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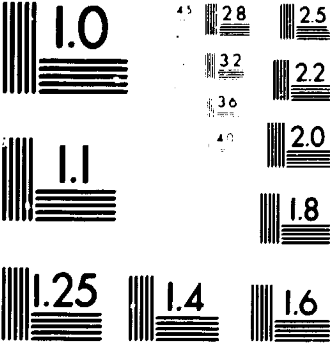

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GUIDELINES FOR THE APPLICATION OF SMALL HYDRAULIC TURBINES *

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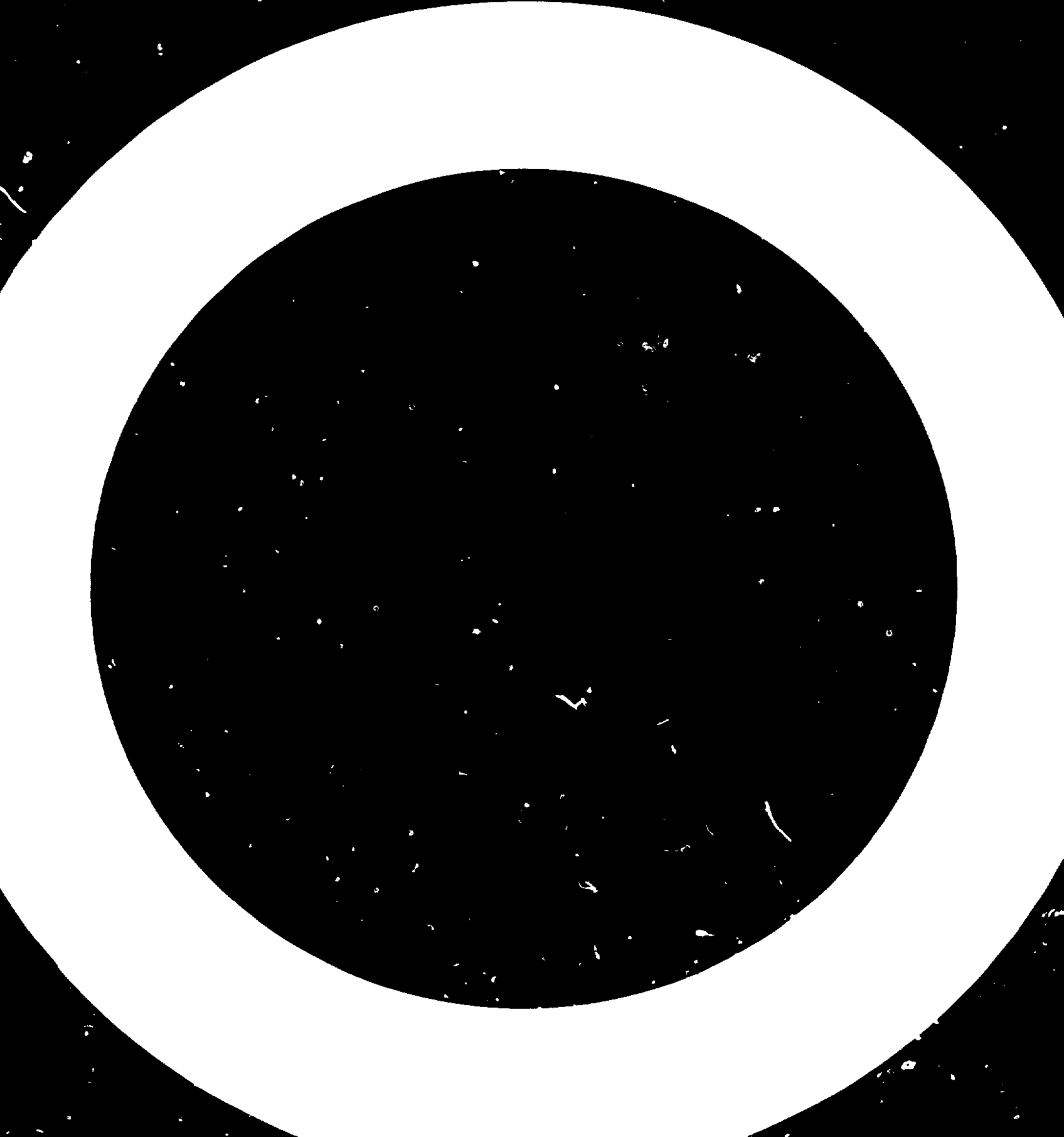
SUMMARY

The small hydraulic turbines application guidelines have been worked out on UNIDO's initiative.

The purpose of the guidelines is to introduce the design features of the turbine equipment to specialists engaged in utilization of water power of small rivers.

The guidelines give a brief desciptive account of turbines and basic turbine configurations ,the main problems and ways of standardization of turbines.The guidelines demonstrate particular examples of development of standard ranges of reaction and impulse turbines.

The guidelines briefly outline experience of standardization and suggestions of small turbine manufacturers.



Contents

I.PREFACE

The present guidelines worked out on UNIDO's initiative contain information about hydraulic turbines and other equipment applied at small hydropower plants.

In recent years rather high interest has been created in different sources of renewable energy due to the rise in prices for organic fuel.Among these sources of energy the sources associated with energy of rivers are of particular interest.The most economic power plants installed at large rivers are of high capacity.In developed industrial countries the most economic sources have been either turned to account or are under development .At relatively low costs of construction of large hydropower plants they demand large capital investments and long periods of construction.

By this reason in the countries with a high level utilization of hydropower resources interest in small hydropower installations has been expressed again.

In developing countries the financial possibilities do not always exists for construction of large hydropower plants.Therefore out of renewable sources of energy in these countries development of energy of small rivers in the form of construction of small hydropower installations is of principal importance .

It should be noted that small hydropower plants may be also profitable when installed at various water control structures associated with water supply,navigation ,irrigation,etc.

Specific cost of energy at small power plants is higher than at large hydropower installations.Therefore during designing and construction of small hydropower plants it is necessary to search continuously for ways of raising profitability through reduction of construction costs by employment of unified structural elements by way of application of the standard mass-produced equipment.

Operating costs of small hydropower plants shall be reduced to a minimum through complete automation of plant operation, employment of remote control systems and rejection of permanent operating personnel. Reliability of the equipment and maintenance intervals shall be high. Industrial methods shall be employed in maintenance of the equipment .

The range of utilization of energy of small rivers and other water control systems depends to a certain extent on the state policy of stimulation of construction of small hydropower installations. In some countries certain finacial stimuli have been given in the form of gratuitous loans for construction of small hydropower installations. In these conditions construction of small hydropower plants appears profitable to meet the demands of isolated consumers and small power systems .

Hydraulic turbines installed at small hydropower plants will be referred hereinafter conditionally as small turbines . At the present time there is no limiting value of parameters of small hydraulic turbines. Various countries and companies have variable limiting parameters of small hydraulic turbines.

It is likely that the maximum capacity of a small turbine may be adopted as 10000 kW though some turbine manufacturing companies cover the range of their small turbines up to 15000 kW. The minimum capacity of small turbines may be adopted as 50 kW .

The highest demands for small power plants are the turbines of 1000-2000 kW capacity . For reduction of their costs it seems necessary to develop standard designs of these turbines with building blocks .

The turbines of high capacity are manufactured by custom-made designs and this fact results in higher specific costs naturally .

The turbines of small power plants operate complete with other equipment including generators, speed governors, control systems and step-up gears. All this equipment must be standardized and mass produced .

The purpose of the small hydraulic turbines application guidelines is to introduce the main conclusions of the theory of turbines which are necessary 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 . The guidelines present the main principles and tasks of standardization of turbines illustrated by particular examples .

In conclusion standards of turbines manufactured by different companies are shown .

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

The author expresses his gratitude to the United Nations Industrial Development Organization and managements of Voith , Voest-Alpine, Kessler , Sanden , Hitachi , Bell , Ossberger , Gilbert Gilkes and Gordon turbine-manufacturers enabling the author to familiarize himself with manufacturing of small hydraulic turbines and to get

necessary information tc be used in the guidelines.However information was limited and consequently may result in some inaccuracies. The author will accept and take into account helpful suggestions and remarks .

2.HYDRAULIC TURBINES.GENERAL INFORMATION .

2.1.Main concepts and definitions .

The hydraulic turbine is a machine converting the energy of the water into mechanical energy by use of the runner which serves as a 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:

$$E = gz + \frac{P}{\rho} + \frac{V^2}{2} \qquad J/kg \qquad (2.1)$$

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

z - elevation of the considered particle of a liquid above the conditional plane of comparison,m:

g - gravity , $m^2\ sec^{-1}$;

P - pressure , N/m^{-2} ;

ρ - liquid density , kgm^{-3} ;

V - liquid particle velocity , $msec^{-1}$.

From the above equation (2.1) 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 .

$$H = Z + \frac{P}{\rho g} + \frac{V^2}{2g} \quad , \quad m \qquad (2.2)$$

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

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

Fig.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

$$H_g = z_A - z_B \qquad (2.3)$$

This value is known as the gross head.

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

During operation of the turbine the water flows through water coveying features,the hydraulic turbine and outlet features of the hydropower plant . Because of losses of a part of mechanical energy of the liquid resulted 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 of specific energies of the flow between these sections is

$$H_n = (z_1 + \frac{P_1}{g\rho} + \frac{V_1^2}{2g}) - (z_2 + \frac{P_2}{g\rho} + \frac{V_2^2}{2g}) \qquad (2.4)$$

and is know as the het 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 (2.4) are average values for sections 1-1 and 2-2 and are equal to

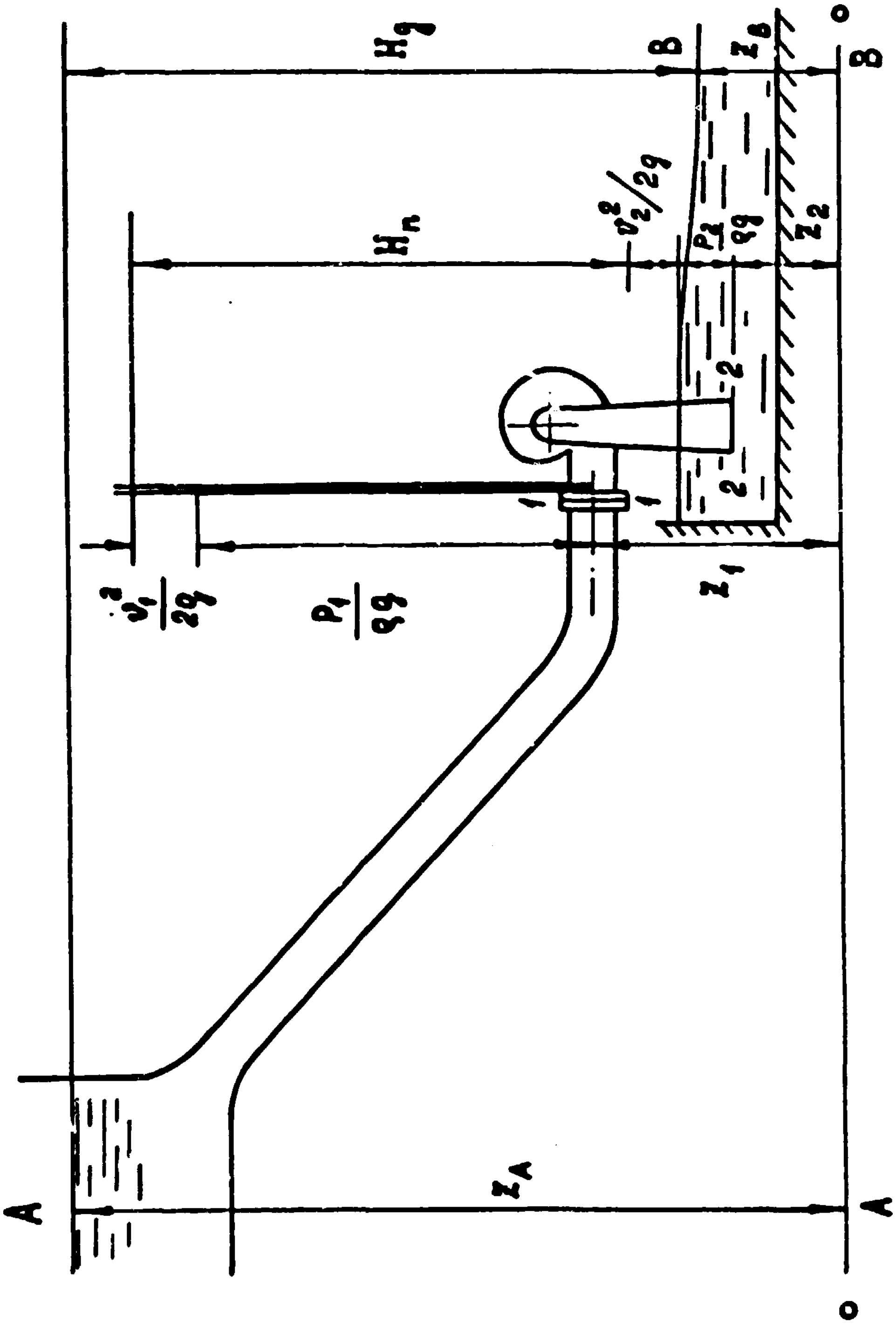


Fig.1

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

where A_1 and A_2 - sectional areas , m^2 .

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

$$P_d = 9.81QH \quad , \quad kW \qquad (2.5)$$

at $\rho = 1000\ kgm^{-3}$ and $g = 9.81\ msec^{-2}$.

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

$$P_t = \eta_t \, P_d \quad ,$$

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 - 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 not do any 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 ,Nm ,

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 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 occurred 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

1) Power - P_t , kW ,

2) Net head - H_n , m ,

3) Discharge - Q , m^3/sec ,

4) 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 combination of working parameters .At a fixed value of n there are head and discharge with which the maximum efficiency of the turbine takes place.This condition is called optimum.At deviation from the optimum conditions the efficiency of the turbine will lower down.

2.2.Similarity laws

The turbines are considered geometrically similar if relative dimensionless coordinates of the water passage of these turbines are similar . To obtain the dimensionless coordinates it is necessary to find the relation of the coordinates to a certain characteristic linear dimension,e.g 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:

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

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

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

Let us denote peripheral velocity by $\vec{U}$,relative velocity by $\vec{W}$ and absolute velocity by $\vec{V}$.

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

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

Fig.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 Fig.2 the triangle of velocities for the similar operating conditions is shown by a dashed line .

It is known that in compliance with the Euler equation

$$g\eta_h H_n = u_1 v_{1u} - u_2 v_{2u} \qquad (2.6)$$

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

$$\frac{V_{1m}}{U_1} = \text{idem} , \quad \frac{V_{1u}}{U_1} = \text{idem} \qquad (2.7)$$

and at the runner outlet

$$\frac{V_{2m}}{U_2} = \text{idem} , \quad \frac{V_{2u}}{U_2} = \text{idem} \qquad (2.8)$$

Since

$$U \sim nD , \qquad V_m \sim \frac{Q_r}{D^2} , \qquad (2.9)$$

then it seems possible to express the conditions of kinematic similarity in terms of operating parameters of the turbine .In conformity with (2.6),(2.7),(2.8),(2.9)

$$\frac{g\eta_h H_n}{n^2 D^2} = \text{idem} , \qquad \frac{Q}{nD^3} = \text{idem} \qquad (2.10)$$

From expressions (2.10) we obtain the known similarity formulas connecting the operating parameters of two geometrically similar turbines operating under similar kinematic conditions :

$$n' = n \frac{D}{D'} \sqrt{\frac{H_n'}{H_n}} \sqrt{\frac{\eta_h'}{\eta_h}} \qquad (2.11)$$

$$Q' = Q \left(\frac{D'}{D}\right)^2 \sqrt{\frac{H_n'}{H_n}} \sqrt{\frac{\eta_h'}{\eta_h}} \frac{\eta_v}{\eta_v'} \qquad (2.12)$$

$$P_t' = P_t \frac{\rho'}{\rho} \left(\frac{D'}{D}\right)^2 \left(\frac{H_n'}{H_n}\right)^{3/2} \left(\frac{\eta_h'}{\eta_h}\right)^{3/2} \frac{\eta_m'}{\eta_m} \qquad (2.13)$$

Let us exclude linear dimensions of turbines from formulas (2.11),(2.13).Thus we obtain at $\rho = \rho'$

$$n' = n \left(\frac{P_t}{P_t'}\right)^{1/2} \left(\frac{H_n'}{H_n}\right)^{5/4} \left(\frac{\eta_h'}{\eta_h}\right)^{5/4} \left(\frac{\eta_m'}{\eta_m}\right)^{1/2} \qquad (2.14)$$

In conformity with formulas (2.11)-(2.14) 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 it is allowable to neglect the relations of volumetric and mechanical efficiencies just taking them 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

$$\eta_h' = 1 - (1 - \eta_h) \left[(1 - x) + x \left(\frac{Re}{Re'}\right)^{1/5} \right], \qquad (2.15)$$

where

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

where ν - temperature dependent coefficient of kinematic viscosity of water. Table I shows approximate values.

Table 1

$10^6 \nu, m^2 c^{-1}$	1.3	1.14	1.0	0.9	0.81	0.75
t^0, C	10	15	20	25	30	40

Coefficient X in the formula (2.15) represents the share of recalculated hydraulic losses. It is equal to

$$X=0.6-0.75$$

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

2.3 Specific speed

The power develeloped by the turbine varies proportionally to 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 500-1000 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

$$n_s = \frac{n\, P_t^{1/2}}{H_n^{5/4}} = 3.65 \ \frac{n\, (\eta_t Q)^{1/2}}{H_n^{3/4}} , \qquad (2.16)$$

where n - speed , min^{-1} ,

H_n- net head , m ,

P_t- turbine power ,h.p. ,

Q - discharge , $m^3\ sec^{-1}$.

