GUIDELINES FOR THE APPLICATION OF SMALL HYDRAULIC TURBINES

Small Hydropower Series No. 4

GUIDELINES FOR THE APPLICATION OF SMALL HYDRAULIC TURBINES

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UNITED NATIONS
INDUSTRIAL DEVELOPMENT ORGANIZATION
Vienna, 1986

ID/SER.N/4
ISSN 0256-727X

CONTENTS

Tables

Page

Figures

INTRODUCTION

The guidelines contained in the present study relate to hydraulic turbines and other equipment applied at small hydropower plants.

In recent years, as a result of the rise in prices for organic fuel, considerable interest has been created in different sources of renewable energy, in particular those associated with the energy of rivers. The most economic power plants installed on large rivers are of high capacity. In developed industrial countries the most economic sources have been exploited or their development has been undertaken. Although their construction costs are relatively low, large hydropower plants demand large capital investments and long construction periods. Hence renewed interest in small hydropower installations has been expressed in countries with a high level of utilization of hydropower resources.

In developing countries the financial possibilities do not always exist for construction of large hydropower plants. The development of energy of small rivers through the construction of small hydropower installations is therefore of primary importance. It should be noted that small hydropower plants may also be profitable when installed at various water control structures associated with water supply, navigation, irrigation etc.

Specific energy costs at small power plants are higher than at large hydropower installations. During design and construction of small hydropower plants it is therefore necessary to search continuously for ways of raising profitability through reduction of construction costs by using unified structural elements and standard mass-produced equipment.

Operating costs of small hydropower plants should be reduced to a minimum through complete automation of plant operation, employment of remote control systems and rejection of permanent operating personnel. Equipment reliability should be high, with regular maintenance intervals and the use of industrial methods in equipment maintenance.

The exploitation of the energy of small rivers and other water control systems depends to a certain extent on State policies designed to stimulate construction of small hydropower installations. In some countries financial incentives have been given in the form of concessionary loans for construction of small hydropower installations. Under such conditions, small hydropower plants appear to be profitable means of satisfying the demand of isolated consumers and small power systems.

Hydraulic turbines installed at small hydropower plants will be referred to hereinafter as small turbines. At the present time there is no limiting value for parameters of small hydraulic turbines, various countries and companies having variable limiting parameters.

The maximum capacity of a small turbine may be adopted as 10,000 kW, although some manufacturing companies include units of up to 15,000 kW within the range of their small turbines. The minimum capacity of small turbines may be adopted as 50 kW.

The highest demands of small power plants are for turbines of 1,000-2,000 kW capacity. To reduce turbine costs it seems necessary to develop standard designs based on building-blocks.

Turbines of high capacity are manufactured according to custom-made designs, thus resulting in higher specific costs. Turbines of small power plants operate complete with other equipment, including generators, speed governors, control systems and step-up gears. All such equipment must be standardized and mass produced.

The purpose of the following guidelines on the application of small hydraulic turbines is to introduce the main conclusions of the theory of turbines required for a better understanding and a proper selection of turbines by specialists. The guidelines show schemes and principles of operation of the main types of turbines employed at small hydropower plants, and present the main principles and tasks of standardization turbines through specific examples. Standards of turbines manufactured by different companies are also described.

The guidelines are intended for mechanical engineers without special training in the field of hydraulic turbines.

A special word of thanks is addressed to the United Nations Industrial Development Organization and the managements of Bell, Gilbert Gilkes and Gordon, Kessler, Hitachi, Ossberger, Sanden, Voest-Alpine and Voith turbine manufacturers, which enabled the author to familiarize himself with the manufacture of small hydraulic turbines and to obtain the necessary information for use in the guidelines. Nevertheless, since information was limited, the text may not be wholly free from inaccuracies. The author will accept and take into account any helpful suggestions and remarks.

I. HYDRAULIC TURBINES: GENERAL INFORMATION

A. Main concepts and definitions

The hydraulic turbine is a machine that converts the energy of water into mechanical energy by using a runner that serves as primary mover. This energy is passed on, via the turbine shaft, to the energy user, which may be either an electrical generator or any other machine.

A liquid at rest or in motion possesses a store of mechanical energy which is characterized by the specific value:

$$E = gZ + \frac{P}{\rho} + \frac{V^2}{2} \ (\mathrm{J\ kg^{-1}}) \qquad (1)$$

which is energy related to a unit of a liquid mass.

Z = Elevation of the considered particle of a liquid above the conditional plane of comparison (m)

g = Gravity ($\mathrm{m\ sec^{-2}}$)

P = Pressure ($\mathrm{N\ m^{-2}}$)

ρ = Liquid density ($\mathrm{kg\ m^{-3}}$)

V = Liquid particle velocity ($\mathrm{m\ sec^{-1}}$)

From the above equation (1) it is seen that energy E is the sum of potential energy determined by the first two terms and kinetic energy.

Usually for practical purposes a more descriptive expression of energy related to a weight unit of a liquid is applied.

$$H = Z + \frac{P}{\rho g} + \frac{V^2}{2g} \ (\mathrm{m}) \qquad (2)$$

In this expression specific energy has the dimensions of one meter of a liquid column.

Utilization of water power in a hydraulic turbine is achieved by the water flow from a high-lying to a low-lying energy level.

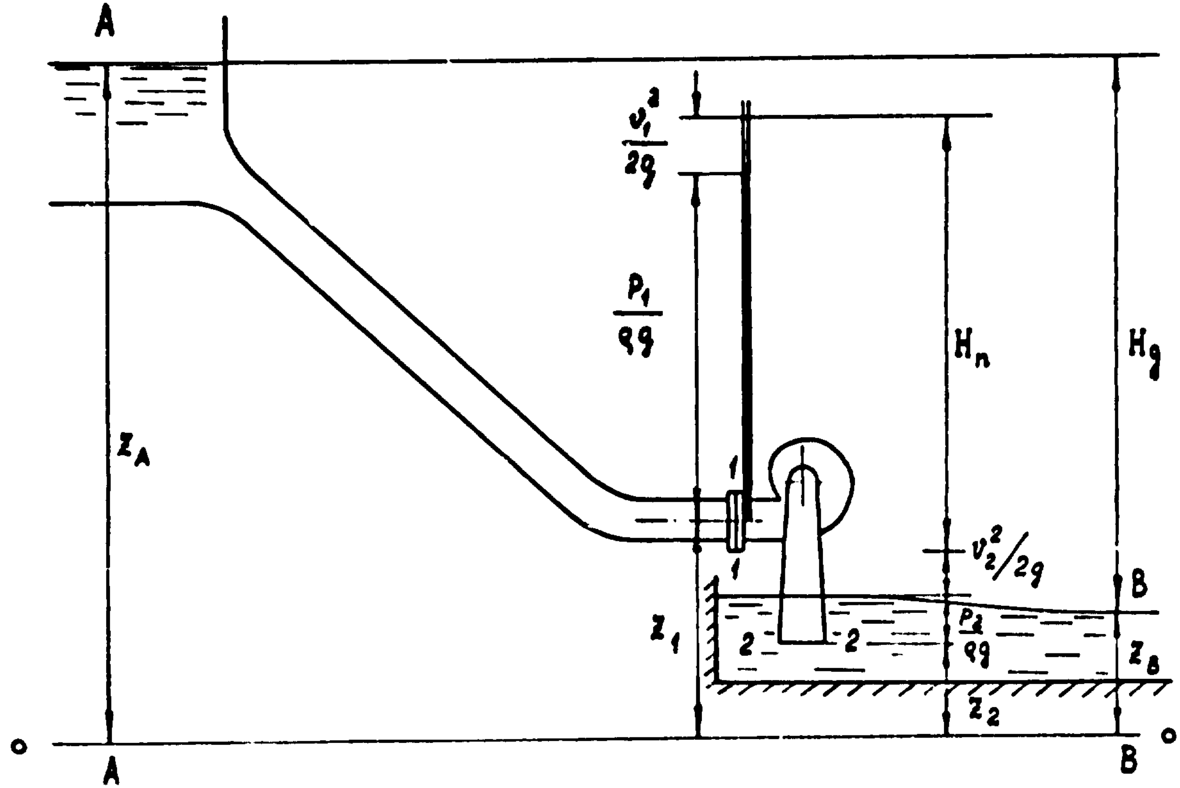
Figure 1 illustrates one of the possible variants of hydraulic turbine installations.

The water in the upstream reservoir in section A-A possesses a higher energy than that in section B-B of the tailrace. In the absence of the water flow the difference of specific energies in these two sections is determined by the value

$$H_g = Z_A - Z_B \qquad (3)$$

This value is known as the gross head.

Figure 1. Hydraulic turbine installation



When locating the elevations the reference point is an arbitrary horizontal line 0-0.

During operation of the turbine the water flows through water conveying channels, the hydraulic turbine and outlet channels of the hydropower plant. Because of losses of a part of mechanical energy of the liquid resulting from friction, the energy which may be transferred to the runner will be less than H_g.

Let us denote the inlet and outlet sections of the turbine by 1-1 and 2-2 respectively. The difference in specific energies of the flow between these sections is

$$H_n = (Z_1 + \frac{P_1}{\rho g} + \frac{V_1^2}{2g}) - (Z_2 + \frac{P_2}{\rho g} + \frac{V_2^2}{2g}) \qquad (4)$$

and is known as the net head.

The volume of water flowing through the turbine during one second is known as the turbine discharge and is denoted as $Q \ m^3 \ sec^{-1}$.

Velocities V_1 and V_2 in the formula (4) are average values for sections 1-1 and 2-2 and are equal to

$$V_1 = \frac{Q}{A_1}, \quad V_2 = \frac{Q}{A_2}$$

where A_1 and A_2 = Sectional areas (m^2).

The energy supplied to the turbine in one second and discharged by it is

$$P_d = 9.81QH \ (kW) \qquad (5)$$

where ρ = 1,000 kgm^{-3} and g = 9.81 m sec^{-2}.

The value P_d is the power consumed by the hydraulic turbine.

In the turbine not all the power P_d may be completely utilized usefully. A part of supplied mechanical energy is converted into heat energy because of internal friction of the liquid, losses in the supports and other internal losses.

The power output of the turbine is

$$P_t = \eta_t P_d$$

where η_t is the efficiency of the hydraulic turbine which is the ratio of the developed power to the delivered power of the turbine.

In the general case

$$\eta_t = \eta_v \eta_h \eta_m$$

where η_v - Volumetric efficiency
η_h - Hydraulic efficiency
η_m - Mechanical efficiency

The volumetric efficiency is

$$\eta_v = \frac{Q_r}{Q}$$

where Q_r is the discharge of water flowing through the runner.

In some systems of hydraulic turbines the discharge through the runner is less than that through the turbine because a part of the liquid flowing through the turbine passes by the runner and does no useful work.

Hydraulic efficiency

$$\eta_h = \frac{P_h}{P_d - P_v}$$

where $P_v = \rho g\ (Q - Q_v)\ H_n$ (the power lost due to the leakage)

$P_h = M_h\ \omega = \rho g Q_r H_{th}$ (the hydraulic power or the power transferred from the water to the runner).

Notation:

M_h = The torque developed by the liquid on the runner (N m)
ω = Angular velocity of the turbine shaft (sec^{-1})
H_{th} = Theoretical head showing the value of specific energy transferred to the runner blades by the liquid

From the expressions above it is seen that

$$\eta_h = \frac{H_{th}}{H_n}$$

The turbine shaft power is less than the hydraulic power by the value of mechanical losses occurring in the turbine shaft bearings and on the outer surfaces of the runner. Therefore the mechanical efficiency is

$$\eta_m = \frac{P_h}{P_t}$$

The main operating parameters of the turbine are

Power = P_t (kW)
Net head = H_n (m)
Discharge = Q ($m^3\ sec^{-1}$)
Speed = n (min^{-1})

From the shaft of the hydraulic turbine mechanical energy is transferred to an electric generator. In some cases of application of small turbines a direct transfer of energy to some mechanical machine of the user is possible.

The power P of the hydraulic unit is usually measured at generator terminals. Usually it is less than the turbine power P_t by the value of losses in transfer and in the generator.

In turbines of small hydropower plants the trend today is usually towards the use of high speed mass-produced electrical generators because of their relatively low costs. For their matching with the low speed turbines step up gears installed between turbine and generator shafts are to be used.

It is obvious that

$$P = P_t \eta_G \eta_{TR}$$

where η_{TR} = Transfer efficiency
η_G = Generator efficiency

During operation of the turbine at the hydropower plant the power developed by the turbine is determined by the generator load and the electric power demand of the system. Depending on different conditions the turbine power may vary within a definite range.

During operation of the turbine at the power plant the speed of the turbine is usually constant irrespective of the load. For variation of the turbine power the water discharge is regulated by the special control system.

During different periods of operation the net head may vary. Operating conditions of the turbine at the power plant are determined by the combination of working parameters. At a fixed value of n there is a head and discharge at which the turbine operates with maximum efficiency. This condition is called optimum. At deviation from the optimum conditions the efficiency of the turbine will be lower.

B. Similarity laws

The turbines are considered geometrically similar if the relative dimensionless co ordinates of the water passage of these turbines are similar. To obtain the dimensionless co-ordinates it is necessary to find the relation of the co ordinates to a certain characteristic linear dimension, for instance the diameter D of the runner.

Thus in geometrically similar turbines for the corresponding points of the water passage the following condition should be met:

$$\frac{X_i}{D} = \text{idem}, \quad \frac{Y_i}{D} = \text{idem}, \quad \frac{Z_i}{D} = \text{idem}$$

Usually the water passage means the space confined between inlet and outlet sections of the turbine through which the water flows.

Kinematically similar operating conditions of the turbine are the conditions in which velocity vectors at corresponding points of the flow form similar angles with co ordinate axes and the ratio of absolute values of velocity for the whole of the flow is constant.

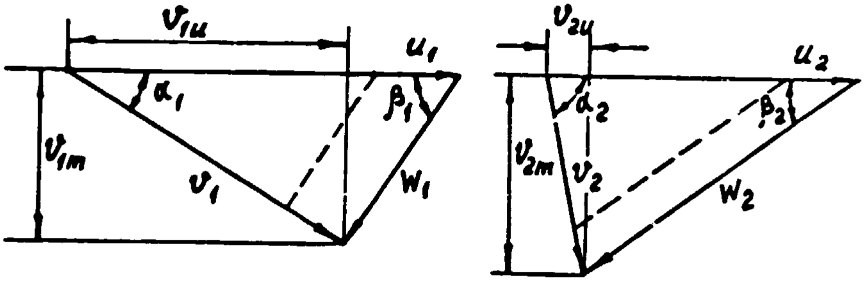
Let us denote peripheral velocity by U, relative velocity by W and absolute velocity by V.

Then $\vec{V} = \vec{W} + \vec{U}$

that is, vectors $\vec{V}$, $\vec{W}$, $\vec{U}$ have constructed a triangle of velocities.

Figure 2 shows triangles of velocities for fluid particles in sections in front of the runner ($\vec{V}_1$, $\vec{W}_1$, $\vec{U}_1$) and after the runner ($\vec{V}_2$, $\vec{W}_2$, $\vec{U}_2$). In kinematically similar operating conditions of the turbine triangles of velocities are similar. In figure 2 the triangle of velocities for the similar operating conditions is shown by a dashed line.

Figure 2. Triangles of velocities for fluid particles



It is known that in compliance with the Euler equation

$$g\eta_h H_n = U_1 V_{1u} - U_2 V_{2u} \tag{6}$$

For the fulfilment of the conditions of similarity at the runner inlet (figure 2) it is necessary that

$$\frac{V_{1m}}{U_1} = \text{idem}, \quad \frac{V_{1u}}{U_1} = \text{idem} \tag{7}$$

and at the runner outlet

$$\frac{V_{2m}}{U_2} = \text{idem}, \quad \frac{V_{2u}}{U_2} = \text{idem} \tag{8}$$

Since

$$U \sim nD, \quad V_m \sim \frac{Q_r}{D^2} \tag{9}$$

then it seems possible to express the conditions of kinematic similarity in terms of operating parameters of the turbine. In conformity with (6), (7), (8) and (9)

$$\frac{g\eta_h H_n}{n^2 D^2} = \text{idem}, \quad \frac{Q}{nD^3} = \text{idem} \tag{10}$$

From expressions (10) we obtain the known similarity formulas connecting the operating parameters of two geometrically similar turbines operating under similar kinematic conditions:

$$n' = n\,\frac{D}{D'}\left(\frac{H'_n}{H_n}\right)^{1/2}\left(\frac{\eta'_h}{\eta_h}\right)^{1/2} \tag{11}$$

$$Q' = Q\left(\frac{D'}{D}\right)^{2}\left(\frac{H'_n}{H_n}\right)^{1/2}\left(\frac{\eta'_h}{\eta_h}\right)^{1/2}\frac{\eta_v}{\eta'_v} \tag{12}$$

$$P'_t = P_t\,\frac{\rho'}{\rho}\left(\frac{D'}{D}\right)^{2}\left(\frac{H'_n}{H_n}\right)^{3/2}\left(\frac{\eta'_h}{\eta_h}\right)^{3/2}\frac{\eta'_m}{\eta_m} \tag{13}$$

Let us exclude linear dimensions of turbines from formulas (11)-(13). Thus we obtain at $\rho = \rho'$

$$n' = n\left(\frac{P_t}{P'_t}\right)^{1/2}\left(\frac{H'_n}{H_n}\right)^{5/4}\left(\frac{\eta'_h}{\eta_h}\right)^{5/4}\left(\frac{\eta'_m}{\eta_m}\right)^{1/2} \tag{14}$$

In conformity with formulas (11)-(14) it is possible to calculate the operating parameters of the turbine. Aside from the known operating parameters of turbines, these formulas include relations of efficiencies as well. For practical calculations in the field of hydraulic turbines of small hydropower plants the relations of volumetric and mechanical efficiencies may be taken to be equal to unity. The relation of hydraulic efficiencies is usually evaluated by approximation formulas which have a wide distribution. In particular it may be taken that

$$\eta'_h = 1 - (1 - \eta_h)\left[(1-X) + X\left(\frac{Re}{R'_e}\right)^{1/5}\right] \tag{15}$$

where

$$\frac{Re}{R'_e} = \frac{\nu'}{\nu}\left(\frac{D}{D'}\right)^{2}\left(\frac{H}{H'}\right)^{1/2}$$

where ν is the temperature dependent coefficient of kinematic viscosity of water. Table 1 shows approximate values.

Table 1. Approximate values of hydraulic efficiencies

$10^6\,\nu$, $m^2\,c^{-1}$	1.3	1.14	1.0	0.9	0.81	0.75
t, °C	10	15	20	25	30	40

Coefficient X in the formula (15) represents the share of recalculated hydraulic losses:

$$X = 0.6 - 0.75$$

Hydraulic efficiency η_h is usually determined under different operating conditions of the turbine during turbine model tests on a special laboratory test rig.

C. Specific speed

The power developed by the turbine varies proportionally with the values of the head and the discharge. The head at the power plant is determined mainly by local conditions. In some cases it is possible to create high heads reaching 500-1,000 m and even higher. In other conditions only low heads up to 1-1.5 m can be reached.

The discharge of the water utilized by the hydropower plant depends on the river flow. The discharge of the water consumed by turbines and the river flow may differ from one another in the presence of a storage reservoir. The storage reservoir will store up the water during the time when the river flow is higher than the discharge of the hydropower plant. The stored water is used for power generation at the hydropower plant during the periods of higher discharge and power.

Frequently during construction of small hydropower plants it is impossible to create a high-capacity storage reservoir. Therefore the discharge at the plant is governed by the river flow.

The water discharge in different rivers varies over a wide range. At the same time this value varies frequently throughout the year. In some periods the discharge of the river differs markedly from the average annual discharge. Approximately it may be noted that the turbine discharge at small hydropower plants ranges between 0.05 and 30 $m^3\ sec^{-1}$ in different conditions.

At high head hydropower plants water discharges are most commonly low. At low head hydropower plants the discharge is usually higher.

In the turbine runner mechanical energy of the water is transferred to the runner and the shaft. In accordance with the relation between the turbine operating parameters relative dimensions of water passages vary over a wide range. There is a complex quantity called the specific speed which in many respects characterizes the relation between dimensions of the water passage.

The specific speed is calculated by

$$n_s = \frac{nP_t^{1/2}}{H_n^{5/4}} = 3.65\,\frac{n(\eta_t Q)^{1/2}}{H_n^{3/4}} \qquad (16)$$

where n - Speed (min^{-1})
H_n - Net head (m)
P_t - Turbine power (hp)
Q - Discharge ($m^3\ sec^{-1}$)

If the power is measured in kW, then

$$n_s = \frac{7}{6}\,\frac{nP_t^{1/2}}{H_n^{5/4}} = 1.167\,n_{skw}$$

If the head is measured in feet and the power in hp, then in compliance with the formula (16) we obtain

n_s (m, hp) = 4.446 n_s (ft, hp)

Formula of specific speed is obtained from (14) if we ignore change of hydraulic and mechanical efficiencies. In these conditions it may be thought that for geometrically similar turbines operating in kinematically similar conditions the value of the specific speed n_s is constant.

Usually the turbine is characterized by the value n_s determined by operating parameters existing under conditions of maximum efficiency.

The specific speed of modern turbines varies over a wide range

n_s = 10-1,200

Turbines with a low value of n_s are used at higher heads. At lower heads speedy turbines with a high value of n_s are applied.

D. Reduced parameters of turbines

The specific speed is a complex measure showing the relation of operating parameters in optimum conditions. It is evident that the coefficient n_s does not depend on dimensions of the turbine and is constant for all geometrically similar turbines.

Under actual conditions the operating parameters and dimensions of geometrically similar turbines vary within wide limits. For comparison of turbines and characteristics of their operating conditions the so-called reduced parameters controlling the operating parameters of geometrically similar turbines with D (diameter) = 1 m and running under the net head H_n = 1 m have found wide use.

If in similarity formulas (11)-(13) we assume D' = 1 m and H'_n = 1 m, then we obtain the reduced values

$$n'_1 = \frac{nD}{H_n^{1/2}} \; \frac{1}{m^{1/2}}, \; \text{min}^{-1} \qquad (17)$$

$$Q'_1 = \frac{Q}{D^2 \, H_n^{1/2}} \; \frac{1}{m^{1/2}}, \; m^3 s^{-1} \qquad (18)$$

$$P'_{t1} = \frac{P_t}{D^2 \, H_n^{3/2}} \; \frac{1}{m^{3/2}}, \; \text{kW} \qquad (19)$$

where n'_1 = Reduced speed

Q'_1 = Reduced discharge

P'_{t1} = Reduced turbine power

In formulae (17)-(19) the ratio of volumetric and mechanical efficiencies is taken to be equal to unity. Coefficient

$$m = \eta_h / \eta'_h$$

is the ratio between hydraulic efficiencies of the considered turbine and its tested model. The formula (15) may be used for calculation of m. In approximate calculations $m = 1$ is allowable.

The reduced values n'_1, Q'_1, P'_{t1} are constant in kinematically similar conditions. These values are defined by model tests.

From formulae (17)-(19) it follows that

$$n = \frac{n'_1 \, H_n^{1/2}}{D} \cdot m^{1/2} \tag{20}$$

$$Q = Q'_1 D^2 \, H_n^{1/2} \, m^{1/2} \tag{21}$$

$$P_t = P'_{t1} \, D^2 \, H^{3/2} \, m^{3/2} \tag{22}$$

From the known values n'_1, Q'_1, P'_{t1} obtained from model tests it is easy to calculate the operating parameters of the geometrically similar turbine of any dimensions running under any specified head.

II. CLASSIFICATION OF TURBINES

A. Classes of turbines

First let us consider classification of hydraulic turbines in terms of distinguishing features of hydraulic action. From formula (4) and the determination of the net head it is seen that the change in mechanical energy of the fluid occurs in the turbine. As it takes place potential and kinetic energies also change. In terms of hydraulic action all the turbines may be divided into two classes, impulse and reaction.

The distinctive feature of impulse hydraulic turbines is that there is no change in potential energy in them when the water gives up its energy to the runner. The water pressure at inlet and outlet of the runner and during movement through runner passages is constant and in most cases is equal to the atmospheric pressure.

In impulse turbines the available energy of the water flow at the inlet of the runner is in the form of kinetic energy. Movement of the water in runner passages takes place with a free surface contacting the ambient air. In some types of impulse turbines the energy takes place at inlet and outlet sections due to variation of elevations Z_1 and Z_2.

Among impulse turbines having applications at small hydropower plants the following three types are worth mentioning: Pelton, Turgo and Banki-Mitchell turbines.

In reaction turbines when the water gives up its energy to the runner the change in potential and kinetic energy occurs under the following conditions. During operation of the turbine the runner passages are completely filled with water. There are no free surfaces with constant pressure in optimum and nearby conditions. The spaces filled with vapour may be formed in operating conditions followed by cavitation.

At the present time two types of reaction turbines, namely Francis and Kaplan turbines, find application at hydropower plants, including small hydropower stations.

Thus five types of turbines find application at small hydropower plants. Each type of turbine is efficient within a certain range of operating parameters, and may have variants differing from one another by relative dimensions of runner passages and other elements of the water passageway.

B. Reaction turbines

1. Francis turbines

Francis turbines find extensive application both at large and small hydropower plants. They are characterized by a large variety of designs and dimensions of the water passage. Figure 3 shows a scheme of the water passageway of the modern vertical-shaft Francis turbine. The water from the penstock or from the low-pressure supply line enters the spiral case (5). The inlet connection piece of the spiral case is displaced with reference to the turbine axis which makes it possible to whirl the water flow in relation to the turbine axis. The dimensions of the spiral case are selected to provide a uniform circumferential water supply of operating devices arranged in series. Radial sections of the spiral case vary in shape and in size diminishing as the distance from the inlet connection piece goes up. The commonly used shape of the spiral case is circular as shown in figure 3. At places of smaller sectional area their shape becomes oval.

From the spiral case the water flow enters the stay ring 2 (figure 3) which comprises a fixed-vane system with vanes which are similar or different in form. One of the vanes of the stay ring at the end of the spiral case is of a specific shape and is called a nose of the spiral case. The stay ring makes for strength and rigidity of the spiral case and associated structures. The stay ring does not perform any hydraulic functions. If strength conditions are satisfied, it can do without the stay ring.

The wicket gate assembly (3) located downstream of the stay ring represents a row of cylindrical gates similar in form, arranged circumferentially and spaced evenly. In most designs the gates are rotated about their pivots at the same rate within the range provided by the design. The limiting position of the vanes when adjacent gates are in full contact blocking the water flow from the spiral case corresponds to a full closing. During opening of the gates the discharge through the turbine and the power increases (see formula 5). Thus the wicket gate assembly is an operating device capable of hydraulic power control.

From the wicket gate assembly the water enters the runner (4) and gives up its energy. Figure 4 shows a meridional projection of the water passage of one of the variants of the runner. The crown 1 of the runner connected to the turbine shaft and the shroud 2 are made integral [illegible] ades 3 arranged circumferentially and spaced evenly. Surface [illegible] blades and the surfaces of the crown and the shroud form a water passage of the runner. Surfaces of blades are of a complicated three dimensional shape.

From the wicket gate assembly the water enters the runner, gives up its energy and changes the direction of its motion along the axis of the turbine.

In modern hydraulic turbines relative dimensions of the runner vary within wide limits. Frequently, the typical dimensions of the runner are adopted to be the maximum diameter of the leading edge of the blades denoted as D_1 (see figure 4). Diameter D_2 of the runner outlet is determined by the maximum diameter of the trailing edge of the blades. Width of the runner water passage is determined in many respects by the height B_0 of the wicket gate assembly. Relative dimensions B_0/D_1 and D_2/D_1 to a large extent depend on the specific speed n_s. In modern Francis turbines B_0/D_1 = 0.08 0.35, D_2/D_1 = 0.65 1.2. The lower values are typical of low speed high head turbines, whereas the higher values are representative of high speed low head turbines.

