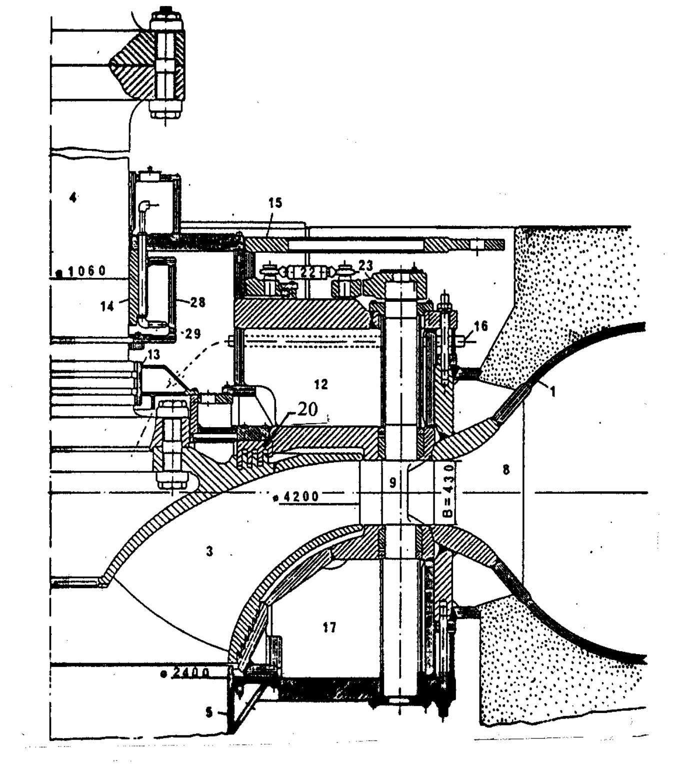


Fig.7.4 Vertical Francis turbine, axial section (Courtesy of Kværner Brug)

In the following considerations only main components/1/ which are vital are dealt with. These components are:

- Scroll casing
- Guide vane cascade
- Turbine covers
- Runner
- Shaft
- Bearing
- Shaft seal
- Regulating mechanism
- Draft tube

7.3.1 Scroll casing

The water from the penstock is conducted through the scroll casing and distributed around the stay ring and the complete circumference of the guide vane cascade.

The scroll casings are normally welded steel plate constructions for turbines at low, medium as well as high heads. An example of this type is shown on Fig. 7.4. This type is made for heights of 600 m to 700 m. However, scroll casings are also made as a combination of cast steel and steel plates in a welded construction.

The stay ring as shown on Fig. 7.4, consists of an upper and a lower ring to which the stay vanes (8) are welded. The stay vanes are given a favourable hydraulic shape to conduct the water towards the guide vanes with minimal losses. The stay vanes also carry the axial forces inside the scroll casing.

The scroll casing is provided with taps for pressure measurements, drain, air vent outlets and a manhole.

Analogous to Fig. 7.3 the scroll casings are partly or completely embedded in reinforced concrete.

7.3.2 Guide vane cascade

The guide vane cascade is represented by pos. (9) on Fig. 7.4. The openings of the guide vanes are adjustable by the regulating ring (15), the links (22) and levers (23).

The vanes are shaped according to hydraulic design specifications and given a smooth surface finish. The bearings of the guide vane shafts are lubricated with oil or grease.

7.3.3 Turbine covers

The covers are represented by pos. (12 and 17) on Fig. 7.4. These covers are bolted to the stay ring of the scroll casing. They are designed for high stiffness to keep the deformations caused by the water pressure at a minimum. This is of great importance for achieving a minimal clearance gap between the guide vane ends and the facing plates of the covers. Between the runner and the covers the clearance is also made as small as possible.

For high head turbines with sand in the water flow an intense wear is expected. Therefore it is common to weld layers of wear resistant stainless steel on the exposed surfaces. The upper and lower cover facings adjacent to the guide vanes are normally lined with high tension stainless steel. This may either be fastened to the covers with screws or clad welded.

The covers are supporting the guide vane trunnion bearings. The upper cover supports also the regulating ring bearing, the labyrinth ring, the turbine bearing and the shaft seal box. The lower cover supports the lower labyrinth ring and the draft tube cone.

The stationary labyrinth rings fixed to the covers are usually made of Ni-Al bronze, and they are replaceable. For large turbines the labyrinth rings are replaceable for the runners as well. The rings for the runners are usually made of stainless steel 13 % Cr 4 % Ni.

7.3.4 Turbine runner

The turbine runner pos. (3) on Fig. 7.4, may either be of cast steel or a welded construction where hot pressed plate blades are welded to the cast hub and ring. In most cases the runner is made of stainless steel.

As examples of runners Fig. 7.5 shows a photograph of a low head turbine and Fig. 7.6 shows a photograph of a high head turbine.

Fig. 7.5 Low head Francis turbine

Fig. 7.6 High head Francis turbine

The manufacture however, may be different from one manufacturer to the other and depends on the size and speed number.

The water flow through the labyrinth seals is a leakage flow and is not utilised by the runner. This flow is depending on the seal clearances. In a new turbine the seal clearances are small and the leakage flow losses lower than 0.5 %. However, during turbine operation the seals are worn, the leakage increases and the turbine efficiency decreases. Sand laden water causes a fast seal wear, and for high head turbines an increase of leakage losses of 2 - 5% has occurred after a relatively short running time.

On high head turbines the leakage water is normally utilised as cooling water for the generator, transformers and bearings. The runner is provided with a pump ring which is indicated on Fig. 7.7. This ring is pumping sufficient power for the cooling water system and prevents water to reach the labyrinth shaft seal. Through outlets in the upper cover filtered water is then provided.

Low head turbines cannot be provided with a similar pumping device because of low rotational speed. Instead the seal water normally runs through holes in the hub and directly into the draft tube.

The runner torque is transferred to the turbine shaft through a bolted friction joint or a combined friction and shear joint as shown on Fig. 7.7. For large dimensions the bolts of this joint are prestressed by means of heat. The bolts are made from high tensile strength steel and provided with a centre bore for measurements of elongation during prestressing.

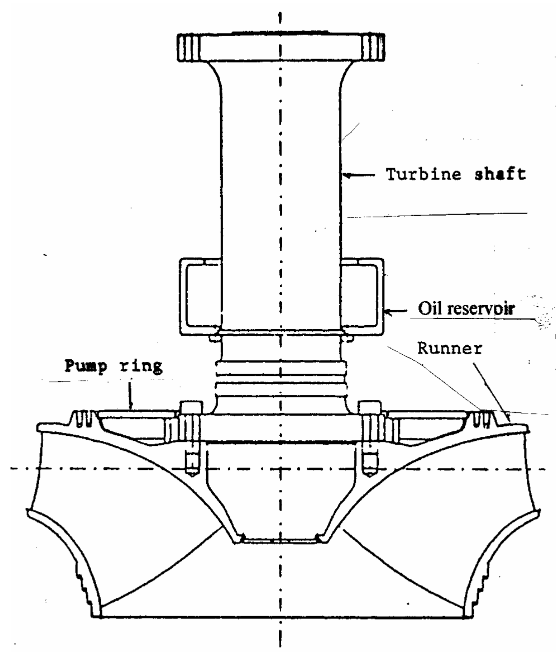


Fig 7.7 Rotating parts of a Francis turbine /1/

7.3.5 Turbine shaft and bearing

The turbine shaft Fig. 7.7, is manufactured from Siemens Martin steel and has forged flanges in both ends[/1/]. The turbine and generator shafts are connected by a flanged joint. This joint may be a bolted reamed or friction coupling where the torque is transferred by means of shear or friction.

An oil reservoir is bolted to the turbine shaft as shown on Fig. 7.7. On Fig. 7.8 this oil reservoir (2) is shown together with the construction of the bearing system.

This bearing is a rather simple and commonly used design and has a simple way of working and a minimal requirement of maintenance. The bearing house (1) Fig. 7.8, is split in two halves and mounted on the upper flange of the upper cover.

The bearing pad support ring (3) consists of two segments bolted together and mounted to the under side of the bearing house. The pad support ring has four babbit metal bearing surfaces with correctly shaped leading ramps ensuring stable centring of the turbine shaft. In the pad support ring there are also four oil pockets. The upper part of the housing has a cover (5) split in two halves with inspection openings.

The cover is provided with a cylindrical extension sleeve around the shaft reaching to the bottom of the housing, but leaving a slit for the oil to access the bearing pads. The extension prevents the oil in the housing from rotating with the shaft. The bearing pad support ring is surrounded by the oil reservoir. A riser pipe connected to an oil scoop passes through the bottom of the stationary bearing house and bearing pad support ring.

Under rotation the centrifugal forces keep the oil as a layer along the vertical cylindrical wall of the reservoir. The stagnation pressure from the rotating oil at the inlet end of the scoop, forces oil through the riser and cooler to the stationary upper oil reservoir.

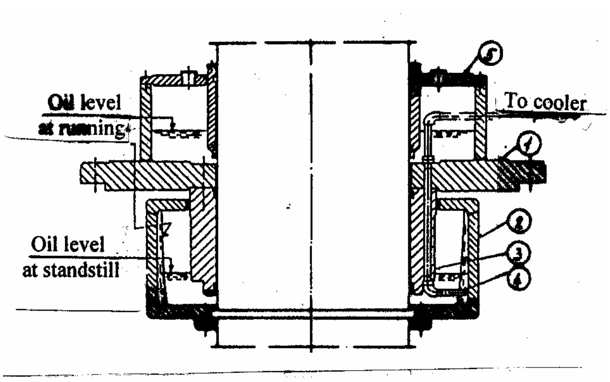


Fig. 7.8 Radial bearing of a vertical Francis turbine /1/

From the upper reservoir cooled oil flows to the pockets in the pad support ring. From these pockets the oil film follows the shaft rotation, enters the bearing segments and establishes the load carrying film in the bearing pads. Finally the oil comes back to the rotating reservoir from where a new circulation round starts.

The oil volume in the rotating reservoir is regulated by positioning the oil scoop at a certain distance from the reservoir wall.

7.3.6 Shaft seal

The location of the shaft seal is shown at pos.(13) on Fig. 7.4. An example of some details of a certain seal design/1/ is shown on Fig. 7.9. This shaft seal is split in two halves and mounted on the upper cover.

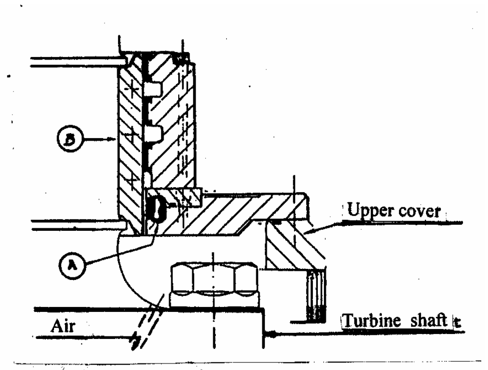


Fig.7.9 Shaft seal of a vertical Francis turbine /1/

The seal surfaces are lined with babbit metal, and depending on speed and size there are as small radial clearances as 0.2 - 0.4 mm between the surfaces of the shaft seal and the sleeve (B). The sleeve is made of corrosion resistant material and fixed to the shaft.

With the special pumping ring system mentioned above, the clearances in the seal box will run without water when the turbine is running. This is why the seal box can be given a design without contact between the babbit lined labyrinth and the shaft sleeve. A labyrinth seal of this type is suitable for operation in sand laden water because no sand will reach the seal while the turbine is running at normal speed.

At stand still a submerged turbine with open draft tube gate, is exposed to a downstream water pressure. A water leakage flow may then penetrate through the upper labyrinths and the seal box. This leakage water is removed from the box by a siphon pipe to the power house drainage pump sump.

For very deep submergence of the turbine an inflatable rubber seal ring (A) is installed in the labyrinth seal box. This ring is inflated during stand still in order to prevent leakage. During operation the air pressure inside the rubber seal is released and the rubber is not in contact with the shaft.

By means of the inflatable rubber seal it is possible to perform the dismantling of the shaft seal box without emptying the draft tube. At standstill the seal is activated by means of air pressure to compress the seal against the rotating sleeve (B) for establishing a droptight seal.

At certain loads unstable flow may occur in the draft tube downstream of the runner. In some cases this can be stabilised by air supply. This air may be supplied through a separate air supply pipe connected to the shaft seal box.

Instead of the design described above, the shaft seal box may also be designed with carbon rings. These are without clearance to the rotating parts and therefore subject to certain wear. Water cooling is necessary to a carbon seal and it cannot run dry without damage.

7.3.7 Regulating mechanism

The regulating mechanism is located by pos. (15) in Fig. 7.4. The design of the mechanism[/1/] is shown in more detail on fig. 7.10.

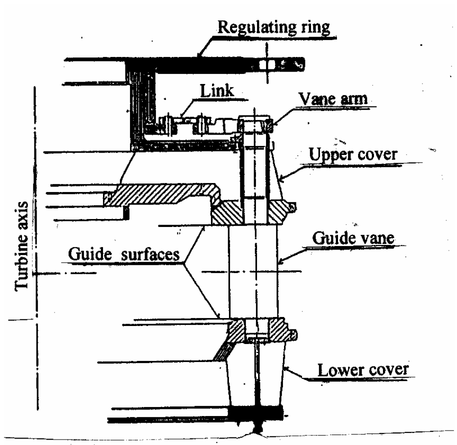


Fig. 7.10 Regulating mechanism of a Francis turbine /1/

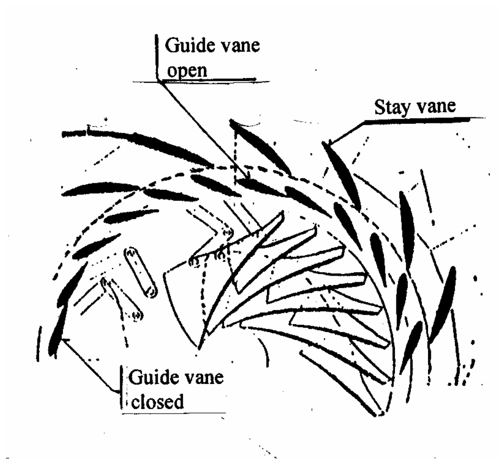


Fig. 7.11 Guide vane regulating system of a vertical Francis turbine /1/

The guide vane mechanism provides the regulation of the turbine output. Together with the governor it is able to maintain a stable speed of the unit and the frequency in the electrical distribution grid.

The turbine governor controls the servomotor which transfers its force through a rod to the regulating ring. This ring transfers the movement to the guide vanes through a rod, lever and link construction. A section through a part of the guide vane cascade, the stay vanes and the runner is shown on Fig. 7.11. The guide vane exit area in flow direction is varied by an equal rotation of each of the guide vanes.

The vane levers are mounted on the upper trunnion and fixed by a wedge, shear pins or pure friction joint.

The guide vane lever and regulating ring are connected through links. These links are connected through selflubricated spherical bearings on trunnions on the regulating ring and the lever respectively.

The trunions are positioned on the guide vanes for achieving minimal regulating forces from the hydraulic forces acting on them.

7.3.8 Draft tube

The draft tube (34) on Fig. 7.3, forms the water conduit from the runner to the draft tube outlet. Fig. 7.12 shows a draft tube in more detail[/1/]. It consists of the draft tube cone and the draft tube steel plate lining.

The aim of the draft tube is also to convert the main part of the kinetic energy at the runner outlet to pressure energy at the draft tube outlet. This is achieved by increasing the cross section area of the draft tube in the flow direction. In an intermediate part of the bend however, the draft tube cross sections are decreased instead of increased in the flow direction to prevent separation and loss of efficiency.

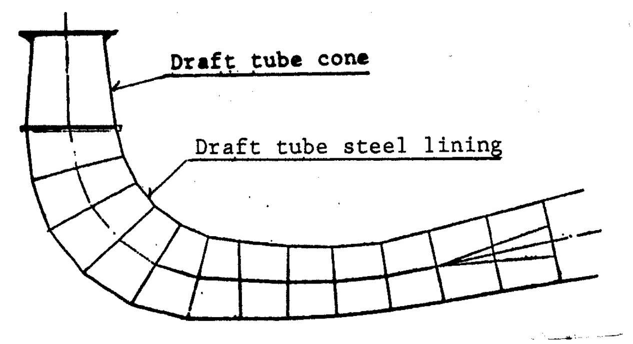


Fig. 7.12 Draft tube of a vertical Francis turbine /1/

The draft tube cone is a welded steel plate design and consists normally of two parts, the upper and lower cone. The inlet part of the upper cone is made of stainless steel. It is normally provided with two manholes for inspection of the runner from below. The lower part is designed as a dismantling piece and is mounted to a flange on the draft tube bend top. This design is always used for units where the runner is dismantled downwards.

For units being dismantled upwards the draft tube cone is made in one piece.

The draft tube lining is completely embedded in concrete.

7.4 Drainage and filling arrangements

A normal drainage and filling arrangement[/1/] is shown in Fig. 7.13.

The penstock cone and the scroll casing of a submerged turbine can be drained to a level corresponding to the tail water level through the draft tube. The draft tube is normally filled tube gate.

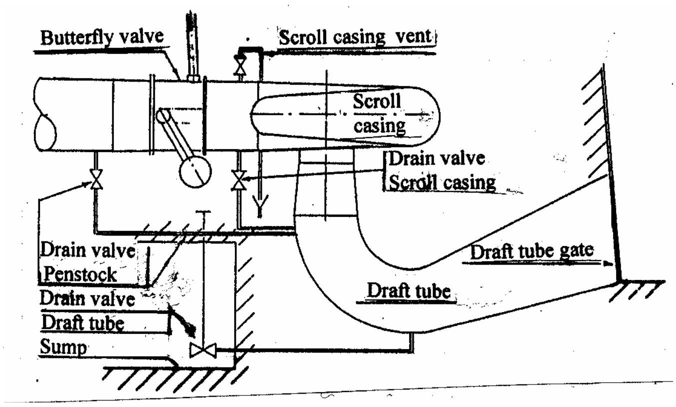


Fig. 7.13 Draining and filling system of a Francis turbine /1/

A turbine that is not submerged, the draft tube and the scroll casing are filled from the penstock.

7.5 Condition control

Routine inspections

Routine inspections means visual inspection of the complete turbine:

- look for possible leakages
- inspect bolted connections
- drain pumps should be inspected and level switches tested.

RPM shutdown curve

The activity is to record the RPM with 30 seconds intervals from the moment of closed guide vanes to standstill of the unit. The RPM shutdown curve should be drawn from nominal RPM to standstill.

Leakage control of guide vanes

The activity is to record the RPM with the spherical valve open and the guide vanes closed.

Shaft alignment

The activity is to record the shaft misalignment by means of a micrometer dial instrument against the shaft at the turbine bearing top. The misalignment should be recorded at different loads on the unit.

Labyrinth seal water flow

The assessment of wear and control of the labyrinth rings is possible by measuring the labyrinth water flow.

Runner

Visual inspection of the runner is required to record possible cavitation and erosion damages as well as cracks in vanes. The inspection of the inlet is done from the scroll casing. Three of the guide vane arms should be dismantled and the guide vanes rotated by hand to open position. The unit should be rotated manually to enable inspection of the complete runner circumference. The outlet side of the runner should be inspected from the draft tube cone.

Scroll casing and draft tube

The activity is to carry out inspection of painting for possible corrosion and ensure that all manometer connection openings are open and that the manhole cover is drop tight after completed inspection.

Guide vane mechanism

The guide surfaces of the covers should be inspected from the scroll casing and possible wear recorded.

Visual inspection of the surfaces of the guide vanes. The clearances between the guide vanes and the cover surfaces and the clearance between guide vanes should be checked.

Look for possible leakages in the guide vane bearings. Look for slacks in the bearings of the guide vane arms and links.

Check the connection between regulating ring and the servomotor.

Shaft seal box

During operation water will normally not flow over the shaft seal box. It may however happen during start and stop. If water runs into the upper cover it is removed by a drain pump.

Turbine bearing

The same activities as for Pelton turbines.

7.6 Monitoring instruments

The turbine is provided with instruments usually as presented in the following list:

- A manometer connected to the scroll casing
- A manometer connected to the penstock upstream of the main valve
- A pressure switch with two minimum settings connected to the penstock upstream of the main valve
- A mano-vacuum meter directly mounted to the draft tube cone
- A manometer connected to the leakage pipe
- A limit switch for signal of stand still seal on

The bearing is provided with:

- A contact thermometer in for high and critical high temperature control
- A remote thermometer for oil temperature measurements in the bearing housing
- A level switch for high and low oil level warning. This hangs together with cooling water leakage

7.7 Assembly and dismantling

The assembly and dismantling is important part of the design of the turbines and the arrangement of the installation. The objective is to make those duties as convenient and easy as possible. Wear will occur sooner or later in the guide vane mechanism as well as in the upper and lower runner labyrinths depending on the water quality and the operating head.

Medium and high head Francis turbines with free access to the draft tube cone provide good opportunities for fast dismantling and assembly of vital parts.

A widely used method is to dismantle the main parts downwards and then lift them up on the floor in the machine hall by using the main crane in the power station. The parts in question are the runner, lower cover and the labyrinth rings. Furthermore the turbine bearing should be easily removable. Dismantling of the upper cover is seldom required. Possible damage to the surface adjacent to the guide vanes is easily repaired on site.

Assembly or dismantling downwards requires a comparatively large opening below the scroll casing. This may be difficult to achieve with low head turbines with large dimensions because the load carrying concrete around the scroll casing occupies space needed for access. These turbines however, are not so much exposed to wear in the labyrinths and other components. The draft tube cone is often embedded in concrete, and the complete turbine must be dismantled upwards after prior removal of the generator rotor.

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CHAPTER 8

Kaplan Turbines

Introduction

Kaplan turbines have been developed to be the most employed type of turbines for low heads and comparatively large discharges when the speed number $^*\underline{\Omega} > 1.0$. The Kaplan turbines are fairly suitable for the purpose of three main reasons:

- relatively small dimensions combined with high rotational speed
- a favourable progress of the efficiency curve
- large overloading capacity

The runner has only a few blades radial oriented on the hub and without an outer rim. The water flows axially through. The runner blades have a slight curvature and cause relatively low flow losses. This allows for higher flow velocities without great loss of efficiency. Accordingly the runner diameter becomes relatively small and the rotational speed more than two times higher than for a Francis turbine for the corresponding head and discharge. In this way the generator dimensions as well become comparatively smaller and cheaper. The comparatively high efficiencies at partial loads and the ability of overloading is obtained by a co-ordinated regulation of the guide vanes and the runner blades to obtain optimal efficiency for all operations.

8.1 Kaplan turbine construction

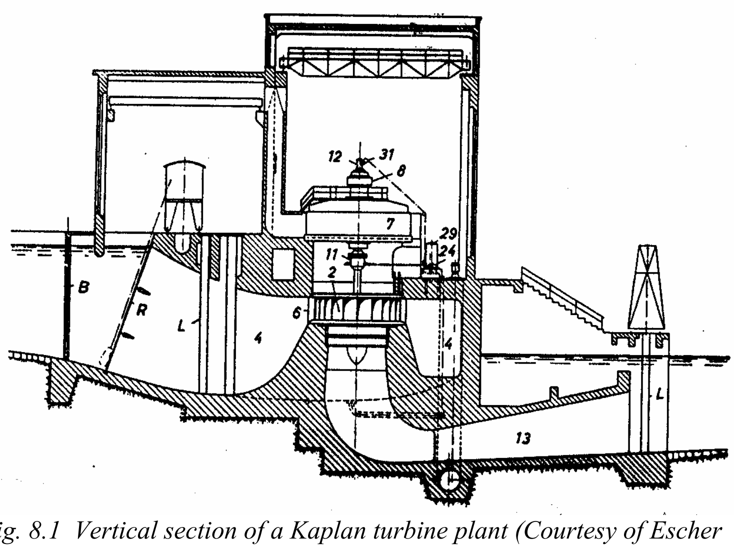

Fig. 8.1 Vertical section of a Kaplan turbine plant (Courtesy of Escher Wyss)

8.1.1 Arrangement

A vertical section through a Kaplan unit is shown on Fig. 8.1. From the upstream basin the water flows into the scroll casing (4). The water flows from the scroll casing through the stay ring (6), the guide apparatus (2), the runner and the draft tube (13) into the tail water basin.

The generator (7) is arranged above the turbine, and in most cases above the highest level of the tail water. The axial thrust bearing (8) is loaded with axial forces from all the rotating parts. In many cases this bearing is arranged upon the upper turbine cover, which then have to carry all the axial forces.

8.1.2 Kaplan turbine construction

A vertical section through a Kaplan turbine unit is shown on Fig. 8.2. The name of the main components are numbered and listed beside and below the figure.