If the power is measured in kW,then

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

If the head is measured in feet and the power in Br.h.p.,then in compliance with the formula (2.16) we obtain

$$n_s\ (\text{m,h.p.}) = 4.446\ n_s\ (\text{ft.Br.h.p.})$$

Formulae of specific speed is obtained from (2.I4) if we ignore change of hydraulic and mechanical efficiencies.In these con - ditions 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 deter- mined by operating parameters existing under conditions of ma- ximum efficiency .

The specific speed of modern turbines varies over a wide ran- ge

$$n_s = 10 - 1200$$

The turbine of 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.

2.4 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 depent 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 formulae (2.11),(2.12),(2.13) we assume D' = 1 m and H_n' = 1 m , then we obtain the reduced values

$$n_I' = \frac{nD}{\sqrt{H_n}} \frac{1}{\sqrt{m}} \quad , \quad min^{-1} \qquad (2.17)$$

$$Q_I' = \frac{Q}{D^2\sqrt{H_n}} \frac{1}{\sqrt{m}} \quad , \quad m^3 s^{-1} \qquad (2.18)$$

$$P_{tI}' = \frac{P_t}{D^2 H_n^{3/2}} \frac{1}{m^{3/2}} \quad , \quad kW \qquad (2.19)$$

where n_I' - reduced speed,

Q_I' - reduced discharge,

P_{tI}' - reduced turbine power.

In formulae (2.17),(2.18),(2.19) 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 (2.15) may be used for

calculation of m .In approximate calculations m=1 is allowable.

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

From formulae (2.17),(2.18),(2.19) it follows that

$$n = \frac{n'_I \sqrt{H_n}}{D} \sqrt{m} \,, \tag{2.20}$$

$$Q = Q'_I \, D^2 \sqrt{H_n} \sqrt{m} \,, \tag{2.21}$$

$$P_t = P'_{tI} \, D^2 H^{3/2} m^{3/2} \tag{2.22}$$

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

3.CLASSIFICATION OF TURBINES

3.1 Classes of turbines

First let us consider classification of hydraulic turbines in terms of distinguishing features of hydraulic action.From the formula (2.4) and 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:1) impulse and 2)reaction .

In terms of hydraulic action 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:1)Pelton turbines,2)Turgo turbines,3)Banki-Mitchell turbines .

In reaction turbines when the water gives up its energy to the runner the change in potential and kinetic energy occurs.In this the change is characterized by 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 find application at hydropower plants including small hydropower stations: 1)Francis turbines,2)Kaplan turbines.

Thus five types of turbines find application at small hydropower plants.Each type of turbines is efficient within a certain range of operating parameters .

Each type of the considered turbines in its turn may have some variants differing from one another by relative dimensions of runner passages and other elements of the water passageway .

3.2 Reaction turbines

3.2.1 Francis turbines

Francis turbines find extensive application both at large and small hydropower plants . They are characterized by a large variety of designs and proportions of dimensions of the water passage . Fig.3 shows a scheme of the water passageway of the modern vertical shaft Francis turbine.The water from the penstock or fromthe 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 with 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 lowering down as the distance from the inlet connection piece goes up.The commonly used shape of the spiral case is circular as shown in Fig.3 . At places of smaller sectional areas their shape changes for oval .

From the spiral case the water flow enters the stay ring 2 (Fig.3) which comprises a fixed-vane system with similar or different-in-form vanes.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 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 is allowed to do without the stay ring .

The wicket gate assembly (3) located downstream of the stay ring represents a row of similar-in-form cylindrical gates 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 ajacent 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 increase (See formula 2.5) .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 . Fig.4 shows a meridional projecture of the water passage of one of the variants of the runner .The crown 1 of

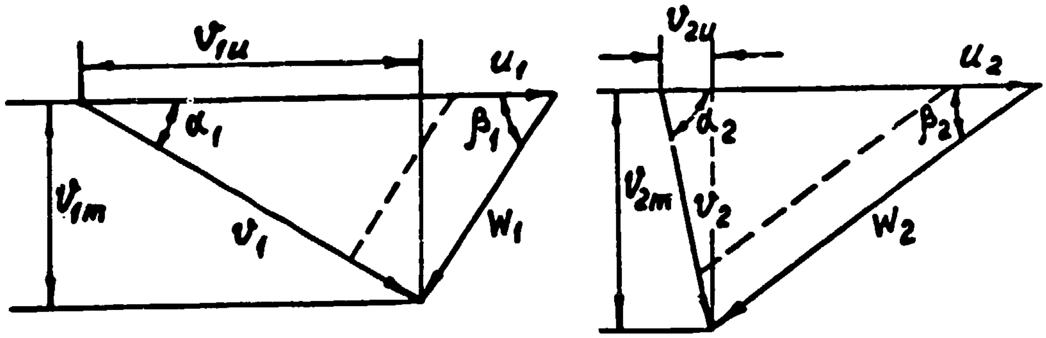

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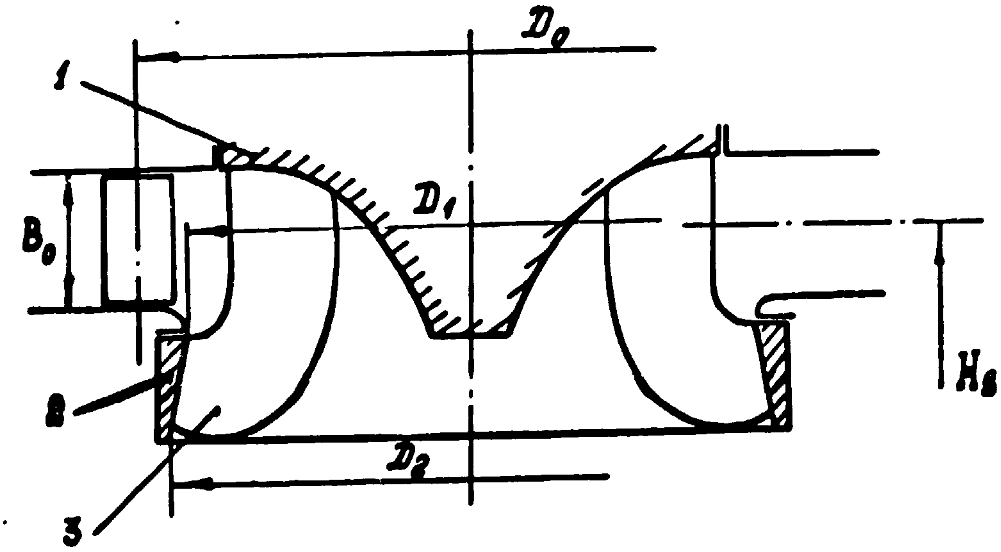
Fig.2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1
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Fig.4

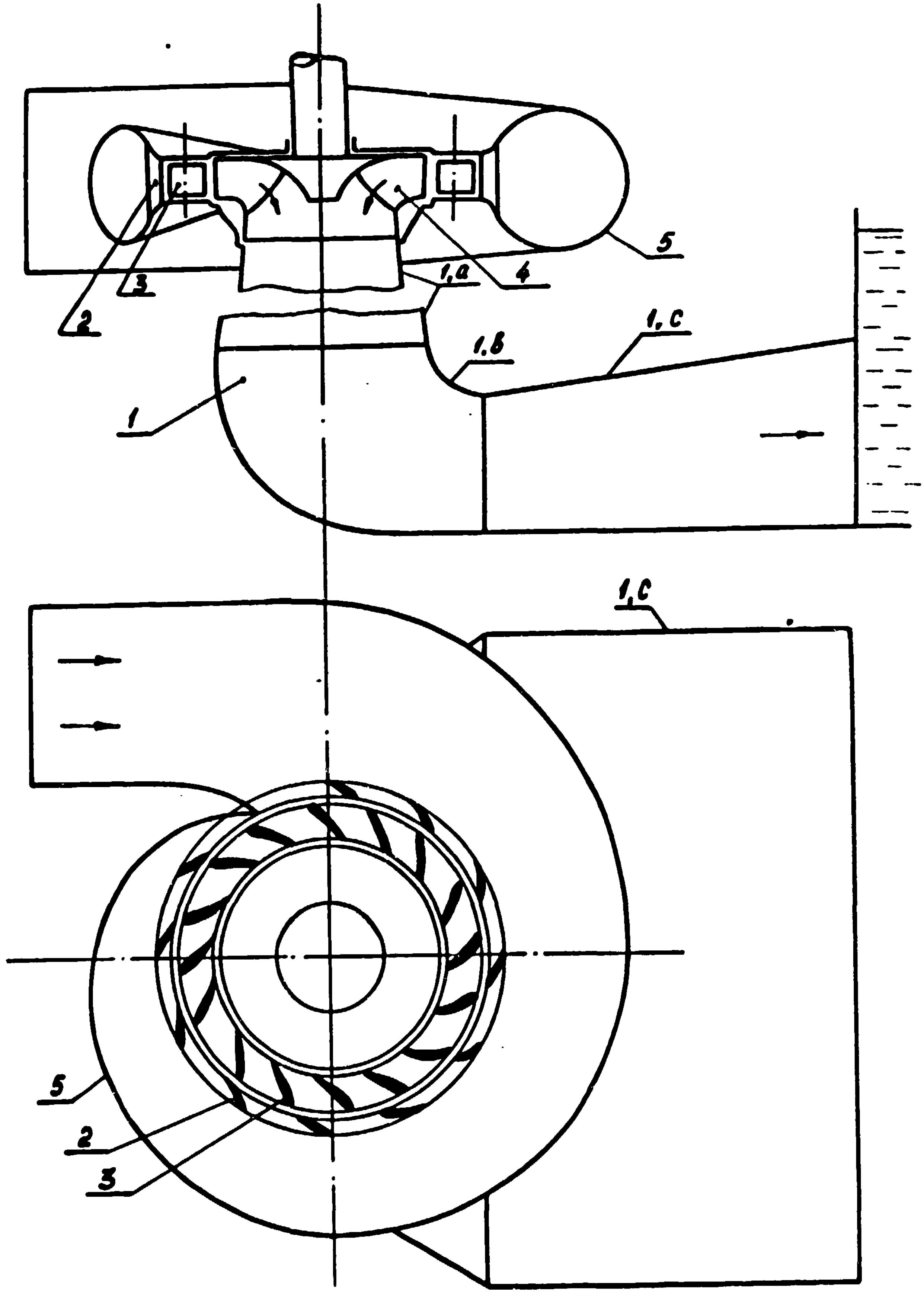


Fig.3

the runner connected to the turbine shaft and the shroud 2 are made integral by blades 3 arranged circumferentially and spaced evenly .Surfaces of 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 Fig.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 .

From the runner the water enters the outlet water passage which directs it to the outlet structures of the hydropower plant. Fig.3 illustrate 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 circlular 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 the given form may be also used in small hydraulic turbines.Besides in the latter case it is possible to use simplified straight conic draft tubes .

The draft tube allows to use 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 with the aim 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 grows up particularly in low-head hydraulic turbines .

Thus the described above 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 the turbine efficiency but at the same time these simplifications reduce the cost of installation .

Now let us consider some practicable and applied variant .

3.2.2 Turbine with single-gate control

There are suggestions associated with simplifications of turbine designs through the system of power and discharge control .

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 which has a pronounced effect on power .

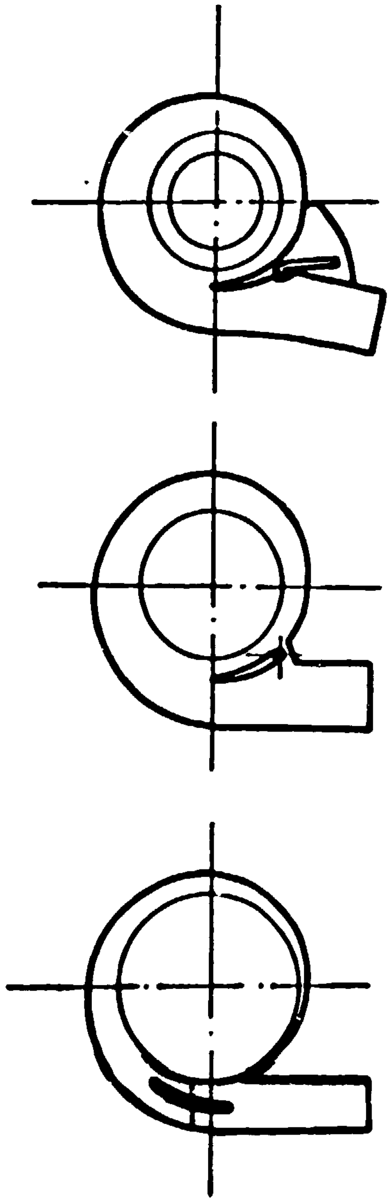
Fig.5 illustrates the scheme of the turbine with the single - -gate assembly .In this case the control is realized by one gate only the rotation of which changes the discharge and the spinning of the water flow at the runner inlet .

Fig.5,a shows the scheme of the single-gate control suggested by Reinfenstein.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 .

Fig.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 .Fig.5,c shows the variant with the unloaded gate .

3.2.3. Open flume turbines

For small capacity hydraulic turbines operating under low heads an open supply of the water to the turbine (Fig.6) finds application .In this case there is no spiral case at the inlet .



a)

б)

c)

Fig.5

It is replaced by an open rectangular flume with a free water level.The stay ring is also omitted.The wicket gate assembly is controllable .

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

The hydraulic efficiency of such turbines is lower .

Fig.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 .There is no straight diffuser at the outlet .

All these alterations compared with a classical design result in reduction of efficiency and reduction of the cost .Simplified designs find application in the turbines of small capacity .

An open flume without a spiral case may be found in horizontal-shaft turbines (Fig.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 is relatively low .

3.2.4 Horizontal-shaft turbines in drum

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

Fig. 8 illustrates the scheme of the turbine with a pressure tank to which the water is supplied through the penstock.From the

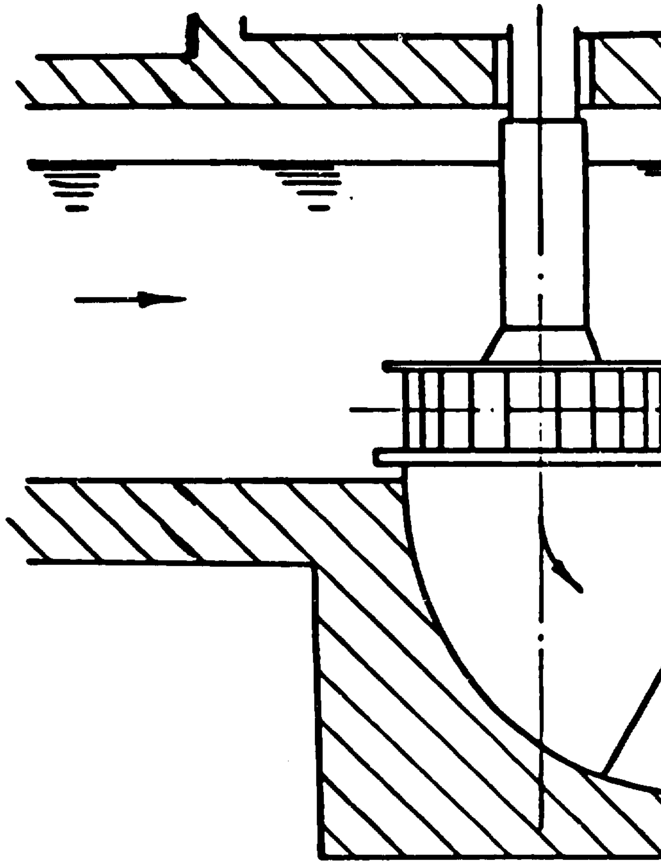
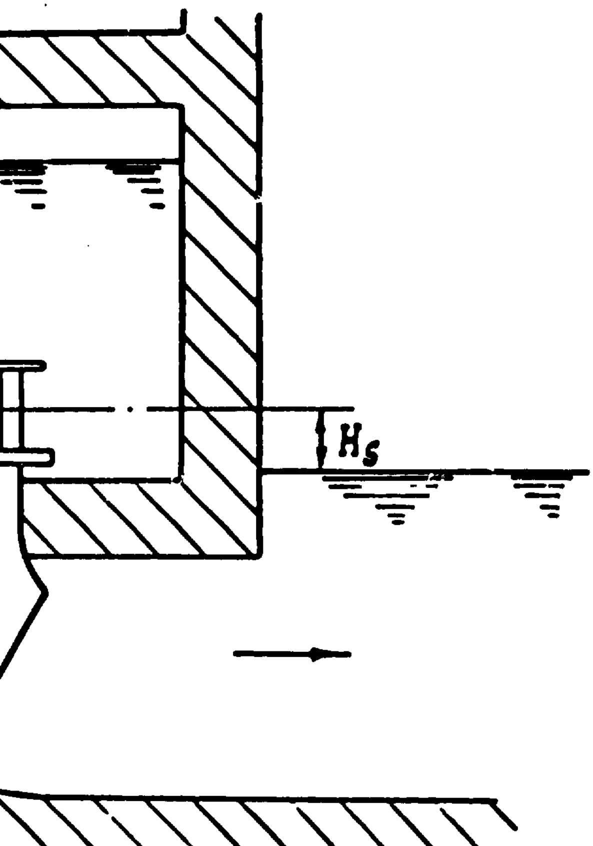