Figure 3. Water passageway of vertical-shaft Francis turbine

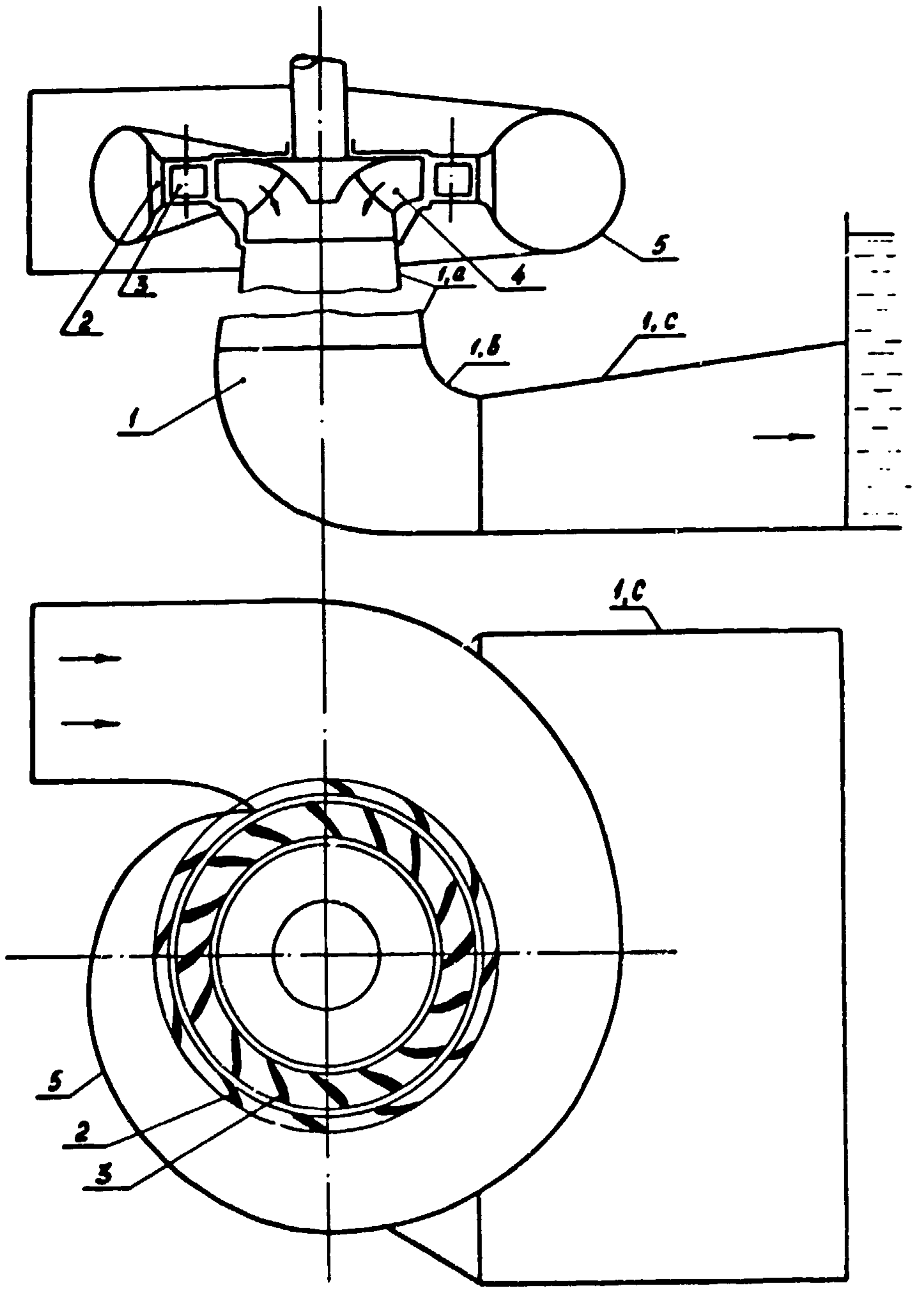
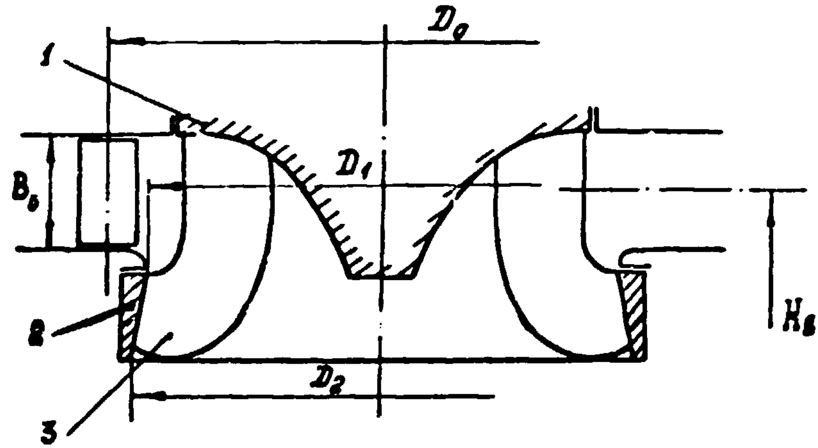


Figure 4. Meridional projection of runner water passage



From the runner the water enters the outlet water passage which directs it to the outlet structures of the hydropower plant. Figure 3 illustrates the outlet water passage constructed in the form of a bent draft tube. The draft tube consists of a conic diffuser (1,a) a bend (1,b) of a circular inlet section and a rectangular outlet section, and a diffuser (1,c) of a rectangular section. The described classical form of the draft tube is typical of large hydraulic turbines. But the given form may also be used in small hydraulic turbines. In the latter case it is possible to use simplified straight conic draft tubes.

The draft tube allows the use of the gross head of the hydraulic turbine more completely. In this case a major portion of the kinetic energy of the water flowing from the runner is used.

From the draft tube the water enters the tailrace. The kinetic energy with which the water leaves the draft tube is not used in the turbine. Therefore it is wise to have a large outlet sectional area of the draft tube to reduce the losses of kinetic energy. Experience recommends the specified dimensions of outlet sections of draft tubes.

The role of the draft tube in the turbine increases particularly in low-head hydraulic turbines.

Thus the above described scheme of the Francis turbine water passage finds application at large vertical shaft hydropower installations. It is also used at small hydropower plants. However, for small hydropower plants some simplifications may be introduced which result in reduction of turbine efficiency, but at the same time these simplifications reduce the cost of installation.

Now let us consider some practicable and applied variants.

2. Turbine with single-gate control

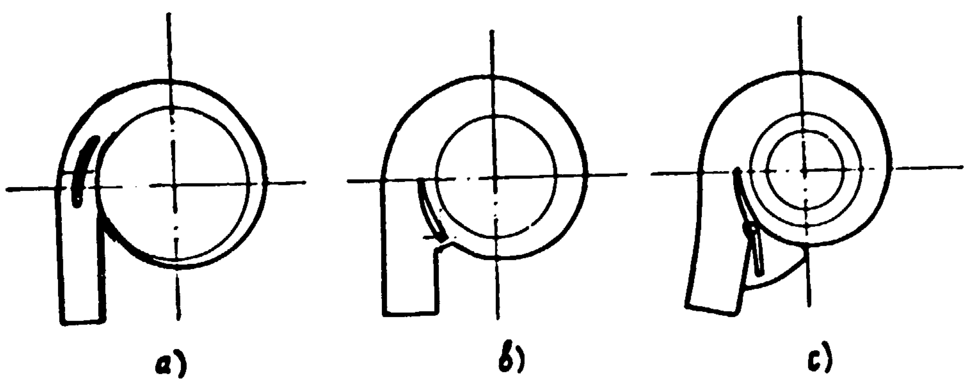
Simplification of turbine design through the system of power and discharge control has been suggested. In the scheme described above the function of control was performed by the wicket gate assembly through adjustment of gates in synchronism. However, the given design is sophisticated. During variation of opening of the gates the change of water circulation takes place at the runner inlet with a pronounced effect on power.

Figure 5 illustrates the scheme of the turbine with the single-gate assembly. In this case the control is realized by one gate only, the rotation of which changes the discharge and the spinning of the water flow at the runner inlet.

Figure 5 (a) shows the scheme of the single-gate control suggested by Reifenstein. In this case the gate is installed at the inlet section of the spiral case. Under optimum operating conditions the gate does not produce appreciable hydraulic losses. Variation of the gate position results in higher losses and reduction of efficiency.

Figure 5 (b, c) illustrates the scheme of control suggested by Kviatkovsky. In this case smaller losses take place when deviating from optimum conditions. At loads differing considerably from optimum ones the Kviatkovsky scheme of control ensures appreciably higher values of efficiency. The gain reaches 10 per cent and above. Figure 5 (c) shows the variant with the unloaded gate.

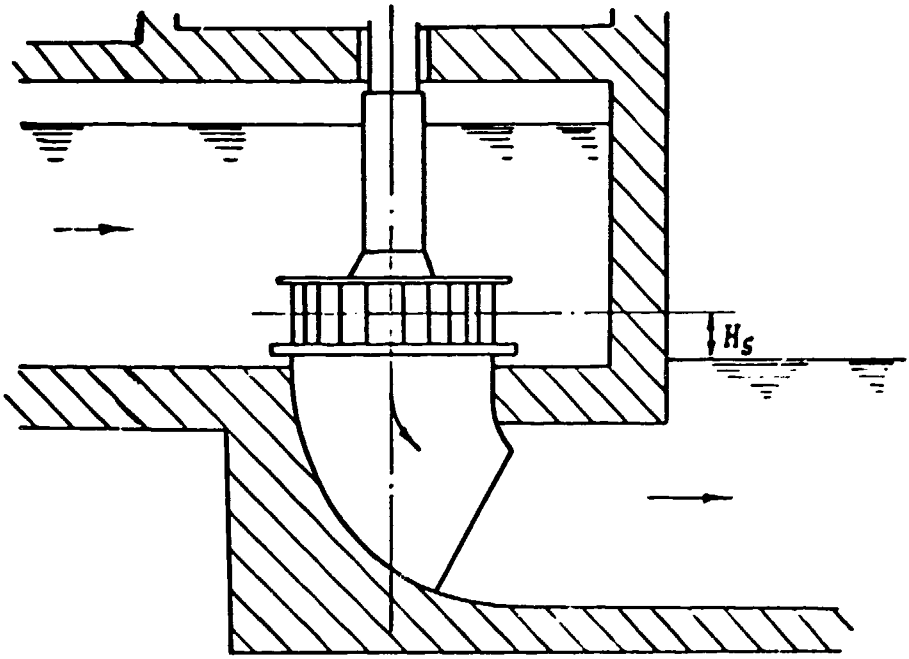
Figure 5. Scheme of turbine wth single gate assembly



3. Open-flume turbines

For small-capacity hydraulic turbines operating under low heads an open supply of the water to the turbine (figure 6) is used. In this case there is no spiral case at the inlet. It is replaced by an open rectangular flume with a free water level. The stay ring is also omitted and the wicket gate assembly is controllable.

Figure 6. Scheme of turbine water supply



Naturally this design is more straightforward and cheaper. However, the conditions of water supply to the wicket gate assembly are worsened, thus adversely affecting operation of the runner.

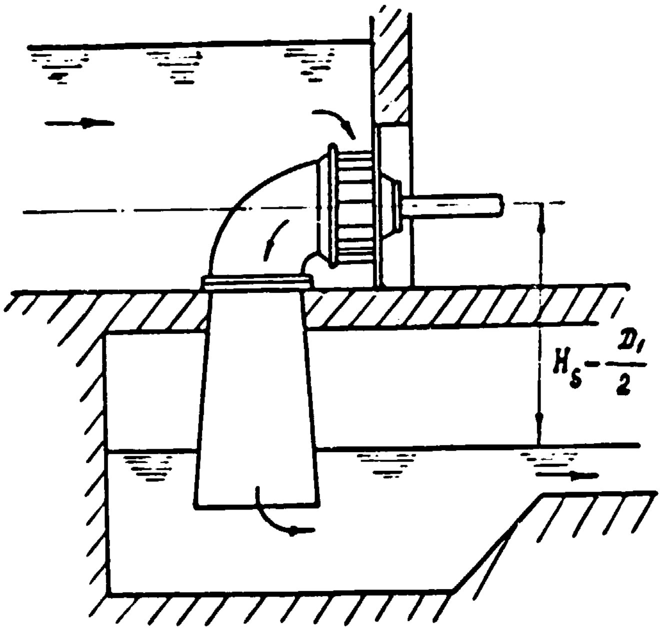
The hydraulic efficiency of such turbines is lower.

Figure 6 shows another simplification of the design with regard to the draft tube. The draft tube in height is less than optimum dimensions. The bend of the draft tube is incomplete and there is no straight diffuser at the outlet.

All these alterations, compared with a conventional design, result in reduction of efficiency and cost. Simplified designs are applied in turbines of small capacity.

An open flume without a spiral case may be found in horizontal-shaft turbines (figure 7). It is seen that the draft tube is provided with a bend at the inlet and a straight diffuser. The design of such turbines is simple and its cost relatively low.

Figure 7. Draft tube of horizontal-shaft turbine



4. Horizontal-shaft turbines in drum

At higher heads the use of open-flume Francis turbines is impossible. In this case horizontal-shaft turbines with a drum in front of the turbine are used.

Figure 8 illustrates the scheme of the turbine with a pressure tank to which the water is supplied through the penstock. From the tank the water enters the wicket gate assembly. The draft tube is arranged integral with the pressure tank.

Figure 9 shows another configuration. Here the turbine shaft is brought out to a dry space through a bend of the draft tube. The design of the pressure tank is simplified as compared with the variant described above.

It is obvious that some other arrangements are feasible.

5. Kaplan turbines

Kaplan turbines are widely used at large and small hydropower plants. A conventional arrangement of the water passage (figure 10) differs considerably from that of Francis turbines. The inlet features incorporate a spiral case (1), a stay ring (2) and a wicket gate assembly (3). Relative dimensions and the shape of the water passage are peculiar.

Figure 8. Turbine with pressure tank supplied through penstock

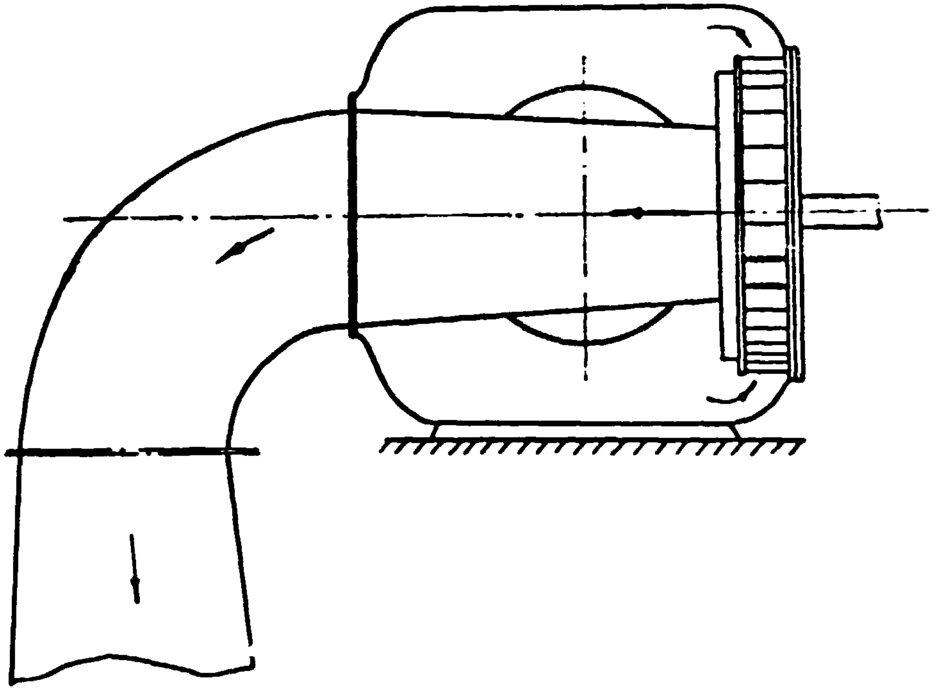


Figure 9. Simplified design of pressure tank

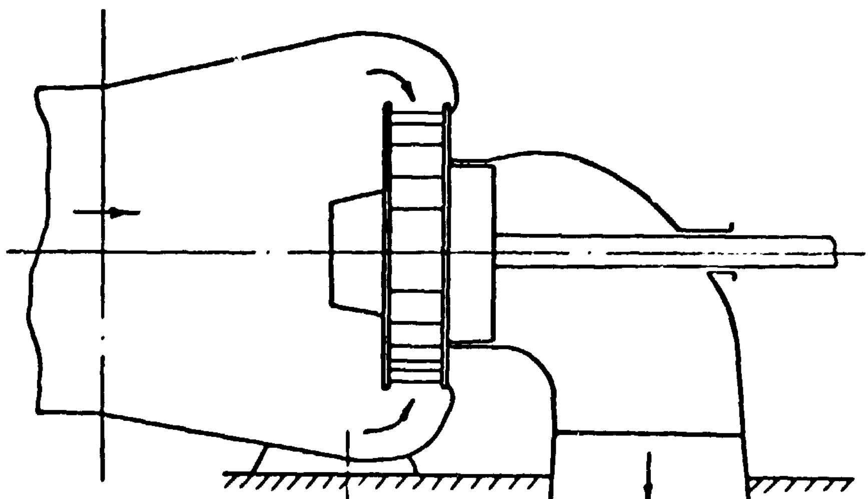
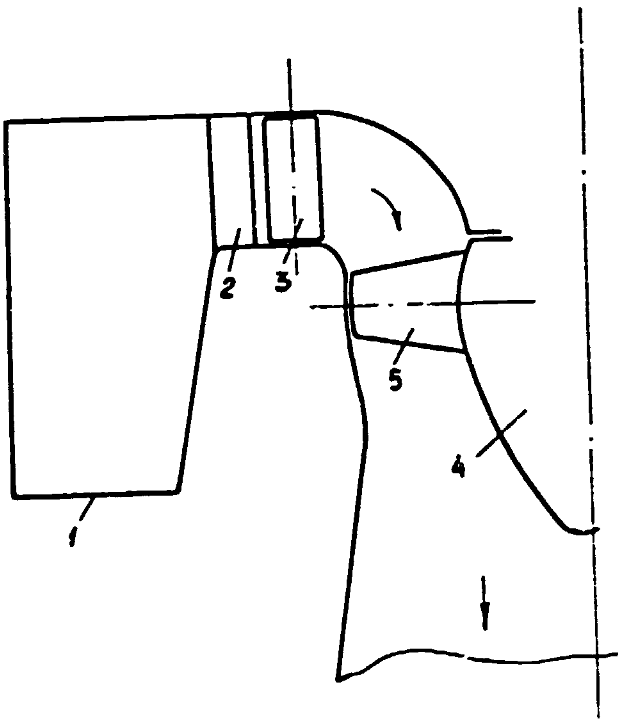


Figure 10. Conventional design of water passage



Radial sections of the spiral case of the vertical-shaft turbine are of a trapezoidal shape. In the plan a spiral section of the water supply line has a smaller wrapping angle. As a rule spiral cases of vertical-shaft Kaplan turbines are made of concrete.

The difference between water passages of Kaplan and Francis turbines is observed at the runner. In the Kaplan turbine (figure 10) the water flow turns after the wicket gate assembly and assumes the axial direction. Near the runner the water flows between two surfaces of revolution which are nearly cylindrical in form.

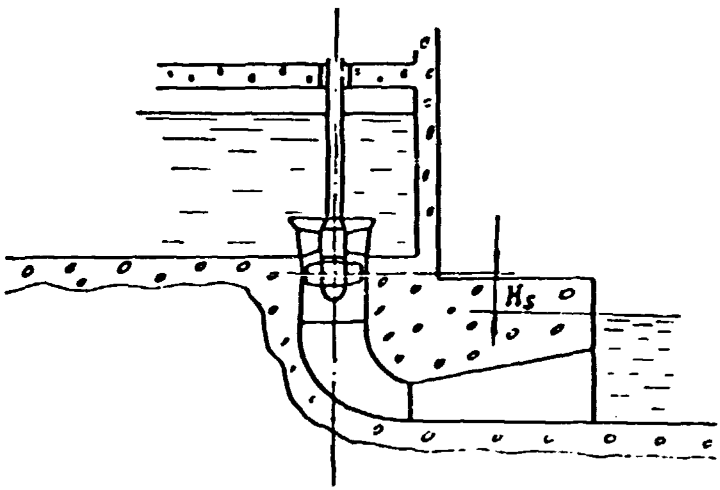
The runner of the Kaplan turbine consists of a hub (4) and blades (5). There is no shroud in the runner.

The main distinguishing feature of Kaplan turbines is that the runner may be provided with adjustable blades. Axes of rotation are normal for the axis of the turbine. Usually the design of the turbine and the runner provides a means of rotation of all blades in synchronism during operation. Thus owing to simultaneous rotation of the wicket gates and blades of the runner it is possible to maintain high efficiency of the turbine over a wide range of operating conditions.

In simplified designs the turbines with fixed gates and adjustable blades or with adjustable gates and fixed blades are used. Characteristics of these turbines as compared with Kaplan turbines with double control are unfavourable when deviating from optimum conditions.

For small hydropower plants vertical-shaft Kaplan turbines may be made of a simplified design similar to Francis turbines. In particular, the water may be supplied to the turbine through an open flume (figure 11). Relatively short straight draft tubes are used. Some simplifications of the governor are feasible.

Figure 11. Turbine with open-flume water supply



6. Horizontal bulb turbines

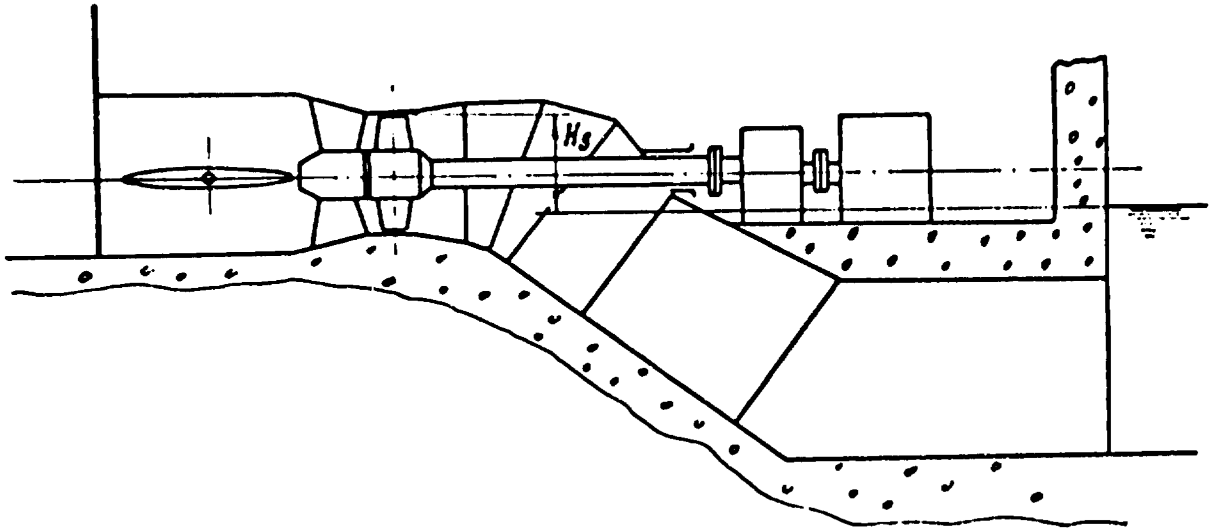
In recent years horizontal bulb turbines have found wide application for low-head small hydropower plants. The design solutions are rather numerous.

Figure 12 shows the scheme of the horizontal bulb turbine with an S-shaped draft tube. In the upstream section there is a stream-lined bulb which houses one support of the turbine shaft. The wicket gate assembly is conical, with adjustable or fixed guide vanes.

The runner is of an axial type with adjustable or fixed blades. At the inlet the draft tube is provided with a cone diffuser which is followed by an S like section with a variable curvature of the mean line. The turbine shaft is brought out to the dry space through the draft tube bend.

Direct-flow bulb turbines with a straight draft tube find some applications. From the shaft through a gear transmission housed in the bulb the energy is transferred via the vertical shaft to the generator installed in the dry room.

Figure 12. Horizontal bulb turbine with S-shaped draft tube



Axial bulb turbines are also employed for installations with vertical or inclined axis.

C. Impulse or free-jet turbines

The classes of turbines described above are reaction turbines in which during transfer of mechanical energy from the water to the runner both potential and kinetic energy change. During operation the water completely fills all the passages of the turbine. Under normal cavitation-free conditions there are no free heater surfaces inside the turbine.

The hydraulic action in impulse turbines differs radically from that of reaction turbines. In inlet passages of the impulse turbine the whole of the store of mechanical energy of water is converted into kinetic energy of a free jet. During interaction of a jet or several jets with the runner blades energy transfer takes place.

There are many different designs of impulse turbines. Below we describe the designs which are of interest from the standpoint of their application at small hydropower plants.

As a rule, impulse turbines are used at hydropower plants characterized by high heads and low discharges. The inlet passage of the turbine terminates in the nozzle set from which the water discharges.

According to the known law of free flow to atmosphere, the water velocity

$$V_j = \varphi (2gH_n)^{1/2} \qquad (23)$$

where = Coefficient of velocity (0.98-0.99 for high quality nozzles)

Under the head H_n = 500 m V_j = 98 m sec^{-1}, and under the head 1,000 m V_j = 138 m sec^{-1}.

Thus the discharge

$$Q = V_j f_j = \varphi f_j (2gH_n)^{1/2} \qquad (24)$$

where f_j = Jet area

The water discharge entering the runner under a constant head is determined by the area of the nozzle opening. The control of power and discharge is effected by nozzles with variable opening area ranging from the maximum value to zero.

The power developed by the impulse turbine is

$$P_t = 9.81 QH_n \eta_t = 9.81\ 2g^{1/2} \varphi f_j \eta_t H_n^{3/2}$$

It is obvious that the power which may be developed by the turbine depends on the head and the opening of the nozzle. Dependence of power on the runner is not explicit. Here only the effect of efficiency is observed.

1. Pelton turbines

Pelton turbines characterized by a relatively simple design find the most extensive application among impulse turbines.

Figure 13 illustrates the scheme of the water supply nozzle (1-needle, 2-casing). The form of the needle and the nozzle plays an important part for operation of the turbine because this form determines hydraulic losses of the jet. With an unfavourable form of the needle and the casing cavitation and associated cavitation failure occur on their surfaces.

For control of the water discharge the needle moves back and forth up to the complete closing when the discharge is equal to zero.

Pelton turbines are used under high heads. Water supply is provided through long penstocks. When controlling the discharge by the nozzle additional loads may take place in the penstock due to pressure rise. A quick closing of the nozzle results in a hydraulic hammer in the penstock due to braking of a large mass of water. For reduction of the hydraulic hammer it is necessary to lengthen the time of closing of the nozzle, which is not possible in all cases.

During operation of the turbine a quick generator load shedding may take place. To avoid excessive acceleration of the runner it is necessary to block the water supply to the runner. Because of a strong hydraulic hammer the nozzle is incapable of such operation. Therefore some designs provide for deflectors.

Figure 13 shows the scheme of a jet deflector. When the deflector enters the flow the direction of the jet changes and the jet passes by the runner.