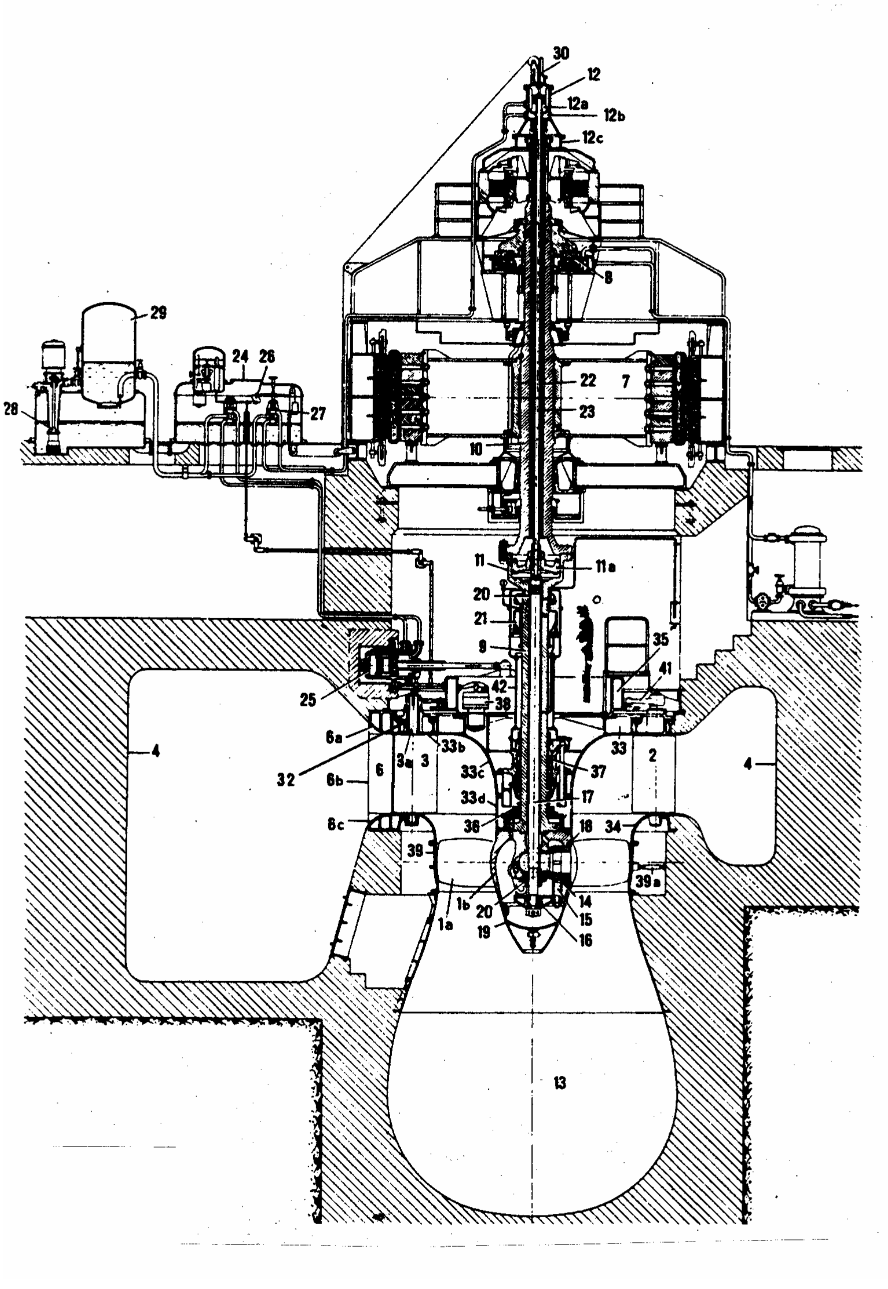

Fig. 8.2 Vertical section of a Kaplan turbine unit (Courtesy of Escher Wyss)

1a. Runner vanes
1b. Crown
2. Guide vane cascade
3. Guide vanes
4. Scroll casing
6a. Upper ring
6b. Stay vanes
6c. Lower ring
7. Generator
8. Axial-thrust bearing
9. Turbine shaft
10. Generator shaft
11. Runner vane servo.
11a. Servomotor piston
12. Oil inlet
12a. and 12b. oil chambers
12c. Oil collector
13. Draft tube
14. Regulating ring for runner vanes
15. Links from (14) to (15)
16. Lever spider for force transfer from (17)
17. Rod for force transfer from (11a)
19. Crown cover
20. Conveying of rod (17)
21. Rotating oil collector
22. and 23. Coaxial pipes
24. Turbine govrnor
25. Servomotor for guide vane cascade
26. Cam
27. Reg. valve for high pressure oil to runner servomotor
28. Oil pump
29. Accumulator
30. Lining for for feed back lever
31. Feed back lever
32. Turbine cover
33b. Outer cover
33c. Inner cover

33d. Base ring for bearin foot
35. Regulating ring
36. Shaft sealing box
37. Radial bearing
38. Outer shroud
39a. Horizontal supports
41. Guide vane lever
42. Stationary cover lining of turbine shaft

8.2 Main components and their functions

The Kaplan turbines have the following main components:

- scroll casing and stay ring
- guide apparatus
- covers
- runner
- runner blade servomotor
- regulating mechanism of the runner blades
- co-operation of regulating the runner blades and guide vanes
- turbine shaft
- turbine bearing
- shaft sleeve and seal box
- runner chamber
- draft tube

8.2.1 Scroll casing

The scroll casings for lower heads 25 - 30 meters are made of concrete. To make these type of scroll casings with the required accuracy, wooden models are used against which the concrete is poured. The manufacturer of the turbine determines the shape and makes the drawings of these models. The quality of a water tight and even surfaces of the scroll casings is required to be the same as for draft tube bends of concrete.

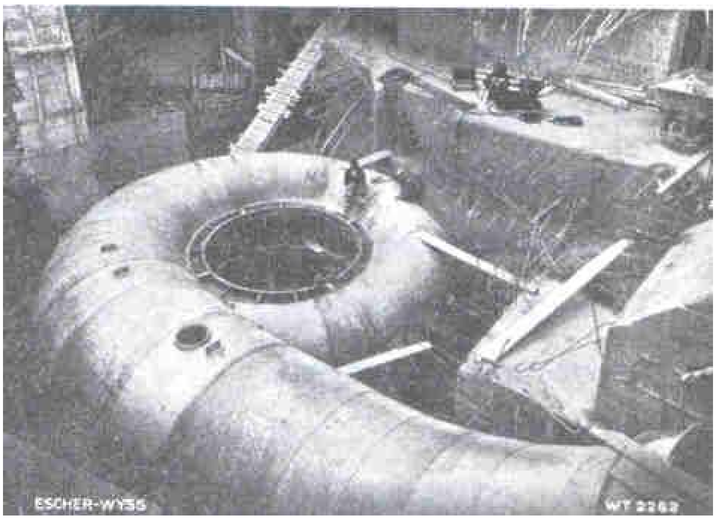

Fig. 8.3 Kaplan scroll casing of steel plates

For higher heads the hydraulic pressure may be too high for the concrete to withstand the load. In such cases scroll casings of steel plates are designed in a way analogous to that of Francis turbines as shown on Fig. 8.3. The cross sections of the scroll casing are normally of circular shape, and the steel plate shells are welded to the stay ring.

The vanes in the stay ring conduct the water towards the guide vanes. In addition the hydraulic forces are transferred through the stay ring and the stay vanes which are anchored to the concrete with large prestressed stay bolts. The stay vanes are normally made of welded steel plates and filled with concrete.

Fig. 8.4 shows a picture of the phase of installation of the stay ring. The lower part of the concrete scroll casing is completed.The upper stay bolts are installed and will be embedded together with the rest of the scroll casing.

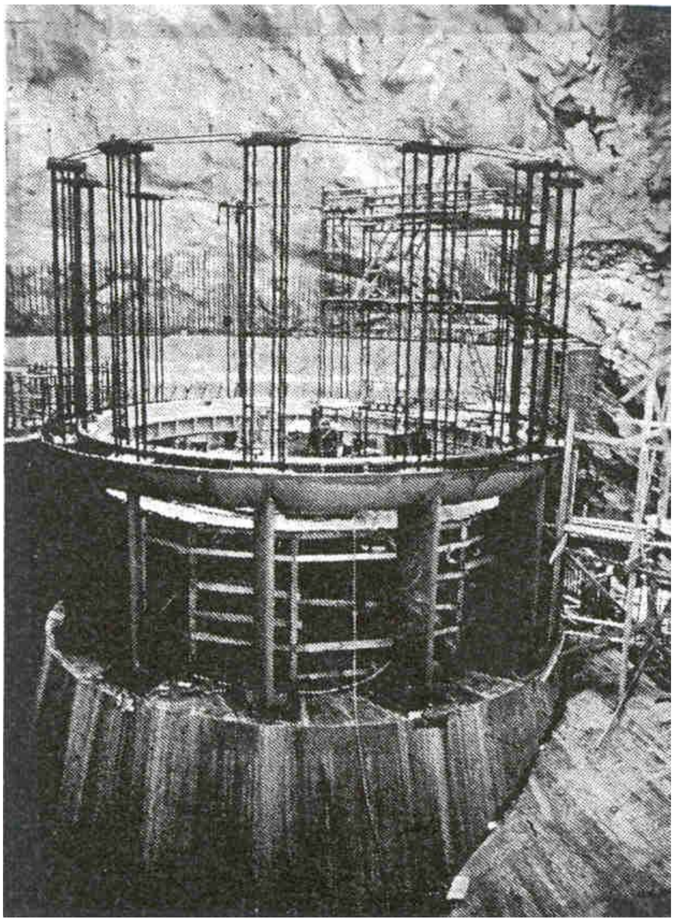

Fig. 8.4 Erection of a stay ring /1/

8.2.2 The guide vane cascade

The guide vane cascade of Kaplan turbines are constructed in the same way as for Francis turbines. In the sense of operation a regulating ring rotates the guide vanes through the same angles simultaneously when adjustments follow changes of the turbine load. The vanes are manufactured of steel plate material and the trunnions are welded to them. The vane design is purposely to obtain optimal hydraulic flow conditions, and they are given a smooth surface finish.

8.2.3 Covers

The Kaplan turbines are usually provided with an inner cover in addition to an upper and a lower cover. The inner cover is bolted to the upper cover and forms a shield from upper flow conducting surface and downwards to the runner. Furthermore this serves as a support for the guide vane mechanism with the regulating ring, the turbine bearing and the shaft seal box with standstill seal.

The lower turbine cover is combined with the runner chamber by a flanged connection.

8.2.4 Runner

The runner in a Kaplan turbine is a very challenging part to design. The details for adjusting the blades can be designed in different ways. Increasing blade number for increasing head may create problems because of lack of space and consequently high stresses in some details of the construction. It is not however, only the head that determines the number of blades. The blade length and shape as well as the specific blade loading and location in relation to the downstream water level, are factors which must be considered. As a general guideline four blades can be used up to heads of 25 - 30 meters, five blades up to 40 meters, six blades up to 50 meters and seven blades up to heads of 60 - 70 meters. Kaplan turbines have also been designed with 8 blades for heads even higher than 70 meters. This increases the hub diameter and the shape of the hub becomes more complicated, and the efficiency may suffer.

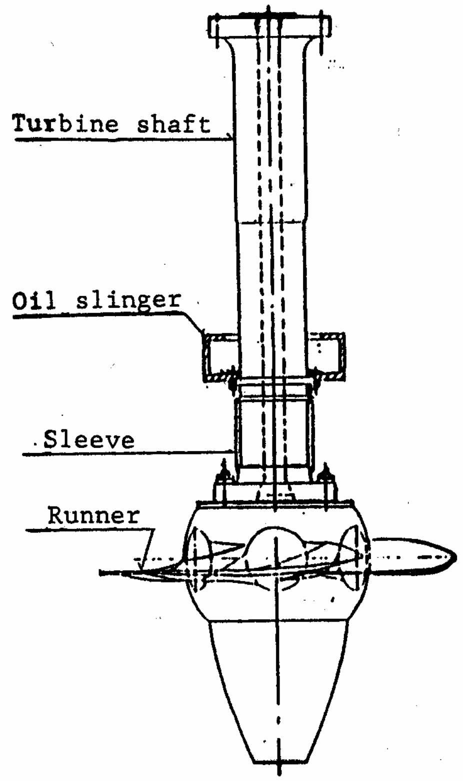

Fig. 8.5 Rotating parts of a Kaplan turbine /1/

The outside of the hub is spherically shaped[/1/] as shown on Fig. 8.5. This is done to keep a small clearance gap between the adjustable blade ends and the hub for all operating conditions. With increasing head the hub diameter is increasing from approximately 40% to 65 - 70 % of the runner diameter.

The torque of the runner is transferred to the turbine shaft through either a pure friction joint connection or through a combined shear bolt and friction joint. The bolts joining the turbine shaft flange and the runner are prestressed by means of heat for the largest bolt dimensions.

8.2.5 Runner blade servomotor

The servomotor for the rotary motion of the runner blades is either a construction part of the turbine shaft or located inside the hub. An example of the former type is shown schematically on Fig. 8.6, which is a concept from Escher Wyss.

There are however, good reasons for localising the servomotor inside the hub, and the details of a such construction are dealt with in the chapter of bulb turbines. This servomotor may consist of a moving cylinder and a fixed piston integrated with the hub. The conversion from axial piston movement to rotating blade movement is carried out by a link and lever construction.

The hub is completely filled with oil to provide reliable lubrication of moving parts. The oil pressure inside the hub is kept higher than the outside water pressure to prevent water penetration into the oil.

8.2.6 Regulating mechanism of the runner blades

An example of the regulating system of the runner blade slope is shown on Fig. 8.6. The slope of the runner blades (1a) are adjusted by the rotary motion activated by the force from the piston (11a) through the rod (17). The cylindrical extension of the upper end of the turbine shaft (9) serves as a servomotor cylinder (11) whereas the lower flange of the generator shaft serves as cover. The rod (17) moves in the two bearings (20).

The oil supply to the servomotor (11) is entered at the upper end of the generator shaft (10). The oil is conveyed to the respective sides of the servomotor through two coaxial pipes (22 and 23) inside the hollow generator shaft. The inner tube (22) conveys oil to and from the lower side of the piston (11a), whereas the annular opening between the pipes (22) and (23) conveys oil to and from the piston top side. The oil is supplied through the entrance arrangement (12) with the two chambers (12a) and (12b) at the top of the unit.

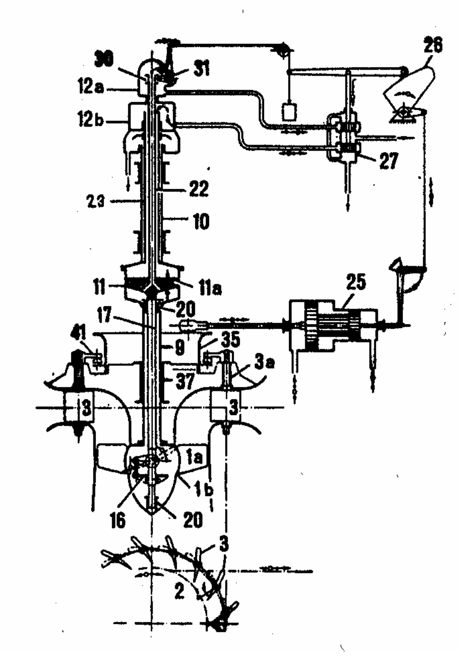

Fig. 8.6 Regulating mechanism

8.2.6 Cooperation of regulating the guide vanes and the runner blades

The turbine governor operates directly on servomotor (25) which executes the movement of the guide vanes (3) shown on Fig. 8.6. The movement of the servomotor thriggs and controls the slope adjustment of the runner vanes. This is carried out by a rod and lever transfer from the servomotor (25) to the cam (26) which is turned according to the movement of the servomotor piston (25). In this way the spool valve (27) is moved out of the neutral position and the servomotor piston (11a) is then put to movement by the oil pressure supply.

The spool valve (27) receives pressure oil either directly from the oil pump or from the accumulator which is energised by an oil pump.

8.2.8 Runner chamber

The clearance gap between the outer blade ends and the chamber wall is essential to keep as small as possible for all blade inclinations. Therefore the runner chamber is made spherical below the rotation centre line of the blade trunnions.

Ideally the spherical shape should have been maintained above the blade rotation centre as well However, on account of installation and dismantling aspects, this part is being made cylindrical as shown on fig.8.7, which is a concept of Escher Wyss. The gap between the runner blade ends and the runner chamber wall is approximately 0.1% of the runner diameter.

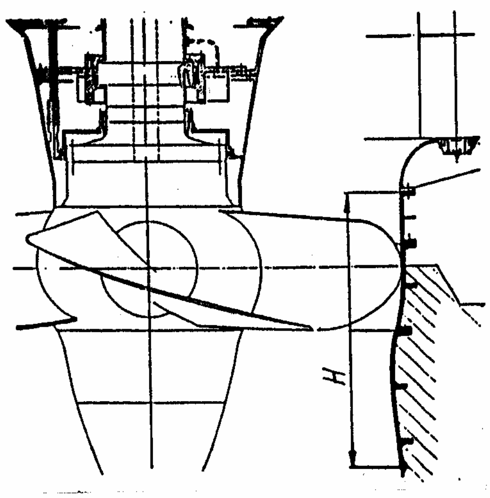

Fig. 8.7 Runner chamber

On fig. 8.7 the length of the runner chamber is indicated by H. At the lower end this chamber is welded to the draft tube via an extruded steel profile.

Cavitation erosion on the runner chamber may occur during the running time of the turbine. To reduce the magnitude of such erosion attacks and to ease the subsequent repair, the runner chamber is normally made of cast or welded stainless chromium nickel steel with higher cavitation resistance than carbon steel. Existing erosions may then be repaired by welding on site. The runner chamber is reinforced by external ribs.

The runner chamber is normally completely or partly embedded in concrete. The turbine shown on fig. 8.7 has an access tunnel around the complete circumference, providing access to the lower guide vane bearings. In combination with this tunnel there is a manhole access to the runner chamber for inspection of the runner blades.

In the lower part of the runner chamber there is a tap for connection of a vacuum meter. In the lower part holes are plugged by means of removable stainless steel plugs.

8.2.9 Turbine shaft

The turbine shaft Fig. 8.5, is made of Siemens Martin steel and is provided with integrally forged flanges in both ends[/1/].

In the area of the shaft seal box, a wear sleeve made of stainless material is clamped around the shaft. The rotating oil reservoir is bolted to the turbine shaft.

8.2.10 Turbine bearing

The bearing design is shown on Fig. 8.8 and the function is described in the chapter of Francis turbines[/1/].

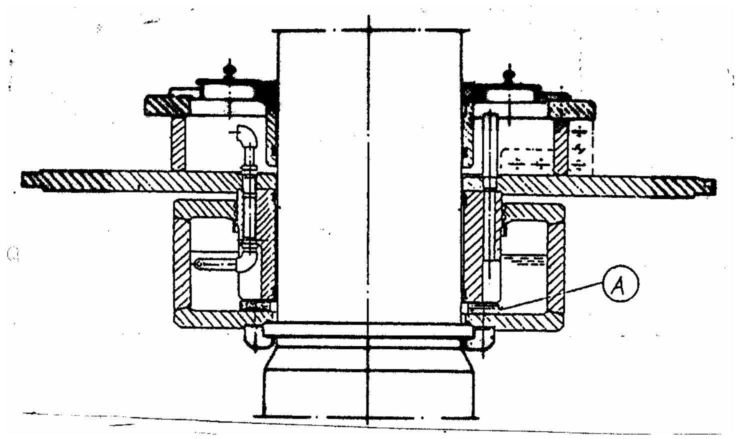

Fig. 8.8 Turbine bearing /1/

The only detail that deviates from the normal Francis turbine bearing is the back thrust slide ring A, which is bolted to the bottom of the rotating oil reservoir. By load rejections the turbine is subject to a vertical force acting upwards and this may cause a lifting of the unit. The back thrust ring then hit the underside of the bearing pads which transfer the vertical force to the base of the bearing.

The back thrust ring is made of bronze and is provided with lubrication grooves ensuring a good distribution of oil, and a load carrying oil film is then attained.

Low head Kaplan turbines are more exposed to lifting than high head Kaplan turbines. These turbines also have a comparatively higher speed increase at load rejections. With the guide vanes closed, the runner is pumping the water against the downstream water pressure. In this way a vertical upward force is created because the pressure above the runner becomes lower compared with the pressure below it. This effect increases with increasing speed.

8.2.11 Shaft seal box

A commonly applied seal box for Kaplan turbines is the carbon ring box of a type as shown on Fig. 8.9. The seal elements against the turbine shaft consist of specially made split carbon rings which are pressed against the shaft by means of spiral springs. The seal box is exposed to a fluctuating pressure from the turbine waterside and the rings must be located with this in mind. The turbine shaft is exposed to certain wear by the seal rings. The shaft which is passing through the seal box is therefore provided with a wear sleeve of stainless steel.

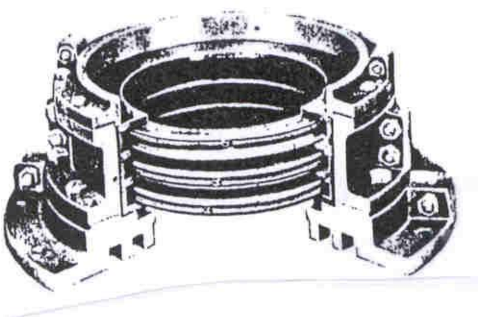

Fig. 8.9 Carbon ring seal box

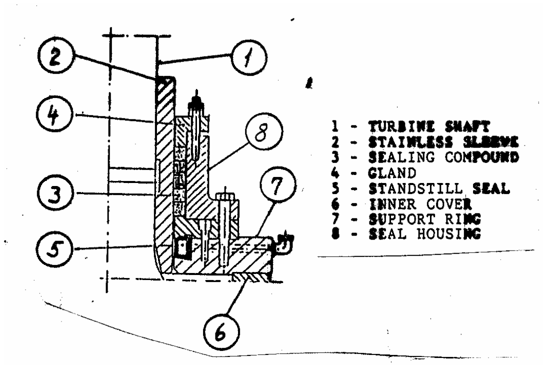

Fig. 8.10 Compound seal box /1/

A simpler seal box design[/1/] is shown on Fig. 8.10. This shaft seal is used for small and middle size turbines with moderate circumferential shaft speed.

The shaft seal box is mounted on the inner turbine cover (6). It consists of the support ring (7), seal housing (8), gland (4) and sealing compound (3). The sealing compound seals against a stainless sleeve (2) which is clamped to the turbine shaft (1).

By means of a gland (4) the sealing compound may be compressed, and a good sealing effect is obtained. Too hard compression may cause heating between the shaft sleeve and the sealing compound. To monitor this a temperature sensor is mounted in the seal housing. The sensor is connected to an alarm in the control room.

The inflatable rubber seal ring (5) which is used as a standstill seal, is described in the chapter of Francis turbines.

Leakage from the seal box is collected in a sump in the inner cover and is pumped to the station sump by float controlled pumps.

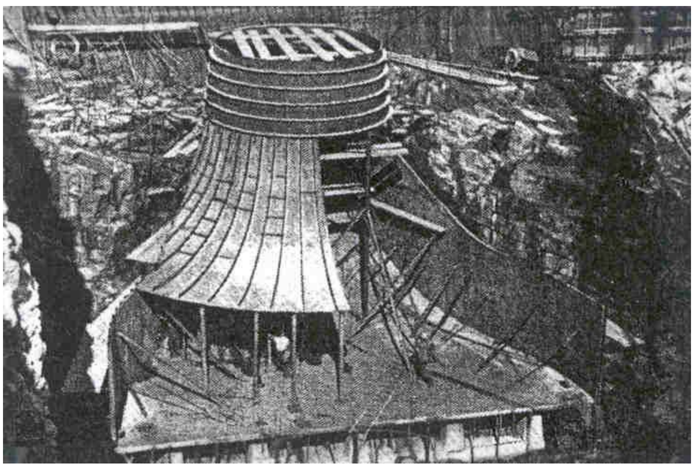

Fig. 8.11 Draft tube erection /1/

8.2.12 Draft tube

The draft tube consists of a draft tube cone and a draft tube plate lining through the bend.

The water has a relatively large velocity when it leaves the runner. This kinetic energy must be converted to pressure energy in the draft tube. To obtain this with a minimum of losses, the outlet velocity at the draft tube outlet should be as uniform as possible. Because the kinetic energy represents a high fraction of the total energy, the shape of the draft tube is of great importance for the hydraulic efficiency.

The draft tube of a Kaplan turbine has a somewhat special shape. The units have comparatively large dimensions and the civil works are expensive. It is therefore a requirement to make the draft tube as shallow as possible. The cone, the upper part and the inner curve surface are always lined with steel plates. The rest is normally made of unlined concrete. The formwork and the pouring of concrete are made as simple as possible by making the walls straight with single curved surfaces only.

The draft tube shown on Fig. 8.11, is welded from steel plate through the bend.

8.3 Drainage and filling arrangement

A normal drainage and filling arrangement is shown in chapter 7 about Francis turbines. A similar arrangement usually exists for a Kaplan turbine plant as well. Because of low head and large dimensions, it is normally not a main inlet valve upstream of the turbine. Instead an intake gate is installed at the top of the penstock. Drainage of inlet pipe and scroll casing is carried out by means of a common drainage pipe after the intake gate is closed. The inlet pipe may however be drained through the turbine down to draft tube level.

8.4 Condition control

The general principles for condition control are the same as for Francis turbines. Further considerations are therefore connected only to a few specific details.

8.4.1 Runner

The runner should be inspected both from above and below. Particular attention should be given to possible cavitation erosion and scratches on the blades as well as leaks around the blade flange against the hub.

8.4.2 Runner chamber

The narrow gap between runner and the runner chamber should be checked if foreign objects may have passed the gap and made scratches in the chamber.

8.4.3 Guide vane mechanism

For guide vane mechanism with individual vane servomotors on bulb turbines it should be checked that the vanes have an identical movement.

8.4.4 Shaft seal box

For bulb turbines at standstill it should be checked that the water does not flow out of the box along the shaft into the turbine bearing.

8.5 Monitoring instruments

The instrumentation consists mainly of the same components as for Francis turbines and is described in Chapter 7.

8.6 Assembly and dismantling

The cone and the runner chamber of a Kaplan turbine are normally completely or partly embedded in concrete. Therefore these parts can not be dismantled. Consequently all dismantling of the turbine therefore must be done upwards through the stator of the generator after removal of the generator rotor.

The parts in question for dismantling are first and foremost the inner cover and the runner. The inner cover is bolted to the outer cover. The smallest diameter of the outer cover is dimensioned to allow the runner to be lifted after dismantling of the inner cover.