Fig.6


Hs

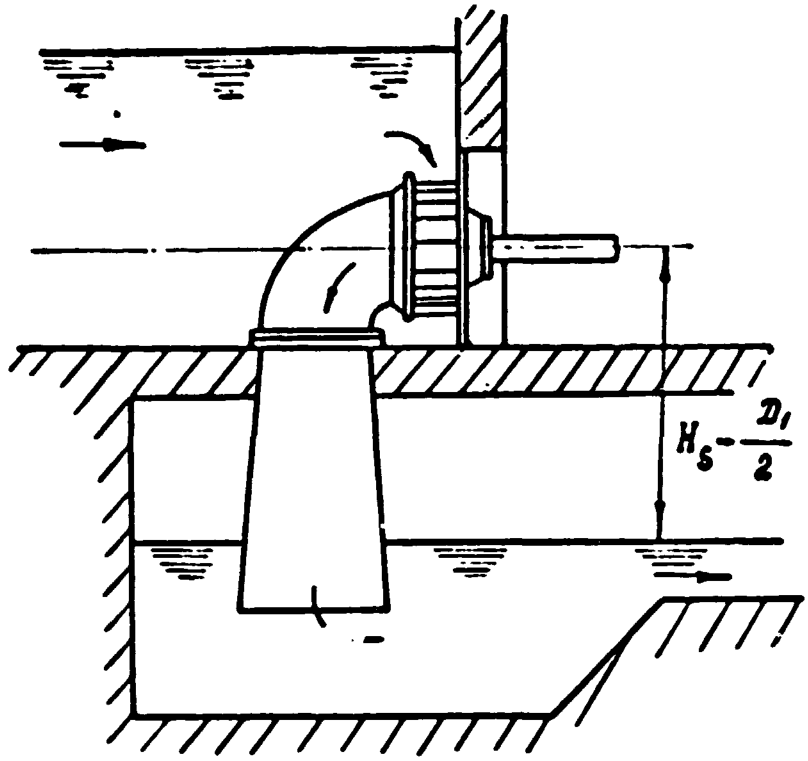


Fig.7

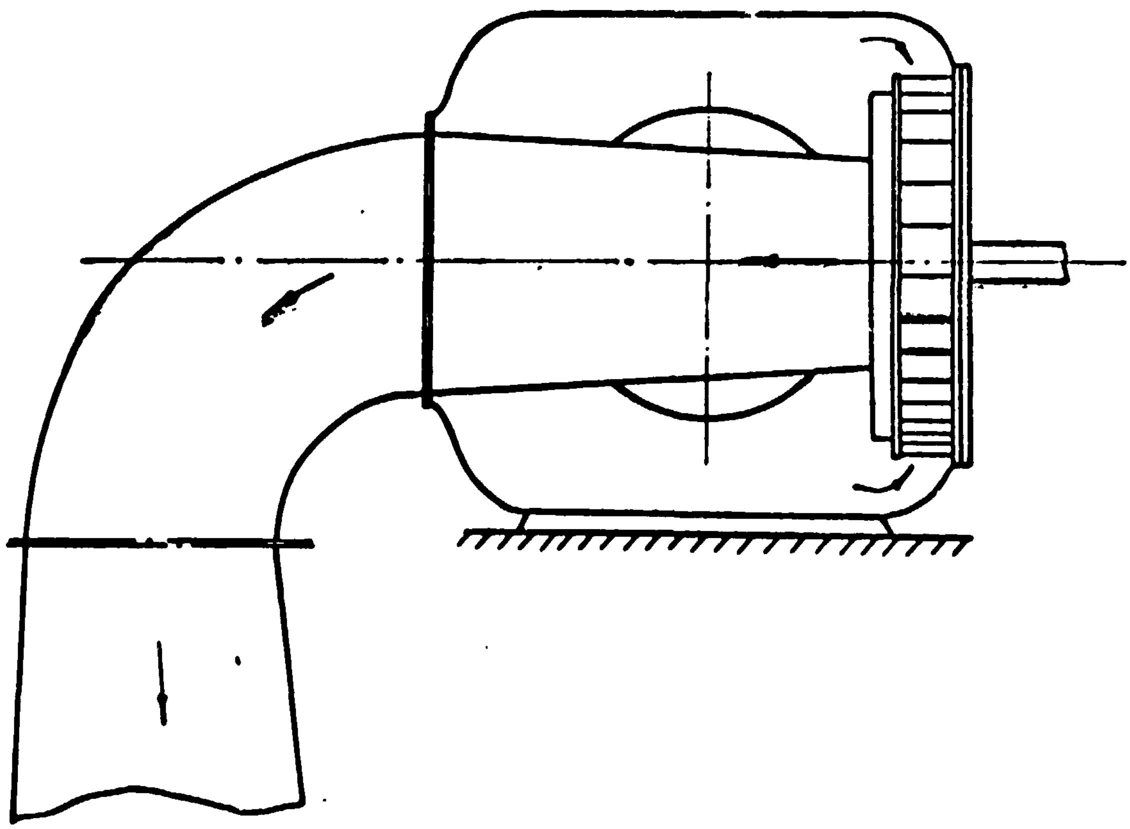


Fig.8

the tank the water enters the wicket gate assembly.The draft tube is arranged integral with the pressure tank .

Fig.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 .

3.2.5.Kaplan turbines .

Kaplan turbines find wide application at large and small hydropower plants.A classical arrangement of the water passage (Fig.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 .

Radial sections of the spiral case of the vertical-shaft turbine are of a trapezoidal shape . In 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 (Fig.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 close to cylindrical in form.

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

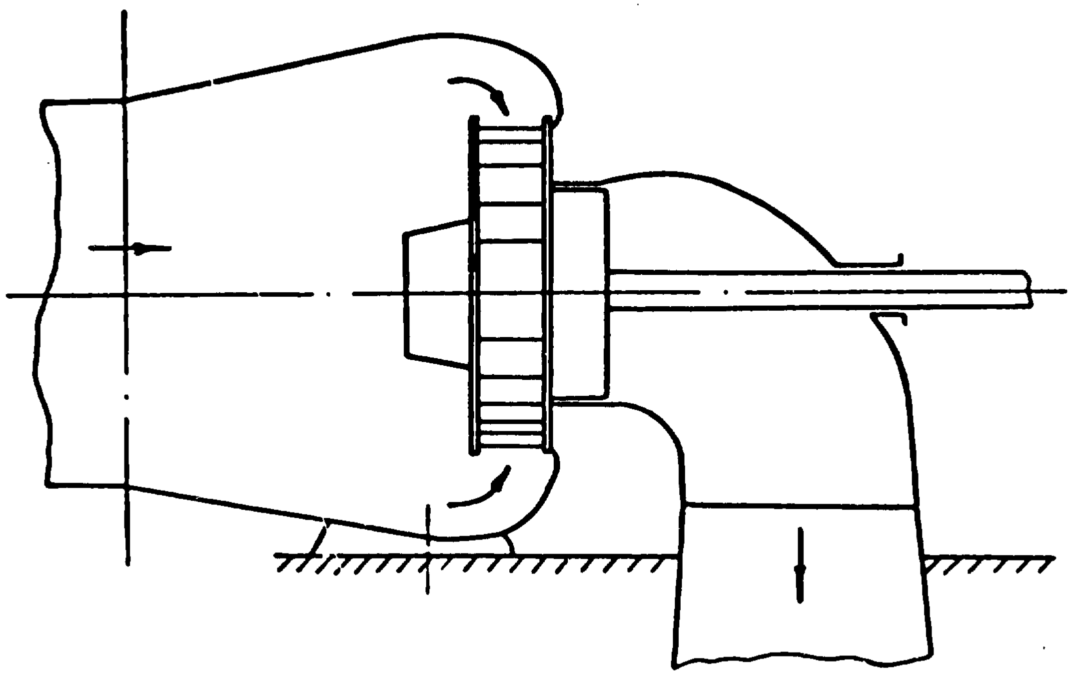


Fig.9

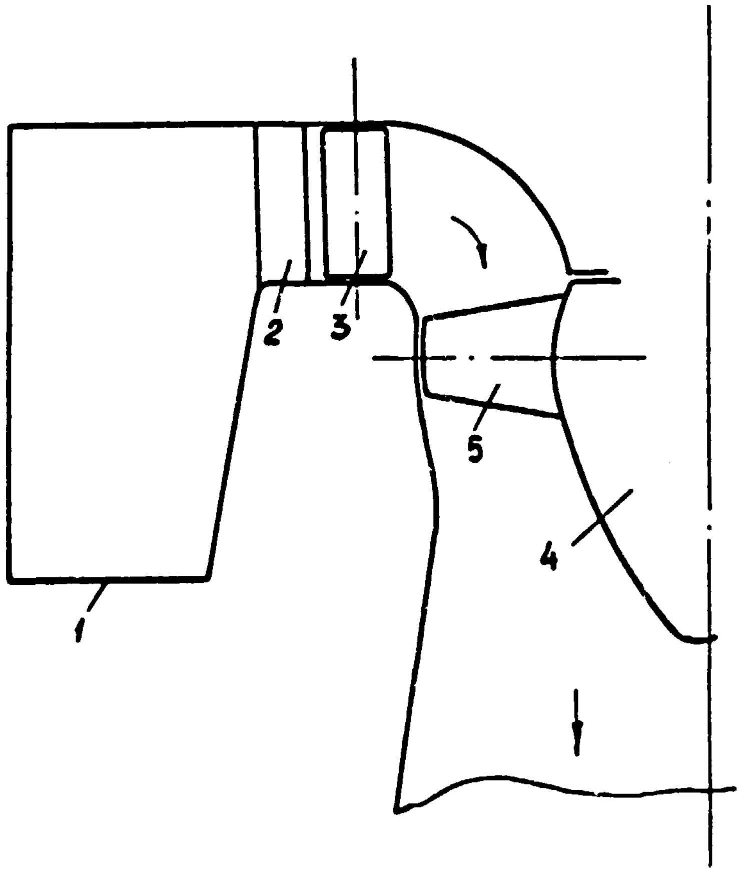


Fig.10

The main distinguishing characteristic of Kaplan turbines resides in the fact that the runner may be provided with adjustable blades.Axes of rotation are normal to the axis of the turbine .Usually the design of the turbine and the runner provides a means of rotation of all blades in syncronism during operation. Thus owing to simultaneous rotation of the wicket gates and blades of the runner it is possible to maintain the 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 worse 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 (Fig.11).Relatively short straight draft tubes are used . Some simplifications of the governor are feasible .

3.2.6.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.

Fig.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

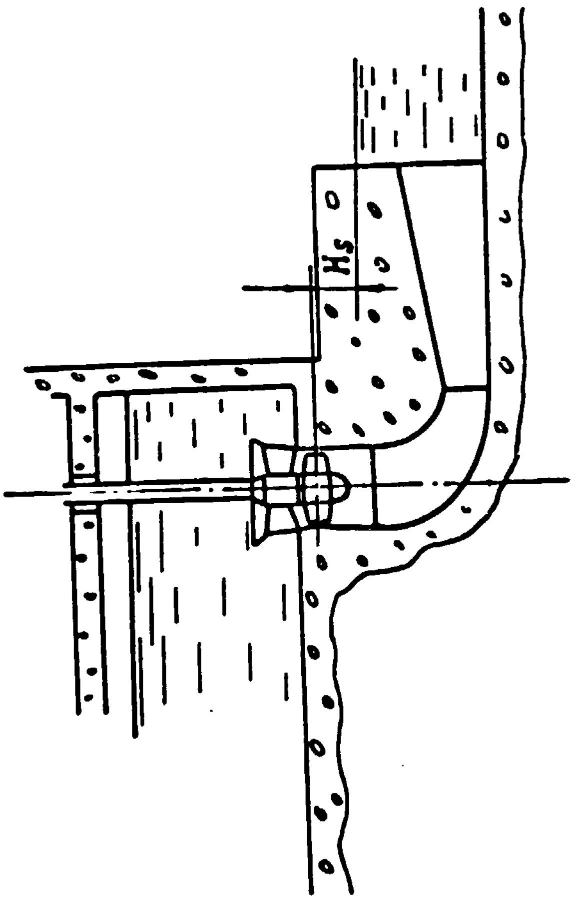


Fig. 11

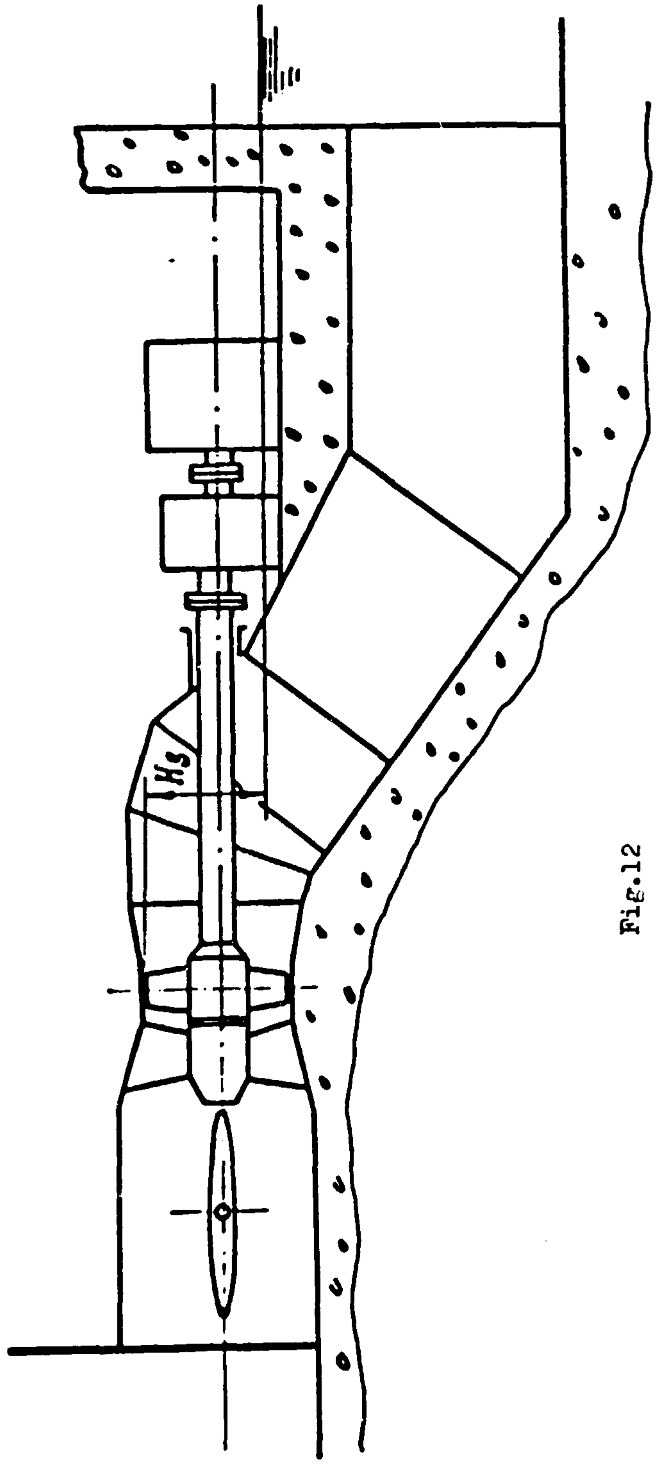


Fig.12

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.

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

3.3. 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 fills all the passages of the turbine completely .Under normal caviation-free conditions there are no free eater surfaces inside the turbine .

The hydraulic action in impulse turbines differs radically from that of reaction turbines.Here in inlet passages of the 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'll 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

$$v_j = \varphi \sqrt{2gH_n} \qquad (3.1)$$

where φ- coefficient of velocity equal to 0.98-0.99 for high quality nozzles.

Under the head H_n=500 m v_j =98 $msec^{-1}$ and under the head 1000 m v_j=138 $msec^{-1}$.

Thus the discharge

$$Q = v_j f_j = \varphi f_j \sqrt{2gH_n} \qquad (3.2)$$

Here 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

$$P_t = 9.81\, Q H_n \eta_t = 9.81 \sqrt{2g}\, \varphi f_j \eta_t H_n^{3/2}$$

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 .

3.3.1.Pelton turbines

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

Fig.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 realized 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 exessive 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 .

Fig.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 .

Fig.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 .