Figure 13. Water supply nozzle

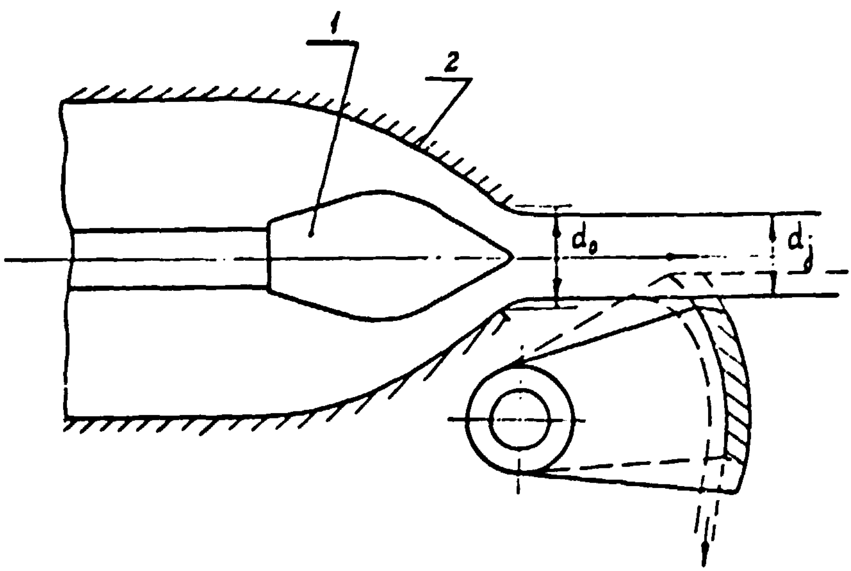
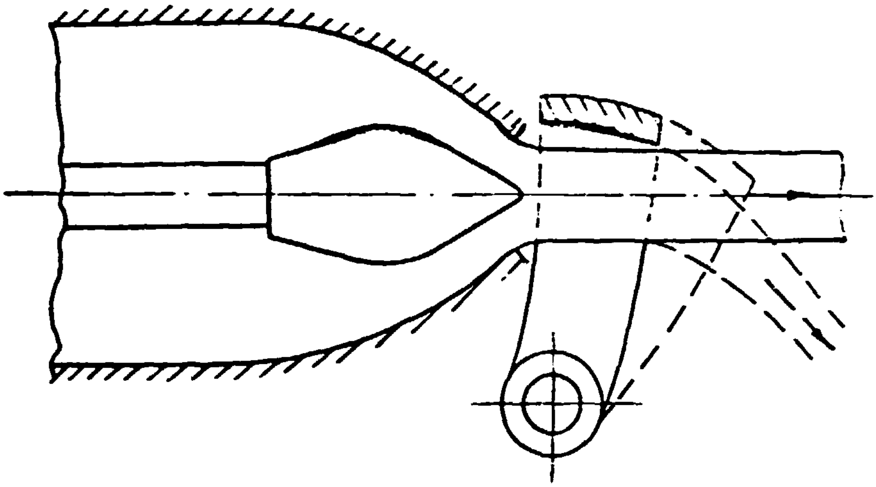


Figure 14 shows the scheme of the deflecting device which may cut off and change the direction of a part or the whole of the jet.

Figure 14. Scheme of deflecting device

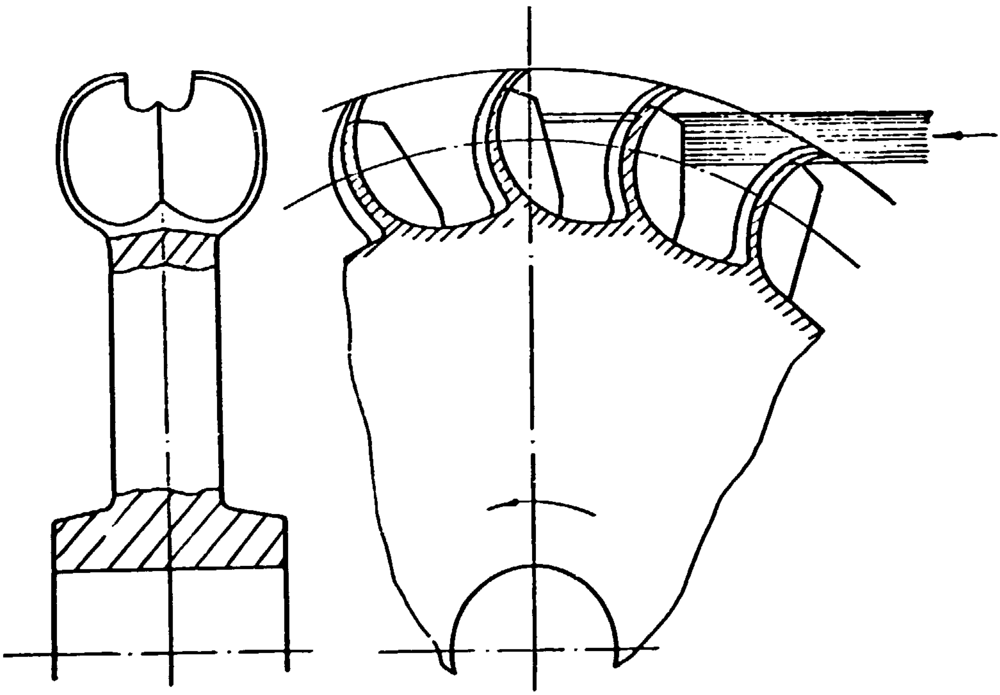


During a speedy change of the turbine power the deflector servomotor shifts the deflector to the required position for a short period of time (2-3 seconds), ensuring the required power with the invariable discharge through the nozzle. Simultaneously, the nozzle servomotor moves the needle to the required position. A full travel of the needle may take 30-40 seconds. During movement of the needle the deflector is withdrawn gradually from the jet and at the end of its motion has no effect on the flow.

The scheme of the Pelton runner is shown in figure 15. The runner is a disk provided with blades and fitted on the shaft. The blades are arranged circumferentially and spaced evenly. The blades have the form of buckets and are symmetric about the plane normal to the turbine axis. The jet from the nozzle enters the jet splitter and is divided into two parts acting on the bucket surfaces. During the flow of the water along curvilinear surfaces of rotating buckets the velocity of the water changes both in magnitude and direction. Due to the change of water momentum the force and the torque are created at the bucket, thus rotating the runner.

During rotation of the runner a successive action upon runner buckets is realized so that the kinetic energy of the jet is used completely. For this certain conditions with regard to dimensions of the runner, dimensions of buckets, their number and rotational speed and velocity of the water in the jet must be satisfied.

Figure 15. Pelton runner



High efficiency requires careful development of bucket geometry and accurate construction. The runner rotates in the air and is always located above the tailwater level.

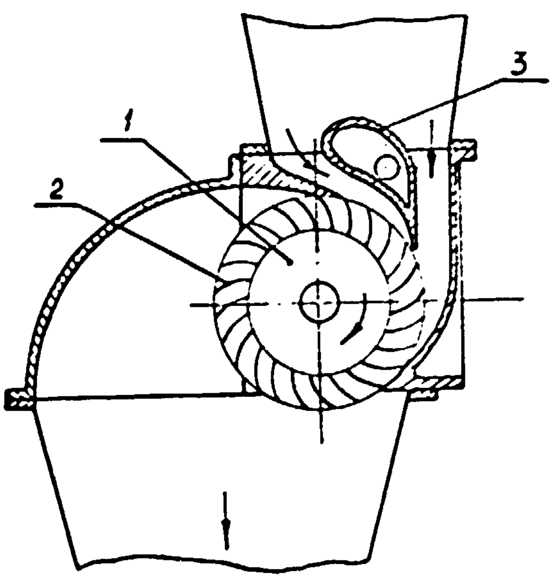
Pelton turbines may be mounted horizontally or vertically. One runner may be provided with several nozzles. An increase in the number of nozzles results, all other things being equal, in a proportional increase in power.

2. Banki-Mitchell turbines

Banki and Mitchell independently evolved the design of the turbine which bears their name. Modifications to the design, such as that of the Ossberger Turbine Fabric Co. in the Federal Republic of Germany, were later introduced.

Figure 16 shows the design of the Ossberger turbine. The horizontal shaft houses the runner (1) with evenly spaced cylindrical bent blades (2). The water supply to the runner blades is effected through a rectangular nozzle the width of which is equal to that of the runner. Some designs provide for a guide vane (3) in the upstream section of the water supply line, by which it is possible to control the water discharge from zero to the maximum value. Thus the power of the turbine is controlled.

Figure 16. Design of Ossberger turbine



The water flows through the runner twice, first from outside and then from inside. During the first cycle 70-75 per cent of the available energy of the flow is used and the remaining portion is used during the second centrifugal cycle.

Among impulse turbines Banki-Mitchell turbines have the highest specific speed. It is well known that an increase in specific speed of Pelton turbines provided with one nozzle n_s = 25-30 is associated with increased losses and lower efficiency. A change to a multi-nozzle design for small turbines is unsatisfactory because of higher costs and complicated maintenance.

The lower limit of application of Francis turbines with regard to the specific speed is 80-100. At n_s < 80-100 efficiency falls substantially.

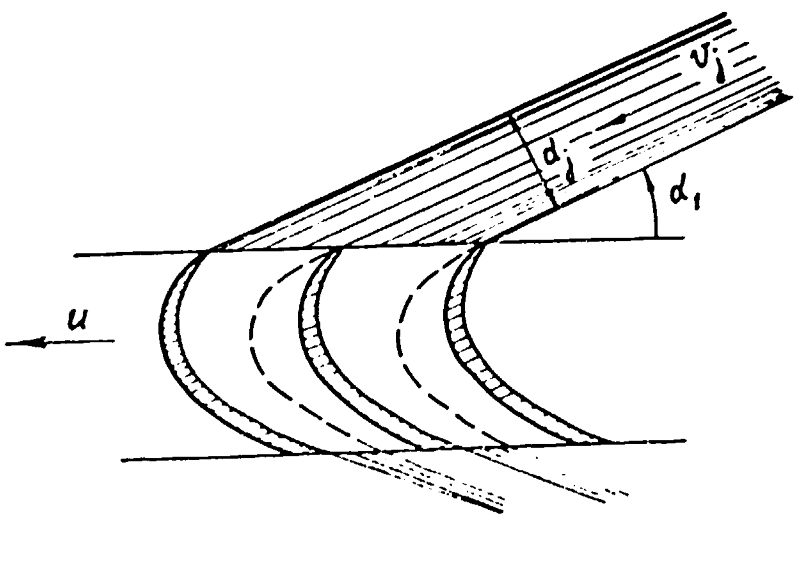
Thus the zone of n_s from 30 to 80-100 cannot be covered by Pelton or Francis turbines.

For turbines of small hydropower plants Banki-Mitchell turbines may be used in the mentioned range of the specific speed.

3. Turgo turbines

One of the variants of impulse turbines is the Turgo or inclined-jet turbine. The design has much in common with the Pelton turbine. Figure 17 shows a flow diagram of the design.

Figure 17. Flow diagram of Turgo turbine



As with the Pelton turbine, a nozzle with an adjustable needle is used in the design of the Turgo turbine. The water supply to the runner is provided at a certain angle α_1 to the plane of rotation of the runner.

The runner consists of blades fixed to outer an ner disks. The blades are of a complicated bucket-line sh the optimum dimensions of which are determined by laboratory stu .

During interaction of the jet with the blades the mechanical energy is transferred to the runner.

Manufacturing methods of Turgo turbines are simpler than those of Pelton turbines.

Investigations demonstrate that the optimum range of application of Turgo turbines is n_s = 30-60, with one nozzle available. Turbines with two nozzles are also used.

III. TURBINE CHARACTERISTICS

A. Power characteristics

Operating conditions of turbines at the hydropower plant are determined by characteristics of the power consumer and characteristics of the turbine. When the hydropower plant operates in a power system where power consumption may vary permanently and continuously, the operating conditions of the turbine change so that the amount of power demand and power supply are equal.

A hydraulic turbine is an energy-converting machine which by its design features and potentialities provides a means for stepless control of power generation over a relatively large range. However, qualitative characteristics of the turbine during operation under different loads expressed in terms of efficiency vary over a wide range.

Since power developed by the turbine is

$$P_t = 9.81QH_n\eta_t \tag{25}$$

then under a constant head the discharge through the turbine has an appreciable effect on the power.

For Francis and propeller-type turbines the discharge is controlled by variation of the gate opening a_0. Usually a_0 signifies the diameter of a circle tangent to two adjacent gates, and in this case it contacts one of the gates at the trailing edge.

During operation of the turbine at the hydropower plant changes in the head with time are possible, but they are relatively slow.

The efficiency η_t depends on operating conditions of the turbine. The complicated interrelations between the main operating parameters of the turbine may be established only through model tests of the turbine in the laboratory.

The model of the turbine must possess geometric similarity with the full-size turbine with regard to all elements of the turbine water passage beginning from the water passage inlet to the outlet of the draft tube.

During power tests of the turbine in steady-state conditions of the test rig the following characteristics are measured simultaneously: net head (H_n); discharge (Q); rotational speed (n); and hydraulic torque on the runner (M_h).

During the change over from one condition to another all the measured values may vary over a wide range. For creation of conditions suitable for comparison of turbine characteristics the latter are reduced to unified conditions.

The reduced turbine parameters are calculated by

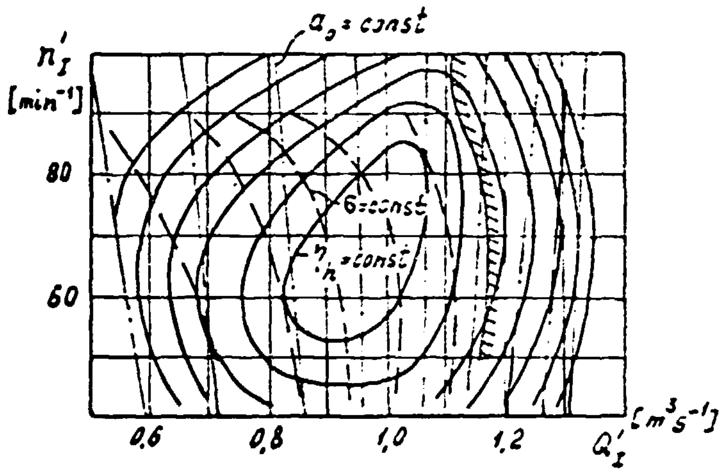
$$n_1' = \frac{nD}{(H_n)^{1/2}}, \quad Q_1' = \frac{Q}{D^2\,(H_n)^{1/2}}, \quad P_{t1}' = \frac{P}{D^2\,H_n^{3/2}} \tag{26}$$

These formulae differ from formulae (17)-(19) because in the reduction of characteristics to the turbine with $D = 1$ m and $H_n = 1$ m the hydraulic efficiency η_h is not scaled.

The turbine model tests are carried out over a wide range of variation of operating conditions at different openings a_0 of the wicket gate and different blade positions Ψ of the Kaplan runner.

Figure 18 shows the hill diagram of the Francis turbine. Here the curves of constant opening of the wicket gate assembly a_0 = const are plotted on the n_1' - Q_1' co-ordinates and the curves η_h = const are plotted as well.

Figure 18. Hill diagram of Francis turbine

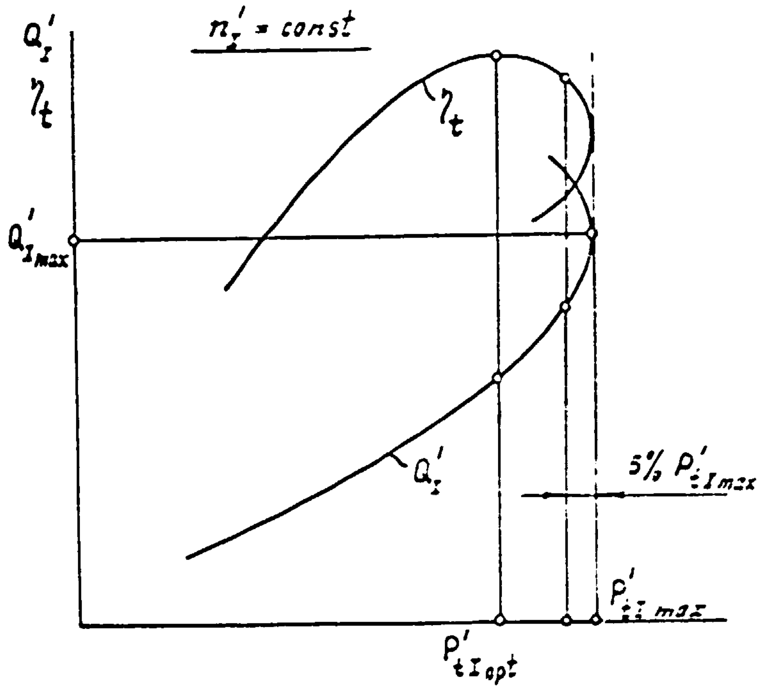


During operation of the turbine at the power plant the rotational speed is maintained constant. In this case if the head is assumed to be constant then n_1' = const.

Of particular interest are the characteristics showing P_{t1}' versus Q_1' at n_1' = const. Figure 19 illustrates the typical characteristics of the Francis turbine. It is seen that an increase in Q_1' through opening of the wicket gate results in a rise in power P_{t1}' up to a certain limit. With a further increase in opening of the wicket gate a_0 and in discharge Q_1' the power P_{t1}' decreases. This is because of a rapid decrease in efficiency. Figure 19 also shows η_t varying with Q_1'.

It is obvious that operation of the turbine at $Q_1' > Q_{1max}'$ is inefficient because it is not associated with a rise in power. Usually the results of model tests determine the conditions in which the turbine has the maximum value of the reduced power. The limiting conditions with regard to power beyond which it is not recommended to operate the turbine are frequently assumed as 0.95 P'_{t1max}. In figure 18 there is a dashed line beyond which turbine operation is inefficient within the range of high Q'_1. This is a power limit line. From the standpoint of the design this limit is reached by the opening of the wicket gate assembly.

Figure 19. Typical characteristics of the Francis turbine



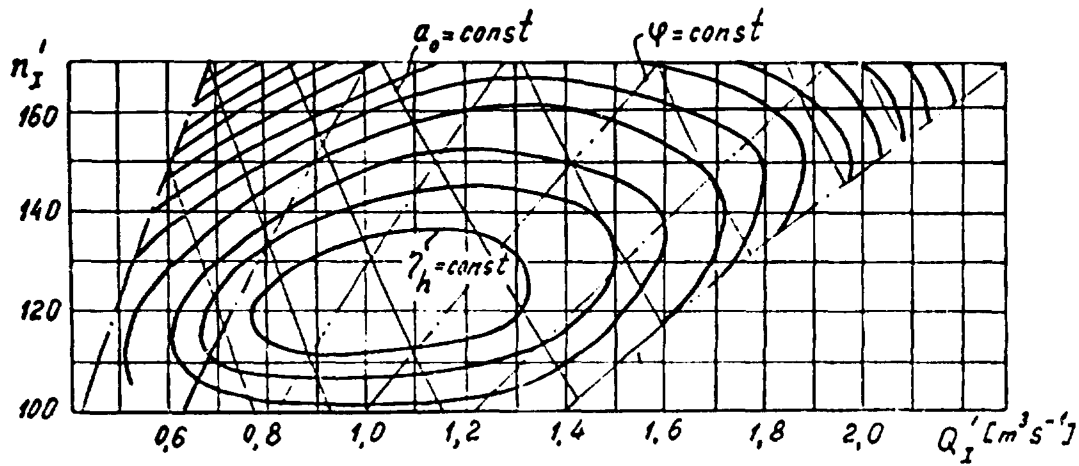
In principle the propeller turbine hill chart does not differ from the Francis turbine characteristics. The difference is that the area Q'_1 in the characteristics with high efficiency is considerably less in propeller turbines.

The propeller turbine is a variant of the Kaplan turbine with fixed runner blades. The propeller characteristics vary with the blade position Ψ. During opening of the runner blades the field of the characteristics covering high efficiency shifts towards higher values of Q'_1.

Thus, with variation of the fixed position of the runner blades and with other unchanged structural elements it is possible to enlarge the field of turbine application.

Figure 20 shows the characteristics of the Kaplan turbine. It was derived from generalization of all propeller characteristics at different blade position. The characteristics of the

Figure 20. Characteristics of the Kaplan turbine



Kaplan turbine demonstrate the optimum relationship between the blade position and the opening of the wicket gate a_0. The design of the Kaplan turbine provides for stepless control of the wicket gate opening and the blade position. With double control use of the Kaplan turbine results in a wide range of values Q_1' at which the efficiency is high.

In modern use of Kaplan turbines at small hydropower plants the design in which the wicket gates are fixed and the runner blades are adjustable finds application as a variant. Sometimes turbines with such a mode of control are called Tomman turbines. The region of high efficiencies of such turbines is wider than that of propeller turbines and narrower than that of Kaplan turbines.

The most simple design of the axial-flow turbine includes fixed wicket gates and fixed runner blades. Power and discharge control under constant head is excluded. The turbine can operate in either mode only.

The power limit curve is not plotted on characteristics of Kaplan turbines. Usually an increase in Q_1' during operation of the turbine is limited by cavitation conditions. At high values of Q_1' differing considerably from optimum values the cavitation factor : rises rapidly. Therefore in conformity with the operating conditions of the particular power plant the maximum value Q_1' is determined by cavitation-free conditions of operation.

It was noted above that the power developed by the Pelton turbine may be controlled by the change of water discharge supplied by the nozzle. By variation of the needle position it is possible to change the discharge from zero to the maximum limiting value governed by the design.

Let us now consider the Pelton turbine hill chart, an example of which is shown in figure 21.

First we note that in Pelton turbines the optimum reduced speed n_1' : 40 min^{-1}. With variation of n_1' the efficiency declines rapidly. Second, with variation of the turbine discharge by means of the nozzle needle the efficiency remains high over a wide range of operating conditions. Third, at constant opening of the nozzle the value Q_1' remains constant irrespective of the value n_1'.

The characteristics of the Turgo turbine with an inclined nozzle (figure 22) are almost similar to those of the Pelton turbine. The difference lies in higher specific speeds due to higher values of Q_1'. As with the Pelton turbine, the reduced speed in optimum conditions is about 39-40 min^{-1}.

Figure 23 illustrates characteristics of the Banki-Mitchell turbine manufactured by Ossberger Turbine Fabric Co. The distinguishing characteristics of this turbine are the ability to control the discharge over a wide range. The water supply nozzle of the turbine is divided into two sections, the relative widths of which are respectively equal to 1/3 and 2/3 of the full width. Each section is provided with an independently driven vane (3) (figure 16). One section may be closed for the flow, while the other one may be full or partially open. In this case it is possible to exercise the step-control of the discharge when the maximum discharge of the turbine is 3/3, 2/3 or 1/3 of the limiting value. Additional control is realized by rotation of the vane.

As may be seen from figure 23, this mode of control ensures high efficiency over a wide range of discharge and power variation.

Figure 21. Hill chart of the Pelton turbine

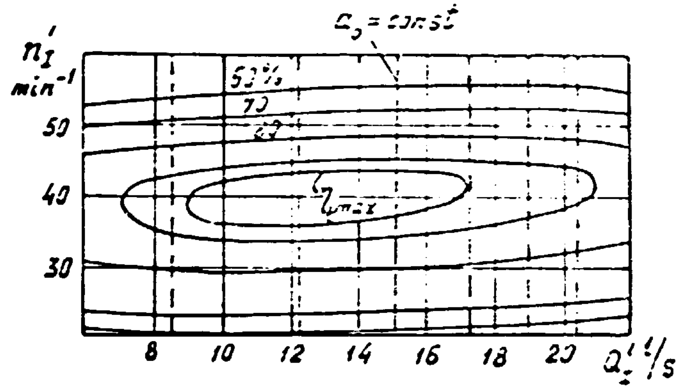


Figure 22. Characteristics of Turgo turbine with inclined nozzle

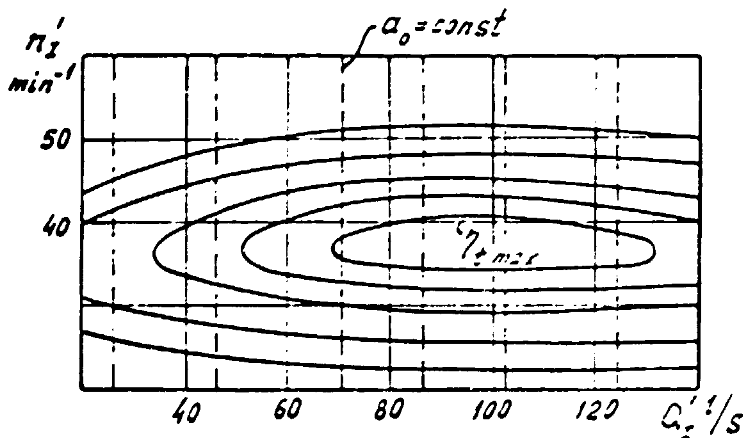
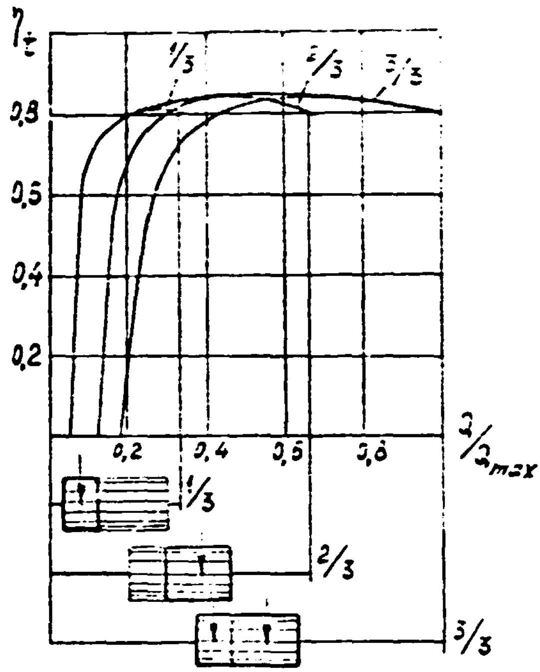


Figure 23. Characteristics of the Banki-Mitchell turbine



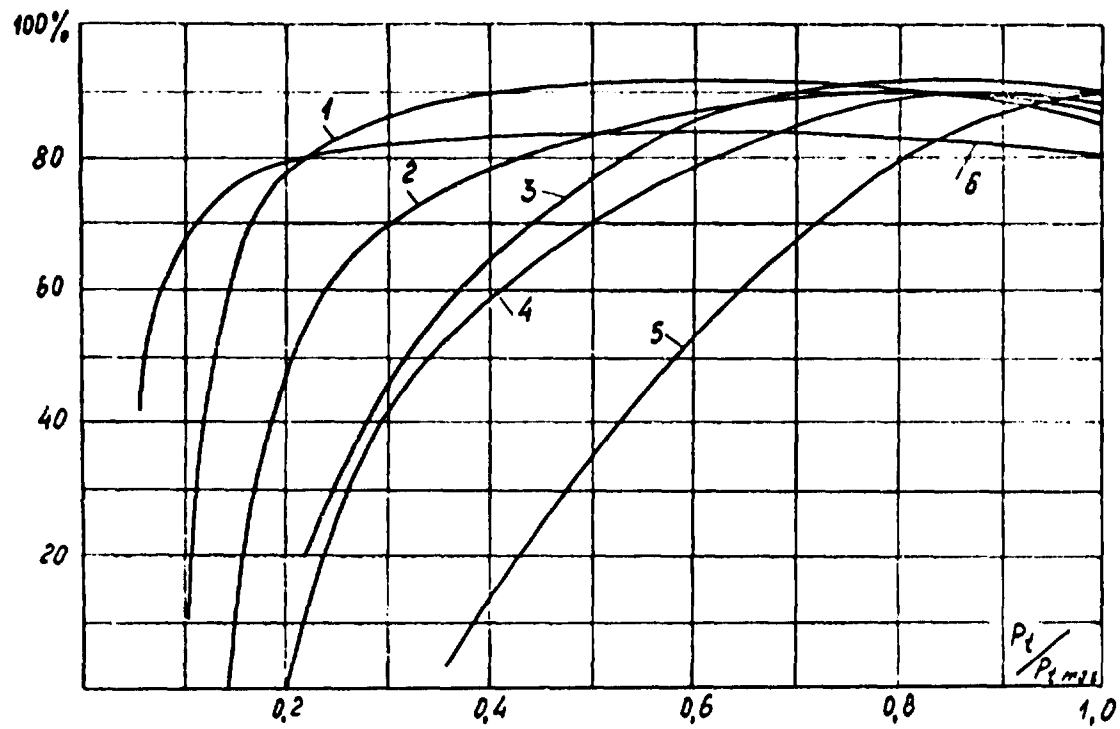
B. Comparison of turbines by their characteristics

Operating conditions of the turbine at the power plant are determined by the generator load in the power system and by the conditions existing for operating parameters. In general, appreciable changes in the net head and discharge governed by the river flow and various other conditions may take place at the hydropower plant. Cross correlation of water conditions of the power plant with the power delivered to the system is the function of the turbine governor. The main idea is to ensure the most efficient utilization of water resources.