Dismantling of the guide vanes may take place after the outer cover is lifted above the longest upper trunnion of the guide vanes.

By assembly or dismantling the runner may hang either by means of the brackets bolted to the runner chamber below the runner or by means of brackets bolted to the lower cover[/1/] as shown on Fig. 8.12.

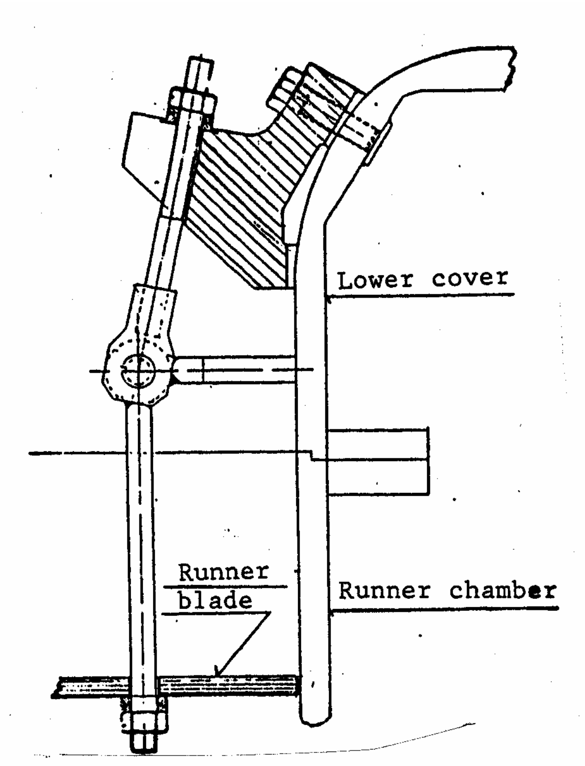

Fig. 8.12 Runner hanging device /1/

To replace a runner blade, it must be possible to dismantle a part of the runner chamber. This part is cut loose from the concrete and is pulled back. The blade may then be dismantled, pulled out and may be taken out through the draft tube.

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CHAPTER 9

Bulb Turbines

Introduction

The Bulb turbine is a reaction turbine of Kaplan type which is used for the lowest heads. It is characterised by having the essential turbine components as well as the generator inside a bulb, from which the name is developed. A main difference from the Kaplan turbine is moreover that the water flows with a mixed axial-radial direction into the guide vane cascade and not through a scroll casing. The guide vane spindles are inclined (normally 60°) in relation to the turbine shaft. Contrary to other turbine types this results in a conical guide vane cascade.

The Bulb turbine runner is of the same design as for the Kaplan turbine, and it may also have different numbers of blades depending on the head and water flow.

9.1 General arrangement

A general arrangement of a Bulb turbine plant/1/ is shown in Fig. 9.1 by a vertical section through the unit. Fig. 9.2 shows the turbine design in more detail.

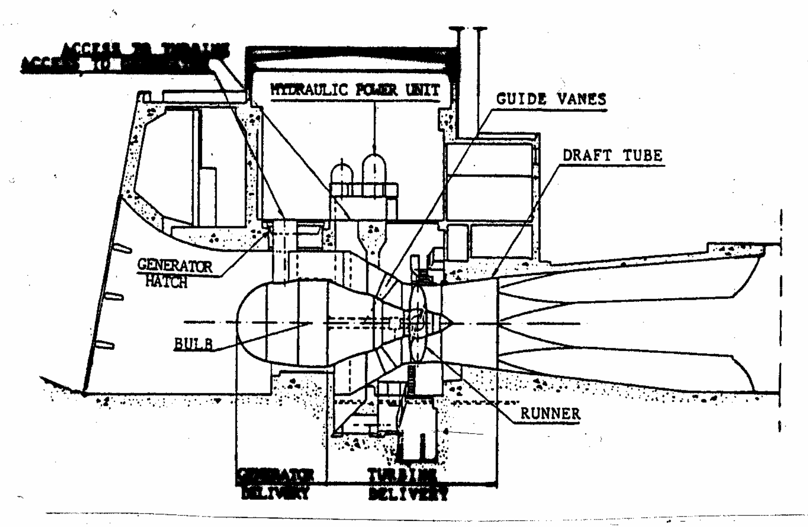


Fig. 9.1 Bulb turbine. General arrangement /1/

The water flows axially towards the unit in the centre of the water conduit and passes the generator, the main stays, the guide vanes, runner and draft tube into tale race channel.

9.2 Main components

The Bulb turbine consists of the following main components:

- stay cone
- runner chamber
- draft tube cone
- generator hatch
- stay shield
- rotating parts

- turbine bearing
- shaft seal box
- guide vane mechanism

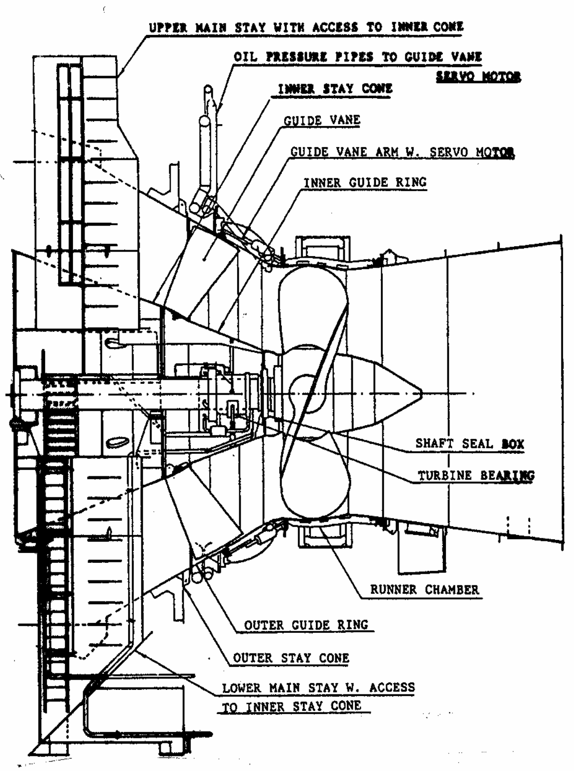


Fig. 9.2 Bulb turbine (Courtesy of Kværner Brug)

9.2.1 Stay cone

A longitudinal section through a stay cone structure and draft tube[1] is shown in Fig.9.3. There are one lower (1) and one upper (5) main stay. An inner stay cone (3) and outer stay cone (6) are welded to the main stays. To the downstream end of the inner stay cone the inner guide ring is bolted as shown on Fig. 9.2.

This outer stay cone forms a part of the outer water conduit contour and is embedded in concrete together with outer parts of the main stays. The generator bulb is bolted to the upstream end of the inner stay cone as indicated on Fig. 9.1. These parts are located in the centre of the water flow and forms the inner water conduit contour together with the runner hub.

Two side stays (4) and (7) on Fig. 9.4, are located on each side of the bulb upstream of the main stays for stiffening the bulb and avoid resonant vibrations.

The total weight and the hydraulic forces are transferred to the surrounding concrete through the stay cone via the two main stay vane structures. The dynamic as well as the static forces from the turbine and the generator are transferred through the structure to the building foundation.

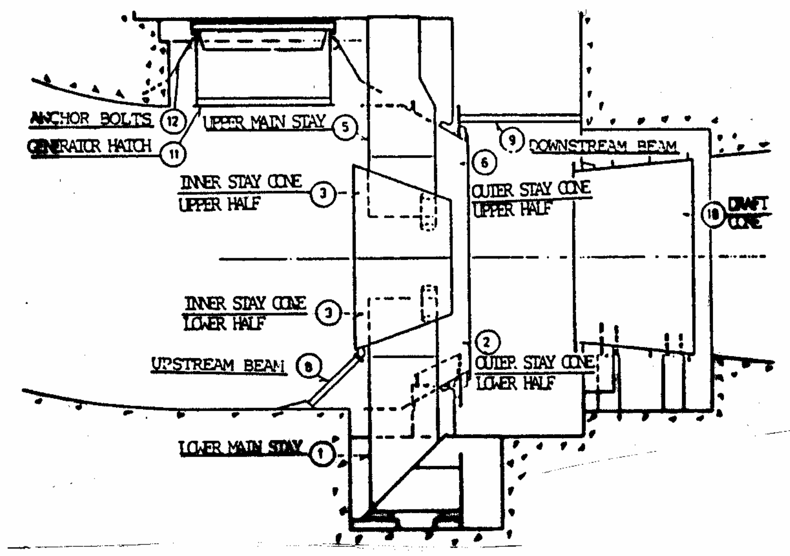


Fig. 9.3 Stay cone and draft tube. Vertical section(Courtesy of Kværner Brug)

9.2.2 Runner chamber and draft tube cone

The runner chamber is the connecting part between the outer stay cone and the draft tube cone, Fig. 9.2. The downstream end of the outer cone is provided with a flange to which the runner chamber is bolted.

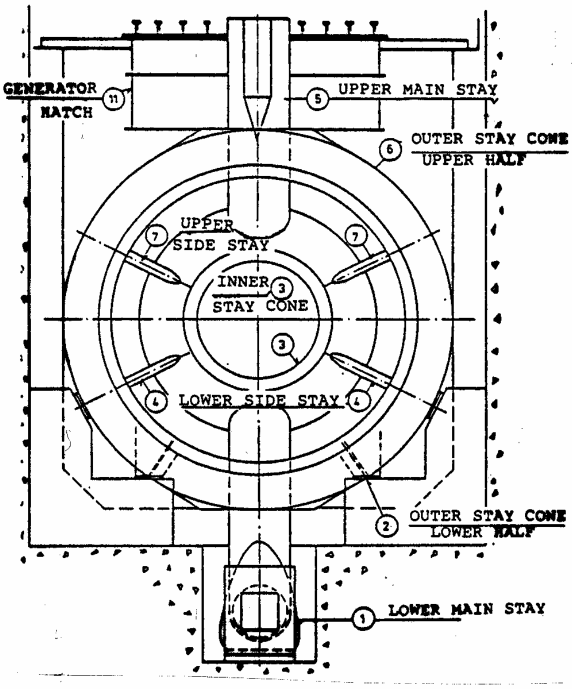


Fig. 9.4 Cross section of stay cone (Courtesy of Kværner Brug)

The draft tube cone consists of two or more straight welded steel cones and is embedded in concrete. The upstream end is connected to the runner chamber through a flexible telescope connection. This type of connection is necessary to allow for a certain axial movement of the runner chamber and the outer guide ring because of elongation/contraction due to temperature changes.

The length of the steel cone lining is determined by requirements to maximum water velocity at the exit and to avoid damage to the concrete.

9.2.3 Generator hatch

The generator hatch (11) on Fig. 9.4 is normally a part of the turbine delivery. It is located above the generator and provides access to the generator for assembly or dismantling tasks.

The hatch consists of a perforated part which forms the outer water conduit contour in the hatch opening. A cylindrical steel mantel with a flange on top is provided for the hatch cover and seal mounting. As the unit's bulb part will rise and lower with filling and draining of the turbine, the seal joint between mantel and hatch cover must allow for a vertical movement of the mantel.

9.2.4 Stay shield

The stay shields are located between the generator access shaft and the turbine main stay. They form an even wall for the water flow and the stay structure is streamlined at the upstream end to prevent undesired vortex formation. The shields are bolted to the bulb and connected to each other by screw stays for stiffness. They are freely supported against the access way and the main stay to allow for axial movements.

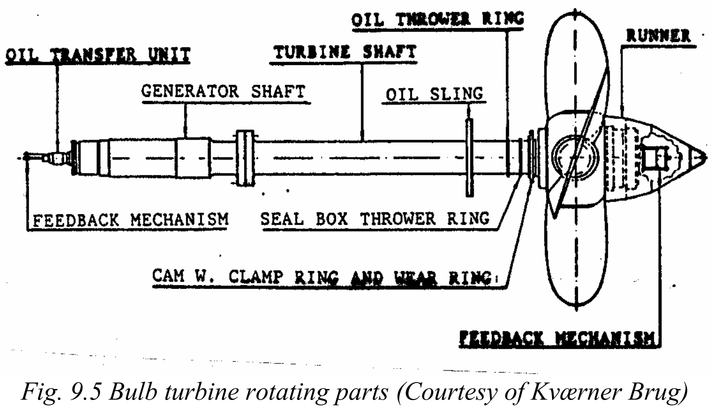


Fig. 9.5 Bulb turbine rotating parts (Courtesy of Kværner Brug)

The shields are provided with a manhole for inspection and possible maintenance of the space between them.

9.2.5 Rotating parts

The rotating parts are shown on Fig. 9.5 and consists of:

-runner
-turbine shaft

- shaft seal cam, clampring, wear ring and oil thrower ring
- turbine bearing oil thrower ring
- feedback mechanism and oil piping
- oil transfer unit from rotating to stationary parts

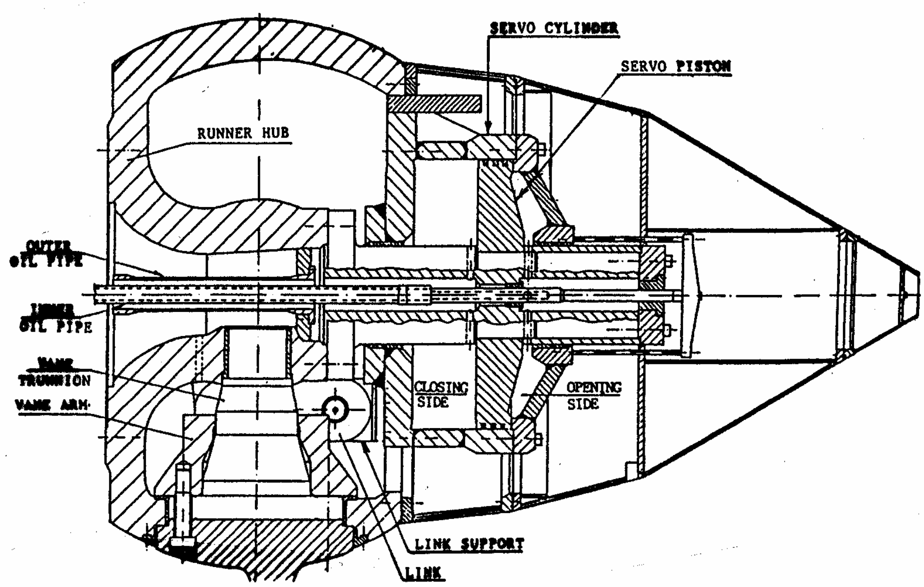


Fig. 9.6 Runner hub (Courtesy of Kværner Brug)

9.2.5.1 The runner

The runner[1] is similar to a Kaplan runner and has normally three to five blades made of stainless steel. The blades are designed with flanges and connected to trunnions and levers.

The servomotor for moving the blades is normally located inside the hub as shown on Fig. 9.6. The servomotor consists of a fixed piston and an axially moving cylinder and link supports, links and blade levers are located inside the hub.

A photo of a bulb turbine runner is shown on Fig. 9.7.

9.2.5.2 The turbine shaft

The turbine shaft is made of forged Siemens Martin steel and has flanges in both ends. One end is connected to the runner hub and the other to the generator shaft. These joints are pure friction joints.

9.2.6 Shaft seal box

Several types of shaft seal boxes are in use. One type as shown in Fig. 9.8 is however, especially for the Bulb turbines[1].

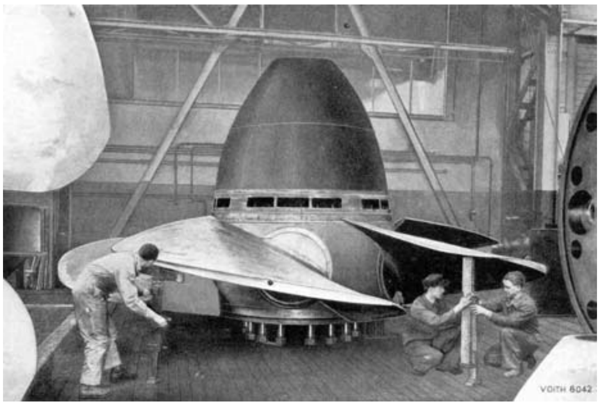


Fig.9.7 Runner assembly in the workshop

This box has radial seal surfaces consisting of a stainless hardened wear disk and two wear rings made of teflon type fibres. The wear disk is bolted to a cam fixed to the shaft. The wear rings are glued to the seal ring.This is movable and supported in the adjustment ring by means of a membrane.

The membrane allows the seal ring to move axially 5 - 6 mm. This is necessary for the shaft movement in the downstream direction when the unit is loaded. In addition allowance must be made for wear of the seal surfaces.

The adjustment ring is bolted to the support ring and may be axially adjusted by means of double acting jacking screws. According to the wear range of wear rings the adjustment range of the seal box should be 8 - 10 mm.

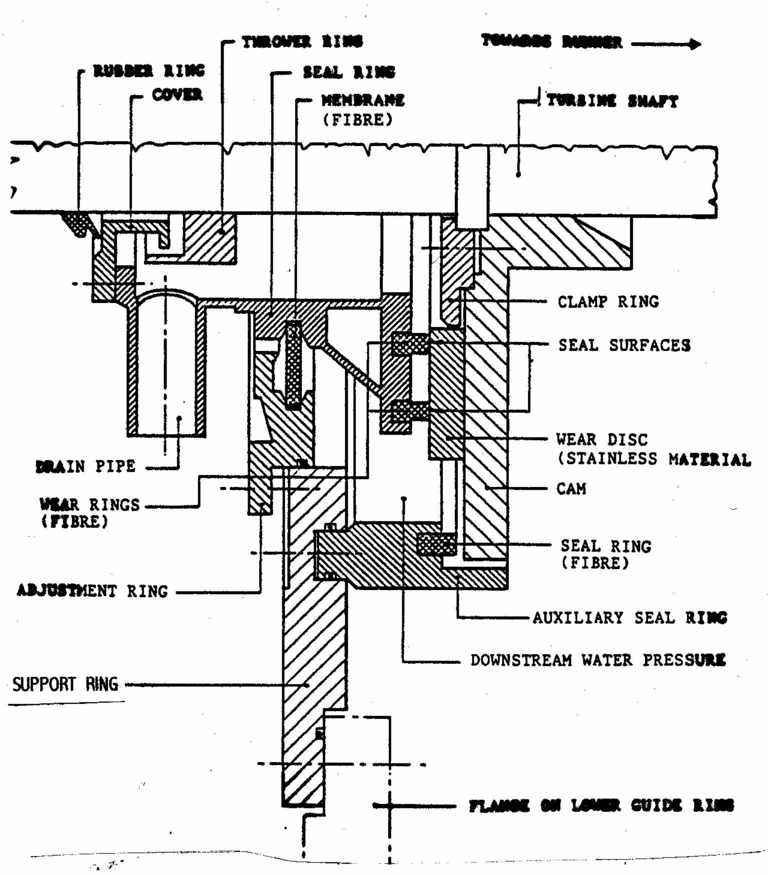


Fig. 9.8 Shaft seal box (Courtesy of Kværner Brug)

The auxiliary seal is located in the support ring. This may be pushed against/pulled away from the cam by means of push/pull jacking screws. When this ring is in contact with the cam the wear seal rings may be dismantled without draining the unit.

Possible water leakage into the seal box is drained through a pipe to the pump sump.

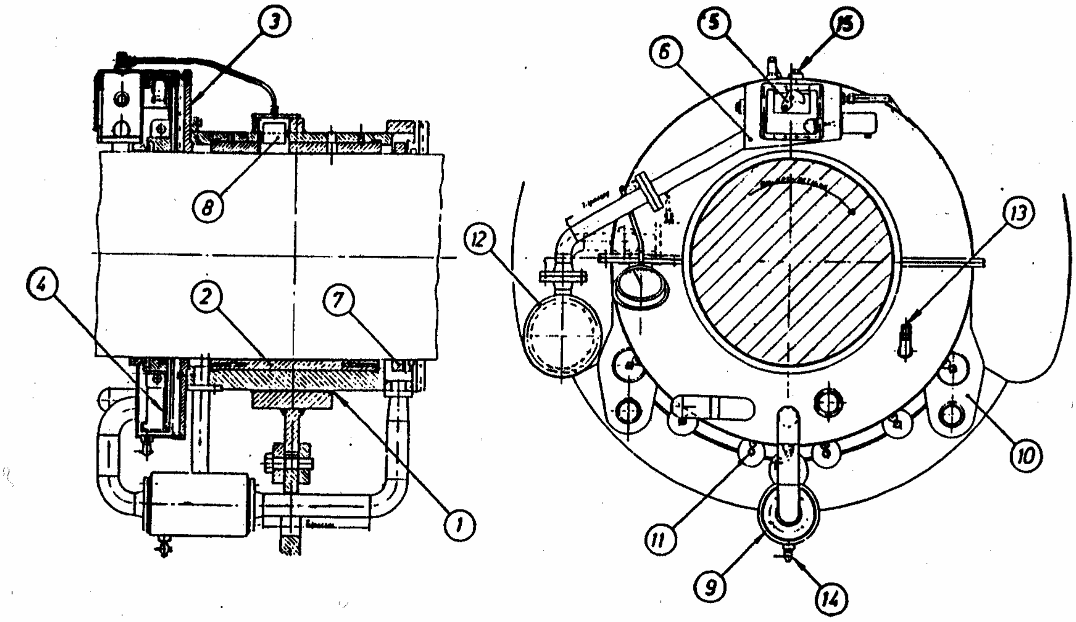
A thrower ring is mounted on the shaft to prevent water leakage along the shaft. A rubber ring is mounted on the upstream end of the shaft and seals against the seal box cover.

The seal box is provided with four springs which are pressing the wear seal rings against the seal surface to prevent leakage when the balance system is out of operation, e.g.,when filling the turbine.

9.2.7 Turbine bearing

An example of a bearing design/1/ is shown in Fig. 9.9. The bearing is sturdy and simple in operation. Maintenance will normally consist of oil change only.

The bearing housing (1) is supported in the inner guide ring Fig.9.2, by means of two yokes and two support stays and rests normally on six wedges. By moving these wedges axially the bearing housing may be vertically adjusted. The bearing housing is split horizontally.



1. Bearing housing	*5. Oil scoop*	*9. Sediment collector*	*13. Filling opening*
2. Bearing pad	*6. Float box.*	*10. Fork*	*14. Drain valve*
3. Oil housing	*7. Oil thrower ring*	*11. Wedge*	*15. Inspection filling plug*
4. Rotating oil reservoir	*8. Autom. start-lubrication unit*	*12. Oil tank*	

Fig. 9.9 Turbine bearing (Courtesy of Kværner Brug)

The bearing pad shell (2), Fig. 9.9, consists of an upper and a lower part. These are "floating" in the bearing housing where a radial locking pin in the upper part prevents the shells from rotation. The surface of the lower supporting shell and a surface in each end of the upper bearing shell are lined with babbit metal.

The oil housing (3) is bolted to the upstream end of the bearing housing. The oil reservoir (4) is fixed to the shaft. The oil scoop (5) and the box (6) are also located inside the oil housing. The float box is provided with a window for observation of the oil circulation while the turbine is running.

An oil thrower ring (7) is fixed to the shaft to prevent oil from seeping out of the bearing along the shaft in the downstream end.

The automatic start lubrication unit (8) is mounted on the top of the bearing housing. This consists of a container which is filled with oil when the turbine is running. When the shaft stops the oil content is kept in the container by a support device held by the shaft. As soon as the shaft resumes rotation the support is removed and the content is tilted. The oil in the container is then distributed on the bearing surface.

When the shaft and oil reservoir starts to rotate, the reservoir draws oil from the lower half of the oil housing. As soon as the oil layer is sufficiently thick, the oil scoop picks up oil and delivers it to the float box, then to the oil tank and the bearing pad. The rotating shaft transports this oil further to the bearing surface.

Normally more oil than required for lubrication is circulating through the oil tank. Therefore a bypass is provided for taking the excess oil back to the oil housing top. This bypass flow is controlled by a float switch inside the float box (6).

To increase the oil volume to more than the volume of the oil housing, an oil tank (12) is located beside the bearing.

The sediment collector (9) is located below the bearing housing. All dirt particles which are trapped in the oil during circulation in the bearing, shall be separated before the oil returns to the oil housing.

The bearing is provided with miscellaneous filling openings, oil level indicators, level float and temperature sensors as well.

9.2.8 The feedback mechanism and oil piping

The feedback mechanism and oil transfer piping are located in the shaft centre. The transfer piping consists of an inner and an outer concentric oil pipe running through the whole shaft length. The inner pipe continues through the oil transfer unit at the upstream end, and it is supported in the outer pipe and connected to the runner servomotor cylinder via an yoke.

The inner pipe is axially movable and follows the servomotor movement. A pointer mounted to the upstream end is moving along a measurement ruler showing mechanically the servomotor position at any time. The outer oil pipe is mounted by means of flange connections to the runner hub, turbine shaft and generator shaft respectively.

9.2.8.1 The oil transfer unit

The oil transfer unit is located at the upstream end of the generator shaft and has one fixed and one rotating part consisting of a distribution sleeve and distribution trunnion respectively.

The distribution sleeve is fixed to the capsule around the generator shaft end and is provided with pipe connections for oil supply and return as well as leakage oil. The distribution sleeve is provided with a bracket having a measurement scale where the runner servomotor position may be read.

9.2.9 The guide vane mechanism

Two different systems have been used for operating the guide vanes. Kværner Brug has designed a system where each particular vane has its own servomotor as shown in Fig. 9.10.

By means of a link ring a simultaneous movement of the pilot valves for the guide vane servomotors is achieved. The movement is governed by the valves controlling the opening/closing of the guide vanes.

How the servomotors are supplied with or drained for oil is shown in section A - A on Fig. 9.10.