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 up 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 Fig.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 flowing of the water along curvilinear surfaces of rotating buckets velocity of the water changes both in magnitude and direction.Due to the change of momentum of water 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

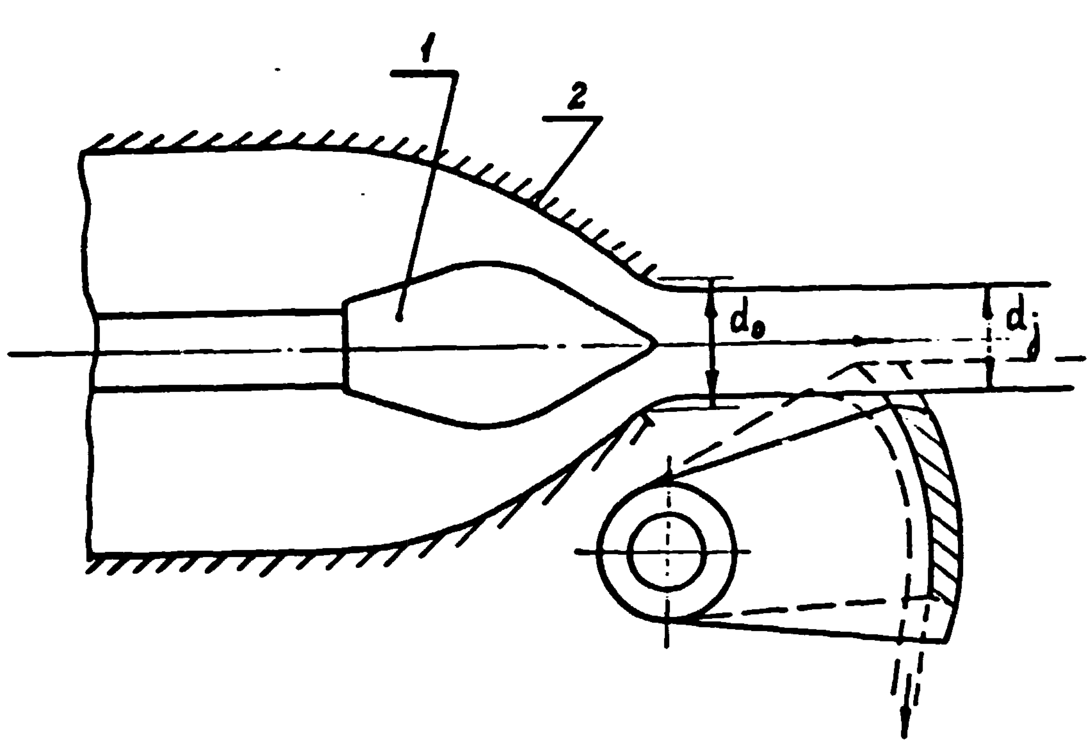


Fig.13

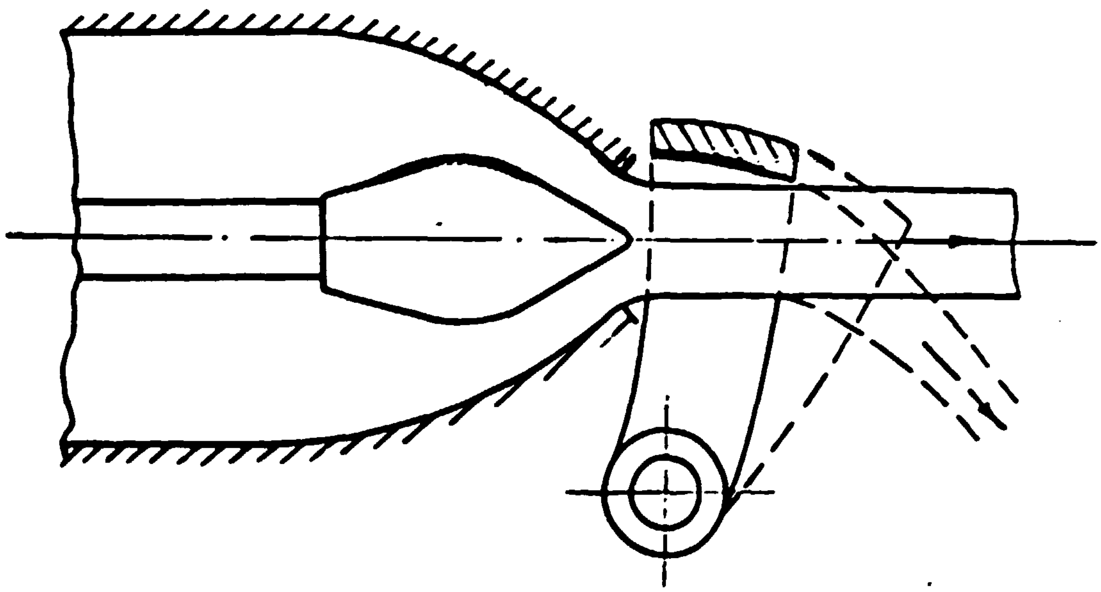


Fig.14

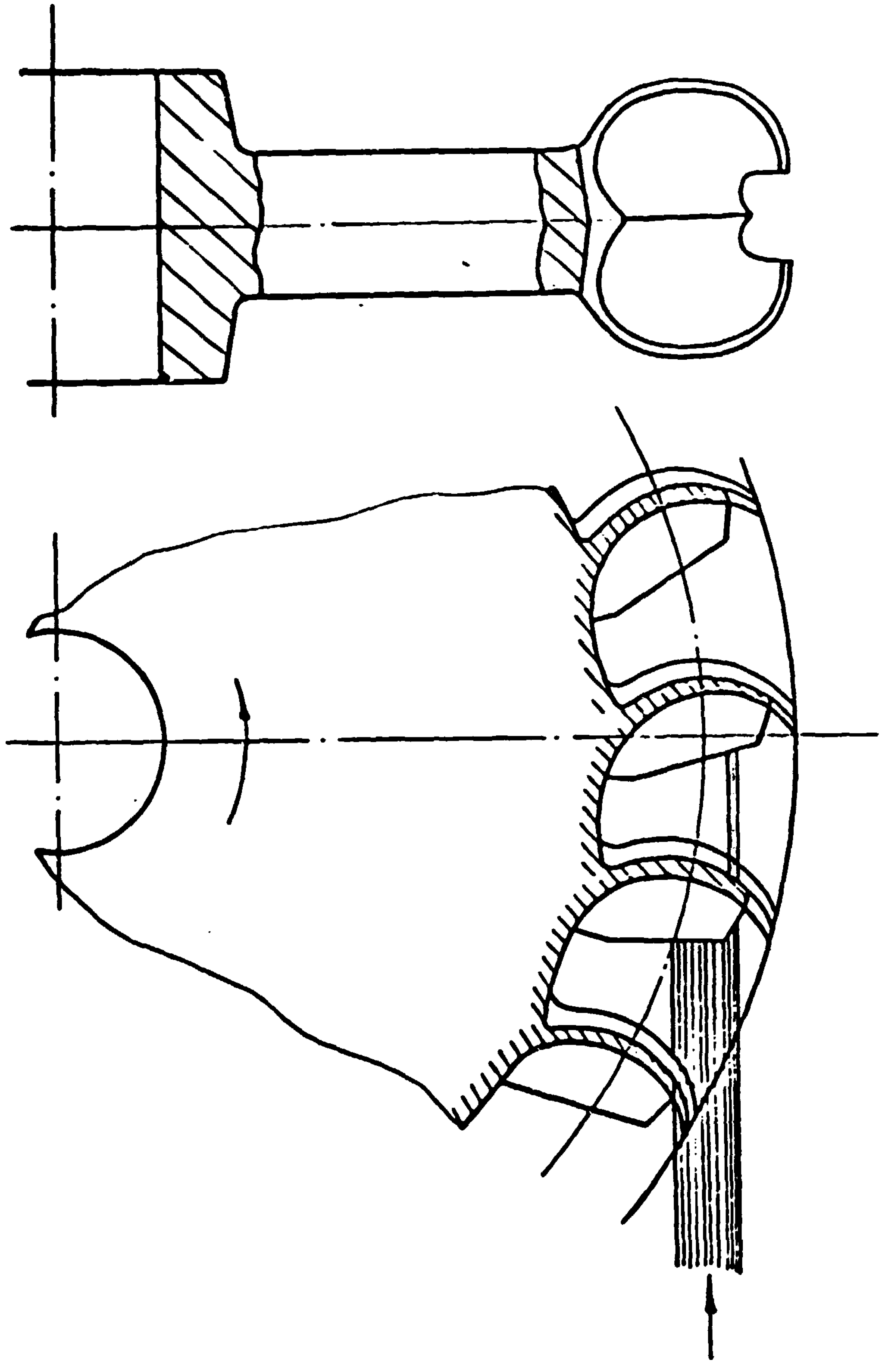


Fig.15

used completely . For this certain conditions with regard to dimensions of the runner ,dimensions of buckets and their number, rotational speed and velocity of the water in the jet must be satisfied.

The high efficiency requires careful and meticulous development of bucket geometry and accurate making .

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 a number of nozzles results,all other things being equal,in proportional raising of power.

3.3.2 Banki-Mitchell turbines

For the first time this idea was introduced by Banki and Mitchell who evolved the design independently.Later some modifications were introduced to the design e.g.by Ossberger Turbine Fabric Co.of Weissenburg,Federal Republic of Germany.

Fig.16 shows the design of 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 .

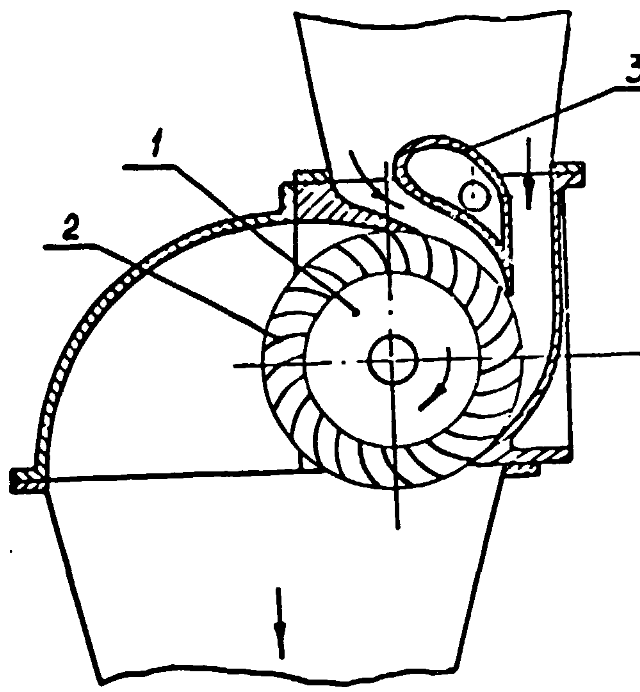


Fig.16

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

Among impulse turbines Banki-Mitchell turbines are characterized by 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 raising of losses and lowering of the efficiency. A change to a multi-nozzle design for small turbines is inexpedient because of higher costs and complicated maintenance .

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

3.3.3 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. Fig.17 shows a flow diagram of the design .

Similarly to the Pelton turbine a nozzle with an adjustable needle is used in the design of the Turgo turbine. Here the water supply to the runner is realized at a certain angle α_1 to

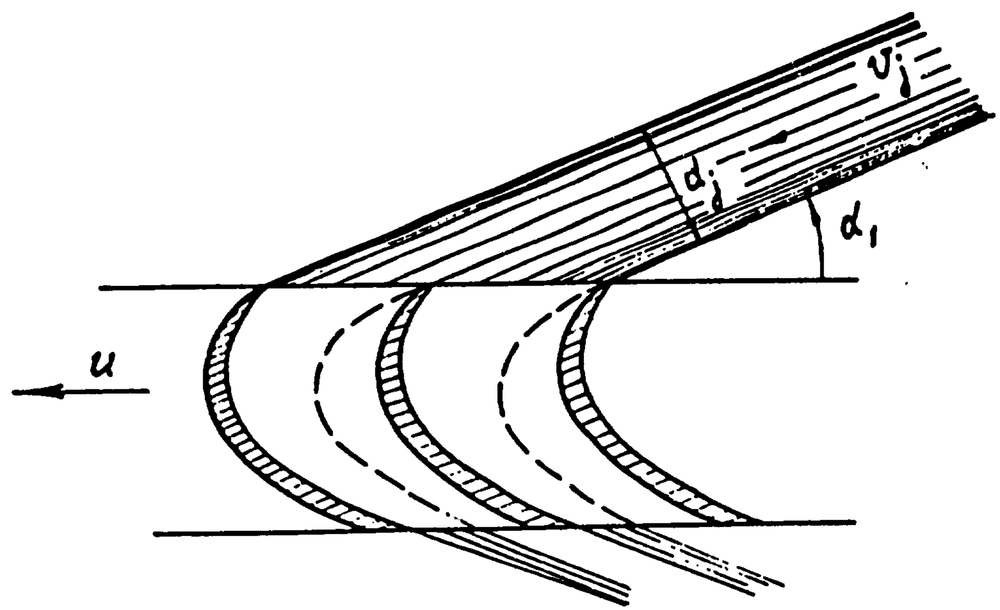


Fig.17

the plane of rotation of the runner .

The runner consists of blades fixed to outer and inner disks . The blades are of a complicated bucket-line shape the optimum dimensions of which are determined by laboratory studies .

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

Manufacturing methods of Turgo turbines are more simple as compared with Pelton turbines .

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

4.TURBINE CHARACTERISTICS

4.1 Power characteristics

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

$$P_t = 9.81 Q H_n \eta_t \tag{4.1}$$

then under a costant 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_o . Usually a_o 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 the change in the head with time is possible but these changes are relatively slow .

In its turn the efficiency η_t depends on operating conditions of the turbine .

The complicated interrelations between main operating parameters of the turbine may be established only by the results of model tests of the turbine in the laboratory.

The model of the turbine shall 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 :

1) Net head - H_n,
2) Discharge - Q ,
3) Rotational speed - n ,
4) 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

$$n'_I = \frac{nD}{\sqrt{H_n}} \cdot Q'_I = \frac{Q}{D^2\sqrt{H_n}} , \; P'_{tI} = \frac{P}{D^2 H_n^{3/2}} \qquad (4.2)$$

These formulae differ from formulae (2.17),(2.18),(2.19) by the fact that at 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 opening a_o of the wicket gate and different blade position φ of the Kaplan runner .

Fig.18 shows the hill diagram of the Francis turbine.Here the curves of constant opening of the wicket gate assembly a_o = const are plotted on the $n_1' - Q_1'$ coordinates and the curves η_h = const are plotted as well .

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.Fig.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 certain limit .With further increase in opening of the wicket gate a_o and in discharge Q_1' the power P_{t1}' decreases.This is because of a rapid decrease in efficiency.Fig.19 shows also η_t varying with Q_1' .

It is obvious that operation of the turbine at $Q_I' > Q_{I\,max}'$ is inefficient because it is not associated with a rise in power .Usually the results of model tests determine the