From the above characteristics of turbines of different types it is seen that efficiency varies with the change of operating conditions. These changes differ from one another in turbines of different types.

Let us compare turbines according to their efficiency for operating conditions corresponding to the optimum value of n_1' for each type of turbine. Figure 24 shows the characteristics obtained at constant H_n and constant n for the relative value of power. It is assumed that the maximum power of all types of turbines is similar.

Figure 24. Turbine characteristics at constant H_n and constant n



Only a qualitative analysis will be attempted because the turbine characteristics within each type also differ from one another. From the plotted qualitative curves it follows that the turbines of different types respond differently to variations of power. From this standpoint the best turbines are Pelton 1 and Kaplan 2 turbines. The characteristics of Francis turbines 3 are lower. Propeller turbines 5 exhibit the most unfavourable characteristics.

From the compared characteristics it is seen that for axial-flow hydraulic turbines the most efficient means of control is variation of the runner blade position (Tomman turbine 4) as compared with operation of the wicket gate assembly (propeller turbine 5).

The modern Banki-Mitchell turbine 6 has adequate control performances. However, the maximum level of efficiency in these turbines is lower than that of reaction turbines.

The dependence of power characteristics of turbines of different types on the head may also be traced. As may be seen from these characteristics, variation of the head entailing the change of n_1' at constant n produces an effect on efficiency. The analysis of characteristics demonstrates that with an increase in the head above the optimum value a decrease in n_1' takes place followed by a decline in efficiency. The reduction of the head results in an increase in n_1'. Here the efficiency declines sharply, especially after a certain value observed within (0.4-0.7) H_{opt} for different turbines.

Kaplan turbines ensure the best performances. In these turbines the value of efficiency remains high over a wide range of head variation. In Francis, Pelton and Turgo turbines efficiency decreases rather quickly with a declining head.

The distinguishing features of different types of turbines govern the conditions of their application at hydropower plants.

Under plant operating conditions where variation of the power and the head is likely to take place over a wide range, it is expedient to use Kaplan turbines with double control. The contraction of the load variation range may result in the justified application of Tomman or propeller turbines. Operation under permanent load makes it possible to use uncontrollable turbines with fixed gates and blades.

Pelton, Turgo and Banki-Mitchell turbines can be controlled rather efficiently. All these turbines maintain high values of efficiency over a wide range of loads.

C. Cavitation in hydraulic turbines

Under certain conditions operation of the turbine at the hydropower plant may be accompanied by cavitation in the flow.

It has been known that water temperature at which evaporation takes place is pressure-dependent. At low pressure vapour bubbles may be formed in the water flow even at the relatively low water temperature which is equal to the surrounding temperature.

In conformity with the fluid motion law the zones of lower pressure in the turbine water passage occur at the point of highest velocity. At this very point the bubbles are most likely to be formed. If the bubbles are numerous the optimum conditions of the flow about the surfaces are violated, which results in higher losses and a decrease in efficiency.

Cavitation includes the formation of gas cavities in the flow which are collapsed after the bubbles have passed to the zone of higher pressure. Condensation in progress at that time is accompanied by an instantaneous rise in pressure and local temperature elevation. A non-stationary rapid process of formation and collapse of cavitation ca.erns is accompanied by high-frequency wave effects resulting in pressure fluctuation, vibration of units and air noise.

During a long period of operation intensive cavitation may bring about the pitting of the material of the runner and other elements of the unit.

Thus, for normal prolonged operation of the turbine conditions must be created under which cavitation would not take place in the flow, while avoiding the failure of streamlined elements and a decrease in efficiency.

In reaction turbines cavitation may develop in runner blades (profile cavitation), in gaps formed between outer surfaces of the blades of axial-flow turbines (gap cavitation), and in the space behind the runner during deviation from optimum operating conditions of fixed-blade turbines (space cavitation).

In Pelton and Turgo impulse turbines cavitation may develop in the needle of the nozzle assembly if its geometry is unfavourable.

D. Cavitation characteristics

Experiments demonstrate that usually cavitation takes place in the water passage where pressure is equal to or below the pressure of saturated water vapour P_v. Dynamic rarefaction of the flow depends on the kinetic head, which in its turn varies directly with the turbine head.

Thoma's sigma in the dimensionless form is

$$\sigma = \frac{H_{sv}}{H_n} \tag{27}$$

which shows the value of dynamic rarefaction in the turbine.

At cavitation free operation of the turbine the minimum pressure in the flow should not be less than the pressure P_v. Therefore it is necessary to meet the condition

$$B - H_s - \sigma H_n > H_v \tag{28}$$

where B = Atmospheric pressure.

H_s is the suction head, which is counted off from the downstream water level to the horizontal plane connected with a certain turbine element, and

$$H_v = P_v/\rho g$$

In axial-flow vertical-shaft hydraulic turbines the representative elevation of the turbine is usually the pivot of runner blades (figure 12), in vertical-shaft Francis turbines it is a mid-plane of the wicket gate assembly (figure 4), and in horizontal-shaft turbines it is the highest elevation of the runner blades (figure 11).

Cavitation coefficient σ is determined during model tests of hydraulic turbines in cavitation test units provided with a system of water circulation around a closed path. Inside the test unit a vacuum pump lowers atmospheric pressure $P_B = \rho gB$ to the level at which cavitation in the turbine is at its highest and the efficiency declines. As a result of the test the critical value of the turbine cavitation coefficient is calculated as

$$\sigma = \frac{B - H_s - H_v}{H_n}$$

using the known measured values of operating parameters.

Cavitation tests of turbine models are carried out for different operating conditions of the turbine over the whole anticipated range of operation. On the basis of test results cavitation characteristics of turbines are constructed with the curves σ = const plotted on the turbine hill charts (figure 18), or they are constructed in the form of individual characteristics (figure 25).

The analysis of cavitation characteristics of the turbines shows that the coefficient σ depends on operating conditions. In particular σ increases with an increase in Q_1' at n_1' = const (figure 25).

E. Cavitation-free conditions of turbine operation

At the hydropower plant the turbine shall be installed so that no cavitation takes place in the turbine in any operating conditions. For this purpose the type of the turbine and its series must be selected for the specified ranges of head and power variation. Besides the suction head H_s shall be defined for the turbine.

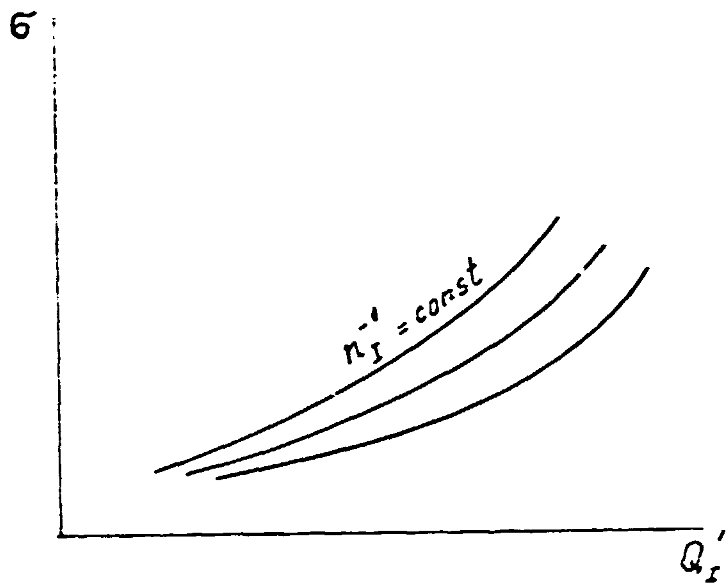
From the formula (4.4) it follows that

$$H_s < B - H_v - \sigma H_n \qquad (29)$$

The atmosphere pressure B depends on local conditions connected with location of the turbine above sea level. Moreover, B depends on climatic conditions in which the reduction of the atmospheric pressure below the average value by 0.35 0.45 m of water column is possible.

The pressure H_v of saturated water vapour is within 0.08 0.43 of water column at water temperature 5°-30°C. For particular conditions of turbine installation B and H_v are usually fixed by the mean level.

Figure 25. Cavitation characteristics



The suction head H_s is selected at the stage of design proceeding from the conditions of the powerhouse layout and construction features. For small hydropower plants the efficient range of values H_s is within 2-6 m. Average design values H_s are 1-3 m. It is obvious that the particular value H_s depends on dimensions of the turbine as well.

Formula (29) must be followed to satisfy cavitation-free operating conditions of the turbine. When selecting the turbine type it should be remembered that the higher the head H_n the less must be the value σ of the turbine.

For turbines operating under low heads σ may reach the limiting values of 1.5-2. For high-head turbines σ = 0.03-0.04.

Cavitation coefficient σ depends on geometrical dimensions of the turbine water passage and in particular on dimensions of the runner passage.

The axial flow hydraulic turbines characterized by high specific speed and the highest values n_1' and Q_1' among reaction turbines feature the highest coefficient σ. The Francis turbines have lower values n_1', Q_1' and σ in the range of optimum operating conditions. The Kaplan and Francis turbines in their turn possess different dimensions of the water passage, each variant of which forms a series of turbines. Among Kaplan turbines several series may be distinguished. Each series is characterized by the optimum field of application. Francis turbines are similar to those of Kaplan turbines in this respect.

Each series of turbines has a limitation of the field of application on the maximum head imposed by cavitation-free operating conditions of the turbine. Installation of the turbine at the hydropower plant with the head exceeding the maximum value requires locating the turbine with high negative values H_s, in other words, a deep setting of the plant and an increase in the construction cost.

The suction head is determined by the following formula:

$$H_s = 10 - \frac{\nabla}{900} - K_s \cdot \sigma \cdot H_n \qquad (30)$$

where ∇ = Tail-water level with respect to sea level (m)

K_s = Safety factor

The safety factor K_s is introduced into the design formula to raise the guarantee against cavitation.

It was stated above that σ is determined by the model cavitation tests through the value of the cavitation coefficient of the unit

$$\sigma_{st} = \frac{B - H_v - H_s}{H_n} \qquad (31)$$

at which the change of power characteristics takes place under the given operating conditions. These operating conditions may be specified with some errors.

When passing from the model to the full size turbine a certain disturbance of similarity of flows connected with the scale effect takes place. Usually in large turbines σ is of a higher value than in the model.

Under conditions corresponding to the critical value σ cavitation develops in the turbine. In full size turbines installed at hydropower plants it may bring about pitting in the runner.

Based on the above considerations, the safety factor K_s, the value of which depends on the series and type of the turbine and power, is introduced. It may be determined from existing structures that

$$K_s = 1.1 - 2.0$$

It is recommended that the value of K_s be increased to 2.5 for complete elimination of cavitation. Application of cavitation-resistant materials makes it possible to reduce the value K_s.

It should be noted that an increase in the safety factor leads to either a deep setting of the powerhouse or the use of low-speed turbines and enlargement of their dimensions.

IV. STANDARDIZATION OF TURBINES FOR SMALL HYDROPOWER APPLICATION

A. Objectives of standardization

The construction of small hydropower plants involves high specific costs. Intensive application of unified mass-produced elements both in civil works and electromechanical equipment would make small hydropower plants more competitive. The recent renewal of interest in developing the hydropotential of small stream flows also calls for upgrading their economic viability both in construction and operation.

At present it is still hard to delineate with sufficient accuracy the limits of application of the terms "small hydroelectric plant" and "small hydroturbine". Further development of small hydropower sources will determine more accurately both the minimum and maximum unit capacity fields.

Let us regard units with installed capacity of 50 kW $\leq$ Pt $\leq$ 10,000 kW as covering the range of small hydropower applications.

Depending on site conditions, small hydropower plants can develop heads in the range of 1 m < H_n < 1,000 m which will be feasible for practical application.

The zone of operating parameters for small hydroturbines can be determined from the following formula

$$P_t = 9.81 H_n Q \eta_t \text{ kW}$$

Figure 26 gives this zone in logarithmic co-ordinates.

Since $$\log H_n = \log \frac{P_t}{9.81\eta_t} - \log Q$$

taking an averaged value of η_t, we obtain the linear relationship between H_n and Q for constant P_t in logarithmic co-ordinates.

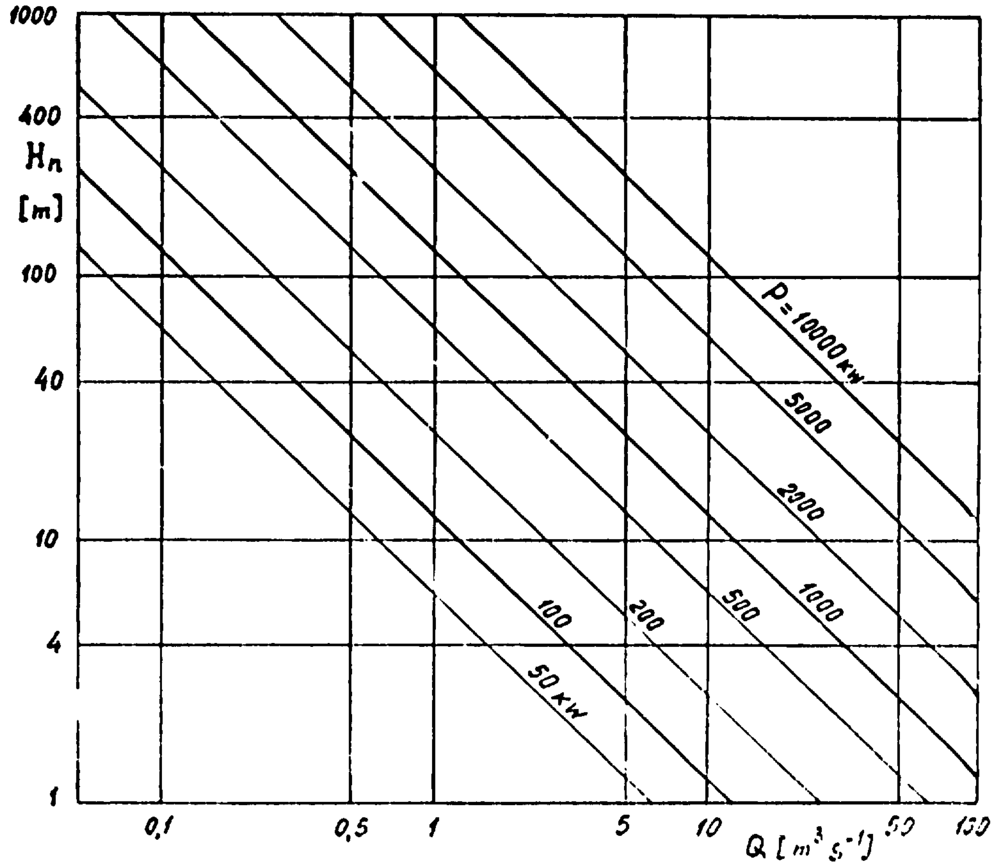
The problem of small hydroturbine standardization consists in development of a number of series and sizes of turbines which could be effective tools in exploitation of water resources for any combination of H_n, Q and P_t fitting into the zone shown in figure 26.

Finding a solution to the problem of turbine standardization will allow development of turbine design types, unification of structural elements, as well as a change-over from the custom-made approach to series production. All this would make it possible to reduce the cost of equipment and delivery time.

B. Line-up of series of reaction turbines

One of the major problems in standardization of small hydroturbines is to establish a practicable line-up of turbine series. The experience gained in hydraulic turbine engineering indicates that the field for potential application of reaction turbines

Figure 26. Logarithmic co-ordinates of operating parameters for small hydroturbines



can be covered by two types, namely the Francis turbine and axial-flow turbines. Axial-flow turbines are used in the low head zone, and Francis turbines in the high-head zone.

The turbine characteristics including parameters $n_1^{\prime}$, $Q_1^{\prime}$, σ under the optimum operating conditions depend on the relative dimensions of the water passage, with the relative dimensions of the runner having a tangible effect. Replacement of the runner results in a considerable change of energy and cavitation characteristics.

Each series of turbines represents a line up of turbines featuring different absolute dimensions with geometrically similar water passages. All the turbines of the same series are assigned a specific designation.

The turbine size is normally characterized by the runner diameter. In the case of the Francis turbine, the diameter D_1 (figure 4) is fixed. This dimension represents the maximum diameter taken at the inlet edges of the runner blades. There are cases where the characteristic dimension is the diameter D_1 taken at the runner crown or the diameter D_2 taken at the shroud (figure 4). Hence each turbine is characterized by the dimension and designation.

When developing the standard for small hydraulic turbines, the number of required series and a normal line-up of runner diameters should be established. An increase in the number of series and sizes of the turbines covered by the standard allows selection of the effective turbine for practically any operational parameters. But manufacture of a wide range of the turbines would decrease production efficiency and increase the cost of equipment involved. The number of standard types should not be extended beyond the economically justified limits.

We now consider some general concepts that should serve as guidelines in tackling the problem. As mentioned above, to provide cavitation-free performance each series of turbines has a limited field of application with regard to the head.

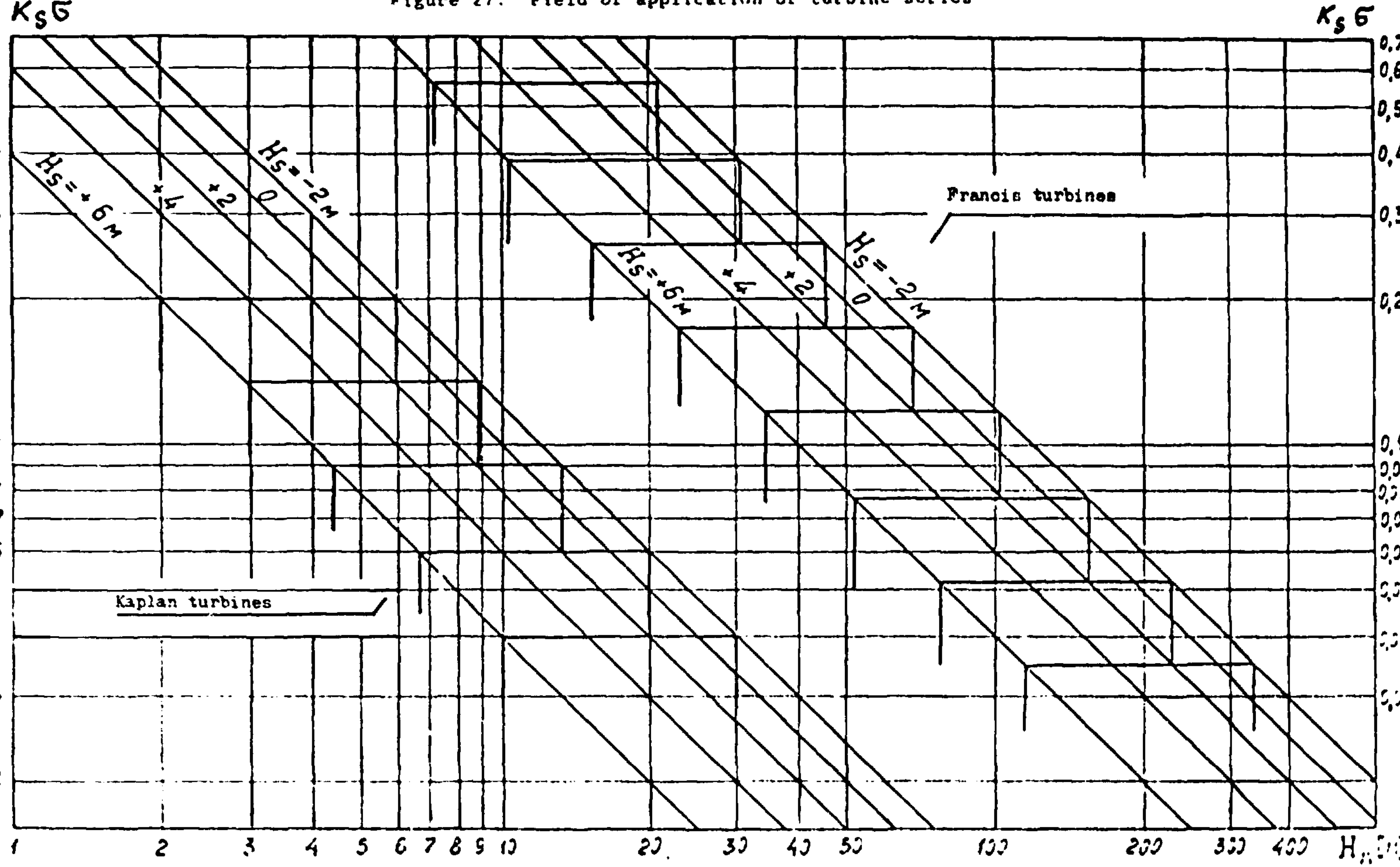
Deep setting is not economically justified for small hydro-power plants. It seems that $H_s = -2$ m might be taken as the limiting suction head.

The relationships between the limiting values of $K_s\sigma$ and the head H_n have been plotted for various suction heads according to the formula:

$$K_s\sigma = \frac{B-H_s}{H_n}, \ \log(K_s\sigma) = \log(B-H_s) - \log H_n \quad \text{(Figure 27)}$$

The minimum value of the cavitation coefficient sigma σ secured for high-head Francis turbines is equal to about 0.03 on the 5 per cent output margin line and at the optimum values of $n_1^{\prime}$. If the cavitation safety factor is assumed to be K_s = 1.15 1.20, then H_s = 400 m would be considered the limiting head for small hydraulic turbines. Increase in the limiting head to 500 m at the assumed safety margin will result in a deeper setting

Figure 27. Field of application of turbine series



of the hydroplant and H_s = -7.5. However, such a solution is not practicable.

We now illustrate one of the possible alternatives for selection of the number of series and the field of application for each series (figure 27).

It is assumed that for the limiting value of the cavitation coefficient sigma σ which is characteristic of the turbine, the minimum suction head should not exceed H_s = -2 m. Taking a minimum cavitation coefficient σ = 0.03 and safety factor K_s = 1.2, we will secure the highest head series for the head range 115-340 m. It is further assumed that the limiting head of the next series for H_s = -2 m is equal to the limiting head of the preceding series for H_s = 2 m. Then the ratio of the limiting cavitation coefficients is 1.5, that is

$$\sigma_{n+1} = 1.5\sigma_n$$

Following this reasoning it is necessary to have 8 series of Francis turbines for the head range H_s = 340-20 m. The limiting heads and cavitation coefficients are given in table 2.

Table 2. Limiting heads and cavitation coefficients of Francis turbines

Series	F8	F7	F6	F5	F4	F3	F2	F1
H_n^{max} / H_n^{min}	20/7	30/10	45/15	70/25	100/35	150/50	260/80	340/115
σ_{max}	0.47	0.32	0.22	0.145	0.095	0.065	0.043	0.03

Five series of Kaplan turbines should be available for the head range H_n = 30-6 m. The limiting heads and cavitation coefficients are given in table 3.

Table 3. Limiting heads and cavitation coefficients of Kaplan turbines

Series	K5	K4	K3	K2	K1
H_n^{max} / H_n^{min}	6/2	9/3	13/4.5	20/7	30/10
σ_{max}	1.7	1.12	0.75	0.5	0.33

The procedure illustrated above for selection of the number of series and the head range for each series is not the only possible way.

When assigning the number of series and specific speed for each series other conditions are also taken into account, and the practical experience of the company working out the standard is a significant factor. The large turbine manufacturers have accumulated considerable experience in development of highly efficient turbines for large-scale installation to meet the different operating conditions. In some cases, the number of developed series of Francis and Kaplan turbines exceeds the figures given in the example.

Increase in the number of series allows selection of a more efficient turbine in terms of energy and cavitation characteristics for each specific case.

The exceedingly small number of series improves the manufacturing process, but in some cases it has a negative effect on the quality of equipment.

C. Normal line-up of diameters

Each series consists of a number of turbines with geometrically similar water passages that differ from each other by the nominal dimension, which is the diameter D_1 of the runner.

Some general aspects related to the selection of a normal line-up of the turbine dimensions are considered below.

Suppose that a series of turbines for which the hill chart is available has been selected for certain rated values of the turbine power P_t and head H_n. Select the rated point on the 5 per cent output margin line of the hill chart (figure 18). Then according to formula (19) and taking into consideration that

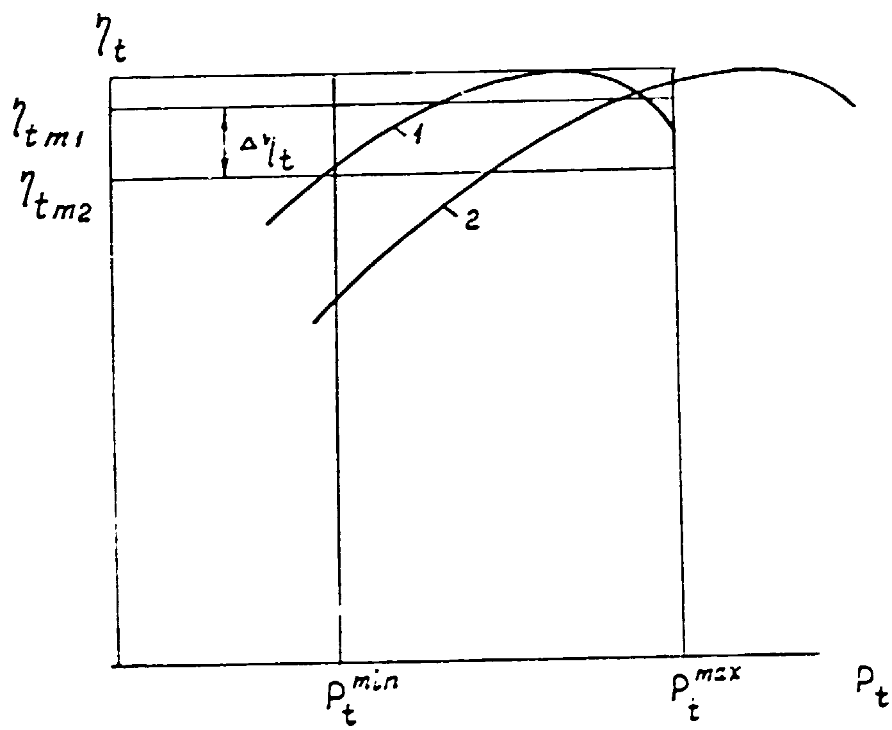
$$\dot{P}_{t1} = 9.81\ \dot{Q}_1 \eta_t$$

the diameter D_1 of the runner can be calculated. The minimum size of the runner will provide the specified power output (curve 1, figure 28). Should a bigger size of the runner be taken for the same power output the rated point on the hill chart would shift to the zone of lower values of $\dot{Q}_1$. Figure 28 shows performance characteristics for turbines of various sizes. Curve 2 corresponds to the turbine with a larger-size runner.

When running the turbine, its power output may vary within wide limits depending on the load demand and flow availability. The generator capabilities hinder the output increase, and a heavy power drop decreases the efficiency value. As seen from figure 28, the average value of efficiency appears as $\eta_{tm1} > \eta_{tm2}$ within the power variation range $P_t^{max} - P_t^{min}$. It implies that at a bigger diameter of the runner the turbine governing is less effective. The bigger the difference in the runner diameters the bigger is the value $\Delta\eta_t$ (figure 28).