The high pressure hoses are connected to the oil pressure system of the unit.

The advantages of this system is that even if one guide vane is stuck, the remaining vanes can be moved without any damage. The same will apply if a foreign object is caught and jammed between two vanes during closing, the remaining vanes can be closed without any damage. If required,

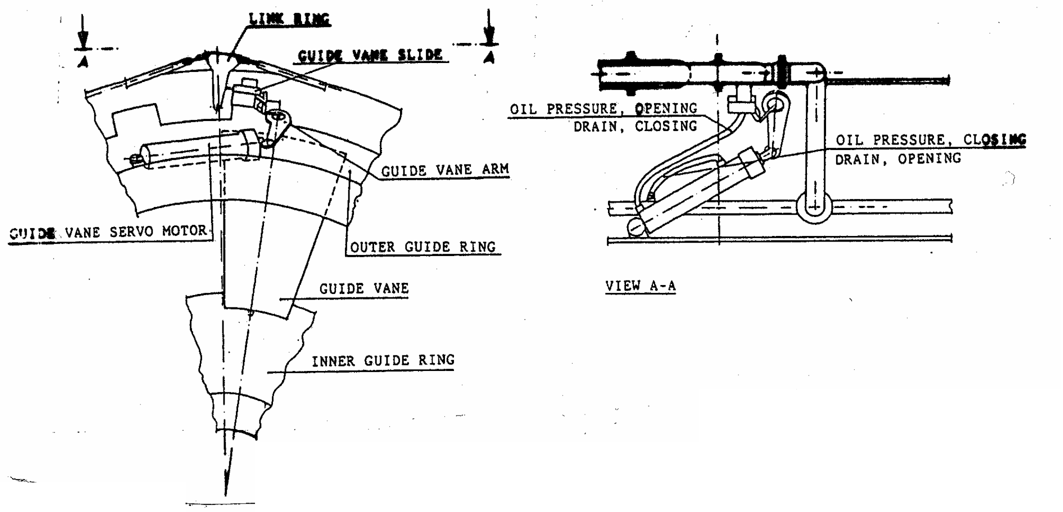


Fig. 9.10 Guide vane mechanism with single servomotors (Courtesy of Kværner Brug)

single vanes may be operated separately and thus jammed object may easily be flushed out of the guide vane system.

The disadvantages of this system are the assembly and extensive adjustment work. However, the total price will approximately be the same as for the regulating ring system which is the other method for moving the guide vanes.

The regulating ring system with links and levers is of the same type as for Francis turbines. A regulating ring with three main servomotors is shown on Fig. 9.11. Due to the conical arrangement of the guide vanes the lever link system must have spherical bearings with large angle movement.

The connection between the levers and the vanes is designed as friction joints. This is done to avoid damage to parts if one or several vanes are stuck or if foreign objects are caught between vanes. The friction joint makes it possible for the vane lever to move with the remaining parts of the guide vanes connected to the regulating ring even if the adjacent vane is stuck.

A disadvantage may be that large weight of the regulating ring and possible slack in bearings can make the governing inaccurate.

9.3 Condition control

The general principles for condition control are the same as for the Francis turbines. Further

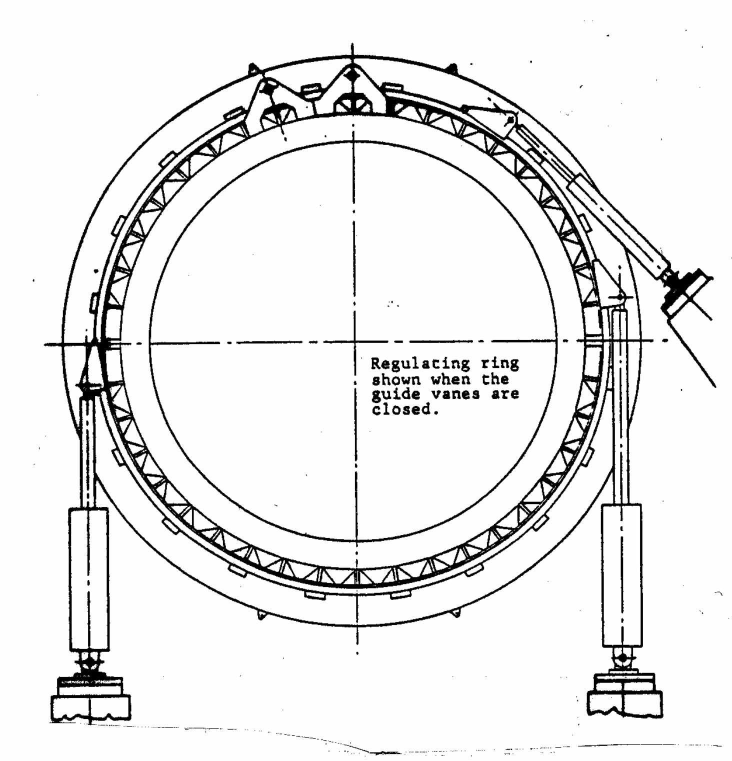


Fig. 9.11 Regulating ring /1/

considerations are therefore connected only to a few specific details.

9.3.1 Runner

The runner should be inspected both from above and below. Particular attention should be given to possible cavitation erosion and scratches on the blades as well as leaks around the blade flange against the hub.

9.3.2 Runner chamber

The narrow gap between runner and the runner chamber should be checked if foreign objects may have passed the gap and made scratches in the chamber.

9.3.3 Guide vane mechanism

For guide vane mechanism with individual vane servomotors on bulb turbines it should be checked that the vanes have an identical movement.

9.3.4 Shaft seal box

For Bulb turbines at standstill it should be checked that the water does not flow out of the box along the shaft into the turbine bearing.

9.3.5 Generally for Bulb turbines

Special attention should be paid to changes in the sound when the unit is in operation.

9.4 Monitoring instruments

The turbine is normally equipped with the following instrumentation:

- A manometer connected to the upstream end of the generator bulb
- A vacuummeter connected to the runner chamber downstream of the runner
- A contact manometer for reading of the pressure in the shaft seal box water pressure pipe, which triggers alarm or alternatively stop signal at too low pressure
- A contact manometer giving alarm:
  Pressurised water in the shaft seal box upon start of the unit
- A float switch for alarm for high water level due to too large leak in the shaft seal box
- A contact thermometer for reading of oil temperature in the bearing housing. The thermometer has two adjustable contacts, one for alarm at high temperature and one for disconnection for the unit on further temperature rise

- A remote indication thermometer with two temperature sensors for reading of temperature in lower bearing pad
- A level float for alarm on low oil level

9.5 Assembly and dismantling

Among the large main parts of the bulb turbine it is only the runner which is normally needed to dismantle. The runner chamber is split axially horizontal in two halves. By removing the upper half access to the runner is obtained.

The guide vanes cannot be dismantled without extensive work. Repairs of these and the guide surfaces should be performed at the plant.

Bearing and seal box can easily be dismantled. By applying the overhaul seal the seal box may be removed without draining the water canal. Then necessary stairs and floors around the guide vane and the runner chamber may be erected.

The stay shields are adapted against bulb and outer water conduit contour. The shields are mounted as soon as the generator bulb and penstock are completed.

Finally the generator hatch dome plate and cover are installed.

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CHAPTER 10

Governors

10.1 Governor system structure

A complete turbine governor system may be divided in three main components:

- *The controller.* This is the unit for execution of control processes. The unit may be of mechanical-hydraulic or electrohydraulic construction
- *Servo system.* The servo system is an amplifier that executes the admission changes determined by the controller.
- *The pressure oil supply system.* The principal duty for this system is at any time to supply sufficient quantities of pressure oil to the servo system.

A structure diagram of a representative turbine governor system is shown in Fig. 10.1.

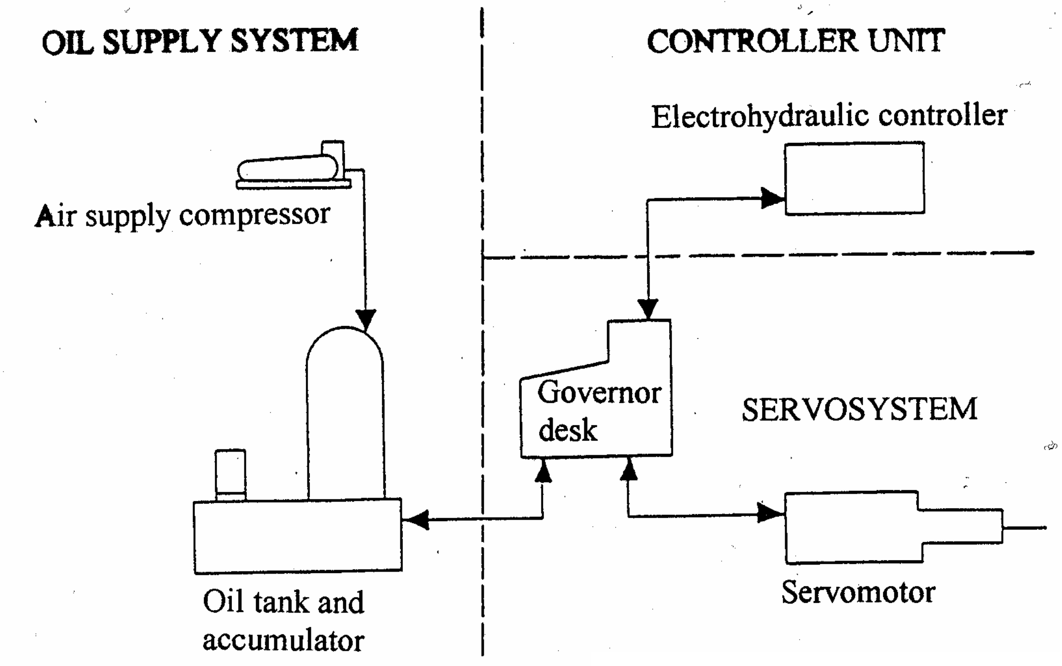

Fig. 10.1 Schematic structure of a turbine governor system /1/

Controller units and servo systems are considered in the following sections while the oil pressure supply system is dealt with in Chapter 12.

10.2 Electrohydraulic controllers

Standard Printed Circuit Boards (PCB) are normally used for electronics in the electrohydraulic controllers. The PCB's are housed in racks, which are mounted in a hinged frame. The power supply and auxiliary equipment are placed inside the governor desk.

10.2.1 Analogous controller

A typical system structure of an PCB-based controller[/1/] is shown in the simplified block diagram on Fig. 10.2. The electric system of the controller consists of the following main components:

- power supply
- frequency measurement circuit for frequency control
- PID-control unit
- servo amplifier
- position measurement of the pilot control actuator
- input/output signal for remote and local control
- sensors for signal transducers (man - machine communication)

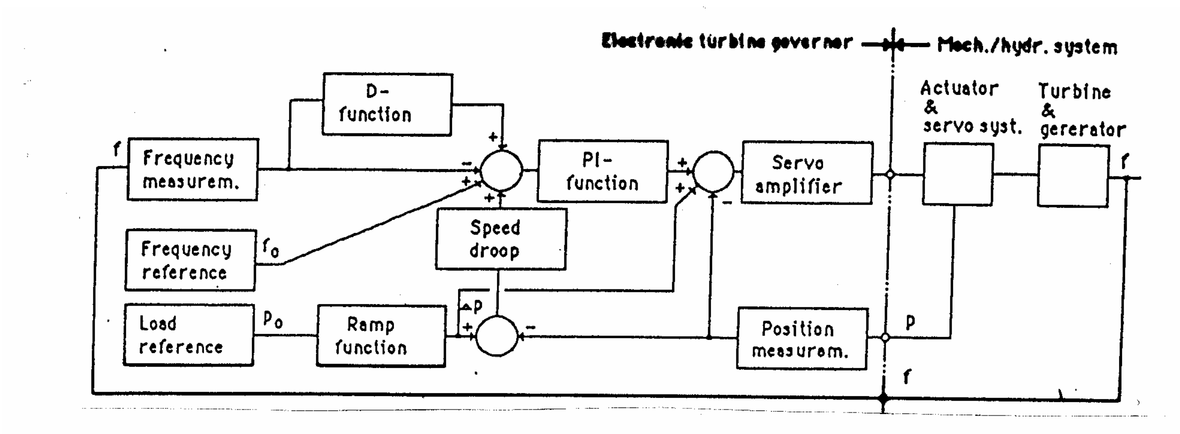

Fig. 10.2 Analogous electrohydraulic controller /1/

The controller has PID functions. Each parameter can be adjusted within a wide range. The adjustment of the amplification can be made without any influence on the time constants and vice versa.

Two inductive sensors close to each other measure the frequency. The sensors are activated by means of a segment disk attached to the turbine shaft. The pulses are received by a digital frequency measurement circuit and transformed to an analogue voltage. The analogue signal is input to the PID function block

The derivative function is influenced only by the frequency input and so giving a smooth changeover between reference changes and the frequency output.

The load reference signal is fed through a ramp function, which limits the rate of change in the reference value and thus the rate of change of the power output. The change will be linear and can be adjusted between 20 and 150 seconds per 100% load change.

The load reference signal also bypasses the PID control unit and is a direct input to the electro hydraulic actuator.

The feedback signal from the electrohydraulic actuator passes through the permanent speed droop circuit and corrects the output according to the actual frequency-load characteristics.

The output signal from the PID control unit is input to the amplifier. The amplifier also receives the feedback signal from the position transmitter on the electro hydraulic actuator.

The actuator will be regulated according to the output signal from the PID control unit. The actuator is connected to the main distributing valve which controls the main servomotor and the turbine admission.

In the “Manual” mode the mechanical-hydraulic load limiter controls the turbine power.

In “Manual” mode the supervision logic receives an alarm signal from the speed monitor at 100% speed, which results in an external stop command from the controller. This signal trips a shut-down function in the main control.

The governor parameters may be changed during operation with no system disturbances. It is also possible to switch between two pre-set parameter values depending on the operation conditions. This is done by means of a remote command.

The governor should be connected to a DC battery supply and 3-phase 50 or 60 Hz system. In addition a separate single-phase input is available for emergency power supply.

10.2.2 Digital computer based controller

A controller based on digital technique is for example a Programmable High-Speed Controller. It is designed to be used normally for all kinds and sizes of water turbines. It is completely autonomous, and is designed with a high inherent flexibility to enable individual requirements to the greatest possible extent.

A digital governor generally contains the following main parts:

- power supply
- frequency measurement
- controller, inclusive sequence control and monitoring
- servo interface
- automatic turbine admission control
- runner blade control
- options as water level control etc.

Two sensors reading a segment disc attached to the turbine shaft measure the frequency. The measurement signals are transformed to a digital code by the frequency measurement circuit. A linear relation exists between the measurement value and the cycle time of the unit frequency.

Alternatively the frequency measurement can be taken from a permanent magnet alternator. Except for the measurement signal there is no difference compared with the analogous controller.

The controller is based on a processor system. The controller hardware is generally built of standard components while the control functions are realised in a specific program.

The controller program includes the following functions:

- controller algorithm

- sequence control for start, stop, interlocking, etc.
- monitoring, process and self monitoring

Fig. 10.3 shows the block diagram/1/ for a typical controller of a Francis turbine. The same block diagram will also be the basic part of a controller for a Pelton and Kaplan/bulb turbine.

The controller has a PID function. Each parameter can be set within a wide range. Adjustment of one parameter makes no influence on the others.

The servo interface is the joint between the electronic part and the hydraulic part of the governor.It is a part of the electrohydraulic position control of the actuator. The loop is closed in the processor.

One or several position control loops are provided depending on the version. Each loop contains a servo interface with a servo amplifier and a transducer for the actuator position measurement.

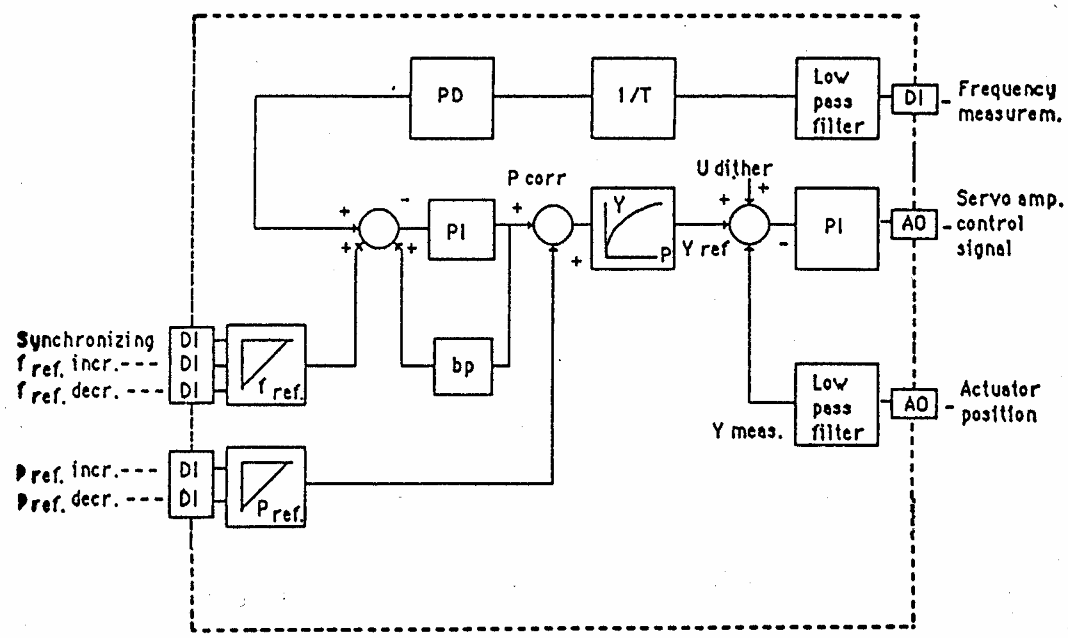

Fig.10.3 Turbine controller block diagram (Courtesy Kværner Brug as)

10.3 Servo system

10.3.1 Governor desk

The system structure in a turbine governor desk/1/ is illustrated in Fig. 10.4.

The main components are:

- electrohydraulic actuator (servovalve and pilot control actuator)
- opening control valve
- rapid shut-down valve
- main control valve
- mechanical-hydraulic load limiter
- mechanical feedback
- control levers
- oil filters

- switches and sensors
- measuring device for the rotational speed

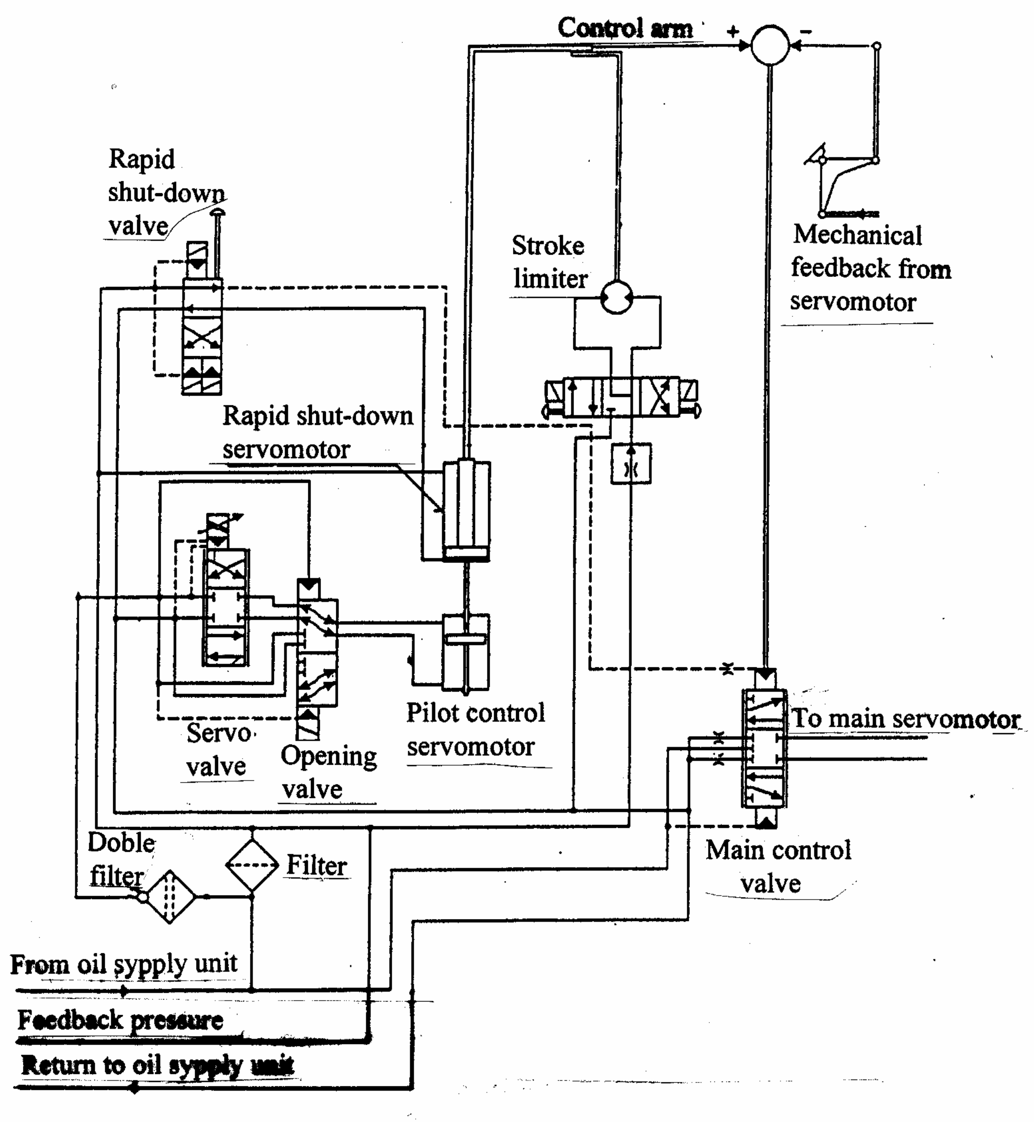

Fig. 10.4 Governor desk, hydraulic circuits /1/

The governor desk is an independent unit located near to the main servomotor. Thereby short pipe lengths between the desk and the servomotors as well as plain feedback mechanisms are obtained.

The hydraulic components in the governor desk are fed from the oil supply system. The output is pressure oil to the main servomotor which is mechanically connected with the feedback and the control lever on the main control valve.

In some new turbine governors it is an electronic feedback signal for the servomotor position. The load-limit function is carried out by the controller electronics and the servomotor is directly governed by the servovalve.

In this way the main governing hydraulics is compressed to a structure containing the servovalve, emergency shut down valves and the servomotor.

10.3.2 Main control servomotor

The main control servomotor consists of:

- cylinder
- piston and rod
- sealbox with bushing and sealbox ring

An ordinary design of the servomotor/1/ is shown in Fig. 10.5

Cylinders are usually made of steel as casted or welded products.

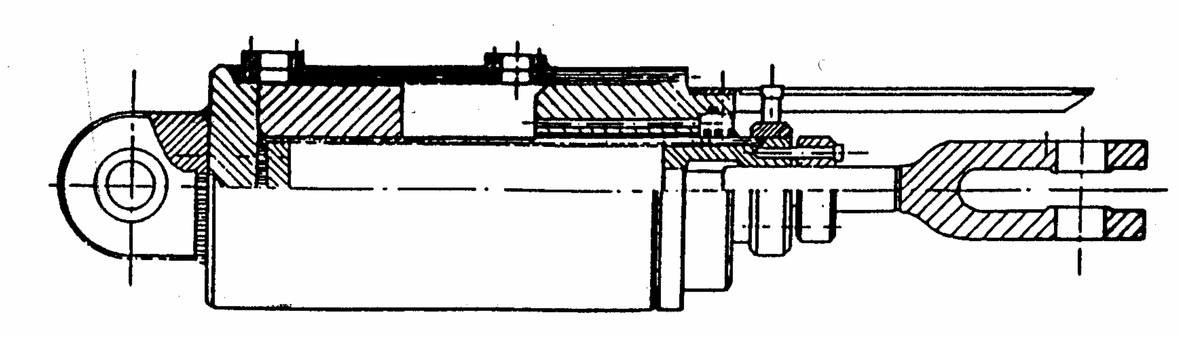

Fig. 10.5 Typical design of a servomotor /1/

10.4 Specific turbine governing equipment

10.4.1 Dual control of Pelton turbines

The governing of Pelton turbines is normally carried out with a dual control, e.g., needles in the nozzles and the deflectors. These components are described in Chapter 6.

By minor load changes the needle adjustment control is satisfying the control requirements alone.

By rapid load rejections however, the rotational speed rise is controlled and limited by activation of the deflector. The servomotor gives the deflector a rotary movement which bends the water jet away from the runner.

The sequence controlled nozzle follows the movement of the deflector servomotor by adjustment of the needle position until the discharge corresponds to the new power/load equilibrium. In this approach to equilibrium the deflector moves gradually out of the jet again to an idle position just outside the periphery of the jet.

The deflector function is controlled from the governor desk via the main control valve. The servomotor movement is transferred via bars, levers and links to the deflectors.

From the servomotor movement feedback signal is transferred to the control valve.

The sequence control of each needle is carried out via a cam disk which is driven by the deflector servomotor. The cam shape accomplishes an input function to the needle control valve which opens for adjusting the needle to correct position.

The feedback from the needle movement controls the valve openings until they reach neutral position and the needle has reached the correct position according to the reference.

The cabinet for the needle control is located as near to the nozzles as possible to obtain a plain feedback system from the needles and short oil pressure pipelines.

In some recent designs of Pelton governors electronics and separate electrohydraulic servosystems carry out the controller functions of the deflectors and the needles. Electronics is also utilised in the feedback systems. In this way the construction of the servomotors with mechanical levers and links have become more simplified.