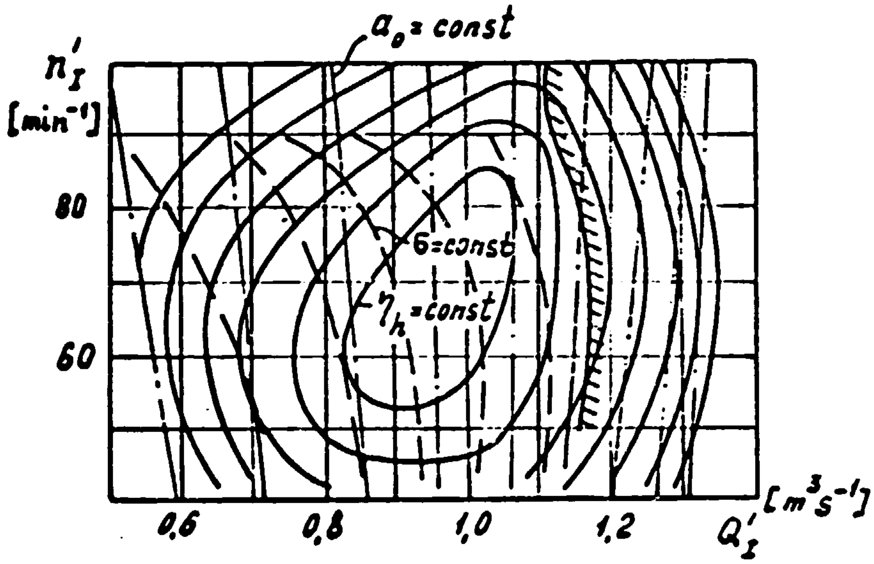


Fig.18

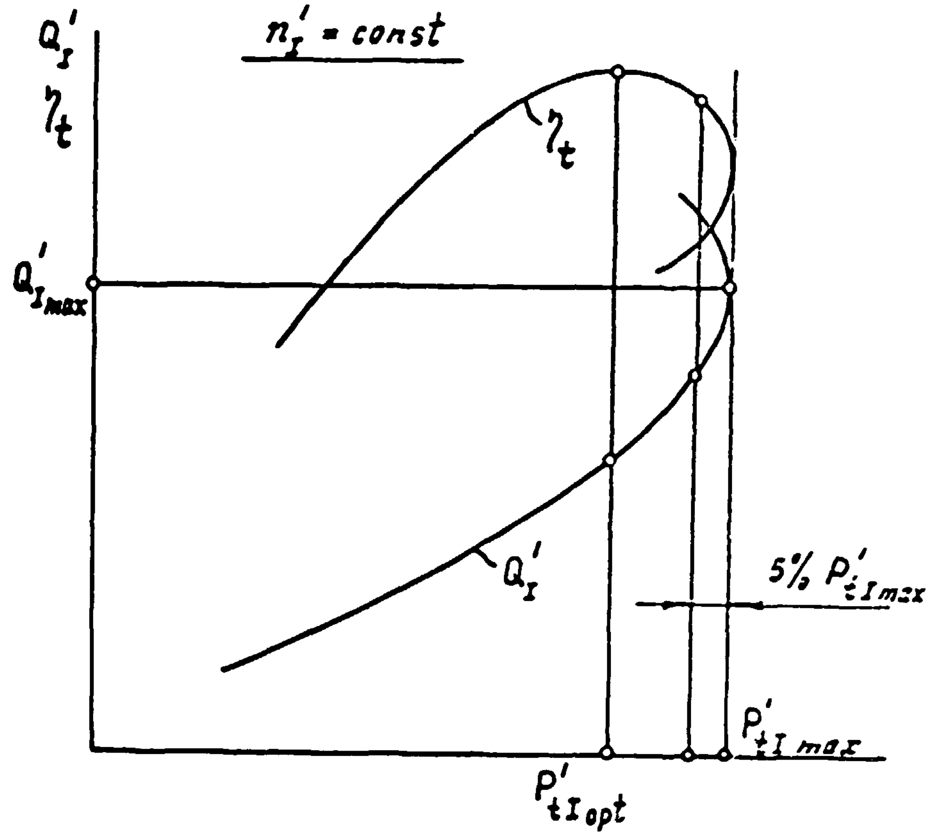


Fig.19

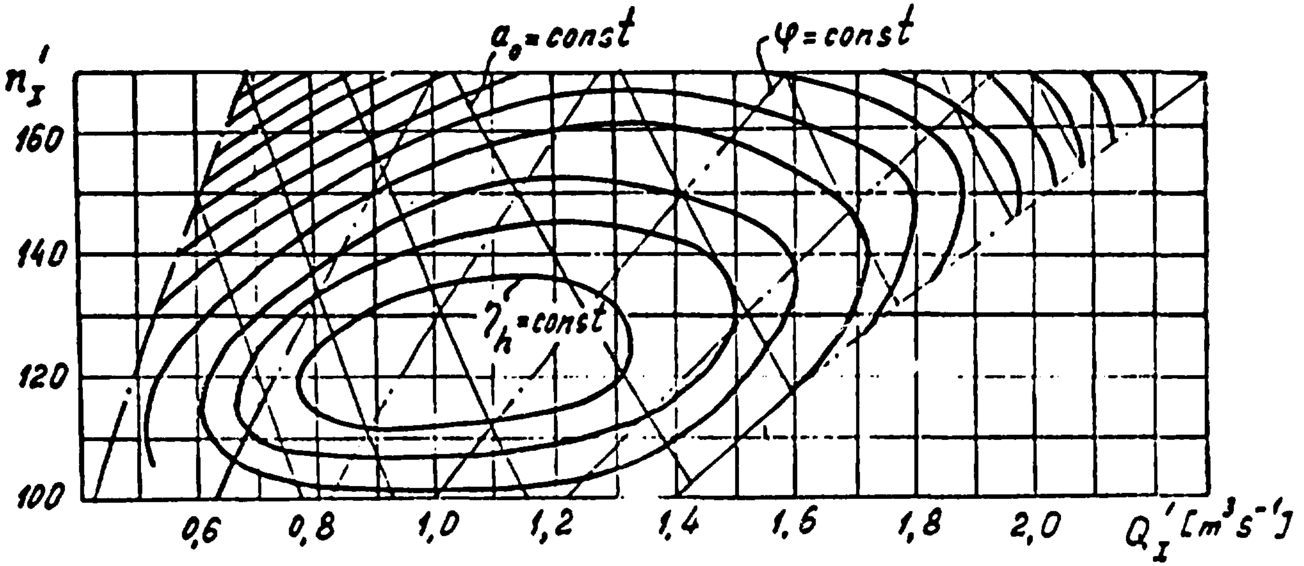
conditions in which the turbine has the maximum value of the reduced power .The limiting conditions with regard to power beyond which it is not recommended to operate the turbine are frequently assumed as $0.95\,Q'_{I\,max}$.In Fig. 18 there is a dashed line beyond which the turbine operation is inefficient within the range of high Q'_I.This is a power limit line . From the standpoint of the design this limit is realized by the limiting opening of the wicket gate assembly .

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

The propeller turbine is a variant of the Kaplan turbine with fixed blades of the runner .The propeller characteristics vary with the blade position φ .During opening of the runner blades the field of the characteristics covering the 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 .

Fig.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 Kaplan turbine demonsrate the optimum relationship between the blade



a₀=const
φ=const
η'ₕ=const
n'ᴵ
160
140
120
100
0,6
0,8
1,0
1,2
1,4
1,6
1,8
2,0
Q'ᴵ [m³s⁻¹]

Fig.20

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

In modern practice of use of Kaplan turbines at small hydro-power plants the design in which the wicket gates are fixed and the runner blades are adustable finds application as a vari-ant.Sometimes the turbines with such mode of control are called Tomman turbines.The region of high efficiencies of such turbines is wider than that of propeller turbines and is nar-rower than that of Kaplan turbines.

The simpliest design of the axial-flow turbine includes the fixed wicket gates and fixed runner blades.Power and dis-charge control under the 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 the optimum values cavitation factor σ rises up 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 opera-tion .

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 .

Now let us consider the Pelton turbine hill chart an example of which is shown in Fig. 21 .

First,let us pay attention to the fact that in Pelton turbines the optimum reduced speed $n_1' \approx 40$ min^{-1} .With variation of n_1' the efficiency reduces rapidly .

Second,with variation of the turbine discharge with the help 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' =const irrespective of the value n_1' .

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

Fig. 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 width of which is equal to 1/3 and 2/3 of the full width respectively .Each section is provided with a vane 3 driven independently (Fig.16) .One section may be closed for the flow

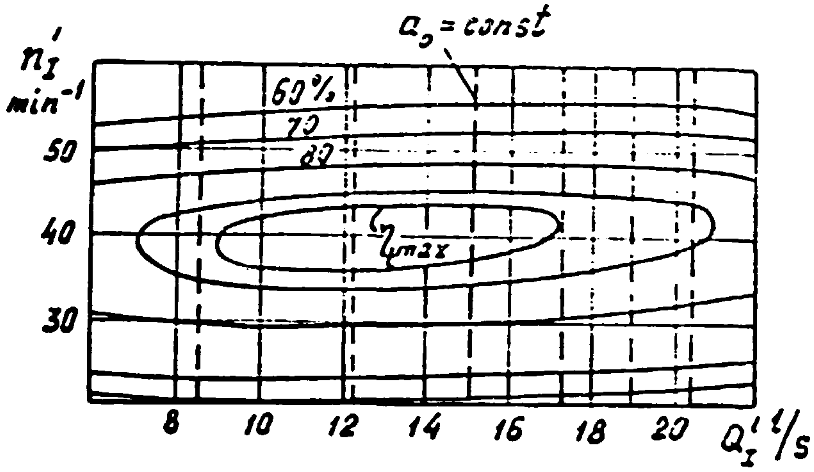

a_0 = const
n_I' min^{-1}
60%
70
80
η_{max}
50
40
30
8
10
12
14
16
18
20
Q_I' l/s

Fig.21

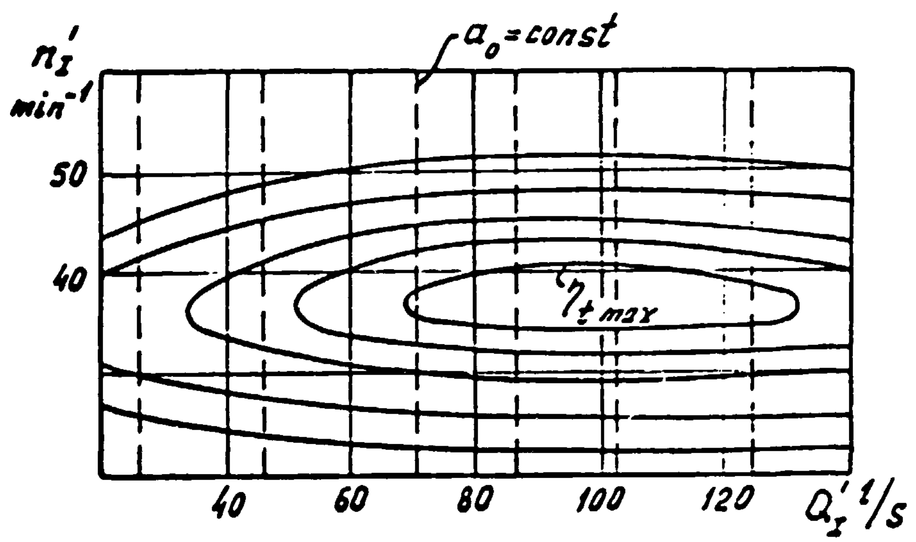

a_0=const
n_I' min^{-1}
$\eta_{t\,max}$
50
40
40
60
80
100
120
Q_I' l/s

Fig.22

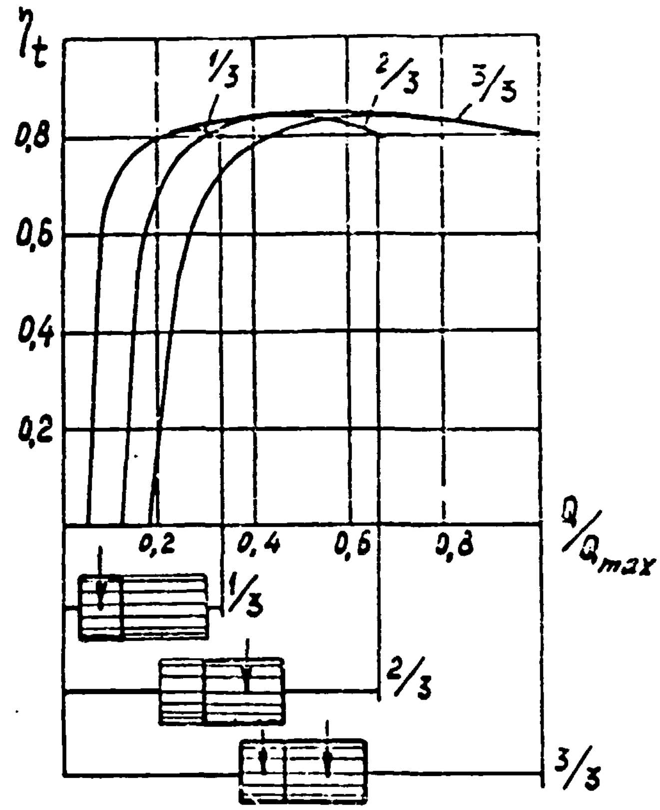


Fig.23

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 3/3,2/3,1/3 of the limiting value .Additional control is realized by rotation of the vane .

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

4.2. 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 parametrs.It is obvious that in the general case appreciable changes in the net head and discharge governed by the river flow and some 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 re-scurces .

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

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

Here the objectives to be pursued are the qualitative analysis only 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 variation 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)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.

Further the dependence of power characteristics of turbines of different types on the head may be traced.As it may be seen from these characteristics variation of the head entailing the change of n_1' at n=const produces an effect on the efficiency .

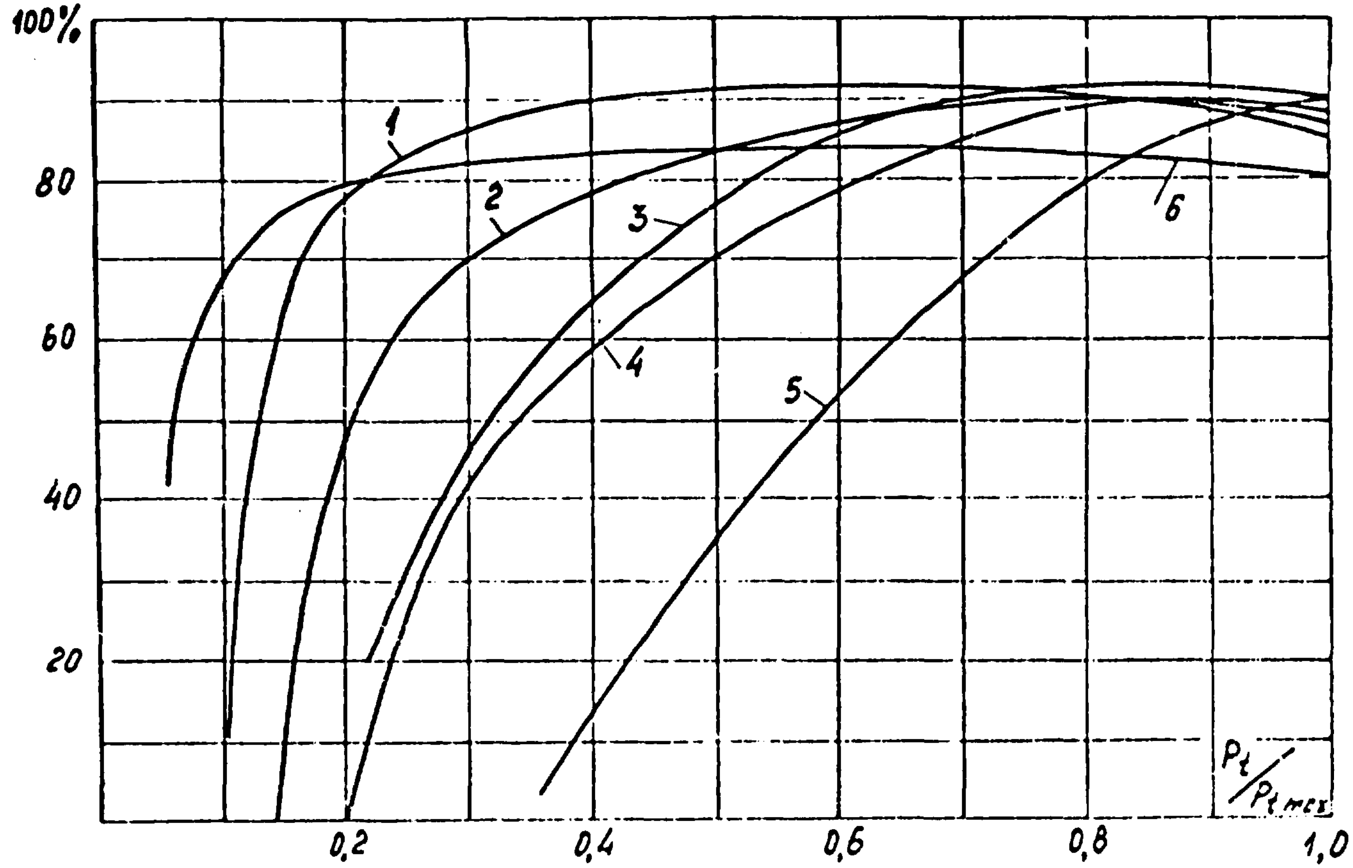


Fig.24

The analysis of characteristics demonstrates that with a increase in the head above the optimum value a decrease in n_1' takes place followed by a fall of the efficiency .The reduction of the head results in an increase in n_1' .Here the efficiency falls down drastically especially after a certain value observed within (0.4-0.7) H_{opt} for different turbines .

Kaplan turbines possess 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 the efficiency decreases rather rapidly with decline of the 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 into the permanent load makes it possible to use uncontrollable turbines with fixed gates and blades.

Control possibilities of Pelton,Turgo and Banki-Mitchell turbines are rather efficient.All these turbines maintain high values of efficiency over a wide range of loads .

4.3.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 the 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 are flown to the zone of higher pressure.Condensation in progress at that time is accompanied by instantaneous hightening of pressure and local temperature elevation .A nonstationary rapid process of formation and collapse of cavitation caverns is accompanied by high-frequency wavy effects resulted 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 the conditions shall be created under which cavitation would not take place in the flow and the failure of stream-lined elements and a decrease in efficiency should not be resulted therefrom .

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

4.4.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

$$\sigma = \frac{H_{sv}}{H_n} \qquad (4.3)$$

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

$$B - H_s - \sigma H_n > H_v \qquad (4.4)$$

where B - atmospheric pressure,

H_s - suction head which is counted off from the downstream water level to the horizontal plane conncted with a certain turbine element, $H_v = P_v / \rho g$

In axial-flow vertical-shaft hydraulic turbines the representative elevation of the turbine is usually the pivot of runner blades (Fig.12), in vertical-shaft Francis turbines it is a mid-plane of the wicket gate assembly (Fig.4) and in horizontal-shaft turbines it is the highest elevation of the runner blades (Fig.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 down atmospheric pressure $P_B = \rho g B$ to the level at which cavitation in the turbine is at its highest and the efficiency falls down .As a result of the test the critical value of turbine cavitation coefficient is calculated $$\sigma = \frac{B - H_s - H_v}{H_n}$$
by 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 (Fig.18) or they are constructed in the form of individual characteristics (Fig.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 (Fig.25).

4.5 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 shall 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

$$H_s < B - H_v - \sigma H_n \qquad (4.5)$$

The atmospheric 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.

The suction head H_s is selected at the stage of designing 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

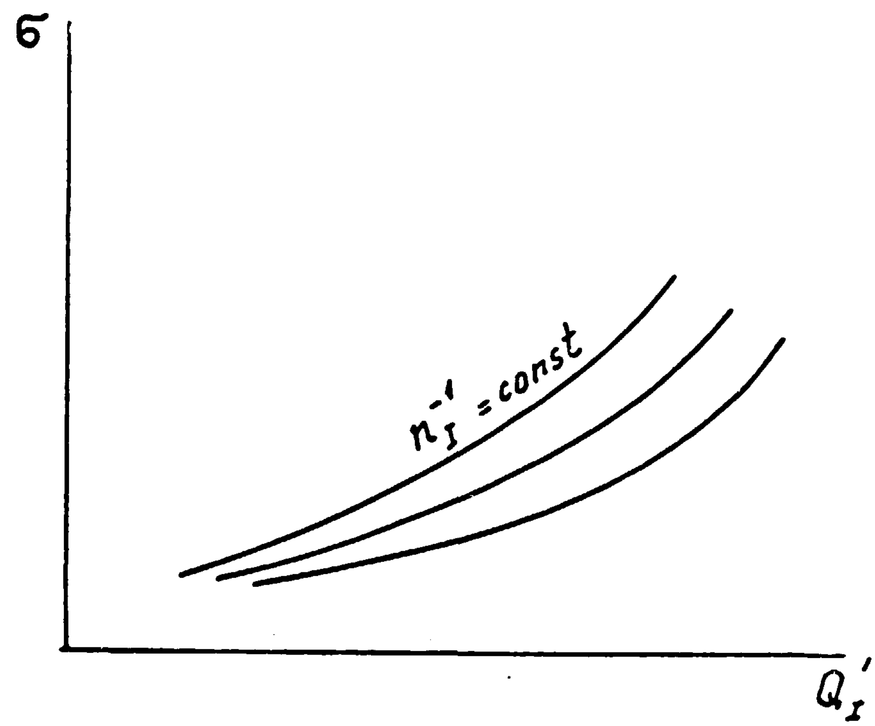


Fig.25

are 1-3 m.It is obvious that the particular value H_s depends on dimensions of the turbine as well .