If the power regulation range and the allowable decrease of η_t are specified it is possible to determine the deviation of diameter for two runners adjacent in terms of their size. Similar work can be conducted for all series of Francis and Kaplan turbines.

Figure 28. Turbine performance characteristics



The following conclusions may be drawn on the basis of current experience of turbine engineering:

(a) Ratio $K_D = D_{n+1}/D_n$

lies within 1.05-1.2 (where D_{n+1} and D_n are diameters of the runners of two adjacent turbines);

(b) For the values of K_D up to 1.07, close correspondence of the designed turbine with its optimum operating conditions is provided;

(c) The lower value of K_D should be taken for larger-size turbines;

(d) K_D can be raised to 1.12-1.15 for very small turbines;

(e) Higher values of K_D are allowable for Kaplan turbines.

It should be noted that the power variation when changing over to another diameter under the assumed conditions is proportional to K_2^D, that is

$$P_{t,n+1}/P_{tn} = K_D^2$$

It means that the maximum power varies by 1.1-1.44 times.

D. Field of application of turbine series

Supplementary charts intended for preliminary selection of the turbine series and size as well as other operational parameters must be plotted on the basis of the development of a standard line-up of turbine series and diameters. Some general concepts which serve as the basis for the required plotting are considered below. We assume that there is a hill chart of the model turbine on which the given series is to be based.

First we take the mode of operation lying on the line of optimum value n'_{1c} as the design value. For the Francis turbine, it is practical to take the design value Q'_{1c} lying on the 5 per cent output margin line or close to this zone. For further computations we assume that the values n'_{1c} and Q'_{1c} are known for the given series.

Neglecting the difference in efficiency of the model and life-size small hydroturbines, from (17) and (18) we have

$$n'_1 = \frac{nD_1}{(H_n)^{1/2}}, \quad Q'_1 = \frac{Q}{D_1^2 \, (H_n)^{1/2}}$$

and from this it follows that:

$$\log H_n = -2\log (Q'_{1c} D_1^2) + 2\log Q$$

$$\log H_n = \frac{2}{3}\log \left(\frac{n}{n'_{1c}}\right)^2 + \frac{2}{3}\log \frac{Q}{Q'_{1c}}$$

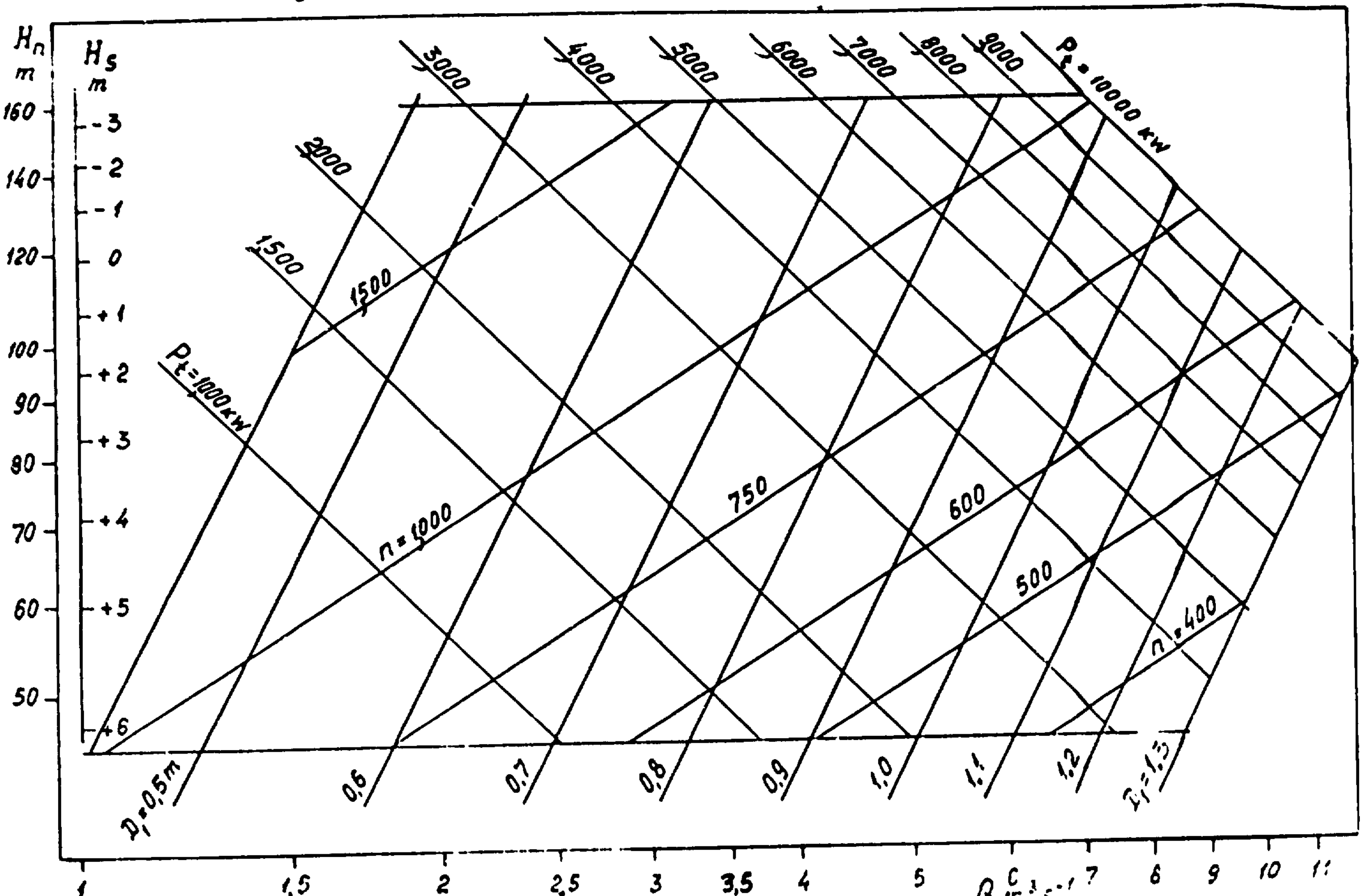
It may be seen from these equations that for D_1 (constant) the relation $H_n(Q)$ is of a linear nature in logarithmic co-ordinates. The lines D_1 (constant) form a family of parallel lines inclined at an angle of [illegible] = arctg2 = 63.43° to the axis logQ.

Relations $H_n(Q)$ for n (constant) are also of a linear nature in logarithmic co-ordinates. The lines n (constant) form a family of parallel lines inclined at an angle of [illegible] = arctg 2/3 = 33.69° to the axis log Q.

In figure 29 is shown, as an example, the field of application for one series of Francis turbines. The model characteristic with

$n'_{1c} = 68$ min^{-1} $Q'_{1c} = 0.75$ m^3 s^{-1} [illegible] = 0.07

Figure 29. Field of application of a series of Francis turbines



in the design mode of operation is taken as the input data. It shows also the lines of the continuous turbine output.

As $\log H_n$ is given by

$$\log H_n = \log \frac{P_t}{9.81 \eta_t} - \log Q$$

the lines of P_t (constant) are inclined to the axis log Q at an angle of -45° in logarithmic co-ordinates.

At the same time, the scale log H_s is plotted on the ordinate, which can be used to determine the allowable suction head.

In the study case, the field of application is limited by the suction heads H_s = -3 to 6 m, turbine power 1,000-10,000 kW and runner diameters D_1 = 0.45-1.3 m.

Preliminary selection of the basic turbine operational parameters can be made on the basis of the nomograph in figure 29. For example, the diameter D_1, speed n and suction head H_s can be determined from the specified head and power. But the values D_1, n and H_s obtained in such a way may turn out to be unacceptable.

The runner diameter may not correspond to the values covered by the standardized line-up. The rotative speed may differ from the synchronous one. When adjusting to the nearest recommended magnitude the values D_1 and n obtained from the nomograph, the operating conditions tend to shift from the design point. But the nomograph in figure 29 can not be used to assess whether the obtained deviations are permissible.

In this connection it is good practice to construct first the nomographs identifying the field of potential application for the turbines of the given series with dimensions corresponding to the standardized dimensions and with synchronous speed.

It should be noted that in current practical applications of turbines for small hydropower production both direct coupling of turbines and generators and step-up gearing are accepted.

Direct coupling is always employed when the turbine has a high synchronous speed allowing the use of a high-speed generator. At low values of n, the turbines can be directly coupled to the generator or through step-up gearing.

For direct coupling and 50 Hz frequency, n = 1,500, 1,000, 750, 600, 500, 428.6, 375, 333 and 300 min^{-1}; at 60 Hz frequency n = 1,200, 900, 720, 600, 514, 450, 400, 360 and 300 min^{-1}.

A cheaper step up gearing calls for development of a standard line up of gear sizes with the fixed transmission ratio. Thus application of high-speed generators together with step-up gearing will govern the standard line-up of turbine speeds that differ from the synchronous ones.

The methods for identifying the field of application for turbines of the given series, the size and speed of which correspond to the standard line-up, will not be considered.

When plotting the field of application shown in figure 29. rather definite operating conditions with fixed values n'_{1c} and Q'_{1c} are involved. As seen from the hill charts for each series of turbines, there is a zone where the turbine features high energy characteristics. Any mode of operation in this zone can be taken as the designed one. Let us limit the zone of favourable turbine application by the values of n'_{1max} - n'_{1min} and Q'_{1max} Q'_{1min}.

The designated turbine output under the fixed head can be obtained using turbines of the given series which have the diameters D'_1 for Q'_{1max} and D''_1 for Q'_{1min}. Subject to the equality of discharges, the following can be written with an accuracy determined by the difference in efficiency under these operating conditions

$$K_D^2 = \left(\frac{D''_1}{D'_1}\right)^2 = \frac{Q'_{1max}}{Q'_{1min}}$$

The ratio K_D controlling the normal line-up of diameters, the values of which are given above, corresponds to the ratio of the adjusted discharges in the zone of optimum operating conditions.

The field of application for the turbine of a certain diameter in the logarithmic co-ordinates log Q - log H_n is limited by two parallel lines inclined at an angle of 63.43° to the axis logQ. In figure 30 are shown these zones for the line up of diameters with K_D = 1.08 from the same series of turbines. The field of application for this series is given in figure 29.

The field of application relative to head at D_1 (constant) and n (constant) depends on the magnitude of n'_{1max} and n'_{1min}. In the study case, n'_{1max} is taken equal to 73 min^{-1} and n'_{1min} equal to 63 min^{-1}.

Figure 30 shows the field of effective application for the turbines of the same series but various diameters D_1 (constant) at n = 1,500, 1,000, 750, 600, 500 and 428.6 min^{-1}. At n (con stant) the field of application is limited by parallel lines inclined at an angle of 26.56° to the axis log Q.

E. General laws for Pelton turbines

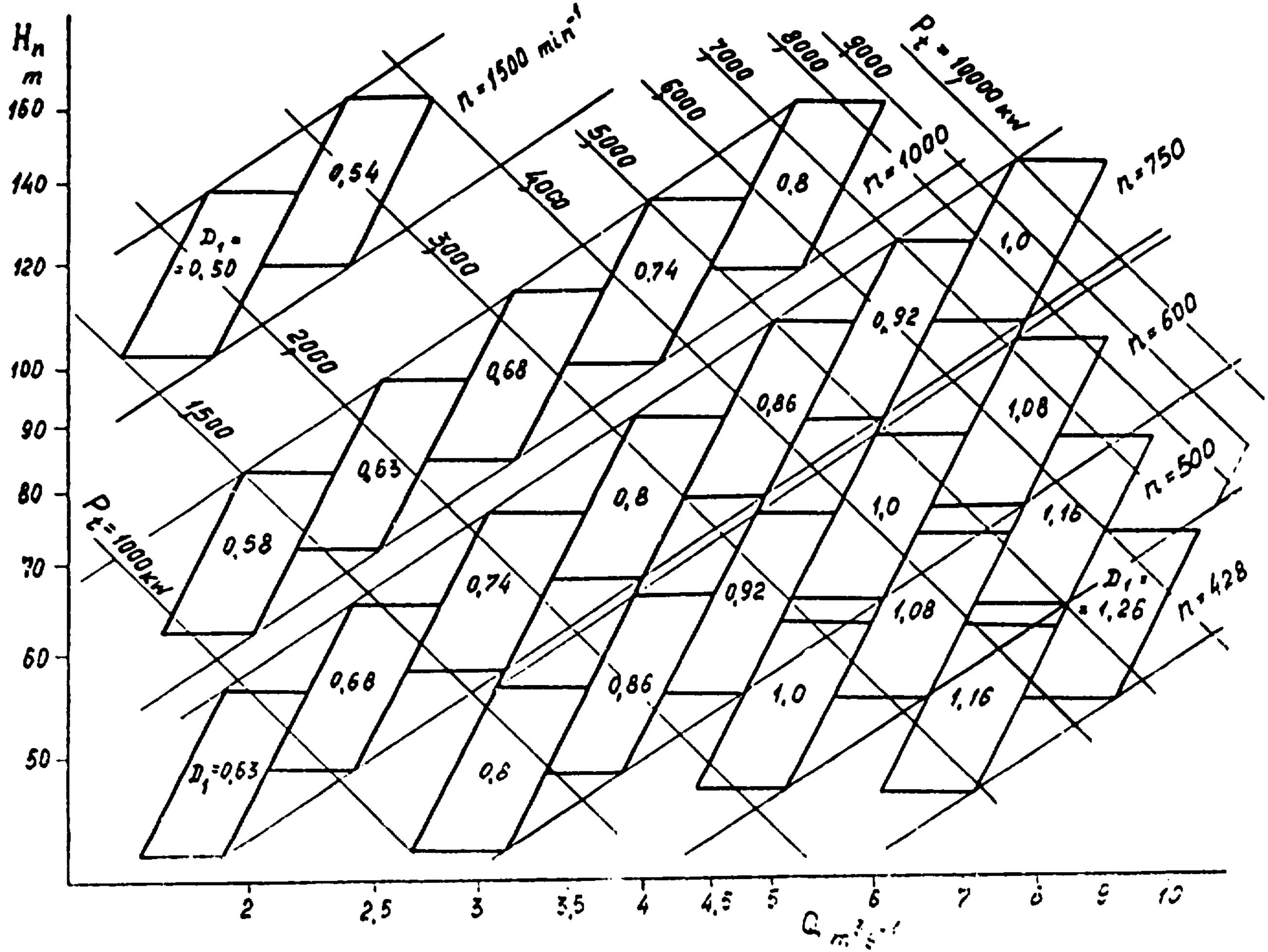
Certain peculiarities in the operation of the Pelton turbine in the zone of maximum efficiencies are considered below.

According to formula (23), the velocity of a jet discharging from the nozzle depends on the head. The interaction of the jet and wheel buckets results in energy transfer to the turbine shaft. The optimum peripheral speed of the runner circle tangent to the central line of the jet should be equal, in an ideal case, to $0.5V_j$ or to $(0.46\ 0.47)V_j$ considering the hydraulic losses.

It follows that

$$u = \frac{\pi D_1 n}{60} = (0.46\ 0.47)V_j$$

Figure 30. Field of effective turbine application for one series



and taking into account formula (23) we obtain the reduced speed

$$n_1' = \frac{nD_1}{H_n^{1/2}} = (0.46\text{-}0.47)\ \frac{60}{\pi}\ (2g)^{1/2} \qquad (32)$$

All Pelton turbines featuring good energy characteristics therefore have in the optimum mode of operation

$$n_1' = 39\text{-}40\ \min^{-1}$$

As seen from the Pelton turbine characteristic shown in figure 21, turbine efficiency tends to drop rather sharply on departure from the optimum values of n_1'.

Applying the general formula (24) to determine the nozzle discharge, we obtain the discharge of the Pelton turbine with a multi-nozzle arrangement Z_j

$$Q = Z_j\ \frac{\pi d_i^2}{4}\ \varphi\ (2gH_n)^{1/2} \qquad (33)$$

Adjusted to D_1 = 1 m and H_n = 1 m, the discharge equals

$$Q_1' = \frac{Q}{D_1^2 (H_n)^{1/2}} = \frac{\pi}{4}\ \varphi\ (2g)^{1/2}\ Z_j \left(\frac{d_i}{D_1}\right)^2$$

or

$$Q_1' = 3.41 Z_j \left(\frac{d_i}{D_1}\right)^2 \qquad (34)$$

This formula indicates that the reduced discharge depends on the number of turbine nozzles and the ratio d_j/D_1.

In the course of discharge control using the needle, d_j varies from zero to a maximum value which governs the maximum turbine discharge.

It has to be noted that the turbines intended for small hydropower application should not have more than two nozzles. Otherwise turbine construction and its operation would become too complicated.

The specific speed of the turbine equals

$$n_s = 3.65\ n_1'(\eta_t Q_1')^{1/2} = (249\text{-}255)(Z_j)^{1/2}\ \frac{d_j}{D_1}$$

The studies conducted show that for d_j/D_1 > (1/6)/(1/7), the efficiency tends to drop significantly. The limiting values of n_s for the single nozzle turbine is 36-42, while for the two-nozzle turbine, it comes to 54-60.

The turbine characteristics become optimum at d_j/D_1 = (1/10)/(1/15). Therefore the specific speed preferable for the single nozzle turbine is 16-26, and that for the two-nozzle arrangement is 22-36 under the optimum operating conditions.

F. Standardization of Pelton turbines

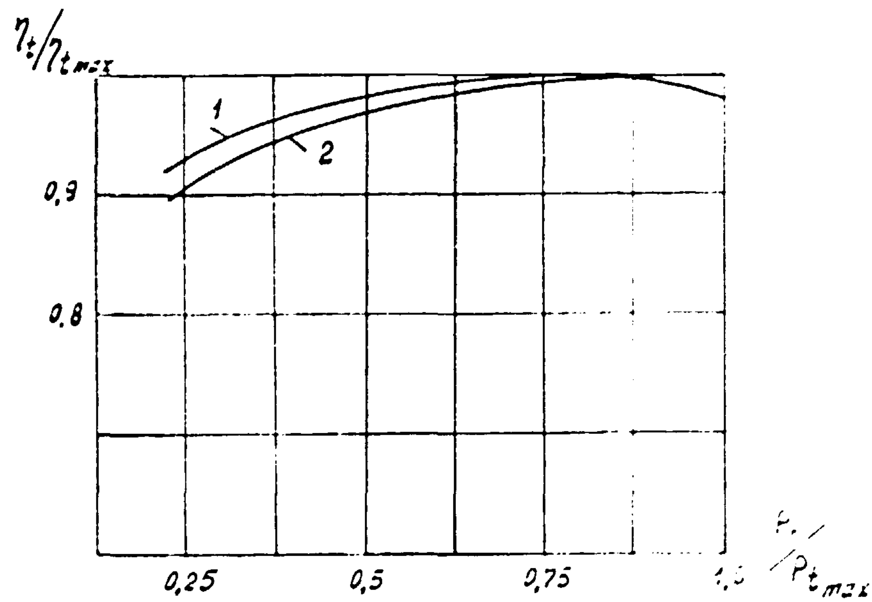
Turbine power is governed by the head and the nozzle dimensions, and does not depend on the runner diameter. In this connection, the jet diameter d_j is one of the major dimensions of the Pelton turbine.

With a change in the diameter d_j, the turbine power, other things being equal, tends to change proportionally to d_j^2.

One of the problems to be worked out in standardization of the Pelton turbine is to establish the line-up of the nozzles differing in basic dimensions, particularly in the maximum jet diameter.

In figure 31 is shown the characteristic of turbine 1 for various loads. It is seen that the turbine features a high efficiency in a wide range of loads. At partial loadings the average operational efficiency η_{tm} 2 tends to decrease, but not significantly.

Figure 31. Turbine characteristics for various loads



It is evident from formula (33) that the relation $H_n(Q)$ in logarithmic co-ordinates is of a linear nature for constant d_j.

Hence

$$\log H_n = -2\log\left(Z_j \frac{d_j^2}{4} \cdot (2g)^{1/2}\right) + 2\log Q$$

In figure 32 are plotted the lines d_j = 0.02-0.25 in the range of possible application for the Pelton turbine with a single-nozzle arrangement at the specific speed n_s = 25.

The lines of equal runner diameter are parallel to the lines d_j (constant). In the given case for d_j/D_1 = 0.1 we have D_1 = 10 d_j. Thus for the entire field Q-H_n the range of runner diameter variation lies within D_1 = 0.2-2.5 m.

Plotted in this field are also the lines n (constant) corresponding to the synchronous speed of the generator. Applying formula (16) we obtain

$$H_n = \left(3.65 \frac{n}{n_s}\right)^{4/3} Q^{2/3} \eta_t^{2/3}$$

$$\log H_n = \frac{4}{3}\log\left(3.65 \frac{n}{n_s}\right) + \frac{3}{2}\log(\eta_t Q)$$

For n_s (constant), the lines n form in logarithmic co-ordinates a family of straight lines inclined at an angle of 33.69° to the axis log Q.

In figure 32 are plotted the lines of constant speed for synchronous values in the range 187.5-1,500 min^{-1}.

The field area Q-H lying above the line n = 1,500 min^{-1} is likely to be excluded from consideration.

The limiting size of the runner is governed by the manufacturing capabilities and its maximum value must be limited.

The manufacturers specialized in production of small hydro turbines can limit the D_1 value by 1 or 2 m.

Assuming D_{1max} = 1.0 m and the field of application with respect to head is limited by 50-500 m, we obtain a field of application for the single nozzle Pelton turbine shown by a thick line in figure 32.

It is seen that the maximum power here does not exceed 3,000 kW. The higher power would require a turbine with double nozzle arrangement. The specific speed of the double nozzle turbine is

$$n_{s2} = n_{s1}(2)^{1/2}$$

provided the other conditions remain unchanged (n_{s1} specific speed of the single nozzle turbine).

The field of application for the double-nozzle turbine tends to shift to the right as the discharge will increase as much as two times. The maximum power is 5,000 MW.

For the study case of Pelton turbine applications, the practical area of standardized turbine application is limited with respect to the head by 50-500 m, to the jet diameter by 0.03-0.1 m, to the runner diameter by 0.3-1.0 m, and to the power by 50-3,000 kW for a single-nozzle arrangement and by 50-5,000 kW for a double-nozzle arrangement.

The turbine featuring higher values of head and power may be custom-made. The preliminary selection can be made on the basis of the nomograph (see figure 32).

One of the basic objectives of normalization and unification of Pelton turbines consists in establishment of a normal line-up of runner diameters, nozzle sizes and wheel bucket sizes. An integrated approach should be followed in working out this problem. The following conditions must be taken into account when specifying the line-up of runner diameters.

A turbine with certain runner dimensions can operate within limited head variations. Figure 21 shows that at departure of n'_1 from the optimum value, the cost effectiveness of the turbines decreases significantly. It is therefore practicable to limit the operating zone by the limiting values of n'_{1max} and n'_{1min}.

Let the area of application for the turbine with the runner D_n be limited by the heads H_n^{min} and H_n^{max} and that for the turbine turbine with the runner D_{n+1} by the heads H_{n+1}^{max} and H_{n+1}^{min}. Assuming further that

$$H_n^{max} = H_{n+1}^{min}$$

and n is equal for both turbines, we will find that according to the formula (32) for reduced (unit) speed,

$$\frac{D_{n+i}}{D_n} = \frac{n'_{1max}}{n'_{1min}}$$

For the specific case considered above, we may take $n'_{1max} = 42.5 \text{ min}^{-1}$, $n'_{1min} = 38 \text{ min}^{-1}$. Then the factor of the diameter line-up is

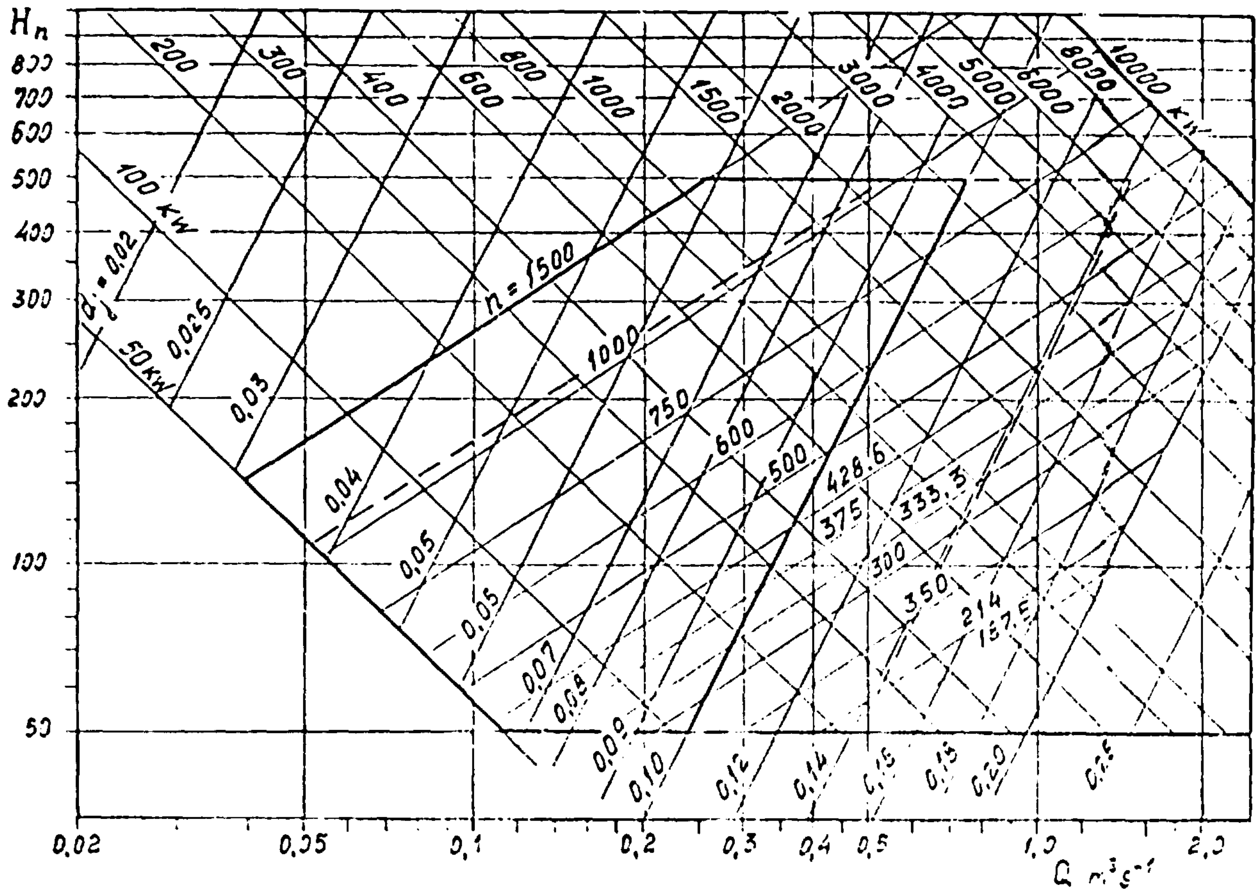
$$K_D = \frac{D_{n+1}}{D_n} = 1.118$$

Since D_{min} is taken equal to 0.3 m and D_{max} - 1.0 m, there must be N of different runner sizes where

$$N = \frac{\log D_{max}/D_{min}}{\log K_D} + 1 = 11.8$$

Assuming N = 11, then $K_D = 1.128$ at $n'_{1max} = 42.6 \text{ min}^{-1}$ and $n'_{1min} = 38 \text{ min}^{-1}$.

F. ure 32. Lines of constant speed for synchronous generator values



Thus we obtain a normal line-up of runner sizes determined from the formula

$$D_{n+1} = 1.128D_n$$

after rounding off

D_1 = 0.30, 0.34, 0.38, 0.43, 0.485, 0.545, 0.620, 0.695, 0.785, 0.885 and 1.00 m.