10.4.2 By-pass control for Francis turbines

10.4.2.1 Function and general arrangement

The regulation following load rejections in high head Francis turbine plants makes it necessary to divert some parts of the instantaneous water flow to the turbine. In this way it is possible to obtain a rapid closure of the turbine admission and to retard the main flow in the penstock to minimise the pressure rise.

The diversion is normally branched off from the scroll casing[/1/] as shown schematically in Fig. 10.6. In this branched off system *the by-pass valve* is installed and its admission is controlled by the turbine governor. The guide vane movement controls the valve opening. This combined control of the guide vane movement and the by-pass through the valve makes it possible to control both the pressure rise and speed rise during load rejection.

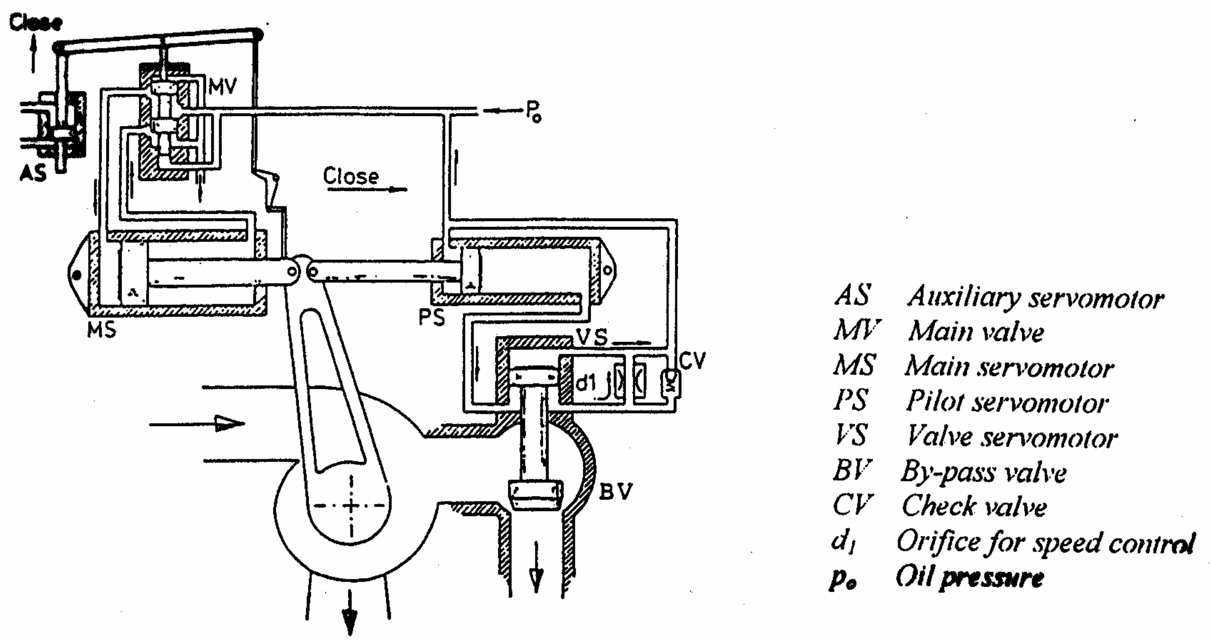

Fig. 10.6 By-pass valve with regulating system /1/

The water flow through the valve is led into an energy dissipater and then into the turbine draft tube as shown in Fig. 10.7.

A Norwegian turbine manufacturer has used this type of by-pass valve system for several years, and a long experience in the use of this equipment has proved a simple and reliable system.

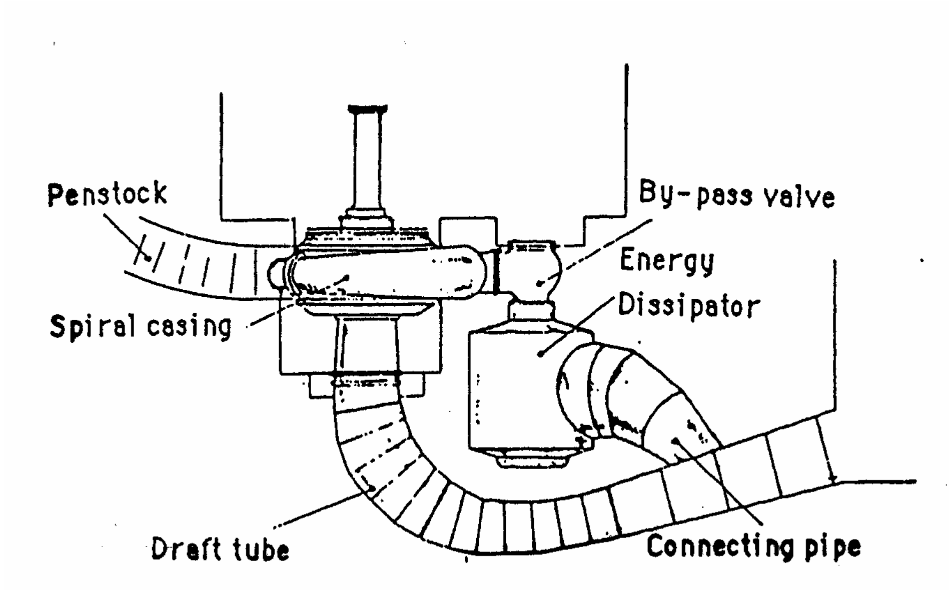

Fig. 10.7 Energy dissipater /1/

A vital point in the design of the control system has been to ensure full control of the pressure rise even if the valve should fail to operate. In this case the closing speed of the guide vanes will correspond to the speed given by the allowable pressure rise. The speed will then be higher than normal, but this is regarded as less serious because the generator is designed to withstand short lasting runaway speed of the unit.

10.4.2.2 The valve control system

A schematic diagram of the valve system and main servomotor control[/1/] is shown in Fig. 10.6. The auxiliary servomotor AS, via the main valve MV controls the guide vane servomotor MS. The double acting servomotor MS, moves the guide vane via the regulating ring. A pilot servomotor PS, is connected to this ring and copies the position according to the movement of MS.

The servomotor PS is pressurised via the orifice d_l from oil pressure supply p_o. The by-pass valve servomotor VS is hydraulically connected to PS. Under stationary conditions it moves to closed position because of the difference between the two sides of the piston areas.

During a closing movement of MS the oil from PS passes through d_l into the accumulator. If the closing speed exceeds a certain value, the pressure on the opening area on VS increases because of the orifice d_l, and VS then opens.

To avoid restriction of the guide vanes a check valve CV, is connected in parallel to d_l.

The size of the volume of VS relative to PS is given by the size of the opening of the by-pass valve at total flow capacity.

In the diagram Fig. 10.8 the relative movement between the guide vane servomotor MS and the by-pass valve BV are shown. Because of the close linearity between servomotor position and the water flow, this diagram also indicates the water flow in the system.

From this diagram it is seen that the guide vanes start closing at maximum speed $1/T_c$. The by-pass valve opens at a speed $1/T_o$ given by the relative servomotor volumes.

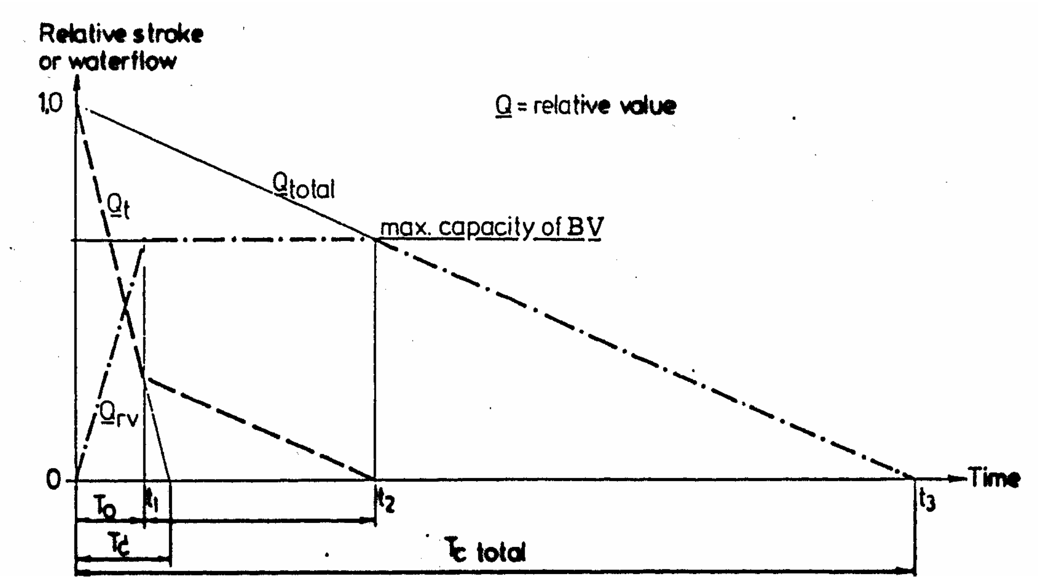

Fig. 10.8 Relative movements and water flow between guide vane cascade and by-pass valve /1/

When the by-pass valve is fully open at t_1, the closing speed of the guide vanes is reduced to a lower value. In closed guide vane position at the time t_2, the pilot servomotor PS stops and the pressure becomes equal p_o on both sides of the piston in valve servomotor VS. Then VS starts closing the by-pass valve because of the piston area difference.

The closing speed of VS shown as $1/T_{c\ total}$ in the diagram, is controlled by d_1. The oil volume from the opening side of VS is forced back to the accumulator.

The total water flow in the penstock is given by the sum of the flow through the guide vane cascade and the flow through the by-pass valve. This controls the pressure rise in the system.

10.4.3 Dual control of Kaplan/Bulb turbines

Optimal efficiencies of Kaplan and bulb turbines are obtained by optimal combination of the functions of the guide vane cascade control and the runner blade control. Examples of the construction of these control systems are described in Chapters 8 and 9.

The combination of the two control functions may be carried out either by mechanical-hydraulic or by electrohydraulic operation. The combination unit is usually located on the top or beside the top of the turbine-generator unit.

A mechanical-hydraulic combination unit is integrated in the runner blade control system and consists of:

- main valve
- feedback mechanism
- combination control function curve
- pipe connections to the oil supply unit

The combination control function is:

- distribute oil for operation of the runner blades via the main valve

- position the runner blades according to the control function curve disk which is governed by the guide vane control
- feedback of the spool position in the main valve.

An electrohydraulic combination unit has an electrical feedback from the position of the guide vane cascade and a combination control function in this case is produced electronically. The servomotor is then operated by an electrohydraulic servovalve.

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CHAPTER 11

Valves

Introduction

The conduits in all water power plants except large low head plants, are ordinarily provided with shut off devices. Generally these components are valves. They exist in different types and design depending on function and requirements.

In any power plant valves for different purposes are usually needed. Normally it is a shut-off valve just in front of the turbine. In this way the turbine may be emptied without emptying the shaft or penstock. In addition the guide vane cascade is depresserised so that leakage flow is avoided.

With a long head race tunnel and surge chamber it is normal to have a shut off valve just downstream of the surge chamber. In this way the shaft or penstock may be emptied without emptying the tunnel.

To prevent large damage at an eventual rupture of the penstock, a pipe break valve is normally installed in the pipe just downstream of the shut off valve. This valve closes automatically when the water velocity exceeds a certain set value.

The most relevant types of valves are:

- spherical valves
- butterfly valves
- gate valves
- ring valves

11.1 Spherical valves

Spherical valves are applied mostly as shut off valves in front of high head water turbines. They are however, used as pipe brake valves as well. Spherical valves are presently covering a pressure range of 160 m to 1250 m water head.

The spherical valves consist of the valve housing with flanges, valve rotor, bearings and seals.

11.1.1 Valve housing and valve rotor

The valve housing has a spherical shape. It may either be axially split permanently in two halves and bolted together with heavy flanges, or these two halves may be welded together after the rotor has been installed.

The rotor inside the valve housing is principally shaped spherically as well. The circular passage through the rotor is equal to the size of the valve inlet and outlet. The length of the passage is dimensioned to fit almost close to the inner end faces at inlet and outlet of the valve housing. In this way the flow losses through the valve become about the same as in a corresponding pipe length.

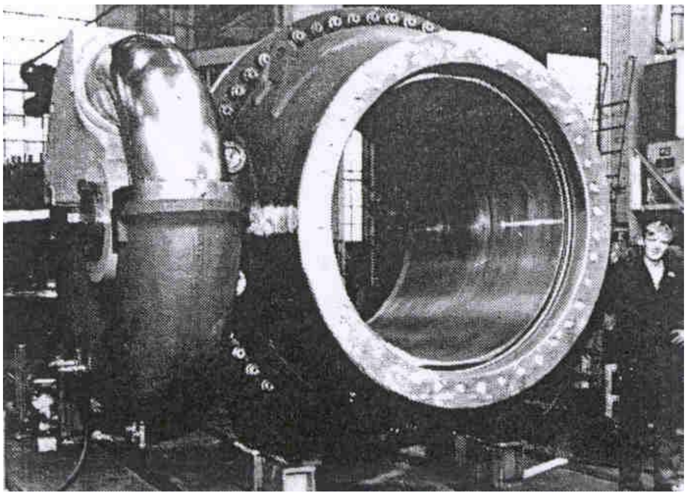

Fig. 11.1 Spherical valve in open position

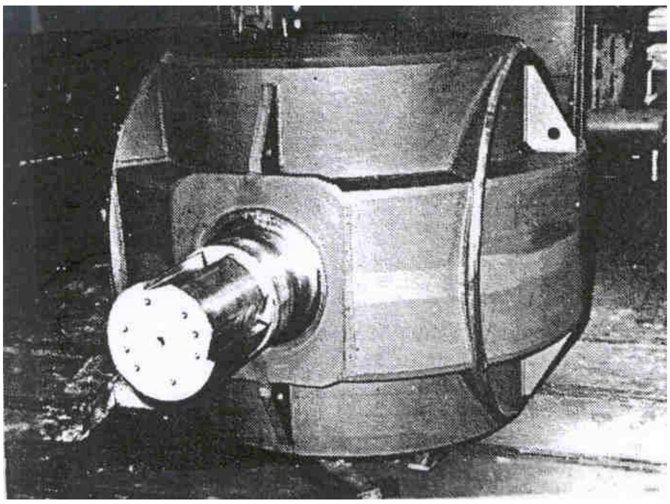

Fig. 11.2 Valve rotor (Courtesy of Kværner Brug)

Fig. 11.1 shows a picture of a complete spherical valve in open position, and Fig. 11.2 shows a picture of a spherical valve rotor.

11.1.2 Valve rotor trunnions and bearings

The trunnions on the valve rotor are horizontally supported in the valve housing. The rotor is turning on these trunnions by an angle 90^{o} from open to closed position.

The valve is normally designed to close under the worst possible conditions. This worst case is to close a valve upstream of the turbine when the guide vane cascade of the turbine is fully open.

The bearings are designed with bushings made of lead-bronze. The bushing is pressed into a bearing housing and provided with two sealings as shown on Fig. 11.3. The inner seal is an O-ring primarily for keeping the bearing free from sand and mud. If this seal after some time is not leak tight, the outer seal is a reserve. This seal is of U-seal type.

Lubrication grease to the bearings is pressed through several borings into a section of grooves in the bearing surface.

11.1.3 Seals for closed valve

Spherical valves[/2/] usually have a main seal downstream and an auxiliary seal upstream of the passage of the rotor. The valve rotor is provided with two seal rings, one downstream and one upstream. These are made of stainless steel and fixed with screws to the rotor.

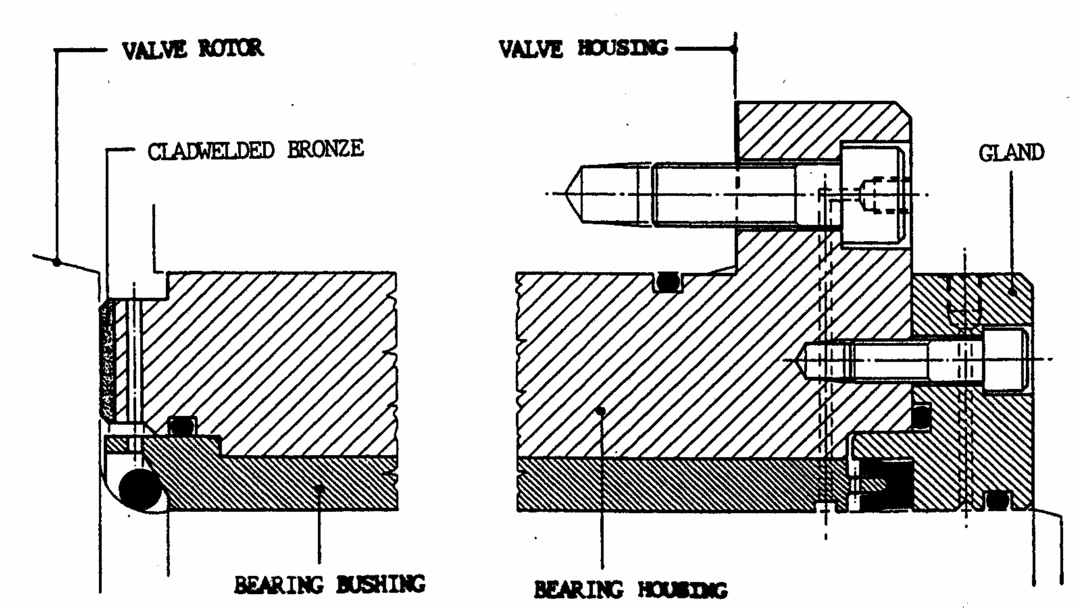

Fig. 11.3 Spherical valve bearing /2/

In closed position the circular passage of the rotor is turned 90° from the open position and the seals are then activated.

The main seal is shown on Fig. 11.4. This consists of a moveable rubber profile contained in a stainless steel seal housing which is bolted to the intermediate ring. The auxiliary seal located on the upstream side of the valve, is shown on Fig. 11.5. This consists of a moveable seal ring made of stainless steel and is operating as a differential piston.

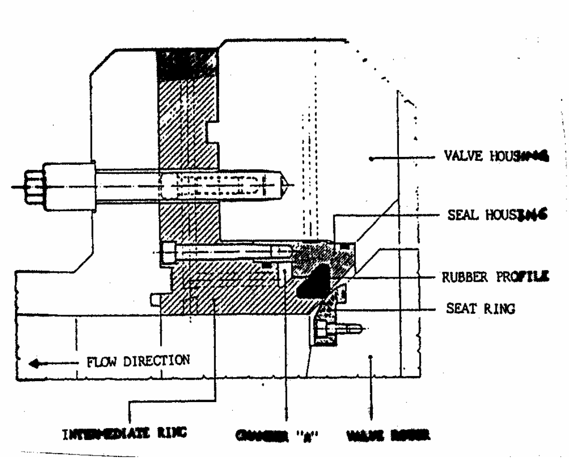

Fig.11.4 Main seal /2/

11.1.3.1 Main seal

The main seal activation is functioning as follows. If water pressure is equal from all sides, the rubber seal ring is resting in its groove as if no pressure was applied. The seal is pressed against the seat rings with a compression of 0.5 - 1 mm when external pressure on the seal is at atmospheric level.

When the valve rotor is in closed position, the rubber seal is pressed against the valve rotor seal surface by applying water pressure to the rear side “A” of the ring, shown on Fig. 11.4. If the downstream pipe in this case is empty and depressurised to atmospheric pressure, the valve is droptight.

To open the valve the downstream pipe is pressurised again through a bypass filling pipe and the rear side of the rubber seal is drained. Then the rubber ring profile is pushed back by the water pressure inside the valve. Thereby a gap between the profile and the seal surfaces is attained, and the valve rotor can rotate without touching the rubber seal profile.

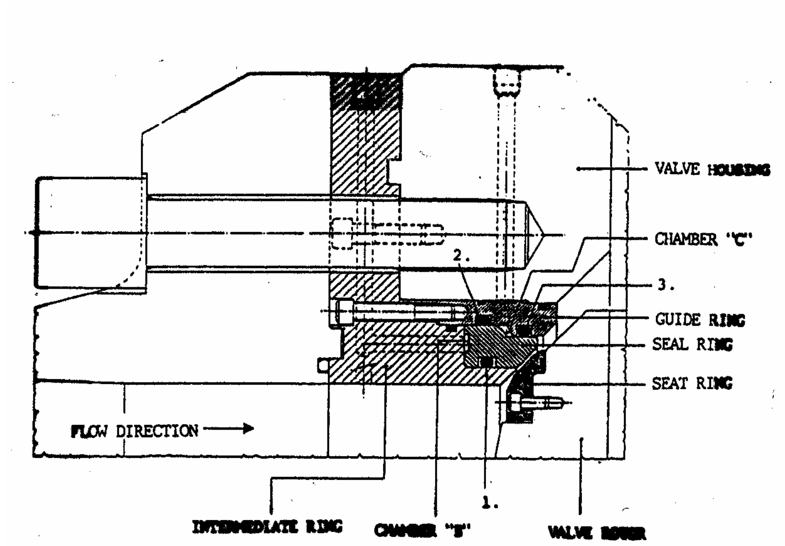

Fig. 11.5 Auxiliary seal /2/

11.1.3.2 Auxiliary seal

The auxiliary seal[/2/] is activated by supplying water pressure to chamber "B" in Fig. 11.5. The seal ring is then moved against the valve rotor seat ring. Chamber "C" is always connected to drain. The water pressure in the penstock and the spherical valve housing is pushing the seal ring back when chamber B is drained.

If the penstock has been drained with the auxiliary seal activated, the seal ring can be withdrawn by supplying water pressure to chamber C.

It is should be notified that the auxiliary seal must be activated only when the valve rotor is in closed position

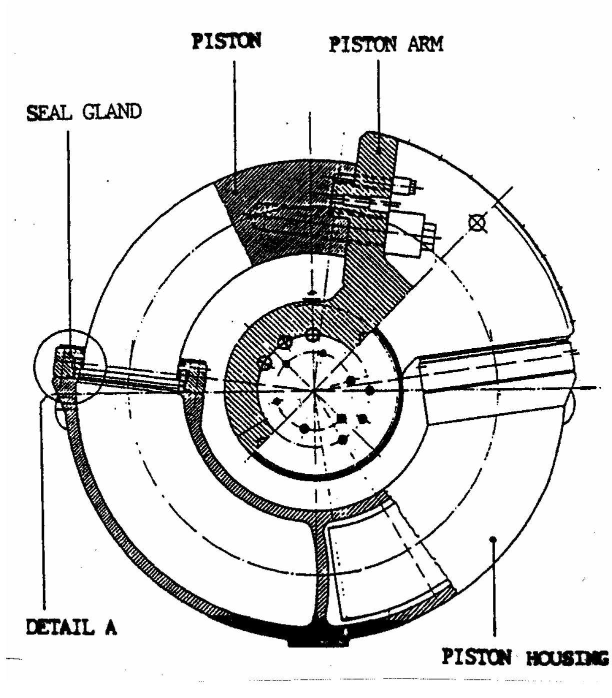

Fig. 11.6 Ring servomotor /2/

11.1.4 Operation mechanism

The opening and closing operation of the valve is carried out by one or two servomotors. Fig.11.6 shows the design of a ring servomotor[/2/] which is mounted solely on the valve housing.

Since the valve is designed to close against the flow at full turbine load, the valve rotor is subject to a large torque. This torque is transferred to the valve housing through the foundation of the servomotor. Therefore the total closing torque is absorbed by the pipe and the valve.

This design of the valve operation facilitates the foundation and dimensioning of flanges and connecting pipes.

The described operation mechanism design is the most applied in Norway.

Another type of the operation mechanism is to use a linear servomotor connected to an arm which is wedged to the valve rotor trunnions as shown on Fig. 11.7. In this case the servomotor and the valve housing have to be anchored to the surrounding foundations.

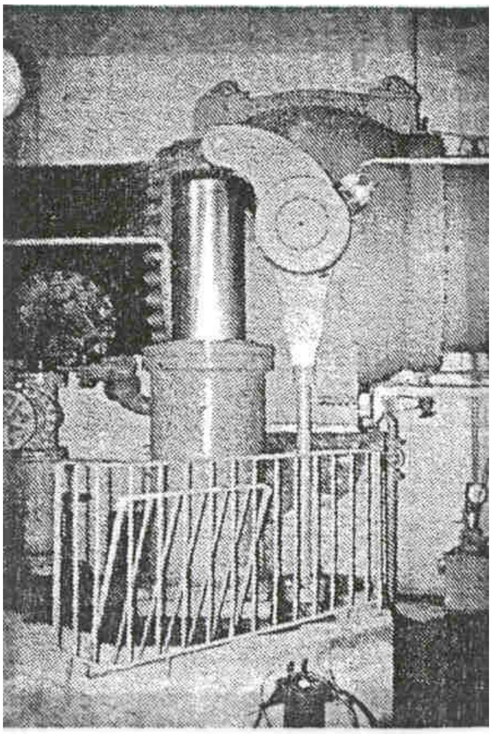

Fig. 11.7 Spherical valve operated by a

With this arrangement the adjoining pipes have to be dimensioned to withstand bending and torque forces in addition to the forces from the closed valve.

During valve closing however, the valve rotor is subject to a closing torque caused by the flow and a bearing friction torque caused by the support forces acting in the opposite direction. The sum of these torques will act partly in the closing direction and partly in the opposite direction. Therefore the servomotor must be operative for action both as a brake and as a motor for the valve rotor.

11.1.5 Control system

The control system is in principle built up/2/ as shown on Fig. 11.8. It is made up by a pilot control system located in "Spherical Valve Control Desk" and a main control system located in the "Control Cabinet".