The formula (4.5) shall be guided 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 the value σ of the turbine must be .

For turbines operating under low heads the limiting values σ may reach 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 proportions of 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 results in the necessity to locate the turbine with high negative values H_s i.e results in a deep setting of the plant and an increase in the construction cost.

The suction head is determined by the following formula:

$$H_s = 10 - \frac{\nabla}{900} - K_s \sigma H_n \qquad (4.6)$$

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

$$\sigma_{st} = \frac{B - H_v - H_s}{H_n} \qquad (4.7)$$

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 that 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 the pitting in the runner.

On the basis of the abovementioned the safety factor K_s the value of which depends on the series and type of the turbine and power is introduced.On the basis of existing structures it may be determined that

$$K_s = 1.1\text{-}2.0$$

It is recommended to raise the value K_s up 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 bring about either a deep setting of the powerhouse or the use of low-speed turbines and enlargement of their dimensions.

5. STANDARDIZATION OF TURBINES FOR SMALL HYDRO APPLICATION

5.I. Objectives of Standardization

The construction of small hydro power plants involves high specific costs. Intensive application of unified mass-produced elements both in civil works and electromechanical equipment would make small hydro power plant more competitive.

Recent renewal of interest in developing hydropotential of small stream flows at the same time calls for upgrading their economic viability both in construction and operation.

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

Let'us regard the units with installed capacity of $50\ kW \leq P_t \leq 10000\ kW$ to be the units for small hydroapplication.

Depending on the site conditions, the small hydro power plants can develop the heads in the range of $1\,m < H_n < 1000\,m$ which will be feasible for practical application.

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

$$P_t = 9.81\, H_n\, Q\, \eta_t \ \ kW$$

Fig. 26 gives this zone in logarithmic coordinates.

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

taking an averaged value of η_t, we are obtaining the linear

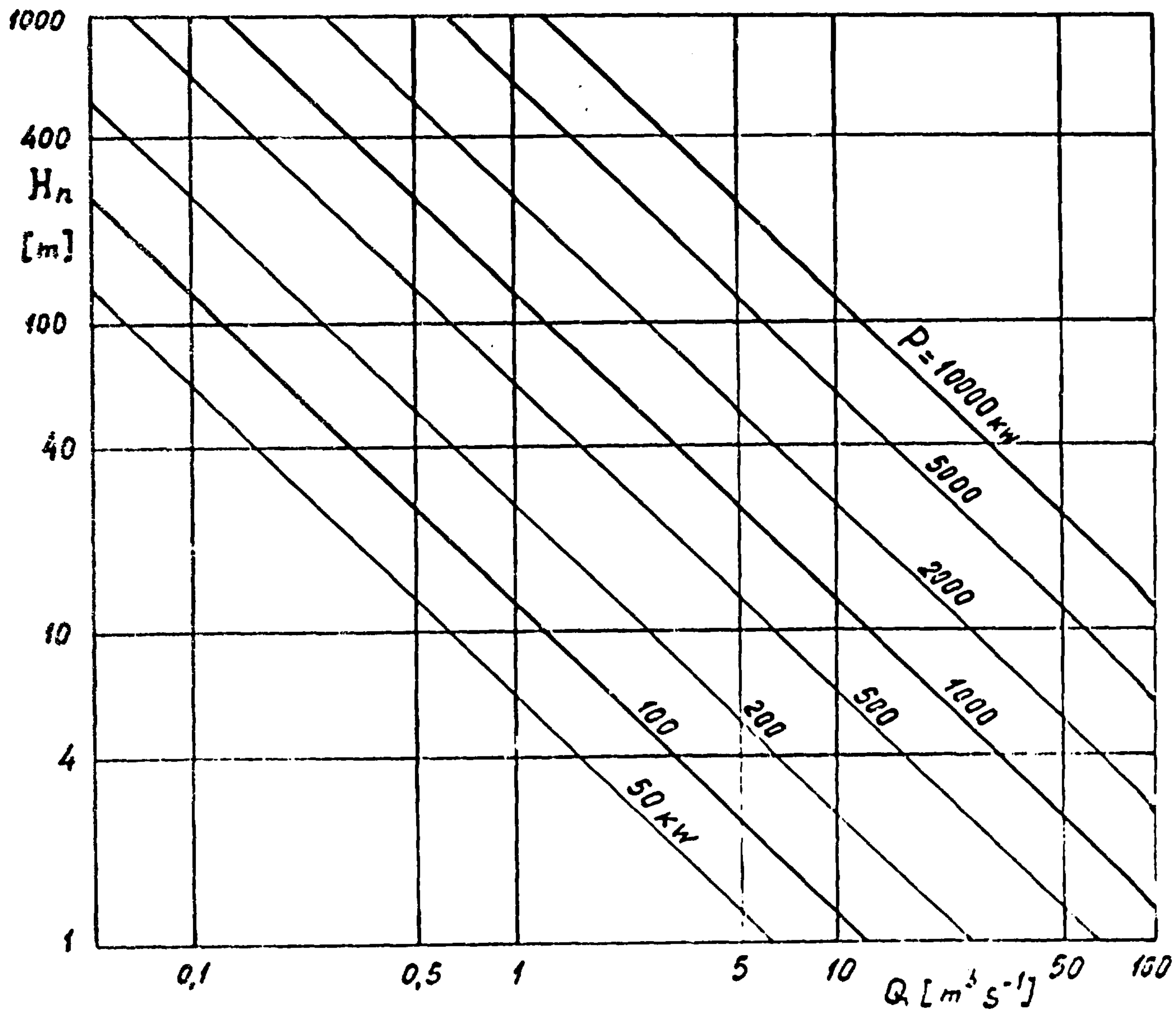

1000
400
H_n
[m]
100
40
10
4
1
0,1
0,5
1
5
10
50
100
Q [m³ s⁻¹]
P=10000 kW
5000
2000
1000
500
200
100
50 kW

Fig.26

relationship between H_n and Q for P_t = const. in logorithmic coordinates.

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

Finding a solution to the problem of turbine standardization will allow development of type-design for turbines, unification of structural elements as wellas change over from custom-made approach to series production. All this would enable the cost of equipment and delivery time to be reduced.

5.2. Line up of Series of Reaction Turbines

One of the major problems in standardization of small hydro turbines is to establish the practicable line up of turbine series. The experience gained in the hydraulic turbine egineering indicates that the field for potential application of reaction turbines can be covered by two types- I) Francis turbine, 2) axial flow turbines. The axial flow turbines are used in the low-head zone. Francis turbines find application in the high-head zone.

The turbine characteristics including parameters n'_1, Q'_1, 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 Francis turbine, the diameter D_I(Fig.4) is fixed.This dimension represents the maximum diameter taken at the inlet edges of the runner blades. Cases are known when the characteristic dimension is the diameter D_I taken at the runner crown or the diameter D_2 taken at the shroud (Fig.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.

Increase in the number of series and sizes of the turbines covered by the standard, allows selection of the effective turbine practically for 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.

Let'us dwell on some general concepts that should serve as guidelines in tackling the problem.

As mentioned above, each series of turbines has a limited field of application with regard to the head to provide cavitation-free performance.

Deep setting is not economically justified for the 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:

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

(Fig.27).

The minimum value of cavitation coefficient sigma σ secured for the high-head Francis turbines, is equal to about 0.03 on the 5% output margin line and at the optimum values of n_1'. If the cavitation safety factor is assumed to be K_s = I.I5 - I.20, than 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 margine will result in a deeper setting of the hydroplant and $H_s = -7.5$. But such a solutions is not practicable.

Let'us illustrate one of the possible alternatives for selection of the number of series and the field of application for each series (Fig.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 $\sigma = 0.03$ and safety factor K_s = I.2, we will secure the highest head series for the head range II5-340 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 I.5 i.e.

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

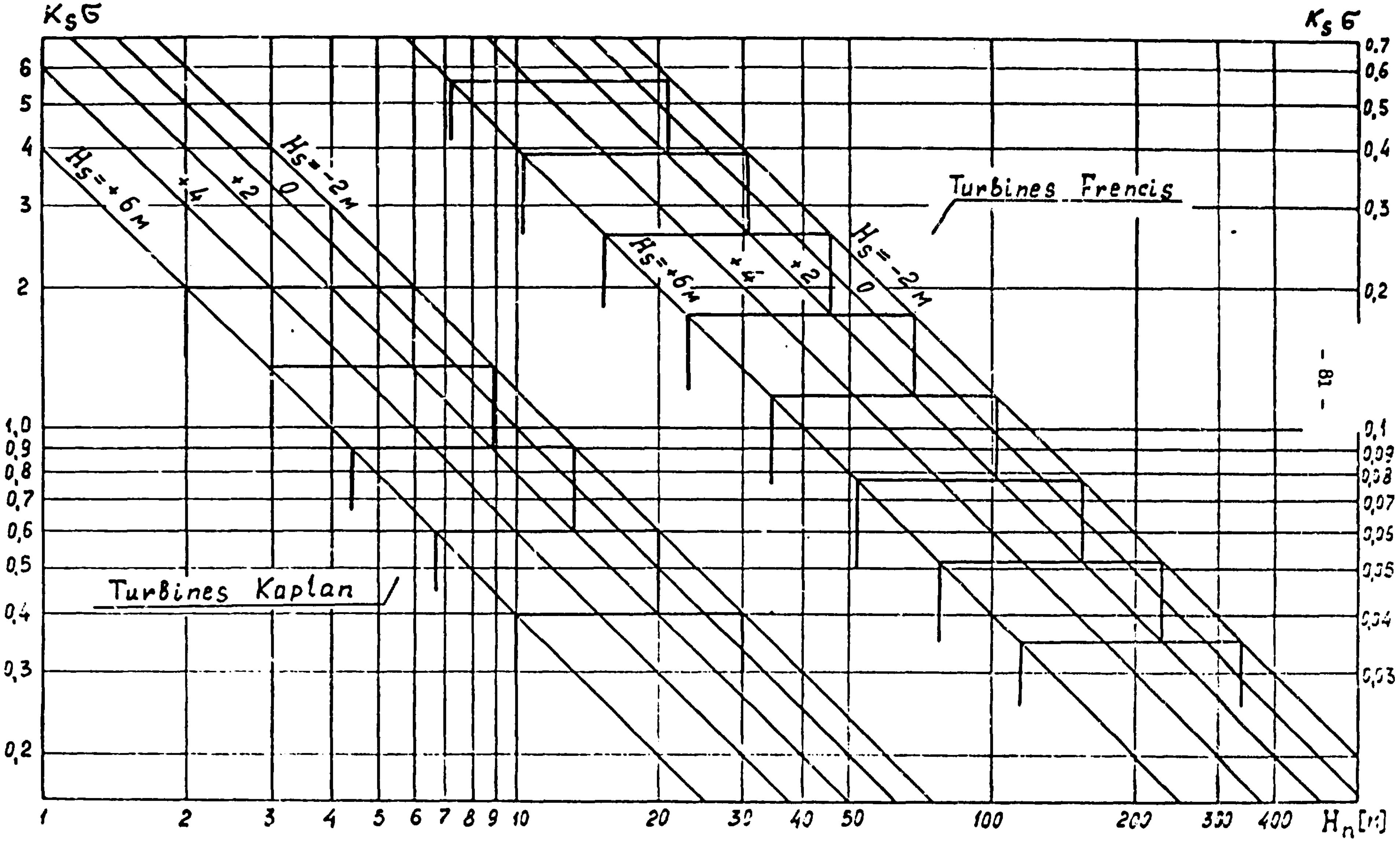

Ks σ
Ks σ
Hs = +6 м
+4
+2
0
Hs = -2 м
Turbines Frencis
Turbines Kaplan
6
5
4
3
2
1,0
0,9
0,8
0,7
0,6
0,5
0,4
0,3
0,2
0,7
0,6
0,5
0,4
0,3
0,2
0,1
0,09
0,08
0,07
0,06
0,05
0,04
0,03
1
2
3
4
5
6
7
8
9
10
20
30
40
50
100
200
300
400
Hn [м]

Fig.27

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

Series	F8	F7	F6	F5	F4	F3	F2	FI
$H_n max / H_n min$	$\frac{20}{7}$	$\frac{30}{IO}$	$\frac{45}{I5}$	$\frac{70}{25}$	$\frac{IOO}{35}$	$\frac{I50}{50}$	$\frac{260}{80}$	$\frac{340}{II5}$
σ max	0.47	0.32	0.22	0.I45	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

Series	K5	K4	K3	K2	KI
$H_n max / H_n min$	6/2	9/3	13/4.5	20/7	30/IO
σ max	I.7	I.I2	0.75	0.5	0.33

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

When assigning the number of series and specific speed for each series other conditions are also taken into consideration. A tangible role is played also by the practical experience of the company working out the standard. The large-size turbine manufacturers have accumulated considerable experience in development of highly efficient turbines for large-scale installation, which meet the different operational 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.

5.3. 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_I of the runner.

Let us dwell on some general aspects related to the selection of a normal line up of the turbine dimensions.

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% output margin line of the hill chart (Fig.I8). Then according to formula (2.I9) and taking into consideration that

$$P'_{tI} = 9.81\, Q'_I \eta_t$$

the diameter D_I of the runner can be calculated. It will be the minimum size of the runner which will provide the specified power output (curve I, Fig.28). Should a bigger size of the runner be taken for the same power output the rated point at the hill chart would shift to the zone of lower values of Q'_I.

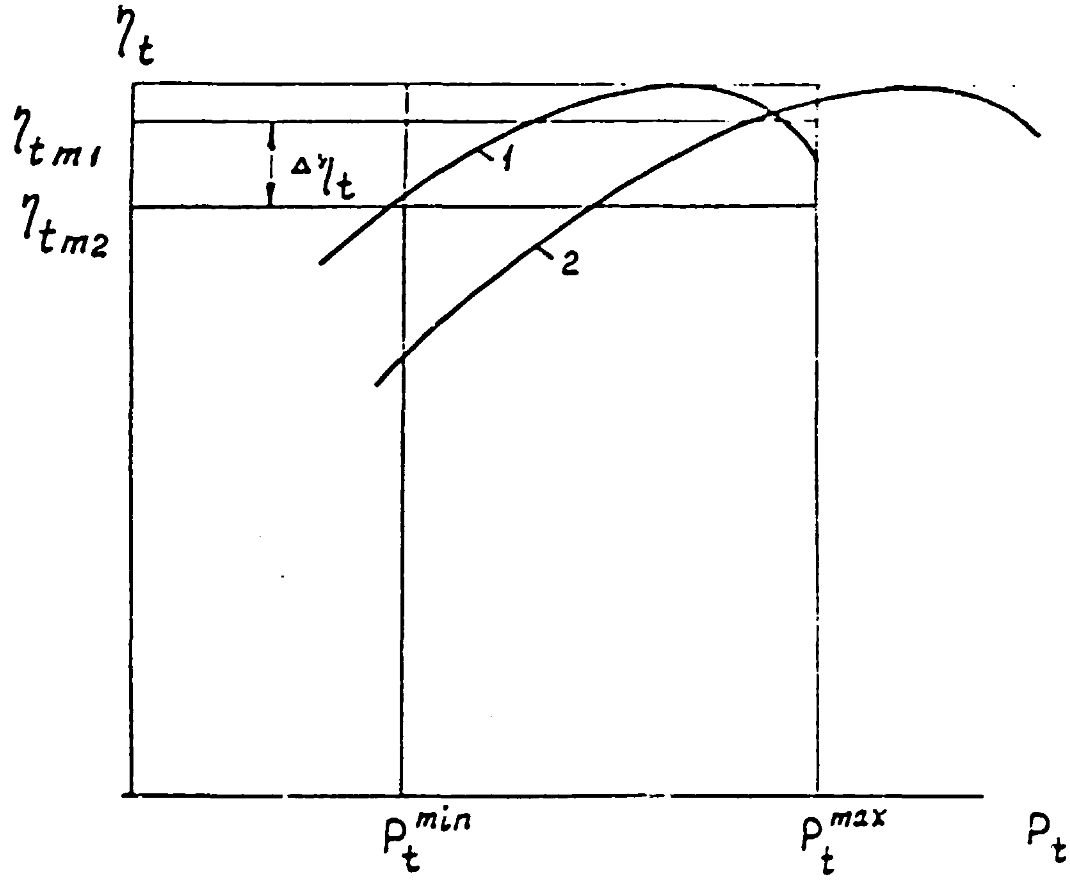


Fig.28

Given in Fig. 28 are 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. A heavy power drop decreases the efficiency value. As seen from Fig.28 the average value of efficiency $\eta_{tm1} > \eta_{tm2}$ 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 $\Delta\eta_t$ (Fig.28).