We now pass to the determination of the line up of nozzle sizes.

As has been assumed above, the minimum jet size d_{jmin} is equal to 0.03 m. Considering the optimum ratio d_j/D given above, we see that the maximum runner diameter to be effectively used with the given nozzle is

$$D \quad 14\ d_{jmin}$$

For the specific case we obtain D_1 = 0.43 m.

Table 4 gives the values of minimum head under which the turbine with the diameter D and speed n should be used.

The computations are made by the formula

$$H_{min} = \left(\frac{D}{n'_{1max}}\ D\right)^2$$

In particular, at n = 1,500 and D = 0.3-0.43, the area of heads 112.0-292.0 m is overlapped (see figure 33).

The maximum jet diameter for the study area is determined to secure the optimum ratio:

$$D_{min}/d_{jmax} \sim 7, \quad D_{max}/d_{jmax} \sim 10$$

assuming at the same time that d_{jmax} = 0.043 m.

Thus for D = 0.3-0.43 and n = 1,500 the nozzle should be applied which provides a maximum jet diameter of 0.043. Closing of the nozzle down to d_j = 0.03 m will result in discharge drop

$$\frac{Q_{max}}{Q_{min}} = \left(\frac{d_{jmax}}{d_{jmin}}\right)^2 = 2.05$$

which will not cause a tangible decrease in efficiency.

As findings of the Pelton turbine tests show, the sizes of the wheel buckets have a certain correlation with the jet diameter. Therefore all runners operating in the study zone must have definite and identical bucket dimensions.

A similar method is used to determine the size of other nozzles, as well as the area of application for the runners of different sizes operating at different shaft speeds.

Table 4. Minimum turbine head values

Number	Dm	1 500	1 000	750	600	500	428.6	375	333	300
1	0.3	111.6	49.6							
2	0.34	143.6	63.7							
3	0.38	179.0	79.6	44.8						
4	0.43	229.2	101.9	57.3						
5	0.485	291.6	129.6	72.9	46.7					
6	0.545	368.3	163.7	92.0	58.9	40.9				
7	0.62	476.6	211.8	119.1	76.2	52.9				
8	0.695	598.9	266.2	149.7	95.8	66.5	48.9			
9	0.785		339.6	191.0	122.2	84.9	62.4	47.8		
10	0.885		431.6	242.8	155.4	107.9	79.3	60.7	47.9	
11	1.00		551.0	309.9	198.4	137.8	101.2	77.4	61.2	49.6

An example is shown in figure 33 which makes it possible to determine the nozzle, runner sizes and speed for any values of H_n, Q or P_t.

In our study case five standard sizes of nozzles and wheel buckets for 10 different runner diameters are required for the area of application of a Pelton single-nozzle turbine at H_n = 50-500 m and P_t = 50-3,000 kW.

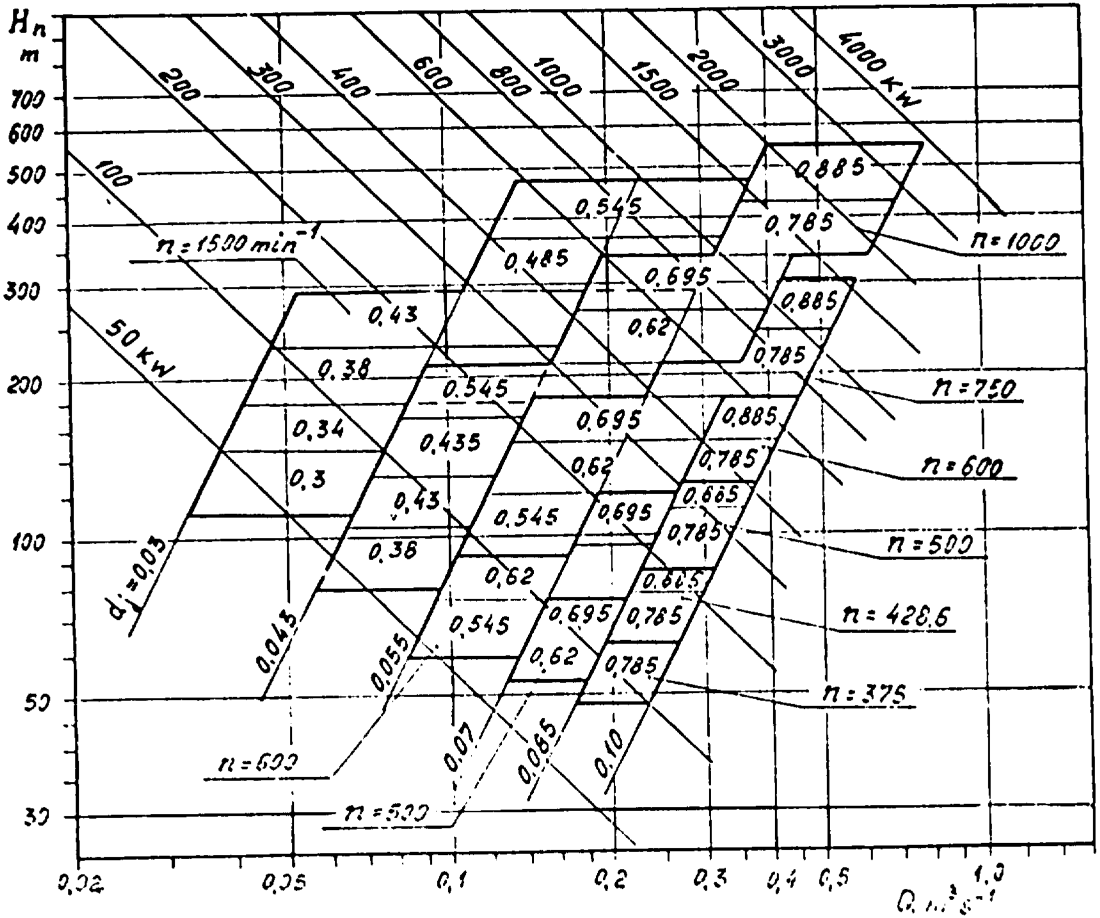
Table 5 gives the required combinations. It is seen that 17 standard sizes of turbines are required to cover the entire area of application given above.

A similar procedure may be applied to solve the problem for a double nozzle turbine.

Table 5. Required combinations of values for a Pelton single-nozzle turbine

D \ d_{jmax}	0.043	0.055	0.07	0.085	0.10
0.3	+				
0.034	+				
0.38	+	+			
0.43	+	+			
0.485		+			
0.545		+	+		
0.62			+	+	
0.695			+	+	
0.785				+	+
0.885				+	+

Figure 33. Area of application for runners of different sizes and different shaft speeds



V. EXAMPLES OF STANDARDIZATION

A. Normalization of small hydraulic turbines in the Union of Soviet Socialist Republics

Normalization and standardization of small hydraulic turbines was undertaken for the first time immediately after the Second World War in the Union of Soviet Socialist Republics (USSR), which was striving to meet the demands of the national economy and power industry and to carry out a programme of construction of small hydropower plants on a large scale.

The mass production of small hydroturbines was preceded by work on the development of standards and unification of turbine construction.

In the first instance, a list of short-term and long-term demands for small hydraulic turbines was drawn up. This list served to work out the line up of turbine types and sizes. In the second phase, the selected systems for most widely spread standard sizes were developed in detail. The design development covered construction not of separate turbines but of the whole turbine series. It then became possible to unify separate parts and components for a number of turbine standard sizes. As a result, completely normalized constructions have been developed.

Standardization helps to reduce the size, weight and cost of the equipment. The turbine construction is more adaptable to the requirements of mass production. Specialized production of turbine equipment can be set up at several factories.

1. Nomenclature of reaction turbines

The range of heads for reaction turbines is limited by H_n = 1.5-250 m. The area of application for reaction turbines in terms of specific speed lies within n_s = 60-1,000.

Each turbine series is characterized by the specific speed n_s related to the optimum operating conditions. When the operating conditions depart from the optimum ones, the efficiency drops. If an allowable value of efficiency loss is specified, the turbine of a certain series can be run within a certain range of specific speeds.

When the allowable efficiency loss is 1-1.5 per cent it is possible to develop a set of 10 turbine series. Of this number, seven will be of the Francis type. The specific speeds of two neighbouring series are related by the approximate ratio:

$$n_{s,(n+1)} = 1.4n_{s,n}-18$$

If a 3-4 per cent efficiency loss is allowed, a line up of 5 series will suffice, and for a 6-7 per cent efficiency drop a line up of 3 series is sufficient.

But selection of the number of series, as indicated above, does not depend on the efficiency drop alone. Cavitation free operation is the most critical consideration.

The following ranges of allowable suction heads have been adopted for the Francis and Kaplan turbines in developing the nomenclature for the small hydraulic turbines: H_s = 0-3 m (for the Francis turbine) H_s = 1-2 m (for the Kaplan turbine).

In compliance with these restrictions it has been found that a bigger number of series are required to meet the cavitation-free conditions. For example, the Francis turbine requires nine series within the range of n_s = 80-300 at the specific speed ratio of two neighbouring series

$$n_{s,(n+1)} = 1.2n_{s,n}$$

and the Kaplan turbine requires five series.

Increase in the number of series will reduce the magnitude of efficiency loss but the number of standard sizes tends to increase.

The adopted nomenclature contains 12 series of reaction turbines. Of this number eight turbine series are of the Francis type and four series are of the Kaplan type.

At the initial phase of practical application of small hydraulic turbines it was found feasible to reduce the number of series to six (four series are Francis and two are Kaplan). But it entails an appreciable efficiency loss of 3-4 per cent.

As indicated above, one of the major objectives of small hydroturbine standarization is to fix the set of nominal turbine diameters making up a single series. To achieve better conformity of the turbine to the specified conditions and higher efficiency, the set of nominal diameters should have a close interval, while the value of the set denominator K_D should be lower.

The optimum conditions are met at K_D = 1.05-1.075. To cut down the number of standard sizes, the denominator K_D can be raised to 1.11 but it entails about a 2 per cent loss of average operating efficiency.

At the initial stage of small turbine normalization, a higher value of K_D = 1.19 was adopted. When changing over from one size to the next larger one, the power increases as much as $1.19^2 \doteq 1.416$. The following line up of diameters is adopted for Francis turbines D_1 = 0.30, 0.35, 0.42, 0.50, 0.59, 0.71, 0.84 and 1.00 m.

For propeller and Kaplan turbines:

D_1 = 0.35, 0.46, 0.59, 0.80, 1.00, 1.20, 1.40 and 1.60 m.

In the practical application of axial flow turbines preference would be given to the propeller turbines because of their simple construction and lower cost. The blade pitch used to be set on the basis of the conditions of application which ensured a high operating efficiency.

It may be noted that the construction of standarized turbines can provide for a minor change (about ± 3 per cent) in the runner diameter without disturbing the construction of major components and overall dimensions. Such a measure allows the loss of an average operating efficiency to be significantly restricted for a relatively high ratio K_D.

2. Reaction turbine configurations

Though great experience has been accumulated in designing, engineering and manufacture of large-size hydraulic turbines, its application to the area of small hydraulic turbines is not possible without considerable adaptation. To be more exact, the designs which are peculiar to the small hydroturbines have such a specific character that they should be singled out in a separate group of hydraulic turbines.

Since the cost-effectiveness of small hydroplants is appreciably lower than that of large-scale hydropower developments, efforts should be taken to cut the cost of hydroelectric plant, including reduction of the equipment cost. There are two possible ways to achieve this.

First, the cost of turbine equipment can be cut through the development of cost-effective designs, unification and application of cost-effective manufacturing processes. Here feasible simplifications in design and manufacturing processes will result in lower cost of equipment without detriment to the energy and cavitation characteristics of the turbines. Secondly, types and configurations unsuitable for large-size installations could be used.

Departure from conventional designs and simplification in configuration and manufacturing processes would lead to turbines somewhat poorer in quality but much cheaper.

From the viewpoint of the organization of cheap turbine manufacture, it is practicable to limit the number of turbine designs. The first nomenclature variants included the following turbine configurations:

(a) Vertical-shaft open-flume turbines with elbow type, or straight draft tube directly connected to generator shaft or through step-up gearing;

(b) Horizontal-shaft turbines in steel case with frontal water supply and shaft passing through draft tube elbow for P_t up to 1,000 kW;

(c) Horizontal shaft spiral turbines for P_t up to 2,500 kW.

3. Nomenclature of impulse turbines

The nomenclature of Pelton turbines will first be considered. The jet diameter is taken as the parameter controlling the turbine power. Normalization of the jet line up is one of the major problems.

If the magnitude of efficiency loss allowable for the Pelton turbine is taken to be equal to about 7 per cent, the maximum power output of the turbine can be established within 0.25 0.1 of its full power to be obtained at the maximum nozzle opening. From this condition it follows that

$$d_j(n+1) = 2d_jn$$

Such a high value of the jet diameter line-up ratio stems from a flat curve of the Pelton turbine. At this ratio only three jet diameters d_j = 0.025, 0.05 and 0.10 m can be obtained in the area of practical interest.

It is desirable from the point of view of manufacturing, but not economically viable because it causes an extra consumption of metal. In the limiting case when installing the turbine, with a capacity exceeding fourfold the required amount, its linear dimensions will be twice and the mass eight times more than those of the turbine designed for the given parameters.

Taking into account feasibility analysis, the following line up of water jet diameters is established:

d_j = 0.025, 0.036, 0.050, 0.065, 0.082 and 0.10 m

The range factor varies from 1.44 to 1.22. When selecting the line-up of runner diameters the minimum value D_1/d_j = 10 is taken. For the smallest turbines with d_j = 0.025 m the greater ratio values are taken, reaching the value of D_1/d_j = 20.

As a result, the following line-up of runner diameters is obtained:

D_1 = 0.036, 0.050, 0.65, 0.82, 1.00 and 1.20 m

for H_n = 40-250 m and P_t = 10-500 kW resulting in Q = 0.012-0.4 m^3/sec^{-1} and n = 250-750 min^{-1}.

The normalization of the Turgo turbines is based on the same principles as that of the Pelton turbines.

For normalized Turgo turbines d_j = 0.025-0.2 m and D_1 = 0.21-0.64 m. These turbines may be applied in the range H_n = 50-400 m and P_t = 10-4,000 kW.

The area of application of the normalized Banki-Mitchell turbines lies within H_n = 10-160 m and P_t = 5-300 kW.

B. Small turbines manufactured by Voith

The Voith company (Austria and Federal Republic of Germany) developed the standard line-up of hydraulic turbines for small hydropower application, including the following types of units: Pelton turbines for high heads; Francis spiral turbines; Francis open flume turbines; and axial flow turbines of tube, pit and bulb type

Standard hydraulic turbines cover the following range of operational parameters:

H_n 2 400 m Q = 0.05-80 m^3 sec^{-1}
P_t 50 10,000 kW

Francis turbines with the capacity P_t > 2,000 kW and axial flow turbines with the capacity P_t > 5,000 kW are custom-designed.

1. Pelton turbines

Pelton turbines are applied in the high-head range with relatively low discharges:

$$H_n = 40\text{-}400 \text{ m} \qquad Q = 0.05\text{-}1.3 \text{ m}^3 \text{ sec}^{-1}$$

The advantages of these turbines are: better efficiency in comparison with Francis turbines in the said area of application; high efficiency in the wide range of loads; simplicity of design; and relatively low cost of installation.

The standard design provides for a turbine with two nozzles. The area of application of each standard size is determined by the head discharge ratio.

Single-nozzle turbines are applied at

$$Q = 0.05\text{-}0.16 \text{ m}^3 \text{ sec}^{-1} \qquad H_n = 100\text{-}400 \text{ m}$$
$$P_t = 50\text{-}500 \text{ kW}$$

Double nozzle turbines are applied at

$$Q = 0.06\text{-}1.3 \text{ m}^3 \text{ sec}^{-1} \qquad H_n = 40\text{-}400 \text{ m}$$
$$P_t = 50\text{-}4{,}000 \text{ kW}$$

A competitive alternative of Francis spiral turbine may be used for the range of

$$H_n = 40\text{-}150 \text{ m} \qquad Q = 0.1\text{-}0.8 \text{ m}^3 \text{ sec}^{-1}$$

The following standard line-up of runner diameters is taken:

D_1 = 0.305, 0.340, 0.385, 0.430, 0.485, 0.545, 0.610, 0.685, 0.770, 0.860 and 0.970 m (K_D = 1.123)

when speed n = 1,500, 1,000, 750, 600, 500 and 428 min^{-1}

It should be noted that the runner is made in the following two alternative designs:

(a) Wheel buckets are cast integrally with disk and hub;

(b) Buckets are bolted to the disk.

The runner, cast as a whole piece, is cheaper than the assembled type. But at the same time the second alternative allows replacement of buckets. The cast alternative is applicable only for small diameters of runner and in case of high specific speed.

The configuration of wheel buckets is established by means of extensive laboratory tests.

The single nozzle turbine case consists of a welded frame and a cover, flanged along the horizontal parting joint. The double nozzle turbine cover is of two parts as well. The seals are mounted on the shaft, coming out from the case.

The turbine shaft bearings are oil-lubricated. One of them takes up the axial thrust. The bearings intended for high loads have a water cooling system.

The turbine is mounted over the outlet pit into which the waste water is discharged. The outlet pit is lined with strong and hard steel plates.

Special needle nozzles are used to supply water to the runner and convert the pressure energy to kinetic energy without losses. Particular attention is paid to proper water supply and its distribution in the double-nozzle turbines.

Pelton turbines are equipped with a speed governor providing the following methods of regulation: nozzle control; deflector control; and dual control by nozzle and deflector. The water discharge is controlled by needle travel. The hydraulic force acting on the needle always tends to close the nozzle. This force is equalized hydraulically or mechanically. Hence a relatively small force is needed for the needle travel. At the nozzle closure the pressure increases in the penstock. The pressure build-up amounts up to 50 per cent at the small hydroelectric plant with small-diameter penstocks.

To reduce the temporary speed drop of the turbine shaft the flywheel may be applied.

2. Francis spiral turbines

Francis spiral turbines with scroll case water supply are used at

H_n = 10-150 m   Q = 0.12-12 m^3 sec^{-1} P_t = 50-2,000 kW

They are also used at lower heads to reduce costs under certain special conditions. They may be advantageous also for turbines of particularly small dimensions.

All turbines have horizontal shafts, since generators with horizontal shafts are cheaper. Vertical turbines are recommended only as an exception.

To meet the said Q-H_n requirements the company applies the runners of 10 series, depending on the pressure head and suction head.

For high-head turbines in the range of H_n = 100-150 m the allowable suction head varies within H_s = -3, +4 m. For lowhead turbines in the range of H_n = 15-20 m the allowable suction head varies within H_s = 3, +7 m.

The following standard line-up of runner diameters has been established:

D_1 = 0.205, 0.235, 0.265, 0.300, 0.345, 0.390, 0.440, 0.470, 0.500, 0.535, 0.570, 0.610, 0.650, 0.690, 0.740, 0.790, 0.840, 0.895, 0.955, 1.015, 1.085, 1.155, 1 230, 1.310, 1.400 m

For small diameters K_D = 1.136 and for larger diameters K_D = 1.066. The turbine shaft speed is n = 1,500, 1,000, 750, 600, 500, 428, 300, 250 and 200 min^{-1}.

The last three values are used for low-head installations with H_n = 4-10 m.

Thus the company standards provide for a wide range of various dimensions of spiral turbines.

A cost-effective turbine with good cavitation-free behaviour may be installed to meet practically any operating conditions.

The following distinguishing features of the spiral turbine design may be noted.

The turbine has a hydraulically optimum spiral case allowing high water velocity and having the dimensions best suited to ensure minimum losses. Being small in size, the spiral case is made of steel casting. For increased dimensions the spiral case is welded from plate steel. The steel-welded spiral cases are provided with a stay ring to increase the structural strength.

The turbines have externally controlled wicket gates. The shifting ring is mounted either on the turbine cover side or on the draft tube side. The last arrangement is seldom applied. Wicket-gate stems are carried in bearings fixed to the turbine cover. The seal is mounted at the end. Specific attention is paid to sealing and its reliable service in operation. Special holes are drilled for removal of leaks so as to eliminate external leaks completely.

The vane of wicket-gate levers and links are made of special high-strength steel castings. The gate operating mechanism may be lubricated during operation of the turbine.

The design of the turbine runners has been worked out in the hydraulic laboratory. The company guarantees the operation of the runners with the highest possible efficiency. The runners are characterized by the head, discharge and speed.

The runners are one-piece castings of steel, bronze, aluminium bronze or chromium steel for turbines of any high specific speed. Runners of medium and high specific speeds are made with hub and rim of steel casting with stamped steel blades lined at the edges. Selection of the material for the runners and spiral case depends on the properties of water. The runner is overhung on the shaft. Access to the runner is quite simple should one move aside the draft tube elbow.

When the water flows through the turbine it exerts hydraulic axial thrust on the runner. Due to this the turbine bearing should take up the thrust as well. The design of bearings is governed by the thrust magnitude and turbine shaft speed.

Voith applies the thrust ring design for ordinary operating conditions. Self-lubricating oil bearings with fixed position of thrust surface and without water cooling of oil are used at negligible hydraulic thrust and speed. The cooling system is arranged at high magnitudes of axial thrust and speed. Pressure lubrication is also provided for heavily loaded bearings.

Protective rings of stainless durable steel in the zone of the wicket gate assembly are provided in case the water carries abrasive bed loads.

The draft tube is of an elbow shape. At the inlet section it has an elbow, which then changes to a straight conical draft tube. For large-size turbines with a small suction head the vertical draft tube becomes very long, which requires considerable earth-moving activities. In this case the length of the tube is extended in the horizontal direction. The tube cross sections change from round at the inlet to rectangular at the outlet.

To suit the type of generator bearings, rigid or elastic couplings for the shafts of generators and turbines are used. If the generator has slide bearings, rigid shafts couplings are used, in which case only one bearing in the turbine is required. For generators with rolling bearings the elastic coupling is preferable, which may make up for slight misalignment of the assembled shafts. In this case the installation of an additional overhung turbine bearing is required, which may simultaneously function as the thrust bearing. When arranging the belt drive, the turbine shaft may have two bearings, the pulley being installed between them. The belt drive is used to drive the speed governor.

The hydraulic turbine is controlled by the company-designed speed governor with a centrifugal pendulum and hydraulic servomotor. Flywheels are used to improve regulation stability. A similar function is intended for the generator rotor. When mounted, the flywheel may simultaneously function as a half of coupling at the end of the generator shaft. When using the rigid coupling the flywheel is often placed between the halves of coupling or fixed on the generator shaft.

The company developed a special design of turbine for installations featuring a wide fluctuation of discharges throughout the year. In such cases a turbine with twin spiral case and two runners is used. A large-size turbine is arranged integrally with a small-size turbine. The two turbines operate together when the water discharge is high. The large size turbine will operate alone at the average discharge, and the small size turbine will run alone at the low water discharge. In such an installation with a discharge approximating one sixth of the full water discharge, water energy may be used with satisfactory efficiency.

High speed operation can sometimes be specified for the turbine. For such installations the twin turbine is a good solution. This turbine is equipped with the twin runner, and each part of the twin runners is designed to pass fifty per cent of water discharge. The turbine has two draft tubes discharging the water from each half of the runner.

3. Francis open-flume turbines

Voith has developed a standard line-up of open flume turbines (figure 6) and recommends it for application when

H_n = 2-10 m Q = 2.5-30 m^3 sec^{-1} P_t = 80-2,000 kW

These turbines cover the range of axial flow turbines. The advantages of their use are as follows:

(a) Vertical open-flume turbines have small dimensions;

(b) The generator may be mounted above the maximum water level, which is often difficult in turbines with a horizontal shaft arrangement;

(c) Turbines of this type have relatively low cost and require small capital investments and cheap equipment;

(d) Application of vertical step-up gear allows the use of high-speed generators.

The area Q-H_n is covered by six series of turbines using 19 normalized dimensions of runners.

The following standard line-up of diameters has been established:

D_1 = 0.84, 0.895, 0.955, 1.015, 1.086, 1.155, 1.23, 1.31, 1.40, 1.49, 1.59, 1.695, 1.805, 1.90, 1.99, 2.09, 2.19, 2.30 and 2.41 m

with D_1 up to 1.805, K_D = 1.0658, and at bigger sizes K_D = 1.0494.

The company developed the nomograph for preliminary selection of turbine type and its dimensions for the given design conditions.

Table 6 shows parameters according to the field of application for series of turbines.

Table 6. Parameters for turbine series

Turbine series	H_n (m)	Q (m^3 sec^{-1})	n (min^{-1})	P_t (kW)
F 160	2-10	1.6-16	40-200	80-1 000
F 190	2-10	1.6-16	45-225	80-1 000
F 225	2-10	2-20	40-200	80-1 000
F 260	2-10	2.4-30	45-225	80-1 000
F 295	2-10	2.4-30	50-250	80-1 000
F 330	2-10	2.4-30	55-275	80-2 000

The adopted series of turbines are of unequal efficiency, the F 160 series being the best. With the growth of turbine specific speeds the efficiency tends to decrease both under optimum operating conditions and especially at partial loadings. The operation of turbines is recommended loads not less than 50 per cent of the maximum value.

Turbine efficiency at full gate is 82-85 per cent. The efficiency also depends on the turbine size and series. For example, under optimum operating conditions the efficiency of the F 330 series is 2 per cent less, and at 50 per cent load it is 6.5 per cent less, than that of the F 160 series.

Turbine efficiency may be lower at modernized old hydroelectric plants, where separate sections of the water passage may remain non-optimum.

The efficiency of single-stage step-up gear is evaluated by the company at about 96.5-98.5 per cent at full load.

The developed turbine series have satisfactory cavitation characteristics. The turbines of the F 160, F 190 and F 225 series at the recommended range of heads may have the suction head up to 7 m, and turbines of the F 260, F 296 and F 330 series up to 6 m.

The company has developed standard designs of turbine installation and the main dimensions are given according to turbine type and size of runners.

The range of supply includes turbines, step-up gears, generators and speed governors. The turbine includes the runner, adjustable wicket gate assembly, draft tube elbow, shaft with bearings as well as protective pipe and coupling.

The runner is usually cast of steel with cast-into steel blades. The material with improved erosion resistanc is used for the corrosive medium with suspensions. The runners have been developed experimentally and the company guarantees the maximum efficiency and good cavitation-free characteristics.

The wicket gates are provided with outside control. The gate operating mechanism is located inside the turbine chamber. The operating mechanism assembly consists of thrust ring, operating ring with levers and links and vertical regulating shaft.

In some cases the company makes the draft tube only in the form of a steel welded elbow, from which the water is discharged directly into the tailrace (figure 6).

Application of Francis turbines for such low heads results in low speeds of the turbine shaft. Use of a high speed generator (500-1,000 min^{-1}) requires step-up gear. For that purpose the company uses planetary coaxial step-up single stage gears with similar direction of shaft rotation. The transmission ratio is within the following ranges:

P_t up to 200 kW	i : 5-31.5
P_t up to 1,000 kW	i : 7-28
P_t > 1,000 kW	i : 2.7-7.3

The transmission gear case is oiland dust proof. Good surface finish of runners secures high efficiency. Depending on the power and thrust on the runner, the thrust bearing is installed either in the gear case or separately. When installed separately the thrust bearing is lubricated by the automatic system. Ball-bearings with spray lubrication system are used for small step-up gears. The big transmission gears have plain bearings and lubrication of bearings and contact places of transmission gears is effected under pressure by the special oil lubrication system.

Generators with a speed range of 500-1,000 min^{-1} are used in hydraulic units. Depending on the type of turbine and head, the runaway speed ranges from 200 per cent up to 250 per cent that of the nominal speed.

Elastic coupling is installed between generator and step-up gear for a synchronous unit, the flywheel being fixed on the generator shaft.

Besides planetary step-up gears it is allowed to use non-coaxial vertical step-up gears as well as transmission gears with the horizontal output shafts. In the latter case the horizontal shaft generator is installed.