These two systems represent the automatic part of the control system. In addition the control system is provided with manually operated valves. By means of these valves it is possible to lock a closed spherical

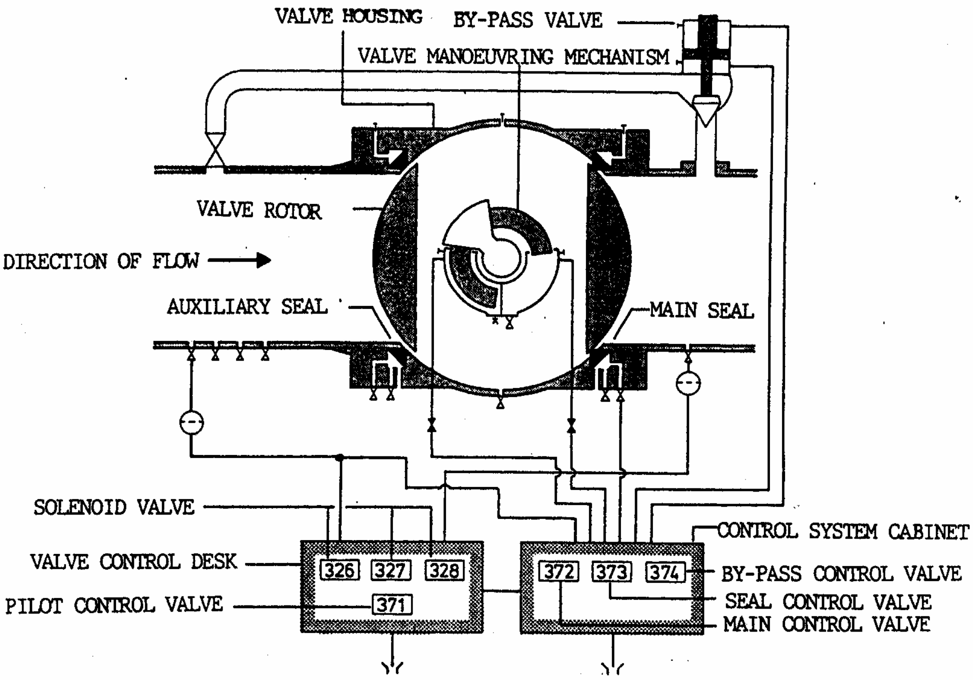

Fig. 11.8 Control system of a spherical valve /2/

valve and to shut off the automatic control system to operate the spherical valve manually.

The control system of a spherical valve is normally operated with the pressure water from the penstock. The water is carefully filtered and only rustproof materials are used in control and shut-off valves. This system of operation has proved to be convenient and reliable.

Alternatively the servomotor may be operated by means of oil hydraulics. This way of operation however, will be more complicated and expensive.

11.2 Butterfly valves

Butterfly valves are normally applied in front of low and medium head water turbines, i.e. heads up to 200 m. For high head power plants the butterfly valve is from time to time used as a closing device in inlet tunnels and alternatively as emergency closure valves.

Butterfly valves consist of mainly of a ring shaped housing, the valve disc, operating mechanism and counter weight.

11.2.1 Valve housing

The butterfly valve housing is a cylindrical ring with the same diameter as the connecting pipe. The housing is either not split or split axially. The valve ring with end flanges is mostly a welded design. In addition bosses for the disc trunnion supports are welded to both sides of the housing ring. Ribs are normally welded to the outer side of the housing ring in order to improve its stiffness.

It is favourable to cone the valve ring housing to a smaller diameter from the upstream to the downstream end. The effect of this cone is lower flow losses through the valve.

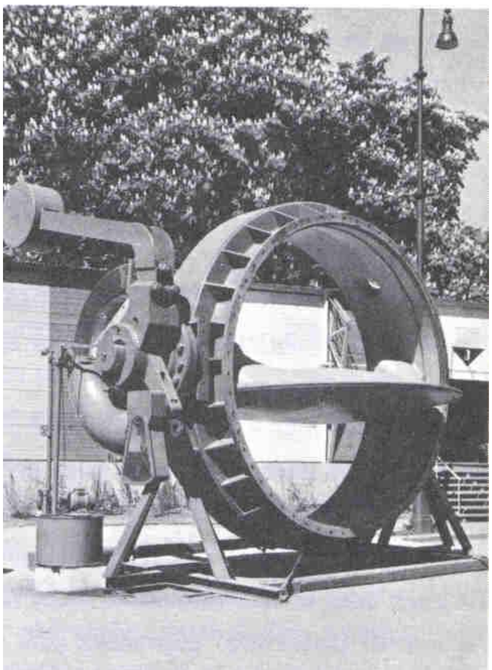

Fig. 11.9 ***Butterfly valve (Courtesy Kvæner Brug)***

11.2.2 Valve disc

Inside the valve ring housing a revolving circular disc with two trunnions, is supported in bearings in the ring housing. This valve disc is a plane with a packing around the cicumference. In the fully open valve the disc plane is parallel to the water flow. Fig. 11.9 shows a picture of a butterfly valve with the disc in open position.

A plane disc plate would however, in most cases be too weak to carry the load of the water pressure when the valve is closed. The disc therefore is often cast with a shape as a lens with internal ribs. For the larger sizes of these valves the disc may be made as a biplane structure to be sufficiently stiff and cause low flow losses. An example of a such design is shown in the butterfly valve on Fig. 11.10. In this case the disc is usually welded from plates with integrally welded boss with trunnions.

The trunnions are offset to the upstream side of the valve disc (considered in the closed position). In this way the seal between the disc and the valve housing ring may be a complete ring without joints at the trunnions. This facilitates the achievement of complete tightness.

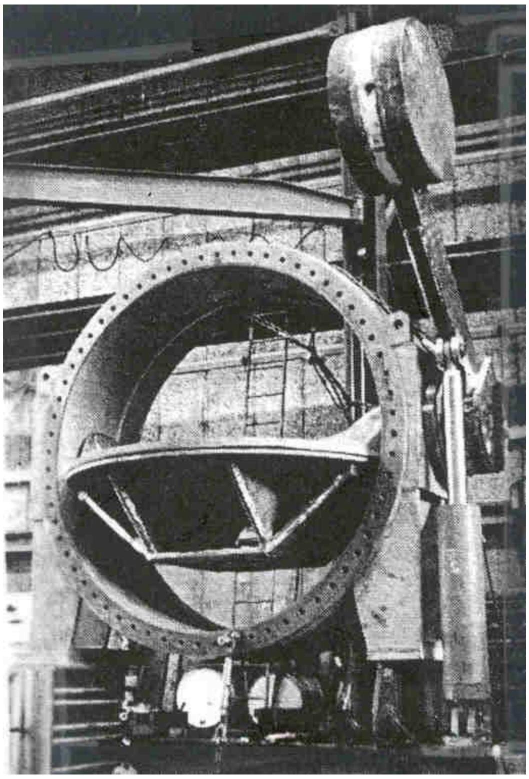

Fig. 11.10 Butterfly valve with a biplane disc (Courtesy of Kvæner Brug)

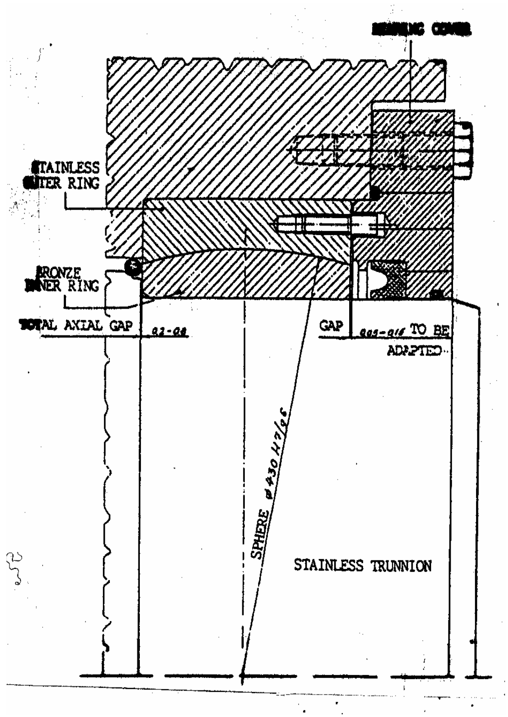

Fig.11.11 Butterfly valve bearing /2/

The disc trunnions are also offset somewhat either below or above the valve ring centre. This is called the butterfly valve's eccentricity.

11.2.3 Bearing

The valve disc has a bending tendency when it is in the closed position. Spherical bearings[/2/] are therefore normally used for the trunnions. They are rotating in a bronze bearing with spherical shape as shown on Fig. 11.11.

The bronze bearing is supported in a bearing ring made of steel or cast iron with internal spherical shape. The bronze bearing is lubricated by grease, and may also be designed on the base of self lubricating material.

The seal around the trunnion is located in the bearing cover. The seal consists of an U-shaped rubber profile and an O-ring located outside this profile. Replacement of the bearing seals requires draining of the penstock.

11.2.4 Seal

The valve housing ring[/2/] contains a machined stainless seat. In a groove in the disc a rubber profile is inserted and kept in position by a clamp ring as shown on Fig. 11.12. With the disc in the closed position, the rubber is pressed outwards against the stainless seat by tightening the clamp ring

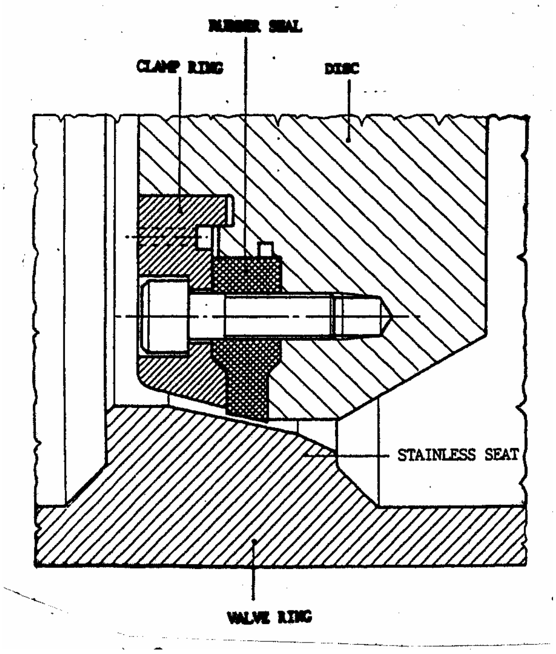

Fig. 11.12 Disc seal of the butterfly valve /2/

screws. In this way the butterfly valve will be completely tight. If the seal should leak, this may be stopped under full upstream water pressure, by tightening the screws in the leaking part of the circumference.

11.2.5 Operating mechanism

The butterfly valve shall be able to open and close under equalised water pressure on the disc sides as well as to close at full turbine discharge. In addition emergency closure valves shall close automatically in the case of penstock rupture.

The opening is done by means of one (or two) servomotors. This may be mounted on the side of the valve and is acting on the counterweight arm which is bolted to the trunnion. The servomotor may also be mounted on the top of the valve housing ring, and in this case the piston rod is acting directly on the valve disc.

The servomotor is operated by means of pressure oil from an external oil hydraulic unit.

The valve is kept in open position by oil pressure in the servomotor when the turbine is running. As the closing function is more important than the opening function, the valves are designed for closing at any conditions without supply oil pressure. The counterweight with the arm bolted to the trunnion, closes the valve when the flow is zero. The servomotor is then acting as a brake and ensures controlled closing.

11.4 Gate valves

Gate valves are also applied as shut off valves in front of turbines. The spherical valves however, have come more and more in use instead. The reason is that the spherical valves need smaller space and cause lower flow losses than the gate valves. Therefore the gate valves are produced merely with diameters up to 750 mm. They are mostly produced with cast housing. For lower heads however, some of these valves are produced with welded housings too.

An ordinary gate valve design with a double acting servocylinder is shown on Fig. 11.13. Common for gate valves is linear operated gate along a perpendicular to the longitudinal axis.

When the valve is fully open the gate is pulled away from the water canal. The movement of the gate is conducted by guides in the valve housing. When the valve is closed, the gate is pressed against the downstream stainless seal surface. The seal surface on the gate is also made of stainless material. In the closed position the water pressure acting on the gate is transferred directly to the valve seal. Increasing water pressure therefore results in increasing seal pressure.

The seal surface is inclined in relation to the gate closing direction. By closing movement the gate meets the seal surface just prior to fully closed position and slides against the seal surface the last part of the movement. To avoid bending between piston rod and the gate, the gate is mounted to the piston with a certain freedom of movement.

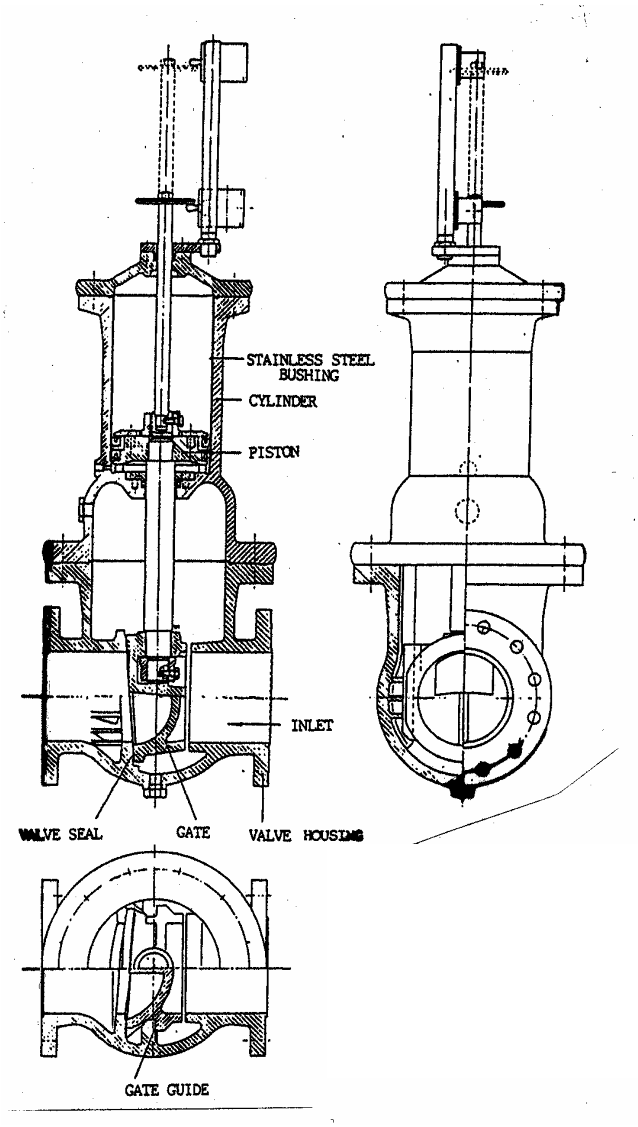

Fig. 11.13 Gate valve with double acting hydraulic cylinder

For opening and closing of the valve the operating mechanism may be hydraulic, electrical or manual. On larger valves it is natural to use hydraulic control due to the large forces involved. For the hydraulic operation water pressure from the penstock or oil pressure from a hydraulic power unit is used.

If water pressure from the penstock is used, the cylinder wall has an inner layer of rustproof material. In this case the piston normally has a leather gasket against the cylinder wall. The water must pass a filter and be carefully cleaned before entering the cylinder. For smaller remote controlled valves electrical operation is an alternative.

11.5 Ring valves

Ring valves have been used in many types of hydro power applications. In pump storage plants they are the most applied valve for closing the outlet of the pumps. However, they are applied also as pipe break valves, drain valves and safety valves.

An example of the design of a ring valve[/2/] is shown in Fig. 11.14. This has a piston shaped closing device which is moved axially when opening and closing. The valve housing is partly conical and partly cylindrical, and together with the closing device it forms a ring shaped flow cross section. The closing device is supported in the housing by ribs.

The closing device consists of a piston with piston rod connected to a cylindrical member. This member has a seal ring fastened to the dowstream end of it. The seal ring is made of stainless steel and designed to adapt to the housing seal ring in the closed position. The piston and piston rod are centered by a bushing at each end.

The design shown on Fig. 11.14 is operated by water pressure from the penstock through the connection A and B, and there is also a connection for emergency closing C. Some ring valves are using oil as operating fluid. For this operation a hydraulic power unit is needed.

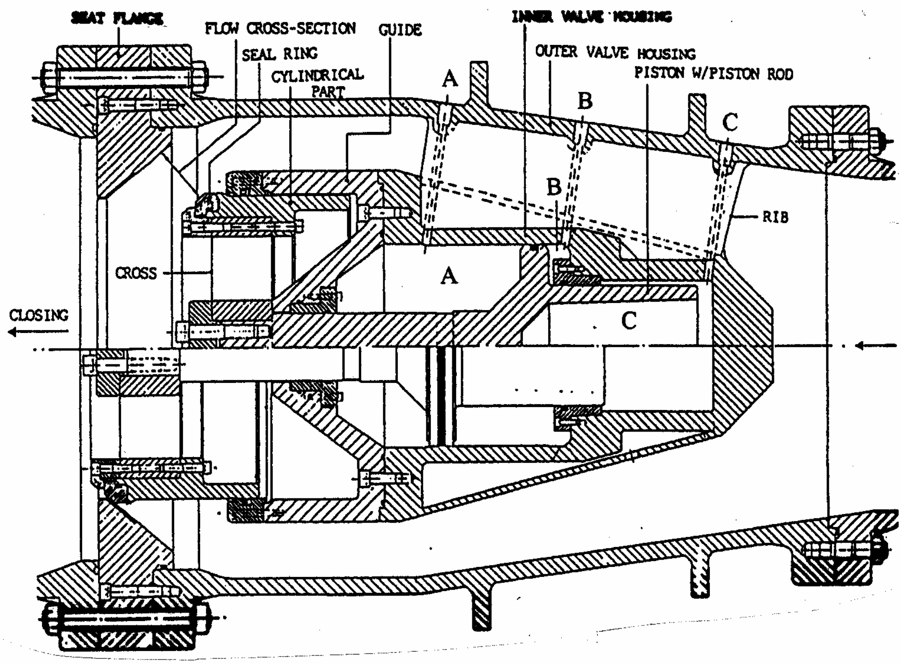

Fig. 11.14 Ring valve section /2/

Ring valves installed in drain conduits[/2/] are equipped with an auxiliary valve upstream and a pressure reducing device downstream of the ring valve. The pressure reducing device is an energy dissipator connected with the downstream end of the ring valve as shown in Fig. 11.15.

The energy dissipator consists of consentric plates with a large number of small axial holes. By this arrangement the pressure drop at any point is kept below a level where cavitation may occur. For lower heads than 75 meters ring valves are delivered without pressure reduction device.

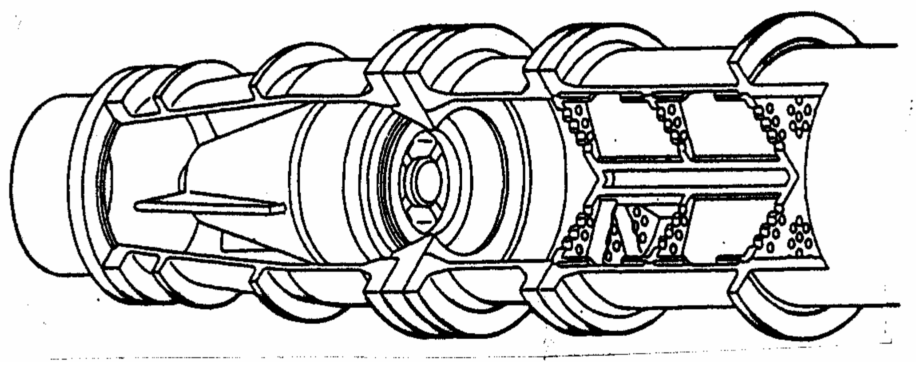

Fig.11.15 Ring valve with energy dissipator /2/

11.6 The bypass valve

A bypass valve is the opening/closing device in a pipe connecting the upstream and downstream side of the valve in the main conduit. The arrangement of a bypass valve is schematically shown in Fig. 11.8. This bypass is used to equalize the pressure on the two sides of the main valve to relieve the valve from large loads during normal opening and closing. The bypass is also used to fill the scroll casing of Francis turbines and the distribution pipe of Pelton turbines

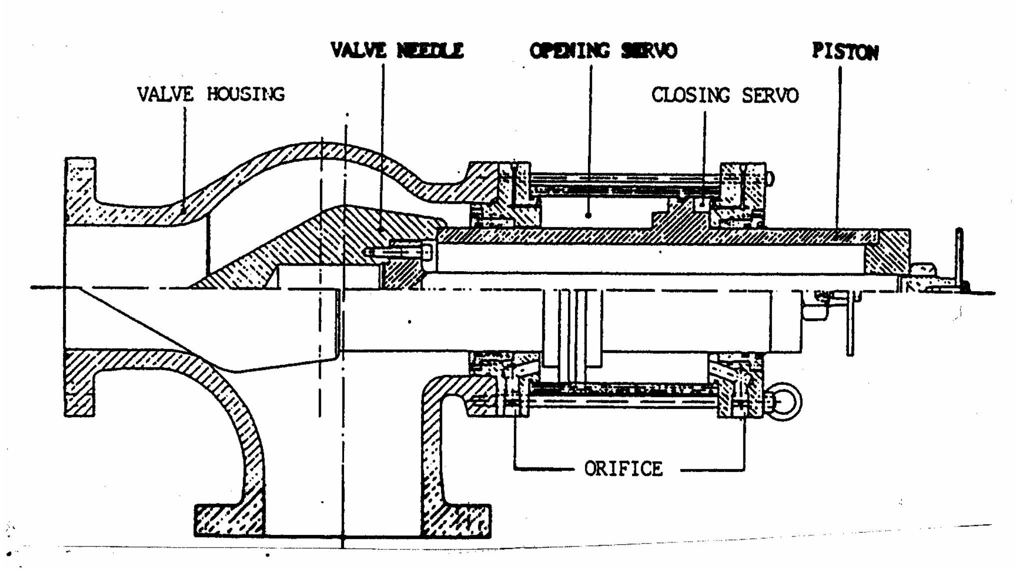

Fig. 11.16 Bypass needle valve /2/

An example of the design of a hydraulically operated bypass valve[/2/] is shown on Fig. 11.16. The valve needle is operated by a double acting servomotor. Opening and closing times are determined by orifices in the supply pipes to the servomotor.

11.7 Guidelines for inspection of valves

On valves and the connecting pipe ends the usual damages to be found are:

- normal wear
- mechanical damages

The valve housing and the closing body as well must be inspected. The access to the upstream side may be difficult to achive, and the time interval between inspections must be determined on the base of the choice of materials, shape, probability of cavitation, vibrations or mechanical wear from sand and so on.

Spherical valves, butterfly valves and gate valves have different properties and patterns of damages. They consist however, of the same main components: valve housing, closing body and operating mechanism. The operating mechanism is normally a hydraulic servomotor operated either by water pressure or oil pressure.

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2. Kvæner Brug: COURSE III, Lecture compendium, Oslo, 1986

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CHAPTER 12

Auxiliary Equipment

Introduction

The auxiliary equipment for the mechanical systems in water power plants comprises

- oil pressure system
- air pressure supply system

12.1 Oil pressure system

The oil pressure system has to supply pressurised oil energy for the turbine governor system[/1/,/2/]. Different types and sizes of these systems are in use and they exist for pressures between 25 and 40 bar. The pressure tank which is designated accumulator, is of the oil-air type for pressures of these levels.

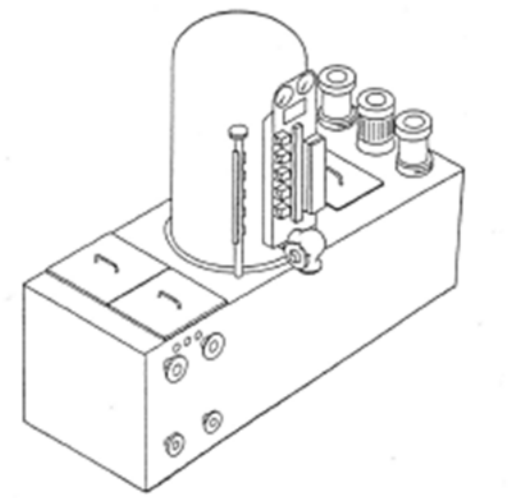


Fig.12.1 Oil pressure system, 40 bar design pressure of the air/oil pressure accumulator /2/

Some small governor systems have been designed also for operation at oil pressures as high as 100 bar. In this case the oil and air in the accumulator must be separated. The usual applications then are off the shelf bladder or piston accumulators. The limitation in the design of these higher oil pressure systems is not only the accumulator size but also the availability of big size control valves.

Oil pressure systems for operation of the shut off valves in main water conduits to the turbines are also found as optional installations in some power stations. These are however, simple power packs which are individually dimensioned and installed for the respective valves. Such power packs are not further described in this book.

An oil pressure system for 40 bar[/2/] which is common and up to date for medium and big size turbine governors, is described in the following section. Fig. 12.1 gives an overlook of a such oil pressure system.

12.1.1 System construction

A general function diagram of the oil pressure system[/2/] is shown in Fig. 12.2. The corresponding hydraulic circuit diagram is shown in Fig. 12.3.