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

The following conclusions can be drawnup basing on the current experience of the turbine engineering -

I. Ratio $K_D = D_{n+1}/D_n$

lies within I.05-I.2 (where D_{n+I} and D_n are diameters of the runners of two adjacent turbines)

2. For the values of K_D upto I.07 adequate close correspondence of the designed turbine with its optimum operational conditions is provided

3. the lower value of K_D should be taken for larger size turbines.

4. K_D can be raised to I.I2-I.I5 for very

small turbines.

5. 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_D^2 i.e.

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

It means that the maximum power varies by I.I - I.44 times.

5.4. 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 basing on the development of a standard line up of turbine series and diameters.

Let us consider some general concepts which serve as the basis for the required plotting.

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'_{IC} as the design one. For the Francis turbine, it is practical to take the design value Q'_{IC} lying on the 5% output margin line or close to this zone. For further computations we assume that the values n'_{IC} and Q'_{IC} are known for the given series.

Neglecting the difference in efficiency of the model and life-size small hydroturbine, from (2.I7) and (2.I8) we have

$$n'_I = \frac{n D_1}{\sqrt{H_n}}, \qquad Q'_I = \frac{Q}{D_1^2 \sqrt{H_n}},$$

from this it follows that

$$\log H_n = -2\log\left(Q'_{Ic} D_I^2\right) + 2\log Q$$

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

It is seen from these equations that for D_I = const, the relation $H_n(Q)$ is of a linear nature in logarithmic coordinates. The lines D_I = const form a family of parallel lines inclined at an angle of $\alpha = \operatorname{arctg} 2 = 63.43°$ to the axis $\log Q$.

Relations $H_n(Q)$ for n = const are also of a linear nature in logarithmic coordinates. The lines n = const form a family of parallel lines inclined at an angle of $\beta = \operatorname{arctg} \frac{2}{3} = 33.69°$ to the axis log Q.

In Fig.29 is shown, as an example, the field of application for one series of Francis turbines. The model characteristic with $n'_{Ic} = 68\ \text{min}^{-I}$ $Q'_{Ic} = 0.75\ \text{m}^3\ \text{s}^{-I}$ $\sigma = 0.07$ in the design mode of operation is taken as the input data. It shows also the lines of the continuous turbine output.

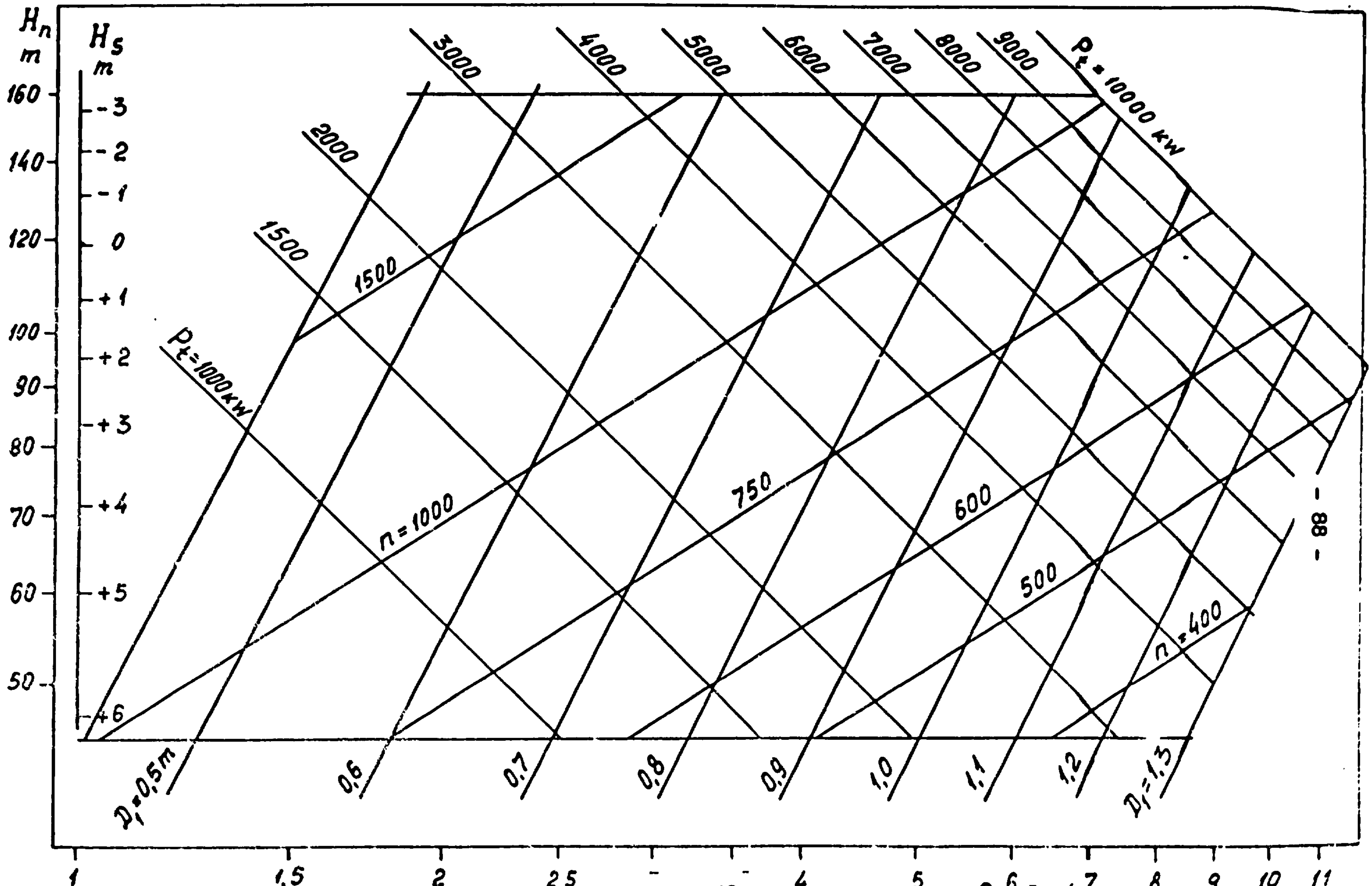
As $\log H_n = \log\frac{P_t}{9.81\eta_t} - \log Q$,

the lines of P_t = const are inclined to the axis $\log Q$ at an angle of $-45°$ in logarithmic coordinates.

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 I000-I0000 kW and runner diameters D_I = 0.45-I.3 m.

Preliminary selection of the basic turbine operational parameters can be made basing on the nomograph Fig.29. For example, the diameter D_I, speed n and suction head H_s can be



H_n m
H_s m
160
140
120
100
90
80
70
60
50
-3
-2
-1
0
+1
+2
+3
+4
+5
+6
P_t = 1000 kW
1500
2000
3000
4000
5000
6000
7000
8000
9000
P_t = 10000 kW
1500
n = 1000
750
600
500
n = 400
D_1 = 0,5 m
0,6
0,7
0,8
0,9
1,0
1,1
1,2
D_1 = 1,3
1
1,5
2
2,5
4
5
6
7
8
9
10
11
Q $m^3 s^{-1}$

Fig.29

determined from the specified head and power. But the values D_I, n, H_s obtained by such a way may turn out to be unacceptable.

The runner diameter may not correspond to the values which have been covered by the standardized line up. The rotative speed may differ from the synchronous one. When adjusting the values D_I and n obtained from the nomograph, to the nearest recommended magnitude, the operational conditions tend to shift from the design point. But the nomograph Fig.29 can not be used to assess if the obtained deviations are permissible or not.

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 whose dimensions correspond to the standardized dimensions and the speed is synchronous.

It should be noted that the current practical experiences with the turbines for small hydro application accept both direct coupling of turbines and generators and step up gearing.

The direct coupling finds always application when the turbine has a high synchronous speed which allows a high speed generator to be used. At low values of n, the turbines can be directly coupled to the generator or through step up gear.

At direct coupling and 50 Hz frequency, n = 1500, 1000, 750, 600, 500, 428.6, 375, 333, 300 min^{-1}; at 60 Hz frequency n = 1200, 900, 720, 600, 514, 450, 400, 360, 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 along side with step up gearing will govern the standard line up of turbine speeds that differ from the synchronous ones.

Further we will discuss the methods for identifying the field of application for turbines of the given series, whose size and speed correspond to the standard line up.

When plotting the field of application shown in Fig.29, rather definite operational conditions with fixed values n'_{Ic} & Q'_{Ic} are taken. 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 certain values of $n'_{I\,max} - n'_{I\,min}$ and $Q'_{I\,max} - Q'_{I\,min}$.

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

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

The ratio K_D controlling the normal line up of diameters (whose values 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 coordinates log Q - log H_n is limited by two parallel lines inclined at an angle of $63{,}43^\circ$ to the axis log Q. In Fig.30 are shown these zones for the line up of diameters with K_D = I.08 from the same series of turbines. The field of application for this series is given in Fig.29.

The field of application relative to head at D_I = const and n = const depends on the magnitude of $n'_{I\ max}$ and $n'_{I\ min}$. In the study case, $n'_{I\ max}$ is taken equal to 73 min^{-I} and $n'_{I\ min}$ equal to 63 min^{-I}.

Fig.30 shows the field of effective application for the turbines of the same series but various diameters D_I = const at n = I500, I000, 750, 600, 500, 428.6 min^{-I}. At n = const, the field of application is limited by parallel lines inclined at an angle of 26.56° to the axis log Q.

5.5. General Laws for Pelton Turbines

Let us first consider some peculiarities of the Pelton turbine operation in the zone of maximum efficiencies.

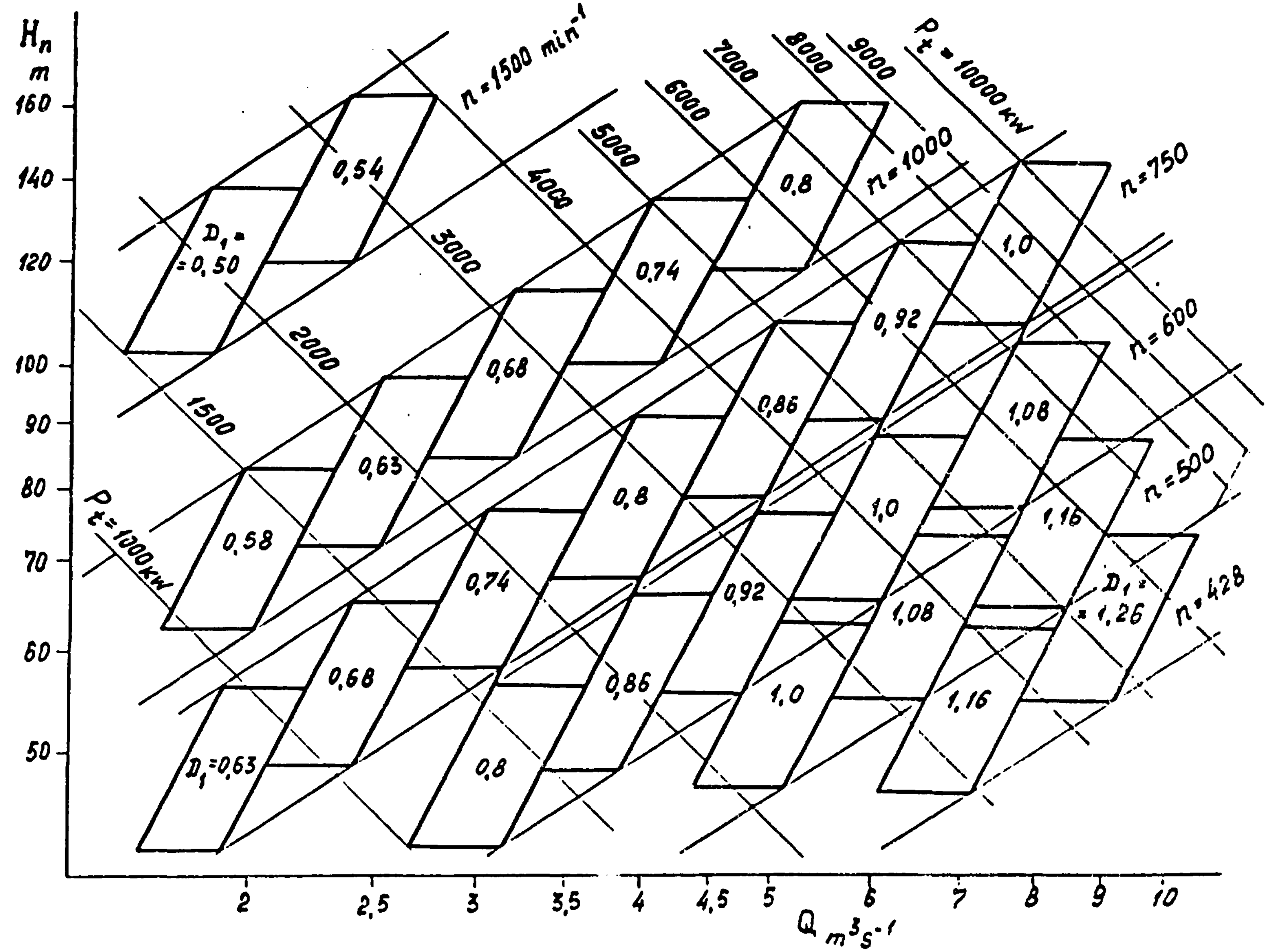
According to the formula (3.I), the velocity of a jet discharging from the nozzle depends on the head. The interaction of the jet and wheel buckets results in energy transfer to the turbine shaft. The optimum peripheral speed of the runner circle tangent to the central line of the jet should be equal, in an ideal case, to 0.5 V_j or to (0.46-0.47)V_j considering the hydraulic losses.

As it follows from the said above

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

with due regard to (3.I) we will obtain the reduced speed

$$n'_I = \frac{n D_1}{\sqrt{H_n}} = (0.46-0.47)\frac{60}{\pi}\varphi\sqrt{2g} \qquad (5.I)$$



H_n
m
160
140
120
100
90
80
70
60
50
n = 1500 min⁻¹
n = 1000
n = 750
n = 600
n = 500
n = 428
P_t = 10000 kW
P_t = 1000 kW
9000
8000
7000
6000
5000
4000
3000
2000
1500
D_1 = 0,50
D_1 = 0,63
D_1 = 1,26
0,54
0,58
0,63
0,68
0,74
0,8
0,86
0,92
1,0
1,08
1,16
2
2,5
3
3,5
4
4,5
5
6
7
8
9
10
Q m³s⁻¹

Fig.30

So all Pelton turbines featuring good energy characteristics have in the optimum mode of operation

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

As seen from the Pelton turbine characteristic shown in Fig.2I, the turbine efficiency tends to drop rather sharply on departure from the optimum values of n'_I.

Applying the general formula (3.2) intended to determine the nozzle discharge we will obtain the discharge of the Pelton turbine with a multi-nozzle arrangement z_j

$$Q = z_j \frac{\pi d_j^2}{4} \varphi \sqrt{2gH_n} \qquad (5.2)$$

Adjusted to D_I = I m and H_n = I m, the discharge equals

$$Q'_I = \frac{Q}{D_1^2 \sqrt{H_n}} = \frac{\pi}{4} \varphi \sqrt{2g}\ z_j \left(\frac{d_j}{D_1}\right)^2$$

or

$$Q'_I = 3.41 z_j \left(\frac{d_j}{D_1}\right)^2 \qquad (5.3)$$

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

In the course of the discharge control by use of 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 hydro application should not have more than two nozzles. Otherwise turbine construction and its operation would get too complicated.