4. Horizontal axial-flow turbines

Voith, like most other hydraulic turbine manufacturers, has developed standards for horizontal shaft axial hydraulic turbines with the upstream bulb arrangement and S-shaped draft tube (figure 12).

Wide propagation of this type of turbine stems from certain advantages in comparison with axial vertical-shaft turbines, including the following: easiest water passage from hydraulic point of view; and small dimensions and convenient arrangement of the power-house.

The turbines mostly have fixed wicket gates and adjustable blade runner. The company has developed nomographs for preliminary selection of main parameters of turbine equipment, depending on water head and discharge:

H_n = 2-15 m   Q = 3-70 m^3 sec^{-1} P_t - 100-7,000 kW
n = 80-300 min^{-1}

The three series of turbines reccmmended are distinguished by the Z number of blades:

where Z - 5 H_n = 5-15 m   Q = 7-70 m^3 sec^{-1} P_t - 200-7,000 kW,
where Z = 4 H_n = 3-10 m   Q = 3.5-60 m^3 sec^{-1} P_t = 100-5,000 kW,
where Z = 3 H_n = 2-6 m   Q = 3-50 m^3 sec^{-1} P_t = 50-2,800 kW

The suction head of H_s = -4 - +6 m in the operational area.

The standards indicate the following normal line up of runner diameters:

D_1 - 0.96, 1.090, 1.16, 1.23, 1.31, 1.40, 1.49, 1.59, 1.70, 1.81, 1.90, 1.99, 2.09, 2.19, 2.30, 2.41, 2.53, 2.65, 2.78 and 2.92 m

$K_D = 1.066$ at small D_1 and $K_D = 1.049$ at big D_1.

The hydraulic turbine includes the step-up gear, generator and governor. Water-conveying and outlet features of the turbine may be adapted to local conditions. The hydraulic turbine includes the runner, its chamber, the fixed wicket gate assembly, the bend draft tube, the shaft with a servomotor and the control rod inside, the guide bearing and the shaft seal.

The runner has 3-5 blades made of steel or bronze casting, which rotate in a bronze bearing housed in the runner hub. The runner blade mechanism, consisting of cross head, links, levers and journals, is installed in the runner hub. The wicket gates are welded to the outer ring.

The runner chamber is a welded structure. The internal surface is machined along the sphere to obtain equal clearance among the runner blades and chamber in any position of the blades. To facilitate installing and dismantling the runner, its chamber is made of two halves and has a removable flange on the draft tube side. The chamber shell may be made of stainless steel for corrosive water with suspensions. The wicket gate assembly is of a welded construction. It consists of an outer cone with flanges and gates welded to it and a hub with removable casing.

The draft tube is very important especially in the high specific speed turbines in question. The shape of the water passage also plays a very important role.

To improve hydraulic properties the draft tube is welded of many separate segments. To increase its rigidity the ring stiffeners are welded to it externally. The inspection hatch is provided in the upper half. The lower part of the tube is concreted after the final alignment of the unit.

The turbine shaft is a steel forging with a bore to accommodate the blade control rod. At the end of the shaft the cylinder of the oil servomotor is forged. The shaft is flanged to the runner and step-up gear shaft.

The guide bearing with oil lubrication is inside the wicket gate assembly hub. To prevent penetration of water a labyrinth seal and a special sealing with the pipe to divert the leaks are used.

The second bearing of the shaft and thrust bearing are installed in the gear transmission unit. At the end of the shaft, where it passes through the wall of the draft tube elbow, the seal is installed. The seal may be made removable without disturbing separate parts of the machine.

The single stage step-up gear with plain bearings is installed between turbine and generator shaft. The speed of the generator is 500 min^{-1} or 750 min^{-1}. The transmission ratio may be easily selected by the ratio of speeds of the turbine and generator shafts and by the number of teeth in gears.

The minimum turbine speed is 95-140 min^{-1} for the maximum size of the runner. For turbines with minimum diameter the speed is 420-750 min^{-1}.

The low speed shaft of the gear is mounted below to house oil pipes supplying the oil under pressure to the runner blade servomotor. The oil head with the indicator of runner blade pitch is fixed to the step-up gear. The oil pump to lubricate the gear and the oil pump of the governor are installed at the free end of the high-speed shaft.

The low speed shaft can be provided with a segment thrust bearing to take up the thrust. The hydraulic unit is provided with a standard cost-effective generator with the speed of 500 min^{-1} or 750 min^{-1}. The maximum runaway speed is higher than the normal one by 250-290 per cent irrespective of the head.

An elastic coupling is installed between the generator and step-up gear. In synchronous units the required flywheel is installed on the generator shaft in combination with the elastic coupling.

5. Vertical-shaft Kaplan turbines

Voith also offers axial vertical turbines of a conventional design for low-head installation.

The nomographs for preliminary selection of the main parameters of hydraulic turbines, depending on installation conditions, have been worked out. The following is the recommended range of application:

H_n = 1.5-8 m Q = 2.5-45 $m^3\ sec^{-1}$ P_t = 50-25,000 kW
n = 80 500 min^{-1}

The following two types of turbines, distinguished by the number of blades, are recommended:

(a) $Z = 3$ H_n = 1.5-5 m Q = 2.5-40 $m^3\ sec^{-1}$ P_t = 50-1,500 kW

(b) $Z = 4$ H_n = 3-8 m Q = 3.0-45 $m^3\ sec^{-1}$ P_t = 100-2,500 kW

The suction head in the operational area is within the range of i = 5-7 m for turbines with 3 blades and within the range of 0-6 m for turbines with 4 blades.

The same line up of diameters as for the horizontal turbines in the range of 1.02-2.92 m is offered for vertical-shaft turbines. A spiral supply passage with a trapezoidal asymmetric form of cross sections is used in the turbine. The configuration and ratio of section sizes are close to those used in large units. The wrapping angle of the spiral case is about 210°. The draft tube is characterized by the height of $h = 2.1D_1$ and the length of $l = 4.85D_1$.

The rotation axis of the blades is below the plane of the wicket gate assembly by the value of $0.6D_1$.

The draft tube, symmetrical in plan, has a slight rise of the invert of the straight diffuser (0-8)°. The pier is installed along the axis of the diffuser. The width of outlet section of the draft tube is $3.2D_1$, the height is $-1.14D_1$.

C. Small turbines manufactured by Voest-Alpine

Voest Alpine (Austria) has developed standards for small hydraulic turbines, including the following well-known types: axial low-head hydraulic turbines; Francis turbines for medium heads; and Pelton turbines for high heads.

The application of standard turbines is limited by the following range of operating parameters:

H_n = 1-1,000 m Q = 0.01-75 m^3 sec^{-1} P_t up to 15,000 kW

Custom made designs incorporating specific structural and cost elements are made for axial-flow turbines at more than 5,000 kW capacity and for Francis turbines operating under the head H_n > 120 m.

1. Axial-flow hydraulic turbines

The existing standards provide for application of axial-flow hydraulic turbines with horizontal, inclined and vertical shafts at low water heads. To cover the range of the head and discharge the standards involve six series of turbines distinguished by the specific speed at 15 different diameters for every type. The standard provides for the average of the whole of the range of application at:

H_n = 1-30 m Q = 1-60 m^3 sec^{-1}

The standard diameters of runners are characterized by the following values:

D_1 = 0.56, 0.63, 0.71, 0.80, 0.90, 1.00, 1.12, 1.25, 1.40, 1.60, 1.80, 2.00, 2.24, 2.50, 3.00 m

where K_D = 1.127

The first series of axial flow turbines is developed for the range:

H_n = 1-5 m Q = 1-50 m^3 sec^{-1} P_t = 12-2,000 kW

The second series:
H_n = 4-12 m Q = 1.6-75 m^3 sec^{-1} P_t = 50-8,000 kW

The third series:
H_n = 8-16 m Q = 1.9-70 m^3 sec^{-1} P_t = 200-10,000 kW

The fourth series:
H_n = 12-22 m Q = 1.9-60 m^3 sec^{-1} P_t = 200-12,000 kW

The fifth series:
H_n = 16-26 m Q = 1.8-60 m^3 sec^{-1} P_t = 250-12,000 kW

The sixth series:
H_n = 20-30 m   Q = 1.8-55 m^3 sec^{-1} P_t = 250-13,000 kW

Each series of turbines is provided with the nomographs, which allow to select the diameter of the runner and the speed by the given values Q, H_n and P_t in design conditions.

The company considers that the most efficient turbine for low heads is a straight-flow turbine of various configurations. The main distiguishing feature of these turbines is the approximately straight flow of the water through the turbine, including the draft tube. This offers the possibility of simplifying the design of the powerhouse, reducing both the distance between the units and the cost. The straight flow design of the unit gives also an increase of the reduced discharge in optimum conditions and an increase in efficiency at low discharges.

Because of the high level of efficiency the possibility of economic operation of the turbine is improved with variation of the head and the discharge, resulting in an increase of power output. The company states that research and standardization of the straight-flow bulb turbines make the application of small water resources with low heads economically efficient.

The geometry of the water passage and the runner determines turbine characteristics. The optimization of the water passage and characteristics has been made on the basis of model studies.

Depending on the specific speed of the turbine the runner is provided with 3-7 blades. The reduction of their number leads to lower discharges and speeds.

The optimum matching of turbine features and local conditions is reached by combinations of gate and blade positions.

Under steady-state operating conditions the turbine with fixed blades and gates is recommended. The Tomman turbine with adjustable blades and fixed gates at partial loads has the most favourable flat curve characteristics with higher efficiency than that of propeller turbines with fixed blades and adjustable gates (see figure 24). For example, the reduction of the efficiency by four per cent in the first case will take place at $Q_{1'}/Q_{opt}$ = 0.5 and in the second case at $\sim$ 0.9 when operating at partial loads. The turbine with fixed blades of the runner has a tendency to vibration and noise. The final decision on the mode of control is taken on the basis of operating and economic conditions.

The following distinguishing features have been realized in turbine designs:

(a) Stainless steel is used for the main components;

(b) There is minimum machining of welded elements;

(c) There is free access and interchangeability of bearing parts.

The design of main components (including the gate and blade operating mechanism) is unified to improve the economic efficiency of manufacture. Such components may be used for various applied series of turbines.

Water conveying features of the water passage, including the inner and outer cones of the wicket gate assembly, are welded. Usually the gates are either cast of stainless steel or welded. Depending on he head, the gates are provided with one or two stems, but in a: · case the stems do not require special servicing.

The hub of the runner is usually made of steel castings. The blades are made of either chromium nickel steel or bronze. The operating ring is welded of sheet stainless steel. The draft tube is also welded.

The turbine shaft is forged with stainless steel surfacing in the area of the sealing and the bearing. The turbine bearing is rubber with water lubrication to avoid penetration of the oil into the water. The main guide bearing at the end of the shaft is provided with oil lul·ication.

For varying local conditions the developed standard allows use of the following alternative arrangements of axial-flow turbines: horizontal; inclined; vertical; with an open headrace; and with water supply syphon. The speed of low-head turbines is usually low, and therefore the standard turbines are provided with the step-up gear to increase the speed of the generator and to reduce its cost.

Free access to and low cost of structural elements are achieved through installation of the turbine and generator above the water level in the tailrace.

Three design alternatives of horizontal bulb turbines are manufactured. These are:

(a) S'·aight-flow turbines with a straight draft tube and a high-speed a nchronous generator, which is connected to the turbine shaft through the angle step-up gear;

(b) Straight-flow turbines with S-shaped draft tubes and the generator, brought out together with the step-up gear to a separate room;

(c) Straight-flow turbines with a double water supply line.

The diameter of the ru·ner does not exceed 3 m. The economic efficiency of the first type is defined by the range:

H_n = 1.5-6 m P_t = 50-1,000 kW

The second type of turbines is used at H_n up to 10 m and P_t up to 5,000 kW.

For heads H_n in the range 10-30 m, in most cases vertical or inclined units are used, in which cases the water is conveyed by means of a bend at an angle of 90-120°. The water outlet is realized by the bent draft tube. The turbine shaft in this case is brought out to the conveying bend.

Vertical turbines are also used in combination with the conveying line in the form of an open flume or a concrete spiral case of T-type section. Alternative design generators are available. In the latter case the step up gear with horizontal output shaft is used.

The distinguishing feature of all the alternative designs is the stability of the water passage in the area of the wicket gate assembly and the runner. In all cases the wicket gate assembly is conical.

The straight-flow turbine with a double supply line is preferable for low and medium capacities, for which free access for servicing is provided. The water flow in these turbines is divided into two portions between the inlet section of the supply line and inlet section of the wicket gate assembly. The two water-conveying pipes may be placed in either a horizontal or a vertical plane. The turbine shaft is brought out to the free space between the conveying pipes. The step-up gear transfers energy to the shaft of the vertical or horizontal generator. The units of this type are manufactured with either horizontal or vertical turbine shafts.

The most economical unit for turbines with diameters of more than 3 m and with a head of up to 20 m is the bulb-type unit.

2. Francis turbines

The main type of turbines for medium heads is the Francis turbine, which finds a wide range of applications for various conditions. The line-up of turbine standard sizes covers the following range of characteristics:

H_n = 15-120 m P_t = 250-15,000 kW Q = 2-30 m^3 sec^{-1}

For this purpose eight series of water passages of hydraulic units with various specific speeds have been developed and eleven standard diameters of runners are used.

For installations at H_n more than 120 m and up to 300 m the hydraulic units may be manufactured by the custom made design.

The standard involves the following diameters:

D_1 = 0.56, 0.63, 0.71, 0.80, 0.90, 1.00, 1.12, 1.14, 1.60, 1.80 m

Table 7 shows the ranges of application of each of eight series of turbines with n_s = 120-450.

A great quantity of various basic models and dimensions of turbines allows to ensure operation over the whole range of application with high efficiency.

For each type of turbines the range of application has been determined and the nomograph constructed for selection of sizes of the runner and the speed. Usually the trend is to install the turbine above the water level in the tailrace. Here cavitation model characteristics must be taken into account.

Table 7. Ranges of application of turbine series

Turbine series	H_n (m)	Q (m^3 sec^{-1})	P_t (kW)
1	15-30	2-27	250-8 000
2	25-40	2.5-30	500-10 000
3	30-50	2.5-30	600-12 000
4	40-60	2.5-30	800-15 000
5	50-70	2.2-25	1 000-15 000
6	60-80	2.0-22	1 200-15 000
7	75-100	2.0-20	1 400-15 000
8	80-120	2.0-18	1 500-15 000

The company has developed some basic typical standardized designs of turbines. In this case the dimensions may vary with the requirements.

The spiral case is welded of sheet steel together with the support stay ring. In its turn the stay ring is welded of two flat steel rings with vanes between them. The stay ring is available in all types and designs of turbines. The turbine head-cover is made also of sheet steel and is either integral with the stay ring or bolted to it. The lower cover is welded and made separately. In both covers the bushings of wicket gate bearings are arranged.

The wicket gate assembly is usually made cast or forged from stainless materials. The bearings of gates and couplings of levers and links require no servicing.

The runner is made cast or welded of chromium nickel steel. The turbine shaft is forged.

The shaft sealing is similar to those of sealings of axial-flow turbines. The draft tube is welded of separate segments.

The Reifenstein turbines with horizontal or vertical shafts are recommended for very small capacities P_t = 10-200 kW and heads H_n = 5-40 m. For heads H_n = 10-40 m and capacity P_t = 120-5,000 kW at the positive suction head open-flume turbines are used.

Spiral packaged turbines with compact arrangement are used at H_n up to 80 m and P_t up to 500 kW. Here the generator of the vertical turbine is installed on the spiral case, with the shaft alignment reduced to the maximum. The generator bearings are reinforced, with the flywheel placed between the generator and turbine head cover.

Spiral turbines of conventional arrangement are made with vertical and horizontal shafts. The advantage of horizontal installations is the low cost of construction elements, simple maintenance and assembly of turbines and generators and the possibility to use the standard generators in combination with the step-up gear.

3. Pelton turbines

Voest-Alpine has developed Pelton turbines for further applications. The turbine is used for high heads and low water discharges. It is used for low heads only under specific conditions, for example, when considerable erosion is expected for water with sand suspensions, or when there is an increased danger of the hydraulic hammer effect.

Pelton turbines are simple, economical and efficient over a wide range of operation.

The standards recommended for the use of Pelton turbines are:

H_n = 80-1,000 m P_t = 250-15,000 kW Q = 0.3-5 m^3 sec^{-1} and

H_n = 20-80 m P_t = 10-150 kW Q = 0.02-0.2 m^3 sec^{-1}

The geometry of hydraulic elements of the runner has been developed experimentally. The high quality of the runner and all water-conveying structures is required to obtain high efficiency.

Pelton turbines are relatively simple in design. In modern turbines the runner is made cast of high-alloy chromium nickel steel and the buckets are cast together with rim and hub. The conveying elements, including the penstock, are welded of sheet steel. The water discharge is controlled by the nozzle (single control) and by the nozzle and the deflector (double control). The nozzle needle is driven by the servomotor.

The turbine has one or two nozzles and a horizontal shaft, but it may be made with vertical shaft as well.

Mass-produced generators are used for small capacities. There are two alternatives for large capacities: mass-produced generators with elastic coupling between the turbine and the generator (four-supports design) or the generator of specific design with overhang arrangement of the runner and the flywheel (two-supports design).

D. Small turbines manufactured by Kessler

Kessler specializes in manufacturing turbines of 10-5,000 kW capacity for hydroelectric stations, operating at heads of 2-500 m.

Typical designs are available for turbines of up to 1,500 kW capacity and for some types only up to 1,000 kW. More powerful custom-made turbines are also produced.

The nomenclature of the equipment offered is rather extensive.

1. Axial-flow hydraulic turbines

The bulb straight-flow turbines are used at H_n = 1-4.5 m.

The design of these turbines is conventional, with a bulb at the inlet, a conical wicket gate assembly and an axial runner. The bulb is provided with a step-up gear with an output shaft normal to the turbine axis. This shaft is linked to the vertical-shaft high-speed generator.

The following line-up of runner diameters is adopted:

D_1 = 1.06, 1.12, 1.18, 1.25, 1.32, 1.4, 1.5 and 1.6 m

(preferred diameters are 1.06, 1.18, 1.32 and 1.5).

The speed is n = 150-333 min^{-1}, and the capacity ranges from 50 to 300 kW.

The company manufactures straight-flow bulb turbines with an S-shaped draft tube. The bulb is arranged upstream of the conical wicket gate assembly. The specific speed runner of the axial type is installed in the turbine. This arrangement makes it possible to bring out the horizontal shaft of the turbine to the isolated dry space. The shaft of the turbine is linked with the input shaft of the step-up gear. The outlet of the step-up gear is linked with the generator shaft. The transmission ratio of the gear is chosen by the conditions of application of the high-speed generator. Designs with direct connection of turbine and generator shafts are available. The flywheel is on the generator shaft, and the turbine shaft is provided with two supports, one of them being in the bulb. The second bearing is brought out from the water passage and is arranged at the draft tube bend together with the sealing.

The design provides for various alternatives of turbine control: double control with adjustable blades of the runner and wicket gates and control by the wicket gate assembly only.

The main design of the unit incorporates the horizontal shaft. At the same time, Kessler can supply the equipment with inclined arrangement of the axis of the straight-flow turbine with S shaped draft tube as well as bulb turbines with vertical axis and curved draft tube.

The following line up of runner diameters is assumed for bulb horizontal-shaft hydraulic turbines with S-shaped draft tube:

D_1 = 0.5, 0.6, 0.72, 0.85, 1.0, 1.15, 1.3, 1.45 and 1.6 m.

The speed of the turbine shaft:

n = 750, 600, 500, 428, 375, 333, 300, 250 and 200 min^{-1}.

Capacity range P_t = 25-1,200 kW

at Q = 1.4-15 $m^3 sec^{-1}$ and H_n = 2-12 m.

2. Francis turbines for medium heads

The Reifenstein turbine is offered for medium heads. It is very simple in design. The turbine shaft is vertical with a direct linking to the generator installed directly on the spiral case

Reifenstein turbines with horizontal shafts and straight connection with the direct generator are also manufactured. The simple design results in the low cost.

The following line-up of runner diameters is adopted for vertical turbines:

D_1 = 0.225, 0.25, 0.3, 0.35, 0.40, 0.45, 0.50, 0.55 and 0.60 m

The turbine shaft speed n = 1,500, 1,000, 750, 600, 500, 428, 333 and 300 min^{-1}.

Capacity range P_t = 5-200 kW at water discharge
Q = 0.08-1.5 m^3 sec^{-1} and head H_n = 5-40 m.

The straight-flow turbines are also used for medium heads. These turbines are characterized by a simple and unique design. With regard to conveying features the turbine resembles the straight-flow axial turbine. The water supply to the turbine is effected through the pipe which further passes into the case.

The turbine shaft is horizontal, linked with the generator shaft by the step-up gear. It makes possible the use of high-speed generators.

The draft tube has the bend at the inlet and is straight further downstream. The straight diffuser of the draft tube inclines at 20° to the vertical axis, which improves the general arrangement of the unit.

For reduction of temporary non-uniform rotation of the shaft at variations of load the heavy flywheel is installed for turbines of capacity up to 150 kW. The turbine has an adjustable wicket gate assembly driven by the hydraulic servomotor.

For turbines of such design the following line-up of diameters is adopted:

D_1 = 0.5, 0.55, 0.60, 0.65, 0.7, 0.75, 0.8, 0.85, 0.9, 0.95, 1.0, 1.15 and 1.3 m

The speeds of turbine shaft:

n = 650, 600, 500, 450, 400, 350, 300, 250, 200 and 150 min^{-1}

Capacity range: P_t = 50-2,000 kW at water discharge
Q = 0.8 10 m^3 sec^{-1} and head H_n = 8-30 m

The design of spiral turbines is distinguished by compactness. The shaft of the unit is vertical. The flange-type generator is installed directly on the flange of the spiral case.

The shaft is common to the generator and the turbine. The generator is of a specific design with flywheel and reinforced supports. The design provides for use of low-voltage generators of a specific design. The flywheel may also be supplied together with conventional high-voltage generators.

The low cost of the unit is the result of a double support arrangement and absence of an intermediate gear.

The turbines of this design are characterized by

P_t = 50-500 kW at Q = 0.145-1.3 m^3 sec^{-1}

E. Banki-Mitchell turbines manufactured by Ossberger

The Ossberger company (Federal Republic of Germany) manufactures small hydraulic turbines of quite simple design known as Banki-Mitchell or divided-type turbines. This turbine was first proposed and studied by a Hungarian, D. Banki, and an Australian, Mitchell. Its design has been modified by F. Ossberger.

The Banki-Mitchell divided-type turbines, because of their design and hydraulic features, are not effective for hydraulic stations with units of medium and large capacities. Application of these turbines, in the manufacture of which Ossberger specializes, is therefore limited by small capacities.

Despite the comparatively low level of maximum efficiency, these turbines possess certain advantages.

1. Range of application

The Banki-Mitchell (divided-type) turbine is related to the system of impulse turbines. During operation the turbine runner is partially submerged. From the conveying structures the water enters the runner and, flowing from the periphery to the centre, gives up some 70-80 per cent of energy.

The blades are shaped so that when the water leaves the blades the flow still possesses a considerable amount of kinetic energy. Flowing inside the runner the water again strikes the blade from inside and in a centrifugal flow gives up the remaining 20-30 per cent of energy. Thus the water jet goes through the runner twice.

Since the Banki-Mitchell turbine is an impulse one it may be used within a wide range of heads.

Ossberger has developed the nomenclature of such turbines to be used at:

H_n = 1-200 m   Q = 0.02-9 m^3 sec^{-1} P_t = 5-1,000 kW
n = 50-2,000 min^{-1}

Comparison with the nomenclature of standard turbines made by Voith shows that the range Q-H_n, covers the range recommended by Voith for impulse Pelton turbines, as well as for open-flume and spiral Francis turbines and axial-flow vertical-shaft and horizontal-shaft turbines.

The nomenclature includes the following line up of runner diameters:

D_1 = 0.3, 0.4, 0.5, 0.6, 0.8, 1.0 and 1.25 m

Thus the eight standard turbine sizes offered by Ossberger cover a rather wide range of application Q-H_n.

2. Design features

The turbine shaft is horizontal, the water being conveyed by either horizontal or vertical supply pipe.

The runner, which is provided with two disks between which the blades are welded circumferentially and spaced evenly, is divided into three equal parts throughout its width by means of intermediate disks. It thus consists of three isolated sections similar in inner configuration.

The blades of the runner, made of sheet steel of constant thickness, are of a cylindrical configuration with a single-valued curvature. Depending on the size, the runner may have up to thirty blades.

The runner is not subjected to axial thrust during operation because of the cylindrical configuration of the blades. The runner is all-welded and balanced after finishing.

The water is conveyed to the runner through the adjustable nozzle of a rectangular section. The vane installed in the rectangular nozzle changes the flow area from maximum to zero when it turns. The vane pivot is selected to reduce the hydraulic moment with respect to the axis.

The opening of the water-conveying line is divided into two sections in width by a special partition according to the ratio 1:2. The guide vane in each section may turn independently.

The flow area may therefore be full or open by 2/3 or by 1/3, and the runner may take full discharge or two thirds or one third of it. During partial discharge only the required part of the whole width of the runner operates. Thus a step-like variation of water discharge is achieved.

The guide vanes may be used for a full blocking of water supply and turbine, shut off at heads up to 50 m. At greater heads the installation of a stop valve between the penstock and turbine is required.

Both guide vanes may turn independently by means of control levers connected with the system of automatic or manual control. The main turbine bearings are fitted with standard self-adjustable roller bearings. These bearings are advantageous if the water or condensate do not penetrate inside them. At the same time, the rotor is aligned in relation to the turbine casing. The sealings of bearings require no servicing. The lubrication material is changed in bearings once a year.

Although the Banki Mitchell turbine is an impulse turbine at medium and low heads ($H_n < 35$ m), the draft tube is installed for more efficient utilizaton of the head. It is considered necessary to have the possibility of controlling the pressure in the draft tube especially in turbines with a wide range of control.

The simple air valve, controlling the vacuum in the turbine casing, helps to achieve this in such a way that a small head of 1 m only may be used with optimum efficiency. The draft tube, made completely of steel and with a bend, reduces the cost of construction for low-head installations in particular.

The above-mentioned design features help to maintain the efficiency of Banki-Mitchell turbines at a high level in a wide range of discharge variations. The maximum efficiency of up to 84-88 per cent is observed in medium and large units, which is lower than in modern specific-speed turbines.

However, a high efficiency of not less than 80 per cent is guaranteed throughout the range of control from 1/6 to 1 of the maximum discharge. The turbine has a very flat characteristics curve. It gives a considerable advantage for installations where the river flow decreases substantially during a number of months.

The Banki-Mitchell turbines of Ossberger are supplied together with all necessary accessories. If the turbine is used to drive synchronous or asynchronous generators a step-up gear is applied. In units with synchronous generators the flywheel is installed to reduce temporary non-uniformity. The governor is belt-driven. In low-capacity installations all supplying equipment is installed on a single frame and supplied as a complete set.

Banki-Mitchell turbines may be directly connected with other units and are used to drive high-head centrifugal pumps.

In conclusion, the advantages of such turbines may be summed up as follows:

(a) A simple design and manufacturing procedure resulting in relatively low cost;

(b) High efficiencies (more than 80 per cent) in a wide range of discharges (0.167-1);

(c) Complete automation and simple servicing;

(d) Guaranteed period of reliable operation of 30-40 years.

All these advantages make such turbines competitive with other modern turbines.