The main components can be summarised as follows:

- oil sump tank
- switches and indicators
- accumulator, e.g., oil pressure tank
- oil pump units
- unloader valve
- check valves
- main oil valve
- relief valve
- oil cooler

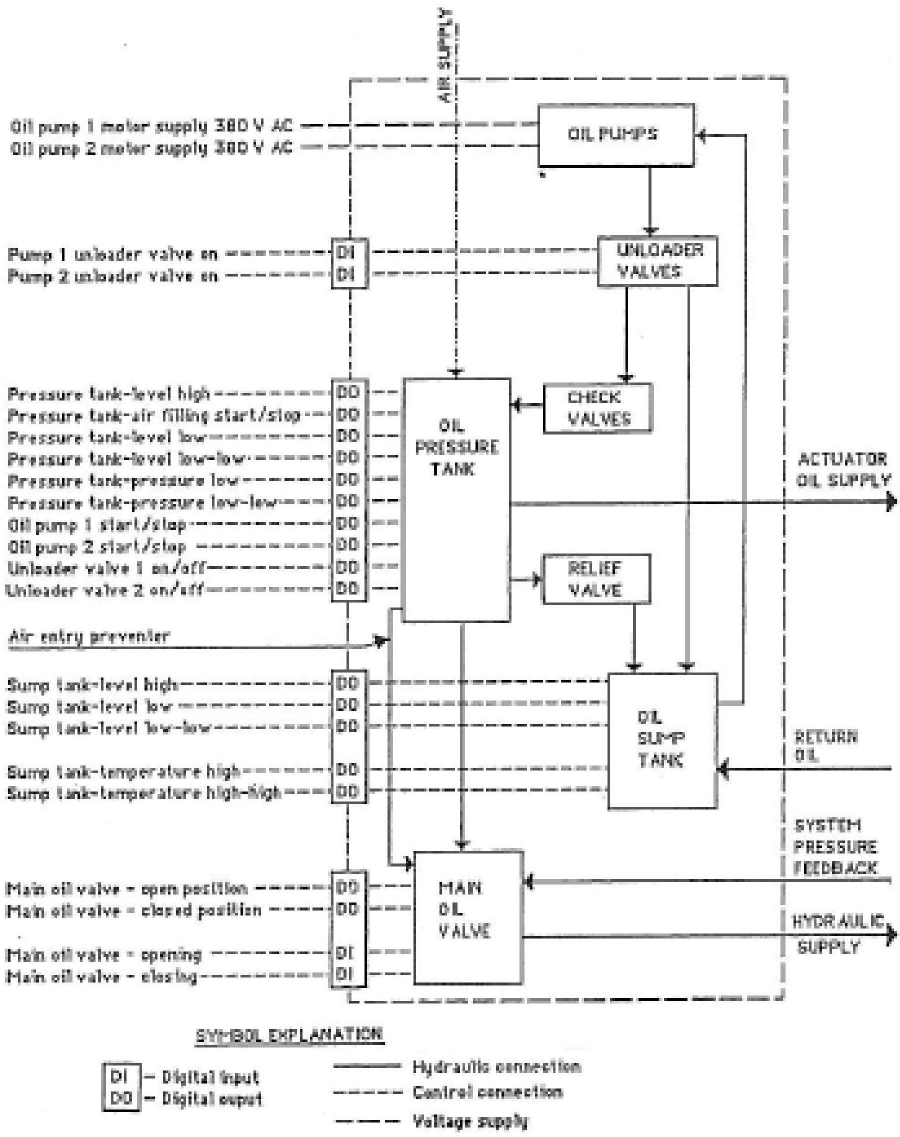
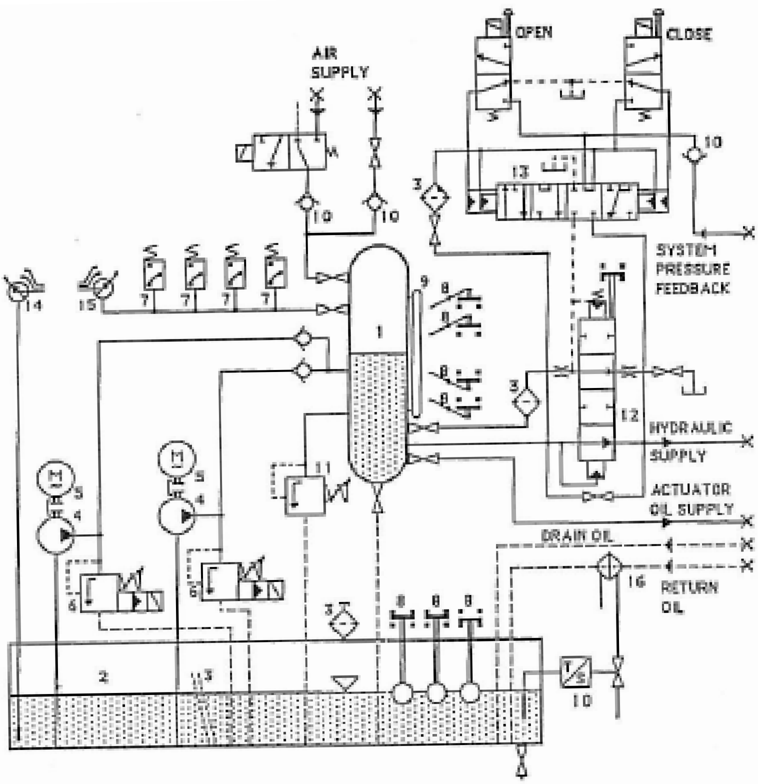


Fig. 12.2 Oil pressure system. Function diagram /2/

General

The accumulator is filled with oil and air under pressure. This energy storage allows a short time high oil consumption which is much higher than the capacity of the oil pumps.

The energy storage is also sufficient for the oil consumption during a stop sequence of the turbine unit without the oil pumps in operation



1. Accumulator
2. Oil sump tank
3. Strainer
4. Oil pump
5. AC motor
6. Unloader valve
7. Pressure switch
8. Level switch
9. Level indicator
10. Thermometer valve for cooling water control
11. Relief valve
12. Main valve
13. Pilot valve
14. Thermometer
15. Pressure gage
16. Oil cooler

Fig. 12.3 Oil pressure system. Hydraulic circuit diagram /2/

The system is designed for a maximum working pressure of 40 bar.

An oil cooler may be connected to the oil return line.

The system is designed for complete remote control with necessary indicators and signal transmitters for safe operation.

Oil sump tank

The oil sump tank is designed with a volume sufficient for the amount of oil in the system. The tank has inspection openings to give access for service and routine inspections.

The oil sump tank contains a special arrangement with strainers to ensure deaeration of the oil. The separation of air entrapped in the oil prevents rapid oxidation and ensures long service life of the oil and also of the components in the system.

Accumulator

The accumulator is mounted on the top of the oil sump tank. A man hole gives access for inspection and service.

The oil level in the tank is controlled by refilling of compressed air. Correct oil level is important to obtain necessary energy storage and necessary amount of oil. Refilling of air is necessary because that pressurised oil absorbs air linearly dependent on the pressure. The absorbed air releases from the oil during the passage of the oil sump tank.

Oil pump unit

The oil pressure system is equipped with two equal oil pump units.

Each pump unit consists of a screw pump, flexible shaft coupling, distance piece, tube/fittings and an electric motor. It can be removed as a separate unit for overhaul. One pump unit may even be dismantled for service and overhaul when the turbine is operating.

The pump is of the self priming positive displacement type. The main parts in the pump are the three rotating screws, e.g., one power rotor and two idler rotors. The power rotor is the only driven element and the idler rotors turn due to the action of the driven rotor. Their thread surfaces are shaped to form a tight seal both in relation to each other and the surrounding rotor housing. The fluid is transported axially and quite uniformly according to the screw rotation.

The pump units are equipped with a fluid trap on the suction side to ensure that the pump remains filled with oil in the stand by mode. This fluid trap shall be filled with oil before the pump is started after overhaul.

The oil pump units are driven by alternating current (AC) motors.

An oil pressure system with two pump units has one pump running continuously while the other pump is in stand by. The stand by pump will start automatically at low pressure in the accumulator. The control of the oil pump units permits the selection of either pump as the preferred unit.

As an option a direct current (DC) motor driven oil pump can be supplied.

A DC motor driven pump is operating only if the AC supply fails. The pump will then start and stop via a pressure switch control which keeps the pressure below but close to the normal oil pressure.

Due to the pump start and stop automatic control the DC motor driven pump unit has no unloader valve system.

Unloader valve

The unloader valve is installed to prevent a fast degradation of the oil quality. A relatively fast

degradation will happen if the relief valve (11), Fig. 12.3, is in continuous operation and thereby cause unwanted oxidation and cavitation.

The unloader valve pos. (6) on Fig. 12.3, is installed in the pipeline between the oil pump unit and the inlet check valve of the accumulator.

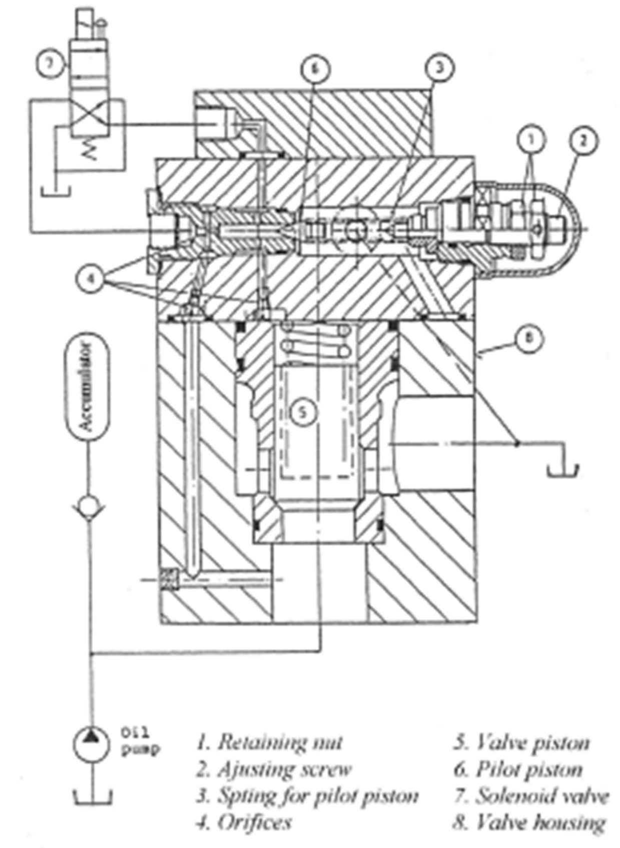


Fig. 12.4 Unloader valve /2/

The design of an unloader valve is shown on Fig. 12.4. This valve is operated when the solenoid valve (7) is being activated by a pressure switch.

At low pressure in the accumulator, the oil is fed from the pump to the accumulator. As soon as the oil pressure has reached the upper set point, the pressure switch activates the unloader valve. The pressure on the top of the unloader valve piston is then decreased, the piston moves upwards and the oil flow is led to the oil sump tank.

In this mode the pressure in the accumulator will decrease due to the oil consumption in the governor system and no oil supply from the pump. When the pressure reaches the lower setting level, the solenoid valve closes. The pressure on each side of the valve piston equalises and the unloader valve closes due to the force from the spring in the valve piston.

The unloader valve works only when the oil pump is running.

Energy is saved as long as the unloader valve delivers the oil directly to the oil sump tank. The unloader valve also reduce the demand for cooling of the oil.

The unloader valve may also be operated as a pilot controlled pressure relief valve.

Check valves

Between each oil pump unit and the accumulator a check valve is installed as shown on Fig. 12.3. These valves prevent pressure oil to flow back to the oil sump tank through the oil pump.

The valve piston is spring loaded to ensure that the valve is closed during pressurising the oil pressure system. The oil pump unit, the oil pipe and other equipment between the pump and check valve may be dismantled without depressurising the accumulator.

Main oil valve

The main valve (12) on Fig. 12.3, is the shut off valve between the accumulator and the main oil piping system.

The design of the main oil valve is shown on Fig. 12.5. The closing/ opening member is a differential piston. For equal hydraulic pressure on the two sides of the piston, the valve therefore remains in closed position. Also at zero pressure on the piston sides the valve is kept closed due to the spring force.

Opening and closing of the main oil valve is controlled by the pilot valve. This valve drains or pressurises respectively the piston chamber.

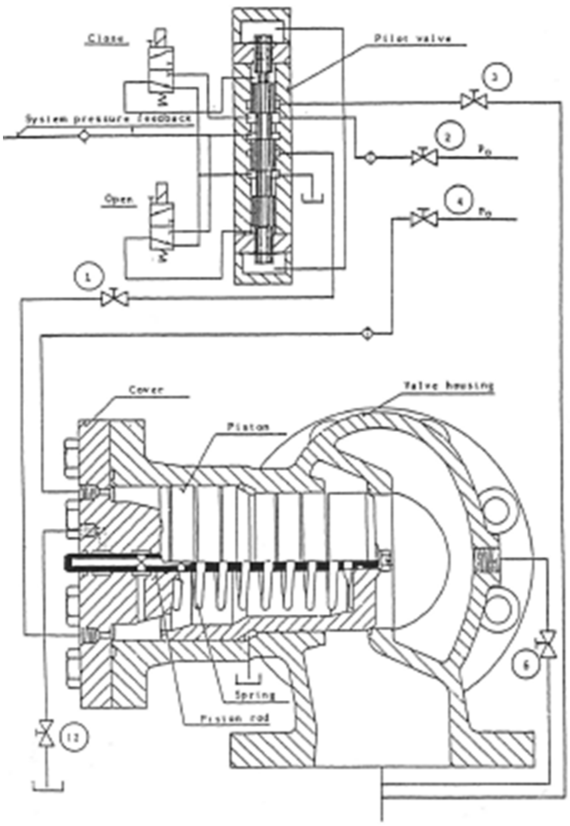


Fig.12. 5 Main valve with pilot valve (schematically drawn) /2/

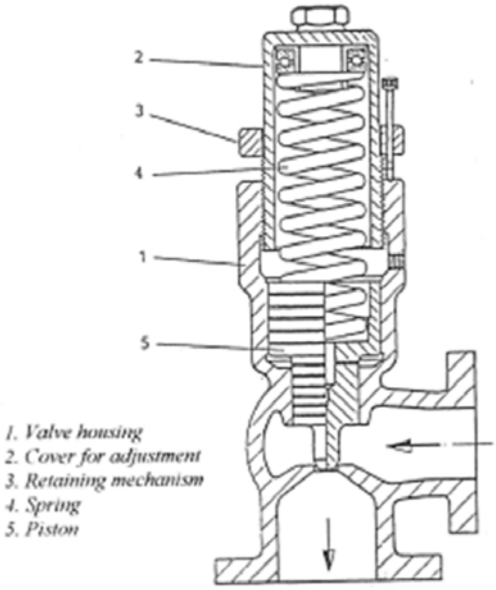


Fig. 12.6 Relief valve /2/

This opening and closing sequence can be done by remotely controlled solenoid valves.

The main valve will be closed automatically at critical low oil level in the accumulator to prevent air from entering the main piping system. This process occurs independent of the manual or automatic sequence control

Manually operated valves are supplied for the by-pass function (6) on Fig. 12.5, and for the opening and closing sequence.

Relief valve

The relief valve (11) on Fig. 12.3, protects the accumulator against unintended high pressure. The design of the valve is shown in Fig. 12.6.

The relief valve shall not open before the pressure in the accumulator exceeds normal working pressure.

The oil begins to flow through the valve when the oil pressure on the lower side of the valve piston balances against the spring pressure.

The design of the valve with a long coil spring and a large valve piston gives a very accurate flow - pressure characteristic. The valve is adjusted by turning the top cover.

If the oil pressure system has no unloader valves or if the valves are out of function, the relief valve can work as a pressure control valve.

Oil cooler

The hydraulic circuit diagram Fig. 12.3 shows the cooling principle for the oil pressure system. The cooling is being done in the return line. Some oil pressure systems do not need oil cooling when the temperature conditions are favourable.

Cleaning of the internal cooler surfaces which are in contact with water, can easily be done by removing the end covers. This may be done during normal operation of the system.

Oil temperature control is accomplished by a thermostatic valve. An automatic control as well is applied. It is based on the oil temperature being sensed in the sump tank to control the water flow through the oil cooler in accordance with the adjustable temperature setting on the valve.

12.1.2 System operation

Opening sequence

The main valve shall not be opened without a preceding careful pressurising and deaeration of the main piping system[/2/]. The pilot valve has this function. When the solenoid valve for opening is activated, the spool in the pilot valve moves to the first stage and oil passes through the by-pass line (3) to the main piping system.

As soon as the system pressure feedback in the main piping system is high enough, the spool in the pilot valve moves to the second stage, the piston chamber in the main oil valve is then drained and the main oil valve opens. In open position the piston chamber will remain drained through the bore in the piston rod. The control voltage to the solenoid valve is switched off and the spool in the pilot valve will move back to the neutral position.

The limit switch on the main oil valve indicates open and closed position.

Closing sequence

When the solenoid valve for closing is activated, the spool in the pilot valve moves to the closed position. Pressure oil flows to the piston chamber in the main oil valve. Together with the inlet oil through pipe (4) the necessary pressure is being built up to close the valve. In closed position the outlet from the valve chamber will be closed and the control voltage to the solenoid is switched off. The spool in the pilot valve moves back to neutral position.

Automatic emergency closing

The orifices in the inlet pipe (4) and outlet pipe (12) in Fig. 12.5, respectively are adjusted so that the main oil valve closes if the oil level in the accumulator becomes extremely low.

Emergency closing

By closing the manually operated valves (1) and (2) in Fig. 12.5, respectively, the outlet from the piston chamber is being closed, the oil pressure increases and the main oil valve closes.

Instrumentation

The oil pressure system is designed for complete remote control with necessary indicators and signal transmitters for safe operation according to up to date standards.

The function diagram Fig. 12.2 shows all in and out going signals and power supply.

The hydraulic circuit diagram Fig. 12.3 shows the instrumentation and the control functions.

Precautions to be taken by start at zero pressure

The oil level in the sump tank has to be checked.

Manual valves for pressure gauge, pressure switches, air supply and pressure to closing side of the main valve piston are to be opened.

Manual valves for air draining, oil draining, pressure to actuator and manual by-pass are to be closed.

One oil pump unit has to be started and run until the oil level can be seen at the lower end of the level indicator.

Air pressure is to be supplied until approximately 2/3 of nominal pressure p_o is reached. It must be checked that the level in the oil sump tank is not too low. One oil pump unit is then started and run until correct level in the accumulator is reached.

Nominal pressure is obtained by filling or draining of air.

Before switching to automatic operation the relief valve, the unloader valves and the instrumentation have to be checked and adjusted if necessary.

12.2 Air supply system

The air pressure system serves to refill the accumulator in the oil pressure system with air.

The air in the accumulator is in direct contact with the oil. With pressurised accumulator, the oil absorbs air linearly with the pressure. In the sump tank the absorbed air releases again during the pressure relief to atmospheric pressure. This process however, takes some seconds to be fulfilled, but the passage time for the oil flow through the sump tank is longer than for the air release process

A function diagram of a typical air pressure system[2] is shown in fig. 12.7. The corresponding pneumatic circuit diagram is shown in fig. 12.8. The system shown is designed with two compressors. However, systems with only one compressor are common as well.

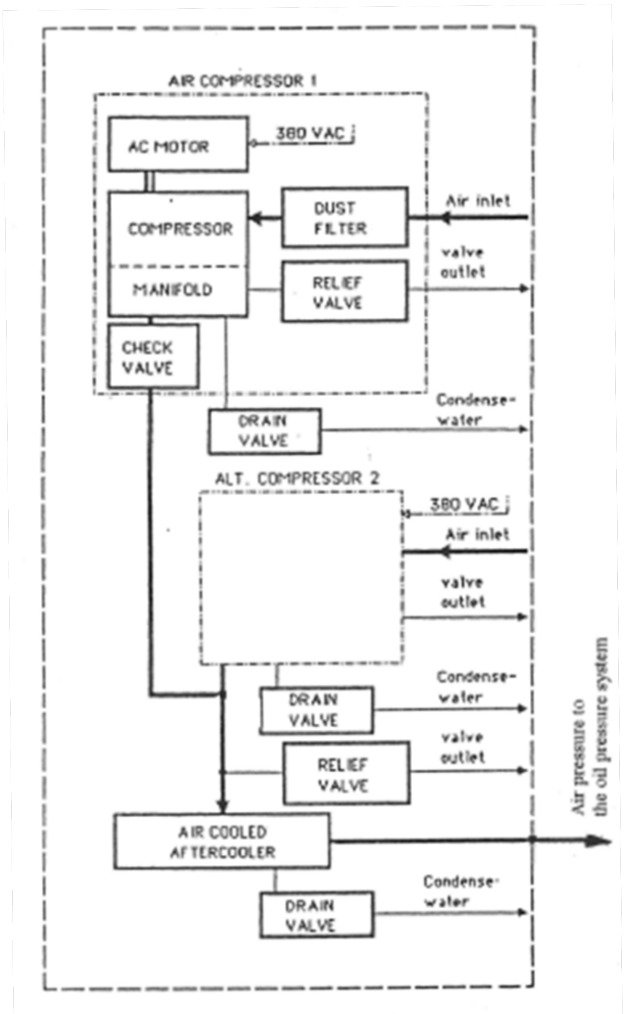


Fig. 12.7 Function diagram /2/

The main components are:

- one or two AC powered high pressure air compressors with inlet filter
- relief valve mounted on the compressor
- air cooled aftercooler with condensate trap
- valves

The start and stop of the refilling cycle is controlled by a level switch on the accumulator.

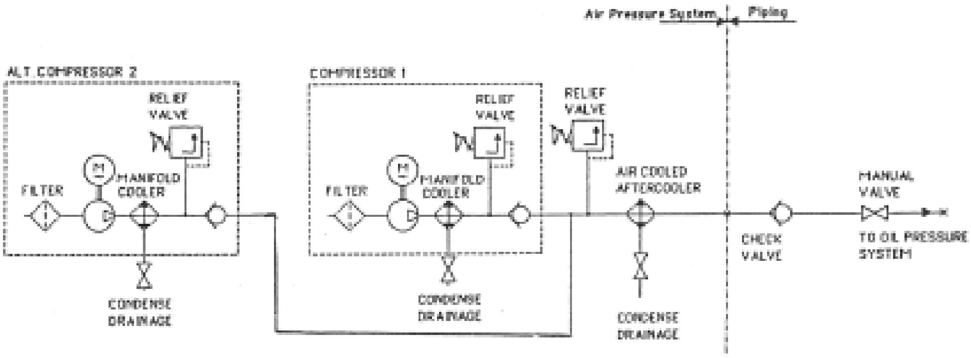


Fig 12.8 Pneumatic circuit diagram /2/

To avoid short cycles for start and stop of the compressor, the stop command is normally delayed to obtain a minimum of working time. This logic is a part of the main control system.

The compressor utilised for the air supply system is an air cooled two stage compressor. Fig. 12.9 shows the compressor with motor assembled on a subbase.

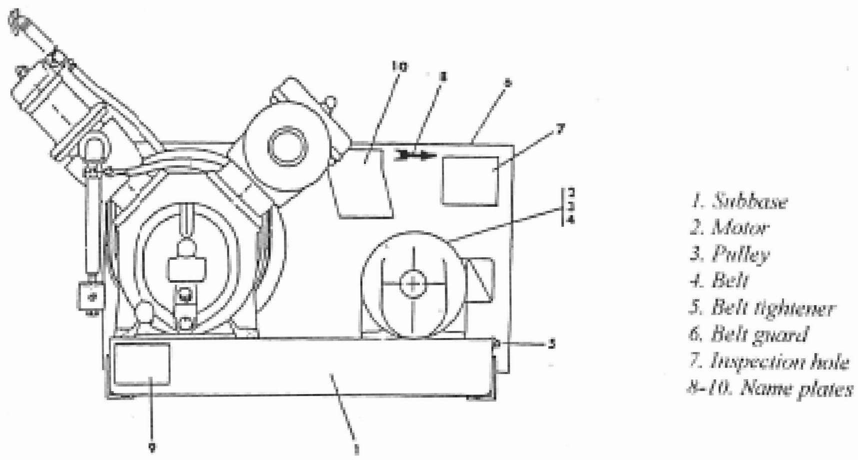


Fig. 12.9 Compressor /2/

The condensate trap on the compressor is equipped with a manual or automatic drain valve for drainage of condensate.

An air-cooled aftercooler is normally installed on the tube between the compressor and the accumulator equipped with a manual or automatic drain valve.

A relief valve on the compressor is a system protection and is adjusted below the maximum allowable pressure of the system.

Air pressure systems with two compressors are equipped with two level switches on the accumulator. The level switch for compressor 1 is placed at normal oil level and the one for compressor 2 is placed 50 to 150 mm above the normal oil level making compressor 2 to start only if the system for compressor 1 is failing.

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CHAPTER 13

Forces transferred to the foundations

Introduction

In water power stations the forces to be transferred to the foundations from the installed units are:

- resulting hydraulic forces from the inlet side of the turbines
- reaction and action forces from jets and jet deflectors respectively
- reaction forces from energy dissipaters
- torque forces from the generator stators
- axial forces from runners and generators via thrust bearings

For Francis and Kaplan turbines with steel scroll casing the main forces are transferred through the scroll casing to the concrete foundation. For vertical Pelton turbines the forces are correspondingly transferred through the distributor.

The transfer and magnitude of these forces are different for horizontal and vertical turbines.

13.1 Horizontal turbines

For horizontal Pelton and Francis turbines normally all the hydraulic forces from the inlet penstock are transferred to and carried by the flange at the inlet end of the penstock via a rigid connection through the inlet shut off valve.

In Pelton turbines the minor reaction forces from the jets and the action forces from the deflectors during rejections, are transferred through the housing to the concrete foundation. The resultant of the jet forces on the runner and the runner weight are transferred to the radial bearings which are bolted to the concrete foundation. The torque forces are transferred to the generator stator foundation.

For a bulb turbine the forces are transferred to the foundation in the river ground through the concrete in which they are embedded. The dominating horizontal hydraulic force is transferred via the outer cone and the main struts. The struts transfer the axial and radial hydraulic forces from the turbine. The torque from the generator is transferred through the generator foundation and partly through the turbine structure. The turbine structure is dimensioned for the total axial force with closed guide vanes.