F. Small turbines manufactured by Bell

Bell (Switzerland) is specialized in the design and manufacture of hydraulic turbines for small hydropower plants. In response to the new interest in small hydraulic turbines, it has developed a standard line up covering the operating range

$$H_n = 2-800 \text{ m} \quad Q = 0.06-86 \text{ m}^3 \text{ sec}^{-1} \quad P_t = 100-2,000 \text{ kW}$$

with the axial flow turbines intended for application at $H_n = 2-25$ m and $Q = 5-86$ m^3 sec^{-1}

Francis turbines at H_n = 6-150 m and Q = 0.3-6 m^3 sec^{-1}

Pelton turbines at H_n = 50-800 m and Q = 0.06-0.8 m^3 sec^{-1}

The following concerns were taken into account in developing the nomenclature:

(a) Optimum utilization of recent advances in scientific research and design;

(b) Supply of complete electromechanical equipment ready for operation;

(c) Application of simple hydraulic solutions for standard basic designs to reduce costs and speed up supply;

(d) Guaranteed service by branches of the company throughout the world.

The unit cost per kilowatt of hydraulic turbines with a capacity of less than 100 kW is relatively high. Turbines with capacities higher than 2,000 kW are manufactured according to custom-made designs.

1. Range of application of axial turbines

Standardized axial hydraulic turbines are used at

H_n = 2-15 m   Q = 2-38 m^3 sec^{-1} P_t = 100-2,000 kW

The nomograph developed by Bell for preliminary selection of axial hydraulic turbines is compiled for a wider range

H_n = 2-25 m   Q = 4.8-86 m^3 sec^{-1} P_t = 100-10,000 kW

that is, the turbines supplied according to custom-made designs are also included.

The following normalized line up of diameters of runners is proposed:

D_1 = 1.0, 1.2, 1.4, 1.65, 1.9, 2.2, 2.5, 2.8, 3.2 and 3.6 m

At heads up to 4.8 m the recommended range of design conditions is within the limits:

n_1' = 150-200 min^{-1} Q_1' = 2.4-30 m^3 sec^{-1}

With the growth of the head the design conditions move to the lower range of discharges under cavitation conditions.

For standardized turbines the suction head is within the limits H_s = -2 to +2 m.

For example, at H_n = 25 m the range of design conditions with respect to discharge at variation of H_s within the given limits is

Q_1 = 0.8-1 m^3 sec^{-1}

From this it may be inferred that several types of water passages were utilized in the axial hydraulic turbines to cover the range of application.

2. Design features

Bell uses only axial-flow horizontal shaft turbines with an S-shaped draft-tube for low heads. As for the general arrangement of units, these horizontal-shaft turbines do not differ from the similar designs of other companies. The main basic design involves the adjustable wicket gate assembly and blades. This is the double control which ensures the optimum utilization of water resources.

In case of simplified operating conditions and regulated water discharge the propeller turbines with fixed blades of the runner and the adjustable wicket gate assembly or the turbines with the fixed wicket gate assembly and adjustable blades of the runner are used, thus simplifying the turbine and the governor. The easiest alternative is the turbine with fixed gates and blades. Its application is possible at constant water discharges and loading. But the start up of such a turbine requires a certain controllable device at the inlet.

The space of the turbine between the inlet of the bulb and inlet of the S-shaped draft tube is made of metal and is not concreted. The design facilitates construction and repair.

The turbine bearing of the shaft is in the bulb. Here the oil head is also installed to feed the oil to the runner servomotor. The single turbine bearing takes up the axial thrust, with the turbine shaft rigidly connected to the shaft of the step up gear. Therefore the gear supports act as a second turbine bearing.

The S shaped draft tube has a straight diffuser with an inclined axis at the outlet. Alternative arrangements of the turbine, involving the whole of the draft tube in the horizontal plane, are possible.

The wicket gate assembly is conical, with the gate supports in the bulb and in the outer shell. The governing system includes the levers in gates, links and a shifting ring driven by the servo motor.

All horizontal turbines are provided with a one stage step up gear and a high speed generator. The flywheel is mounted in the generator shaft.

Comparison of various governing systems shows that a decrease in efficiency by 20 per cent takes place in the following turbine types: propeller type at $0.85Q_{max}$; with fixed gates at $0.35Q_{max}$; and with dual control at $0.2Q_{max}$.

The number of blades in the runner, depending on specific speed, varies from 3 to 5. The number of gates in the wicket gate assembly is 16. Turbines with fixed wicket gate assembly require the additional stop device.

Depending on the value of the head, two types of installation of horizontal turbines are recommended. At low heads the open headrace $2.4D_1$ wide is applied. The powerhouse is integrated with the dam. At high heads the penstock of $1.4D_1$ diameter is used. Here the powerhouse is separated from the dam. The width of the tailrace is $2D_1$; the width of the powerhouse with one unit is $3.6D_1$; and the length of the powerhouse along the axis of the unit is $8D_1$.

Apart from the given arrangement the company offers other possible alternatives:

(a) A horizontal shaft and a draft tube, with a vertical diffuser at the outlet;

(b) An inclined shaft of the unit with the generator installed above the turbine and the bottom of the headrace and the tailrace approximately at the same level;

(c) Vertical unit with an open flume, a bulb and conical wicket gate assembly and a bent draft tube with the generator installed either above the upstream water level or in the pit under the bend of the draft tube

The small hydraulic turbines supplied by Bell are provided with automatic control, including the following features: an electric system of the automatic start up; hand control; remote control of automatic operation; and local or remote control of the governor.

The following devices ensure reliable operation:

(a) Devices triggering the closing of the gates at any time under the action of the special weight suspended to the shifting ring;

(b) A mechanical centrifugal pendulum which eliminates the possibility of speed rise;

(c) Automatic devices preventing any undesirable operating conditions.

In addition to the hydraulic turbine, the governor is also supplied.

The electronic speed governor is a part of the governing system, which maintains the a.c. frequency, produced by generator for an isolated line or the power grid

The electronic speed governor includes the hydraulic actuator (the main distributing valve, the main servomotor and the oil supply system).

G. Small turbines manufactured by Sanden

Sanden is a leading Norwegian enterprise specializing in the design and manufacture of turbines for small hydroelectric power-stations.

The range of standard hydraulic turbines of the company covers:

H_n = 3-1,000 m Q = 0.05-30 m^3 sec^{-1} P_t = 100-10,000 kW

The range of high heads is covered by Pelton turbines of standard design, adjusted to local conditions. The same is true of Francis turbines with a capacity exceeding 1,000 kW. For capacities up to 1,000 kW there are standard turbines with the predeveloped design.

Axial turbines are recommended for the range:

H_n = 3-18 m Q = 3-27 m^3 sec^{-1} P_t = 200-2,800 kW

Francis turbines are used at:

H_n = 4-400 m Q = 0.4-25 m^3 sec^{-1} P_t = 100-10,000 kW

Pelton turbines cover the range:

H_n = 80-1,000 m Q = 0.05-2 m^3 sec^{-1} P_t = 100-10,000 kW

1. Spiral Francis turbines

The spiral turbines are manufactured to the basic design. To ensure high efficiency over a wide range of heads and discharge Sanden uses twelve series of runners. The water passages of one series are approximately geometrically similar. The main products are standardized, but some modifications of sizes depending on the head and discharge are possible.

Table 8 shows the ranges of application of turbines of various series. The following is the normal line-up of diameters:

D_2 = 0.3, 0.35, 0.4, 0.45 0.5, 0.6, 0.7, 0.8, 0.9, 1.0, 1.2, 1.4, 1.6, 1.8 and 2.0 m

The diameter D_2 was adopted as the typical dimension of the turbine at the outlet of the runner.

The design operating conditions of the turbine are ensured at the suction head H_s = -2 to +2.5 m at the upper boundary in respect to the head and H_s = 6.5 to 7.5 m at the lower boundary in respect to the head.

The efficiency of turbines at D_2 = 1 m has been measured as follows: type C (91.7 per cent); type H (92.3 per cent); and type N (92 per cent).

The turbines of the spiral type are made with horizontal and vertical shafts.

Table 8. Ranges of application of turbine series

Turbine series	H_n (m)	Q (m^3 sec^{-1})	n_s
B	110-400	0.5-2.8	75
C	100-350	0.5-3.2	80
E	85-300	0.5-4.0	100
F	70-250	0.5-5.5	120
G	60-230	0.5-6.5	130
H	50-200	0.7-9.0	140
J	45-170	0.8-10.0	170
K	35-130	0.9-15	200
M	25-95	1.2-18	250
N	18-65	1.6-23	370
O	14-55	2.5-26	400
P	8-40	2.5-33	475

The turbine parts at the inlet and outlet may be changed and their design adapted for specific conditions of the particular installation.

In the standard turbines with horizontal shafts the runner is overhung on the generator shaft. In vertical installations the turbine has its shaft and bearing.

In low-speed and low-capacity turbines of less than 5,000 kW a step-up gear between the turbine and generator is allowed. At very low heads the vertical turbines with an open flume are used to reduce costs.

Other design and manufacturing features should be noted. The spiral case in sheet steel is welded to the stay ring. The gates of the adjustable wicket gate assembly are welded of stainless steel. The levers and links are of conventional rolled steel. The bearings of gates are self-lubricated.

The turbine covers are made of sheet steel. Stainless steel is welded in the places of labyrinth seals.

In turbines with fixed wicket gate assembly the welded gates are supplements to the stay vanes.

The runner is a welded construction. The blades are stamped of plate steel. The material of the runner is stainless or low- alloy carbon steel. The smallest runners may be cast of nickel- aluminium bronze.

The runner is fitted to the shaft hydraulically to allow a simple assembling and dismantling of the given connection. This method eliminates the use of the key or some other device loosening the shaft and entailing metal fatigue.

The runner has removable rings of a slit seal on the drive and the driven disks. The case of the shaft seal is welded. Within the limits of the seal the shaft is plated with white metal. The seal, from which there runs a drainage pipe, requires no servicing.

The air is supplied behind the runner through the seal case and holes in the runner hub. The draft tube consists of an outlet bend and a conical diffuser. To prevent cavitation failure the stainless steel is used at the inlet of the bend.

The turbine installations, which always operate in a big low-voltage system, and which are not designed for frequency control, may be fitted with the control system. These control devices are designed for hand and automatic control or remote control of the start-up, loading and shut-down of the turbine.

The automatic control is not designed for speed control when the unit operates for an isolated line, but it ensures adequate connections with output loads and with the water-level regulator.

The installations, which operate to an isolated line, require the application of the speed governor, which is designed for automatic speed control, depending on the output developed by the generator. Depending on the turbine type and its dimensions, the governor may be either hydromechanical or electrohydraulic.

In order to determine whether the characteristics of the installation will be stable in the isolated line, the relevant data and sizes of the penstock or the headrace are required.

The turbine shut-off valve is usually installed just in front of the turbine. The throttle valves are used for heads up to 160 m. Closing valves are effective with the auxiliary, compensating weight or when self-closing. The opening is effected by a servomotor.

The spherical gates are used at heads exceeding 160 m. The opening and closing are effected by the water control system or the oil pressure plant. At low heads in hydroelectric plants with short penstocks, the shut-off valve in front of the turbine may be omitted.

2. Francis turbines in drum

This type of turbine has no definite designation. It differs substantially from the spiral hydraulic turbines mainly through its water conveying features. Eight standard sizes were developed for such turbines.

Table 9 shows the main operating parameters.

All the dimensions of these normalized turbines are determined. Probably only 5 runners of various specific speeds were used. The normal suction head shown in the table is allowable at the nominal head and the minimum value H_g at the maximum head. The maximum efficiency of the given turbines is within the range of 85 90 per cent, which is considerably less than in turbines of the spiral type.

Table 9. Main operating parameters of turbine series

Turbine series	H_n (m)	Q (m^3 sec^{-1})	H_s (m)	n_s
I	8-25	2.2-4.5	1.4-6.5	380
II	8.2-34	1.4-3.6	4.0-6.5	280
III	8.5-40	1.2-3.0	2.8-6.5	280
IV	10-55	0.9-2.5	3.6-6.5	225
V	13-55	0.7-1.8	2.3-6.5	235
VI	16-82	0.55-1.55	4.5-6.5	155
VII	22-105	0.4-1.2	2.8-6.5	155
VIII	28-110	0.31-0.8	3.9-6.5	120

Water is conveyed to the turbine through the tube, which is connected to the cylindrical drum. The axis of the drum coincides with the axis of the turbine and the water conveyance is normal to the axis. The cylindrical drum is welded of ordinary carbon steel and provided with a convex bottom. The hatch and the drain valve are also provided.

The adjustable wicket gate assembly is provided with cast gates. The levers and links are made of carbon steel. The gate bearings are self-lubricated. The turbine covers are made of carbon steel. The lining of stainless steel is welded to the cover in the area of the runner seals. The seal at the outer disk of the runner is removable.

The runner is welded and the blades are stamped of plate steel. The material of the runner is stainless steel or plain carbon steel. Small runners are cast of bronze.

The runner is pressed into the shaft and has replaceable sealing rings.

The box of the shaft sealing is welded. In the area of the sealing the shaft is plated with white metal. The leakages from the sealing are designed to be drained. The air is supplied to the runner through the box of the sealing and special passages in the runner hub.

In the installations, which always operate for power grids and which do not require the application of frequency regulators, the ordinary system of turbine control, including manual, automatic and remote control of the start-up, loading and shut-down is used.

The hydroelectric stations, operating for an isolated line periodically or constantly, require the installation of speed governors. The governor is designed for automatic control of speed, depending on the generator loading. In this case hydro-mechanical speed governors are used.

The shut-off valve is usually installed just in front of the turbine. The butterfly valve is used in all normalized hydraulic turbines. Either self-closing throttles or throttles with balancing load are used. The throttle valve is fitted with an oil servomotor.

3. Axial turbines

The axial hydraulic turbines manufactured by Sanden are provided with horizontal shafts and S-shaped draft tubes. The nomograph for preliminary selection of the axial turbine includes information on normalized designs. The normal line up of diameters of runners with 0.2 m pitch is as follows:

D_1 = 1.0, 1.2, 1.4, 1.6, 1.8, 2.0, 2.2 and 2.4 m

The use of the normalized turbines is recommended for:

H_n = 3.5-18 m   Q = 3.5-27 m^3 sec^{-1} P_t = 400-3,000 kW

Two types of axial turbines are produced, one with fixed wicket gate assembly and runner blades, the other with fixed wicket gate assembly and adjustable runner blades. The turbine has no speed governor if it operates for the interconnected power grid. The turbines operate without servicing.

The first type of turbine is used at hydraulic stations with relatively constant discharge and head, when it is not necessary to control power. The second type is used at stations with variable discharge and power control. These turbines are characterized by a good level of efficiency in the range of discharges from 40 to 100 per cent of the full maximum discharge.

The turbine has the following main components: a butterfly valve; an inlet element with guide bearing and wicket gate assembly; a runner; a draft tube with guide bearing; and a governor.

The valve housing and valve disk are welded of plate steel. The disk is installed on the two eccentric supports with stainless journals and self-lubricating bushings. There is a rubber sealing ring adjusting the stainless sealing ring of the valve housing. The sealing may be regulated when the valve is closed and, if necessary, it may be replaced without dismantling the valve. The valve housing is flanged to the inlet section of the turbine. The connection with the penstock is usually welded.

The disk is controlled by the hydraulic cylinder, to which oil is supplied from the governor. The piston shifts the valve for the opening. The throttle is self-closing, in other words, it is closed automatically when the oil is drained from the cylinder.

The inlet section of the turbine is welded, and the blades are welded to the outer shell and the bulb. The sliding bearing of the turbine shaft is arranged in the bulb. The bearing is water lubricated, does not require water proof seals, and is resistant to impurities and solids in the water.

During the start-up and shut down of the turbine the lubricating water goes to the bearing through the bypass. The part of the inlet section which adjoins the runner is made of stainless steel. The draft tube is flanged to the inlet section, which acts as the runner chamber and is made of stainless steel. The rest of the tube is welded of ordinary plate steel. The configuration of the tube is favourable from the standpoint of hydraulics. Therefore high efficiency and minimum vibration are achieved during operation.

For the purpose of disassembly of the turbine the draft tube is divided into two flange connected sections. The lower section is concreted into the foundation.

The housing of the thrust and guide bearing is welded to the draft tube on stiffeners. The bearing is of a spherical roller-type with oil lubrication and temperature monitoring. The draft tube has a hatch for inspection of the runner and the tube. The runner is fitted into the shaft and transmits the moment through friction. The runner blades may be cast of bronze, which is well resistant to rust and cavitation. During repairs they may be welded up without preheating.

The runner blades may be adjustable. The force required for adjustment of blades is transmitted from t e two hydraulic cylinders, housed in the thrust bearing case through the pipe, wrapping the turbine shaft and connected to the runner hub. The pipe rotates together with the shaft and is driven by the servo-cylinder. The hydraulic seal with the leakage drain to the drainage system is located at the pipe outlet.

The governing system consists of the oil pressure plant and the governor with solenoid valves controlling the position of runner blades and the valve disk.

The system is intended for manual and automatic start-up, putting the unit on line, shut-down and remote control of loading. It is not intended for speed control, but is well fitted for load and head control.

The governing system of the turbine consists of the standard parts. The standard high speed generator is usually applied to reduce costs. Therefore the gear is installed between the turbine and generator.

H. Small turbines manufactured by Hitachi

Under the impact of the energy crisis Hitachi (Japan) began to pay much attention to design and manufacture of small electric power stations. The main obstacle to wide use of the energy of small rivers is the relative high cost of small hydroelectric stations and the equipment required for them. For profitable construction it is necessary to reduce equipment costs, which is only possible when there is wide standardization of the main and auxiliary equipment.

Hitachi has developed standardized ranges of application for various types of hydraulic turbines used in small hydroelectric power stations with capacities of from 50 to 10,000 kW. Table 10 shows the ranges of application of various types of turbines.

Table 10. Various types of turbine: ranges of application

Type of turbine	H_n (m)	n_s
Pelton	200 m and higher	9-35
Francis	10-600	55-400
Kaplan	5-100	230-1 050
Horizontal axis	3-20	450-1 190

Detailed information on different types of turbine is given below.

1. Francis turbines

The nomographs for preliminary selection of turbines were developed for countries using frequencies of 50 or 60 Hz in electric lines.

Use of the normalized turbines is proposed at:

H_n = 20-180 m Q = 0.8-25 m^3 sec^{-1} P_t = 300-10,000 kW
n = 375-1,000 min^{-1} (for frequency of 50 Hz)

The following line-up of the diameters of runners for these turbines was taken:

D_1 = 0.375, 0.40, 0.425, 0.450, 0.475, 0.50, 0.530, 0.560, 0.60, 0.63, 0.67, 0.710, 0.750, 0.80, 0.85, 0.9 , 0.95, 1.00, 1.060, 1.120, 1.180, 1.250 and 1.320 m

Altogether, 23 normalized sizes were adopted. It should be noted that the minimum diameter at the leading edge of the runner blade was expressed through D_1. Fourteen different models of the runner are used for the considered range of H_n and Q.

Table 11 shows the range of application of each type of turbine (for frequency of 50 Hz). A total of 151 standard sizes of turbines are recommended for use.

Similar information is available for a frequency of 60 Hz. Here 137 standard turbine sizes are used.

The distinguishing feature of vertical turbine design is its distinct difference from conventional designs for large and medium units. It has been possible to reduce considerably the length of the shaft and the height of the unit through the use of the common shaft for the turbine and the generator. The shaft is provided with two supports. One of them is a radial support in the area of the top spider of the generator, and the other is on the turbine cover, where the radial and axial thrusts are taken up.

The stator rests on the flange of the turbine stay ring, which is cast together with supports. The shells of the spiral case are welded; the design of the hydraulic part of the unit is conventional; and the adjustable wicket gate assembly has an outer shifting ring.

Table 11. Operating parameters of various turbine types with frequency of 50 Hz

Turbine type	n_s	H_n (m)	Q (m^3 sec^{-1})	P_t (kW)	n (min^{-1})	D_1 (m)
A	105	95-120	1.8-2.7	1 500-3 000	750	0.8-0.85
B	135	68-160	1.0-9.0	750-10 000	1 000-600	0.6-1.32
C	150	60-150	0.9-9.0	500-10 000	1 000-600	0.5-1 18
D	165	48-160	0.8-12	300-10 000	1 000-500	0.475-1.32
D_o	170	98-115	11-13	900-1 000	500	1.32
E	192	35-62	0.8-1.5	300-750	1 000	0.425-0.475
F	205	29-115	1.0-14	300-10 000	750-500	0.5-1.25
G	235	30-45	0.85-1.2	300-450	1 000	0.4-0.425
H	245	22-88	0.9-14	250-10 000	750-500	0.45-1.18
I	255	65-85	14-18	1 000	428	1.32
J	290	21-76	1.2-22	300-10 000	750-428	0.425-1.25
L	325	18-55	1.3-25	300-10 000	750-375	0.375-1.18
M	375	19-48	2.0-26	300-10 000	750-375	0.4-1.12

The design of the unit makes possible a considerable reduction in the period of assembly. For example, for turbines of capacity 3.5 MW of conventional design, the period of assembly usually took 75 days. This period has been reduced to 35 days, thus by nearly one half, for turbines of normalized design. The assembly of such a unit may be performed by means of a mobile crane.

The hydraulic turbine may be connected with a synchronous or an induction generator. The induction generator has no exciter and governor if the electric line is sufficiently extensive and high-powered. Thus the induction generator allows the use of simplified equipment, which results in cost reduction. The largest generator of this type in Japan at the beginning of 1979 was supplied by Hitachi. Its capacity was 7,850 kW.

The turbine governor is either mechanical or electrical, the latter type being smaller and easier to service.

The mechanical governors are standardized for turbines of small hydropower stations. If the induction generator is used, then the speed is not controlled by the governor. In this case its function is to control the unit, depending on variations of the upstream water level and the load.

Properly designed control and protective equipment are supplied. The electrical instrumentation, protection devices, the exciter and the electrical speed governor are installed in the common block in the standard installations with the synchronous generator of 5,000 kW or lower capacity.

Apart from the vertical turbines the standardized horizontal spiral turbines are used at:

H_n = 20-100 m and P_t = 50-1,000 kW

The hydraulic part of these units is of a conventional design with the shifting ring of the wicket gate assembly installed from the side of the draft tube. One guide turbine bearing and two generator bearings are installed. The turbine and generator shafts are rigidly flanged and the flywheel is installed between the flanges.

2. Axial turbines

The company recommends the following two types of horizontal axial turbines for practical use at small hydroelectric power-stations: straight-flow bulb type; and bulb type with an S-shaped draft tube. There are nomographs for preliminary selection of hydraulic turbines for frequencies of 50 and 60 Hz.

The normalized hydraulic turbines cover the following range in relation to the head and discharge for the frequency of 50 Hz

H_n = 3-20 m   Q = 2-28 m^3 sec^{-1} P_t = 100-5,000 kW
n = 214-750 min^{-1}

The unified line-up of runner diameters includes the following sizes:

D_1 = 0.70, 0.75, 0.80, 0.90, 1.00, 1.12, 1.25, 1.40, 1.60, 1.80 and 2.00 m

Altogether there are 11 normalized sizes.

Forty-eight standard sizes of axial hydraulic turbines are proposed to cover the established ranges in relation to the head and discharge.

1. Small turbines manufactured by Gilbert Gilkes and Gordon

Gilbert Gilkes and Gordon (United Kingdom of Great Britain and Northern Ireland) manufactures turbines of all types, with the exception of axial-flow turbines. Equipment intended for small hydroelectric power stations is one of its main products.

The approximate range according to head and capacity falls within the limits:

H_n = 3-24 m P_t = 8-6,000 kW

Seven types of water passages of Francis turbines and five types of impulse turbines are available to cover the given range of application.

Table 12 shows the specific speeds for all types of turbines.

Table 12. Specific turbine speeds

Type	n_s
C	311
2	302
R	244
III	187
β1	185
IV	142
V	109
SJ	65
TJ	65
TI	51.5
HCTI	65
PELT	22.6

The runners with the highest speed are used for low heads within the limits H_n = 2.8-4.2 m. Only Francis turbines are recommended for heads up to H_n = 3-13 m.

Francis and impulse Turgo turbines are used for the range of heads H_n = 15-35 m. Francis turbines are not used for heads higher than 150 m. Impulse inclined-jet hydraulic turbines are recommended for heads H_n = 15-400 m and the impulse bucket turbines for the heads H_n = 40-1,200 m.

Hydraulic turbines with an open flume for low heads are of simple design. The turbine shaft may be either vertical or horizontal. The shaft is connected with the step-up gear in vertical turbines, the output horizontal shaft of which is connected with the high-speed generator. The flywheel is installed on the horizontal intermediate shaft.

The belt drives are used to drive the generator; the spiral radial-axial hydraulic turbines are used for low heads; and the water is conveyed through the penstock. In all cases the draft tube is straight and conical.

The horizontal shaft rests on two bearings in the spiral hydraulic turbines. It is connected either with the a.c. (synchronous) generator or with any other unit by means of a rigid coupling. The third bearing is added if the belt drive is used.

The cast or welded steel spiral cases are used for medium heads. The impulse inclined-jet hydraulic turbines are also made for medium heads. There are two types, with one or two nozzles, the specific speed of which is about 65. The turbine is simple in design and easy to manufacture.

The efficiency of such turbines is higher than that of low-speed radial-axial hydraulic turbines. In particular, turbines with small seal clearances of the runner wear considerably in the presence of abrasive suspended matters in the water. The inclined-jet turbine is always characterized by higher efficiency at loads of less than 5/8 or 3/4.

Double control by means of the nozzle, needle and deflector is realized in the inclined-jet turbines. The efficient control system is particularly important in long penstocks.

The impulse bucket and inclined-jet hydraulic turbines are used for high heads, except in the case of small capacities. The bucket hydraulic turbines are not used at heads of less than 100 m. The bucket hydraulic turbines are made for heads up to 800 m. For high heads these turbines may be supplied according to custom-made designs.

The impulse turbines are mainly made with one nozzle; turbines with two runners are also used. For the most part, turbines with 22.6 specific speed are employed.

The runner is installed with its own bearings for turbines with a capacity of less than 750 kW. The runner is arranged at the generator shaft end for higher capacities. In this case the supports and the generator shaft are of special reinforced design, making it possible to reduce costs.

The generators are of custom-made design; they are designed for speed increases of 70-200 per cent in runaway conditions.

The synchronous generator, operating for the line or the isolated load, is driven by the turbine at conventional hydro-electric power-stations. The generator and the governor should be designed in such a way as to ensure the required frequency in parallel and isolated modes of operation.

Asynchronous or induction generators are also used on a wider scale. These generators are of simple design and cheaper in comparison with synchronous generators. Speed is controlled by the line frequency on which the generator operates. The efficiency of the asynchronous generator is often less than that of the synchronous generator.

Gilbert Gilkes and Gordon does not manufacture its own governors. The low-capacity turbines are controlled by electric governors.

SOMMAIRE

Les directives ci-après sur l'utilisation des petites turbines hydrauliques ont pour objet de présenter les caractéristiques de cet équipement à des spécialistes de l'utilisation de l'énergie hydraulique fournie par les petits cours d'eau.

Les directives comportent une brève description des turbines et de leurs configurations de base, ainsi qu'un examen des principaux problèmes et méthodes de normalisation. Elles donnent également des exemples spécifiques de mise au point de types courants de turbines à réaction et à impulsion, un aperçu de l'expérience acquise dans le domaine de la normalisation et des propositions émanant des constructeurs.

EXTRACTO

El objetivo de las siguientes pautas para la aplicación de pequeñas turbinas hidráulicas es presentar las características de diseño del equipo de turbina a especialistas en el aprovechamiento de la energía hidráulica de los ríos pequeños.

Las pautas comprenden una breve descripción de las turbinas y las configuraciones básicas de la turbina y un examen de los principales métodos de normalización y problemas que esto plantea. Se dan también ejemplos especiales de desarrolo de gamas normalizadas de turbinas de acción y de impulsión, una exposición general de la experiencia adquirida en la esfera de la normalización y sugerencias de fabricantes de pequeñas turbinas.