In addition to the axial forces the outer cone and plate cylinder in bulb turbines are exposed to radial expansion. This effect requires a heavy reinforcement of the concrete in the area around the struts.

13.2 Vertical turbines

The vertical Pelton turbines installed in cavern power houses normally transfer the axial forces from the distributor through the concrete to the rock wall on the downstream side of the power house.

The maximum magnitude of this force is

$$F = \rho g H_{max} \frac{\pi D_c^2}{4} \tag{13.1}$$

where D_c is the outer diameter of the dismantling pipe (of telescopic type with seal ring) located downstream of the inlet shut off valve. The reaction forces from the jets are negligible compared with the static hydraulic forces on the distributor. This is easily stated by calculation of the maximum reaction force of each jet

$$F_j = 2\rho g \frac{\pi d_j^2}{4} \tag{13.2}$$

where d_j is the maximum jet diameter.

However, to transfer the large axial forces to the power house an extension of the inlet flange of the distributor must be welded to the inlet flange. This extension which is indicated by (E) on fig. 13.1, must be introduced to reduce the specific pressure on the concrete and avoid crushing and cracking. This purpose comprises also the necessary reinforcement bars which are laid in to avoid cracking of the concrete during pressurising of the distributor.

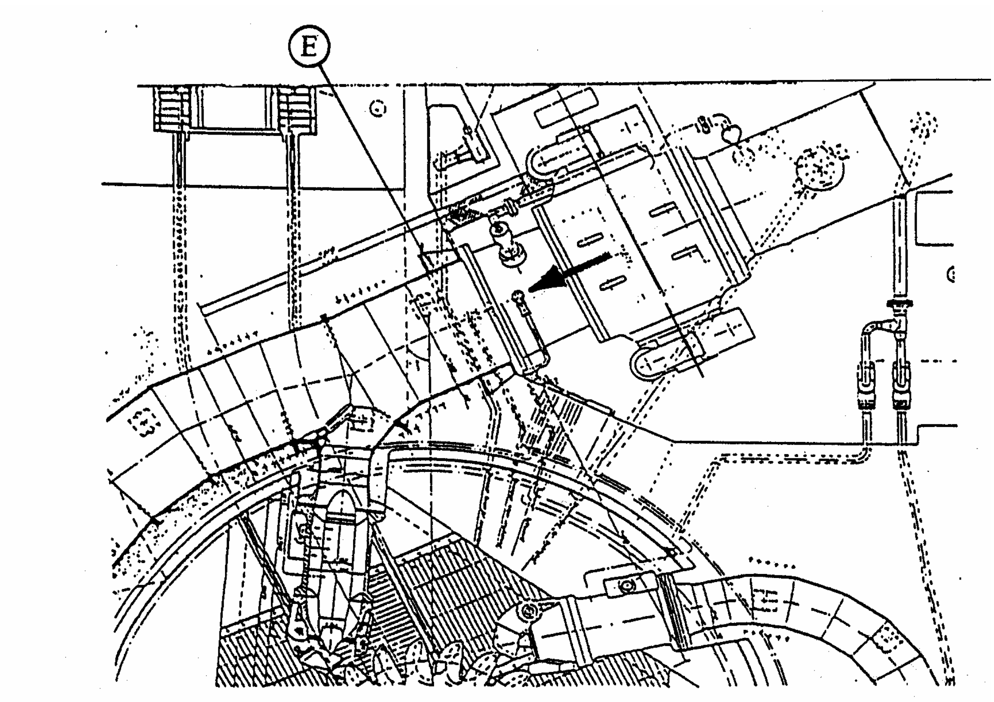

Fig. 13.1 Reinforcement collar for transfer of axial forces on a Pelton distributor.

An extension of the same type and an installation analogous to that for the Pelton turbine is applied for high head Francis turbines as well.
For high head plants the hydraulic forces carried by the tangential stresses in Pelton turbine distributors or scroll casings of Francis turbines are so large that only a negligible part of the hydraulic load is to be carried by the concrete. The reinforcement bars are laid in only to avoid cracking. Moreover, five millimetres soft material are also put in on the steel plates where the deformation is large in axial direction.

The spherical inlet shut off valve is normally free to slide horizontally on its foundation plate on the top of the concrete foundation. Vertically however, the valve is anchored to avoid vibration during closure with open needles.

For high head Francis turbines the hydraulic forces transferred to the concrete foundation are similar to the forces from a vertical Pelton turbine.

However, reinforcing bars which are located at each bifurcation in a distributor will not be used for a Francis scroll casing.

It should however, be emphasised that no welding is allowed of reinforcement bars to the high tensile strength steel plates in the stress carrying parts of high head Francis and Pelton turbines. The reason is that dangerous hardening and cracks may occur.

The vertical forces on the foundation, i.e., in the axial direction of the turbine shaft, are the weights of the turbine and the generator stator and the axial forces from the thrust bearing. In addition on Francis turbines there is a by-pass from the scroll casing through an energy dissipater. During rejections the valve in the by-pass opens according to the governor regulation. The corresponding reaction flow force is transferred vertically as well. In the rotational direction the torque of the unit transfers twisting forces to the foundations.

The twisting force and all the vertical forces which are of minor magnitude compared with the horizontal hydraulic force from the inlet conduit, are transferred through the reinforced concrete structures to the rock wall and bottom of the cavern.

The forces transferred to the power house from vertical turbines described so far, are valid only for cavern power plants with solid rock on the downstream side.

For vertical turbines in open air power houses the horizontal axial forces must be transferred to the penstock flange by a rigid connection through the inlet valve as described for the horizontal turbines.

For Kaplan turbines with scroll casing there is normally no inlet valve. The inlet of the scroll casing is then welded to the oulet of the conduit and also in this case the axial forces are transferred to the penstock.

On account of large dimensions and flexible design the vertical forces from the scroll casing will partly be transferred to the concrete foundation by utilising the weight of the generator.

Especially for Kaplan turbines with unlined concrete scroll casings a large support from the generator weight may be utilised in addition to the anchoring of the stay ring.

The trust bearing is often located in the turbine pit below the generator. In this way the weight of the rotating parts are transferred via the upper turbine cover to the stay ring.

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CHAPTER 14

Causes to damages

Introduction

Damages concerning water turbines are caused mainly by cavitation problems, sand erosion, material defects and fatigue. Damage problems occur primarily in turbines for higher heads than about 250 m. These problems are consequences essentially of high pressures, pressure variations and high water velocities which to some extent depend on the ever prevailing search for a minimising of the costs of the investments.

To cope with these problems, studies and research of the phenomena as well as the properties of materials have been carried out and are going on. Precautions to be taken to avoid or minimise damages have been recommended, criteria for choice of materials and methods for estimation of durability of stressed components have been developed.

In the following section a survey of aspects of the main damage problems and how to reduce or avoid these, are considered.

14.1 Cavitation

The turbine parts exposed to cavitation are the runners and draft tube cones for the Francis, Kaplan and bulb turbines and the needles, nozzles and the runner buckets of the Pelton turbines.

Measures for combating erosion and damage under cavitational conditions are improvements in hydraulic design and production of components, search for erosion resistant materials and arrangement of the turbines for operations within the good range of acceptable cavitation conditions. The further considerations are focused merely on the search for materials with high resistance against erosion from cavitation.

Through the three last decenniums extended research is done on developing new alloy steel qualities. Successively those materials which have performed the best properties, have been tested in relevant applications. As a conclusion it has been an extensive and successful advance in the development of materials with properties as high strength, wear resistance, corrosion resistance, weldability and reduction of defects and brittleness in heat affected zones. This development of material properties is in further progress.

In harmony with these attainments certain types and qualities of materials are chosen according to the prevailing conditions for the different components exposed to cavitation. At present guide vanes and runners exposed to high flow velocities with danger of cavitation or turbulence corrosion are made of stainless steel 13% Cr 4% Ni and 16% Cr 5% Ni respectively. The 16% Cr 5% Ni is used in the runner due to good weldability without preheating.

The upper part of the draft tube cone is also made of stainless steel 16% Cr 5% Ni. The surface of the covers against the guide vane end faces are normally clad welded with stainless steel 16% Cr 5% Ni with hardness of about 300 Brinell. This hardness is about 70 Brinell different from the hardness of the guide vanes and that is chosen to prevent tearing of the surface when the guide vanes are moved.

The parts exposed to high flow velocities such as needle tips and nozzles are made of hardened stainless steel of 13% Cr 4% Ni or 16% Cr 5% Ni.

14.2 Sand erosion

Sand erosion problems in Norwegian water power plants are generally limited to plants at net heads above 200 - 300 m. That implies experiences limited to Francis and Pelton turbines.

Sand erosion is designated as abrasive wear. This type of wear will brake down the oxide layer on the flow guiding surfaces and partly make the surfaces uneven which may be origin also for cavitation erosion. Sand erosion therefore may be both a releasing and contributing cause for damages which are observed in power plants with a large transport of wearing contaminants in the water flow.

The *erosion damages* are to some extent different for Pelton and Francis turbines.

On *Pelton turbines* it is the needle tip, the seal rings in the nozzles and the runner buckets which are most exposed to sand erosion. The wear on needle tips occurs as streaks and indentations forward to the tip. On the buckets the sand contaminants in the water flow is wearing the bucket gap backwards. That causes a delayed contact with the jet and makes an erroneous splitter. The bucket edge is usually exposed to an even wear and tear and becomes wider, and extended indentations may occur just behind the edge.

It should be emphasised that sand erosion even from silt with grain size less than 60 μm have made severe damage on needles and nozzles of a Pelton turbine at 800 m head in Norway while the runner buckets have had negligible damage. This phenomena may be caused by the strong turbulence in the high jet velocity bringing the grain particles to oscillate and rotate in circles causing collisions with the steel surface.

If coarse sand is present, the Pelton buckets are also severely eroded while the damage of the nozzles may be less serious. The reason may be explained by the extreme acceleration of a particle passing the Pelton bucket. A flow analysis shows that an acceleration of 100 000 m/sec^2 may occur for small buckets in a high head turbine.

In *Francis turbines* the guide vane cascade and the labyrinth rings are the major parts which are exposed to wear. In some serious cases also the surfaces of the covers against the guide vane end faces and the runner blades have been damaged. Usually the wear of the guide vane cascade and the labyrinth rings have the most significant influence on the efficiency of the turbine.

Materials resistant to sand erosion

The real mean to minimise sand erosion in the turbines is to apply the most suitable material with properties that match the highest possible resistance against erosion, and contemporary satisfy manufacturing and operating conditions. Against sand erosion stainless steel does not really show acceptable resistance. The materials stellite and titanium show better resistance, but only by a factor of two to three. However, promising work in developing coating processes of ceramic material as wolfram carbides is going on.

At present however, the materials with the best properties against cavitation erosion are also used even if sand erosion is expected. That means guide vanes and runners are made of stainless steel types as 13% Cr 4% Ni or 16% Cr 5% Ni. These same materials are used also in the rotating seal rings on Francis runners, while the static labyrinth seals in the covers are made of Ni-Al bronze which has a hardness different from that of the rotating ring.

For needle tips and nozzles in Pelton turbines which are exposed to high flow velocities, *hardened* stainless steel of 13% Cr 4% Ni or 16% Cr 5% Ni is recommended. If however, severe sand erosion is expected, a ceramic coating on the needle tip and the nozzle ring has been used successfully.

So far ceramic coating of Pelton runners is not commercialised with sufficient success, but improvements are made in recent years. Coating of facing plates of Francis turbines is under development as well, while the guide vanes may be treated in the same way as the needles and nozzles of a Pelton turbine.

A general problem with the surface coating is however, that it will be more rough than a polished steel surface. This means a decreased efficiency of the turbine. Grinding of the coating is also regarded as a problem because the coating may be too thin or even penetrated.

14.3 Material defects

For high head turbines the stress carrying parts are made of fine grain high tensile strength carbon steel. The base for the dimensioning criterias of these parts is the maximum stress and the number of pressure pulsations cycles. To allow for acceptable material and weld defects within realistic values for production, materials with a yield point value above 460 MPa are not recommended.

For large vertical Pelton turbines the development of high tensile steel has allowed for increasing size of turbines for high heads in the range of 800 to 2000 m. With basis in fracture mechanics it is essential to design a turbine to fulfil the requirement that unstable fracture from a crack shall not occur until the crack has penetrated the whole plate thickness. This condition is designated *Leakage Before Rupture* (LBR). The reason is that a crack through the thickness will be easy to detect by the leakage. As long as a crack has not penetrated the whole plate thickness an unstable rupture will be prevented. This requirement limits the maximum size of the turbine depending on the toughness of thick materials.

On the basis of the above mentioned requirements the maximum main stress must be limited to about 200 MPa for any material quality developed untill now.

The most important requirement for the manufacture is the weldability to avoid defects and brittleness in the heat affected zones of the material. To fulfil this requirement limitations in the chemical composition is made. These limits are:

Carbon C < 0,13%, Sulphur S < 0.01% and Vanadium V < 0.09%

Welding defects

Defects will always occur in a weld. Cracks and lack of fusion are the most serious defects because they are two dimensional. Three dimensional defects such as slag and gas cavities are not so dangerous except if sharp edges of slag is connected to a lack of fusion or a crack.. Gas cavities may also indicate hydrogen which is very dangerous due to micro cracks.

The most dangerous defect is a propagating crack with a very sharp edge, e.g., a two dimensional defect or a flaw.

Crack propagation based on fracture mechanics

As defects always occur in a weld or in the heat affected zones in the base material, it is therefore essential to have acceptance criterias for defects which will not grow to rupture under the operational conditions for the regarded structure.

An acceptance criterion for material defects may be based on the theory of fracture mechanics. An applied criterion is briefly described as follows.

The stress in front of a crack tip in a *complete elastic* material is expressed by

$$\sigma = \frac{K}{\sqrt{2\pi r}} \qquad (14.1)$$

where σ is the mean stress which cause the crack to propagate
K is the stress intensity factor
r is the distance to crack tip

By means of this theory one will find that the stress at the crack tip will reach an infinite value. This indicates that the crack should propagate even for low average stress values σ in the surrounding of any small crack with depth a in a complete elastic material with no ductility.

However, all materials suitable for structural design will undergo a *plastic* deformation at a certain stress level which limits the stress peak value. If the load or stress level and crack depth are within certain limits the crack will stop in the plastic zone. The material will have a certain crack arresting ability depending on the ratio between the elastic energy and the elastic energy stored in front of the crack tip.

As a conclusion:

A crack will propagate only if the loss of elastic energy is equal or greater than the energy needed to form new fracture surface.

The critical elastic energy at the crack tip is expressed by the stress intensity factor K found in the following way.

If the average stress in the uncracked material is σ, the stress intensity factor becomes

$$K = \sigma \sqrt{\pi a} f(\phi) \qquad (14.2)$$

where a is the crack depth
$f(\phi)$ is a factor which is a function of crack length and geometry of the material surrounding the crack

An unstable crack propagation will occur if $K \geq K_C$. The constant K_C is a material constant determined by testing of the respective material quality or welding joint. Unstable crack propagation may occur as a brittle fracture with sound speed.

However, if the material cross section is thick another value K_{IC} instead of K_C is valid. But for ductile materials K_{IC} values are not valid.

Materials for turbines are of the ductile type. Depending on the strength of the material and the level of the working stress the material includes both elastic and plastic deformation in front of the crack tip before propagation.

Therefore, another method than to find a critical stress intensity factor must be used to find the crack arresting ability and the critical size of a ductile material. This method is called Crack Tip Opening Displacement (CTOD).

For any material there will be a critical crack size which leads to unstable fractures. This crack size may be based on the CTOD determined experimentally for the base materials and welds.

It is proven however, by small scale CTOD-tests that high tensile strength steel with high working stress level allows for a smaller crack size than a low tensile strength steel with a lower stress level. A critical crack size may be calculated, but this is not further considered here.

14.4 Fatigue

Defects of critical size rarely occur in a new turbine. Such defects will in any case be detected by a leakage or rupture during the pressure test which is a very important type of tests at the end of the manufacture. Smaller defects in a new turbine may however, grow to critical size during life time due to fatigue propagation caused by a certain number of load cycles.

Basically the stress carrying parts in a Francis turbine are statically loaded. A turbine in typical peaking operation however, may be stopped and started, i.e., loaded and unloaded three times or more a day. For a life time of 50 years that leads totally to about 50 000 cycles. In modern fracture mechanics theory this is in the low cycle fatigue domain. In addition minor stress amplitudes caused by pressure oscillations from the turbine regulation will be superimposed.

On the base of this conditions the stresses must be limited to avoid small fabricated cracks and other material defects to grow to critical size. Safety margin to yield stress or ultimate stress is not sufficient as safety criterion if high tensile strength steel is used.

Unfortunately the fatigue crack propagation speed has not decreased by the development of high tensile strength steel even if the toughness and strength have increased. On the contrary some research works have indicated a slight improvement of fatigue lifetime of high cycle fatigue in materials with low yield stress. On the other hand the crack propagation speeds in brittle materials and tougher materials show only negligible differences.

From these experiences the following conclusions are drawn:

- For turbine parts exposed to a number of loading cycles above 20 000 in the lifetime it is not reasonable to increase the maximum working stresses above 200 MPa because acceptable defects then will be too small to be detected.
- In heat affected high tensile strength steels hardening effect and residual stresses may occur. For all thick plates therefore, welds should be heat treated after the welding. Otherwise after a number of years in operation, crack growth may lead to unstable fracture starting from cracks which do not penetrate the whole plate thickness.

Critical area in scroll casings is the joint between the shell and the stay ring where both the stress and wall thickness have the maximum value. For Pelton turbine distributors the bend joint on the inside is the critical point.

Parts exposed to high cycle fatigue and erosion

Runners and guide vanes especially in reversible pump turbines, will be exposed to high cycle fatigue up to the range of 10^{10} - 10^{11} cycles with relatively low stress amplitudes. Pelton runners are normally casted of 13% Cr 4% Ni alloy steel. The same material is used also in Francis runners. However, for Francis runners also the 16% Cr 5% Ni has been used. This steel shows a somewhat better cavitation resistance. Moreover, it may be welded with no preheating except for a moderate temperature of 50 ^{o}C to keep the material well above the dew point during welding. The fatigue threshold limit for a number of cycles higher than 10^{11} indicates infinite lifetime for stress amplitudes $\Delta\sigma \leq 45$ MPa and a surface crack less than one mm in depth and two mm in length.

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CHAPTER 15

Condition Control

Introduction

Condition control of water turbines and additional mechanical equipment is the primary basis for organizing and carrying out

- *preventive maintenance* - a continous process which is taking place with certain time intervals and at planned dates.
- *overhauls* - being performed to improve the operation conditions, rectify wear and leakages on the plant according to plans adapting to the plant operation.

If an accident occurs or preventive maintenance and overhauls are not carried out properly, situations requiring repairs may arise. A *repair* is here defined as an unplanned overhaul.

Continous monitoring and condition control is of great importance to all types of turbines. The content of the condition control activities are however, somewhat different from one type to another of the turbines. Therefore, condition control is briefly described specifically for the respective turbine types.

15.1 Activities for Pelton turbines

Turbine guide bearing

The first oil change should be done after 3 - 6 months of operation. The later oil changes are to be done as required by evaluating oil sample tests.

To empty the bearing for oil it has to be done at standstill by pumping through the oil level pipe in the bearing housing.

If babbit metal particles are found in oil samples, the bearing should immediately be dismantled for inspection.

Runner

The runner should be regularly inspected to record possible damages from foreign objects in the water. The time interval between each inspection is dependent on sand content in the water.

The runner inspection is done visually by means of magnaflux and/or dye penetrant. Particular attention should be paid to the area between the buckets.

If minor cracks or defects have been formed, these should be removed by grinding and polishing according to advices from the manufacturer.

The special shape of the runner buckets makes it difficult to detect material defects just below the surface. These defects may penetrate to the surface during the first operation time period. To prevent an extensive crack propagation they must be rectified as soon as possible.

Main injector with needle servo motor

The needle tip and the nozzle should be inspected with respect to cavitation damages and damages from foreign objects. If the water contains fine silt or sand, the needle tips may loose their original shape.

It is of great importance that the nozzle is inspected from the inside.

The needle servomotor should be run to neutral mid position and the oil pressure taken off for safety reasons. The leakage indicators should be regularly inspected.

Seal ring in deflector bearing

Leakage in this seal do not require immediate replacement of the seal ring, but replacement should be done as soon as possible to avoid bearing corrosion. The seal ring must be provided with a spring of stainless material.

Filter

Filter for breaking system control water supply should be checked and cleaned if necessary. It may be cleaned after closing of the valves for this water supply system from the penstock.

15.2 Activities for Francis turbines

Routine inspections

Routine inspections means visual inspection of the complete turbine:

- look for possible leakages
- inspect bolted connections
- drain pumps should be inspected and level switches tested.

RPM shutdown curve

The activity is to record the RPM with 30 seconds intervals from the moment of closed guide vanes to standstill of the unit. The RPM shutdown curve should be drawn from nominal RPM to standstill.

Leakage control of guide vanes

The activity is to record the RPM with the spherical valve open and the guide vanes closed.

Shaft alignment

The activity is to record the shaft misalignment by means of a micrometer dial instrument against the shaft at the turbine bearing top. The misalignment should be recorded at different loads on the unit.

Labyrinth seal water flow

The assessment of wear and control of the labyrinth rings is possible by measuring the labyrinth water flow.

Runner

Visual inspection of the runner is required to record possible cavitation and erosion damages as well as cracks in vanes. The inspection of the inlet is done from the scroll casing. Three of the guide vane arms should be dismantled and the guide vanes rotated by hand to open position. The unit should be rotated manually to enable inspection of the complete runner circumference. The outlet side of the runner should be inspected from the draft tube cone.

Scroll casing and draft tube

The activity is to carry out inspection of painting for possible corrosion and ensure that all manometer connection openings are open and that the manhole cover is drop tight after completed inspection.

Guide vane mechanism

The guide surfaces of the covers should be inspected from the scroll casing and possible wear recorded.

Visual inspection of the surfaces of the guide vanes. The clearances between the guide vanes and the cover surfaces and the clearance between guide vanes should be checked.

Look for possible leakages in the guide vane bearings. Look for slacks in the bearings of the guide vane arms and links.

Check the connection between regulating ring and the servomotor.

Shaft seal box

During operation water will normally not flow over the the shaft seal box. It may however happen during start and stop. If water runs into the upper cover it is removed by a drain pump.

Turbine bearing

The same activities as for Pelton turbines.

15.3 Activities for Kaplan and Bulb turbines

The general principles for condition control are the same as for the Francis turbines. Further considerations are therefore connected only to a few specific details of Kaplan and Bulb turbines.

Runner

The runner should be inspected both from above and below. Particular attention should be given to possible cavitation erosion and scratches on the blades as well as leaks around the blade flange against the hub.

Runner chamber

The narrow gap between runner and the runner chamber should be checked if foreign objects may have passed the gap and made scratches in the chamber.

Guide vane mechanism

For guide vane mechanism with individual vane servomotors on Bulb turbines it should be checked that the vanes have an identical movement.

Shaft seal box

For Bulb turbines at standstill it should be checked that the water does not flow out of the box along the shaft into the turbine bearing.

Generally for Bulb turbines

Special attention should be paid to changes in the sound when the unit is in operation.

References

Kværner Brug: COURSE III, Lecture compendium, Oslo 1986

CHAPTER 16

Quality Assurance (QA)

The organisation for building a hydropower plant performs a quantity of specifications, expectations and goals for the project with respect to economy, time and quality. The requirements for carrying out the project are that it is equally important that all sub suppliers are capable of delivering their supply in accordance with demands as those performed by the project organisation.

There are several ways to ensure the requirements above. The most common however, is to apply and follow an international quality standard. The most applied standard in Europe is EN ISO 9001 which specifies requirements aimed primarily at preventing nonconformity at all stages from design through servicing.

The requirements may be divided in two parts. The one part is aimed at organisational matters and the other specifically at the project.

Examples of organisational requirements are:

- Policy, objectives and commitment to quality shall be documented by the management.
- Responsibility, authority and interrelation of all personnel who manage, perform and verify work affecting quality shall be defined.
- A management representative shall, irrespective of other responsibilities, have defined authority and responsibility for the Quality System and its maintenance.
- Quality audits shall be carried out to verify whether quality activities comply with planned arrangements and to determine the effectiveness of the quality system.

The requirements through a project life are shown in the flow-chart defining period, activities and end results of different QA - activities.

One may say that the picture differs depending on whether You are a customer or a supplier. However, whatever role an organisation have in a hydropower project, there is always a customer of Your services. The Operational Organisation will be customer of the Project Organisation as the consumers will be customers of the Operational Organisation.

Hence, the need for a Quality system in order to avoid nonconformity is equally important to everybody involved in a hydropower plant.

QUALITY ASSURANCE ROUTINES

PERIOD	**ACTIVITY**	**RESULT**
	STARTING UP MEETING	***WORK DISTRIBUTION OFFER SCHEDULE***
OFFER	OFFER REVIEW	
	CONTRACT REVIEW	***PROJECT ORGANIZATION SCHEDULE***
	DESIGN REVIEW	***DESIGN VERIFICATION QUALITY PLAN***
ORDER	PRODUCTION FOLLOW UP SUB-CONTRACTOR CONTROL	***VERIFICATION***
	ERECTION	***VERIFICATION***
TAKING OVER	COMMISSIONING	***TEST RESULT COMMISIONING REPORT***
GUARANTEE PERIOD	ASSESSMENT OF PRODUCT QUALITY	***DELIVERY ASSESSMENT***
DELIVERY FINALIZATION	GUARANTEE INSPECTION	***REPORT***

References

Kværner Brug: COURSE III, Lecture compendium, Oslo 1986