The specific speed of the turbine equals

$$n_s = 3.65\, n'_I \sqrt{\eta_t Q'_I} = (249\text{-}255) \sqrt{z_j} \left(\frac{d_j}{D_1}\right).$$

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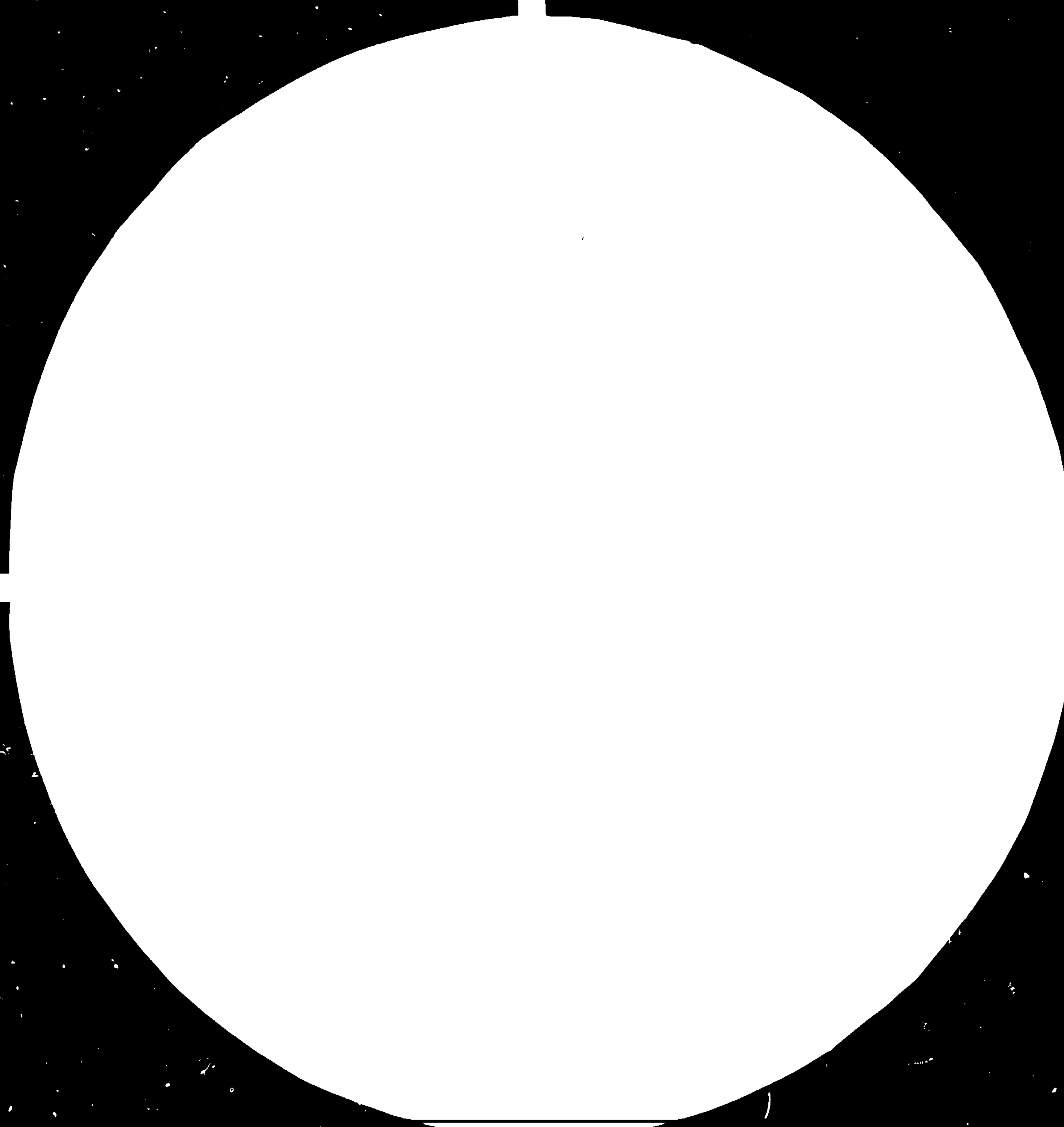
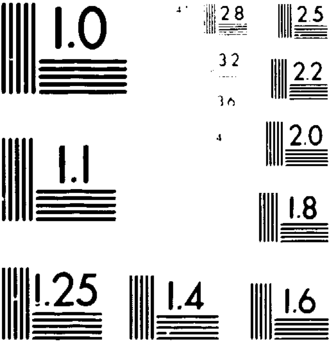




MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS
STANDARD REFERENCE MATERIAL 1010a
(ANSI and ISO TEST CHART No. 2)

The conducted studies show that for $d_j/D_1 > (1/6 \div 1/7)$, the efficiency tends to drop rather intensively. 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 $d_j/D_1 = 1/10 \div 1/15$. Therefore the specific speed preferable for the single-nozzle turbine is I6-26 and that for the two-nozzle arrangement is 22-36 under the optimum operational conditions.

5.6. Standardization of Pelton Turbines

The turbine power is governed by the head and the nozzle dimensions. The turbine power 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.

At change in the diameter d_j, the turbine power, with the other thing 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 its basic dimensions, in particular, the maximum jet diameter.

In Fig.3I is shown the characteristic of turbine I for various loads. It is seen that the turbine features a high efficiency in the wide range of loads. At partial loadings the average operational efficiency η_{tm} 2 tends to decrease but not intensively.

It is evident from the formula (5.2) the relation $H_n(Q)$ in logarithmic coordinates is of a linear nature for d_j=const.

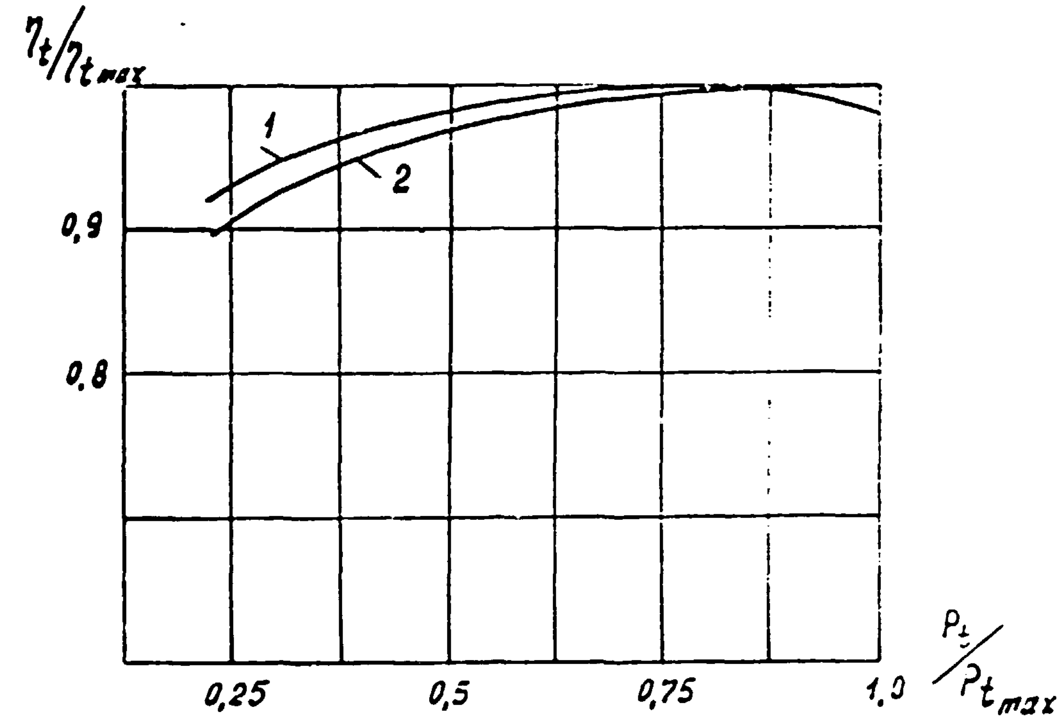


Fig.31

Hence

$$\log H_n = -2\log\left(z_j \frac{\pi d_j^2}{4}\varphi\sqrt{2g}\right) + 2\log Q.$$

In Fig.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 = const. In the given case for d_j/D_I = 0.I we have D_I=I0 d_j. Thus for the entire field Q - H_n the range of runner diameter variation lies within D_I = 0.2 - 2.5 m.

Plotted in this field are also the lines n = const corresponding to the synchronous speed of the generator. Applying the formula (2.I6) we obtain

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

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

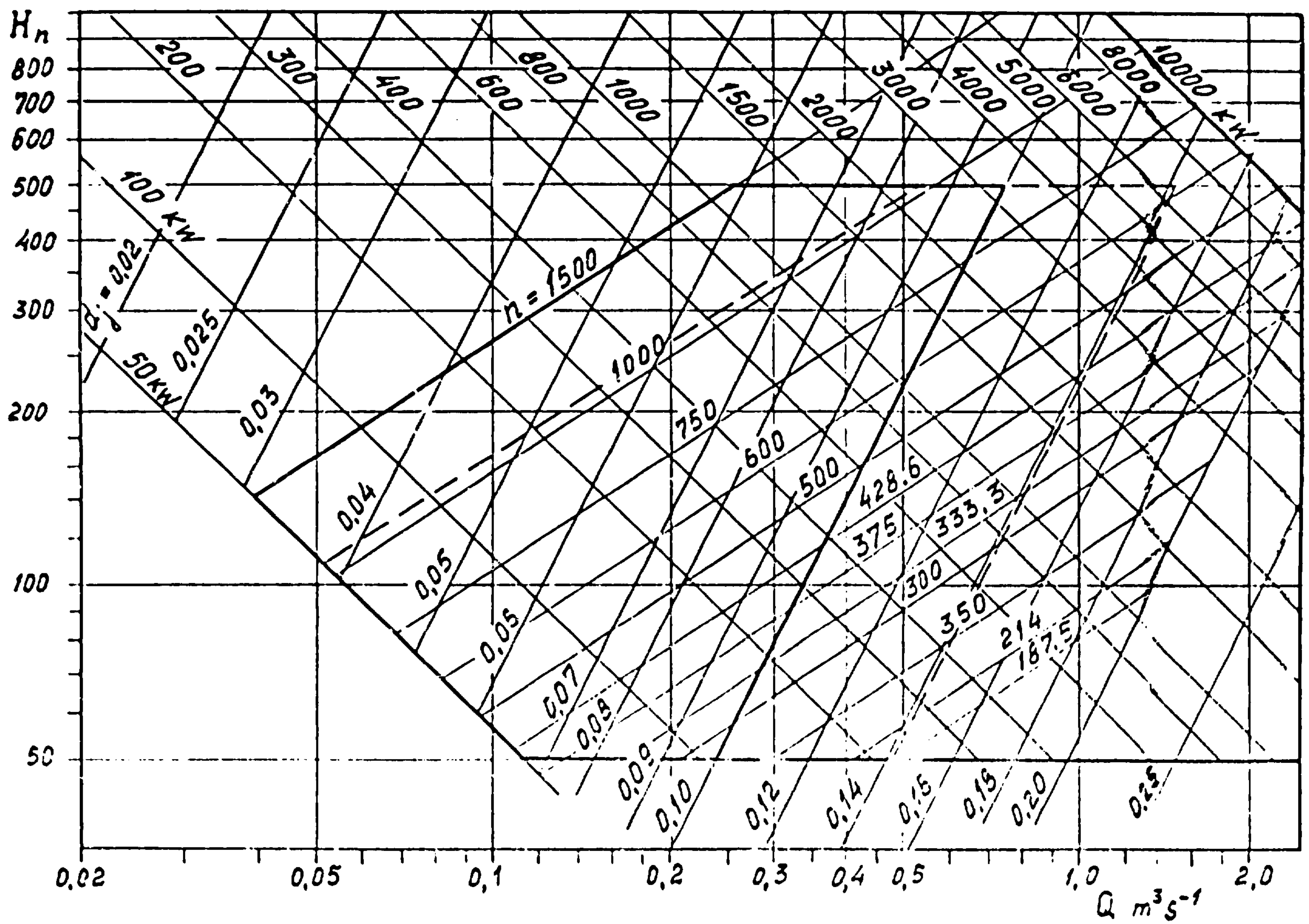
For n_s = const, the lines n = const form in logarithmic coordinates a family of straight lines inclined at an angle of 33.69^o to the axis log Q.

In Fig.32 are plotted the lines of constant speed for synchronous values in the range I87.5 - I500 min^{-I}.

The field area Q - H lying above the line n = I500 min^{-I} 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 hydroturbines can limit the D_I value by I.0 or 2.0 m.



H_n
800
700
600
500
400
300
200
100
50
200
300
400
600
800
1000
1500
2000
3000
4000
5000
6000
8000
10000 kW
100 kW
50 kW
$d_j = 0,02$
0,025
0,03
0,04
0,05
0,06
0,07
0,08
0,09
0,10
0,12
0,14
0,16
0,18
0,20
0,25
n = 1500
1000
750
600
500
428,6
375
333,3
300
350
214
187,5
0,02
0,05
0,1
0,2
0,3
0,4
0,5
1,0
2,0
$Q\ m^3 s^{-1}$

Fig.32

Assuming $D_{I\ max}$ = I.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 Fig.32.

It is seen that the maximum power here does not exceed 3000 kW. The higher power would require a turbine with double-nozzle arrangement. The specific speed of the double-nozzle turbine equals

$$n_{S2} = n_{S1}\sqrt{2}$$

, provided the other conditions remain unchanged (n_{sI} - 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 equals 5000 MW.

So, for the study case of the Pelton turbine application, the practical area of the standardized turbine application is limited with respect to the head by 50 - 500 m, with respect to the jet diameter by 0.03 - 0.I m, with respect to the runner diameter by 0.3 - I.0 m, with respect to the power by 50-3000kW for a single-nozzle arrangement and by 50-5000 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 basing on the nomograph (ref to Fig.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, wheel bucket sizes.

An integrated approach should find application in working out this problem.

The following conditions must be taken into account when specifying the line up of runner diameters.

The turbine with certain runner dimensions can operate within a limited head variations.

It is seen from Fig.2I that at departure of n'_I from the optimum value, the cost effectiveness of the turbines decreases tangibly. Therefore it is practicable to limit the operating zone by the limiting values of $n'_{I\,max}$ and $n'_{I\,min}$.

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 with the runner D_{n+I} by the heads H_{n+I}^{max} and H_{n+I}^{min}.

Assuming further that

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

and n is equal for both turbines, we will find that according the formula of reduced (unit) speed (5.I),

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

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

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

Since D_{min} is taken equal to 0.3 m and D_{max} = I.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 = II, then K_D = I.I28 at $n'_{I\,max} = 42.6\ min^{-I}$ and $n'_{I\,min} = 38\ min^{-I}$.

Thus we obtain a normal line up of runner sizes determined from the formula $D_{n+1} = 1.128\,D_n$

after rounding off

D_I = 0.30; 0.34; 0.38; 0.43; 0.485; 0.545; 0.620; 0.695; 0.785; 0.885; I.00 m.

Let us pass over now to the determination of the line up of nozzle sizes.

As has been assumed above, the minimum jet size $d_{j\,min}$ 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 equals

$$D \sim 14\, d_{j\,min}$$

For the specific case we obtain D_I = 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

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

In particular, at n = I500 and D = 0.3 - 0.43, the area of heads II2.0 - 292.0 m is overlapped (see Fig.33).

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

$$D_{min}/d_{j\,max} \sim 7, \qquad D_{max}/d_{j\,max} \sim 10$$

Assuming at the same time that $d_{j\,max}$ = 0.043 m.

Thus for D = 0.3 - 0.43 and n = I500 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

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

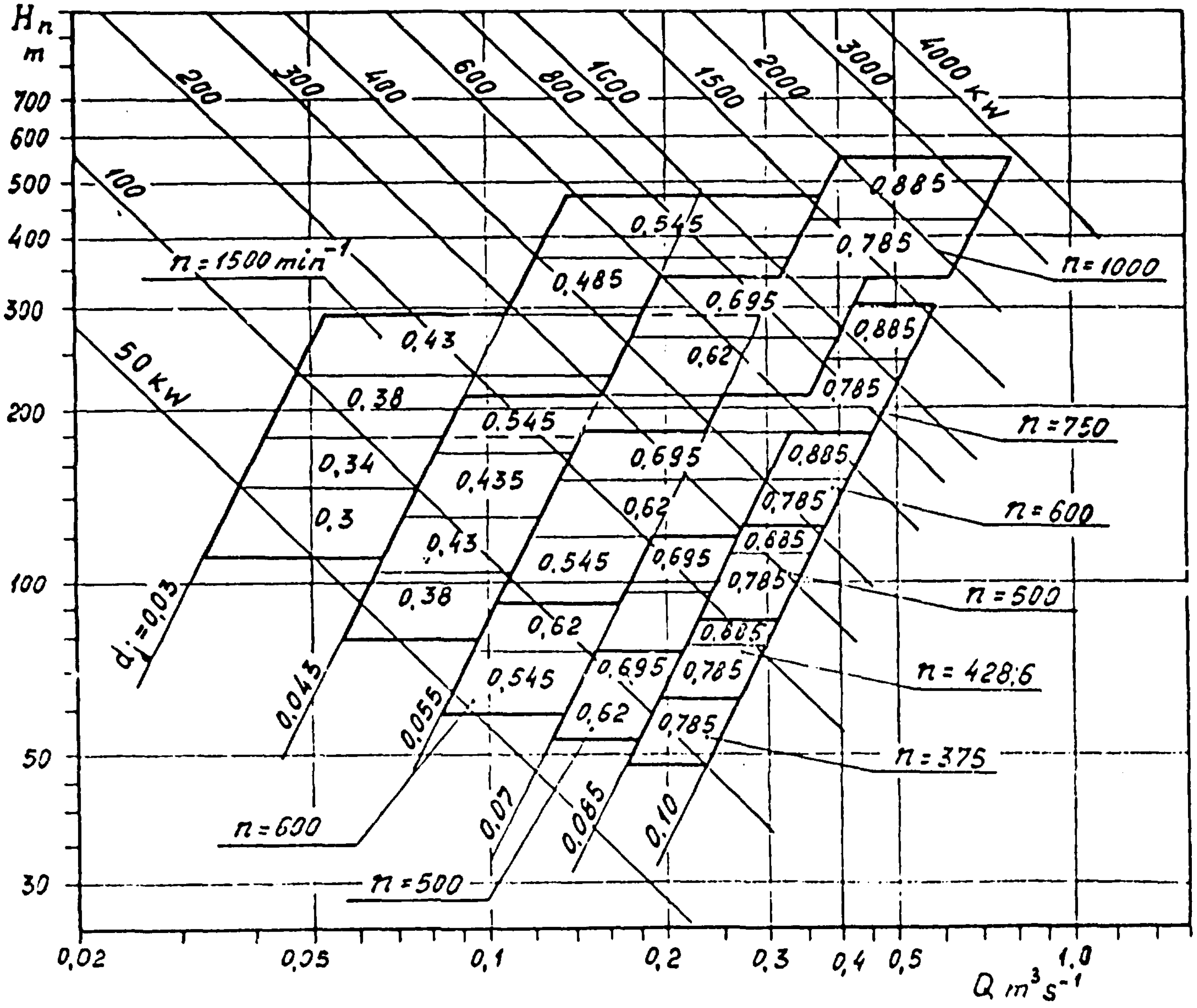

Hn
m
700
600
500
400
300
200
100
50
30
0,02
0,05
0,1
0,2
0,3
0,4
0,5
1,0
Q m³s⁻¹
50 kW
100
200
300
400
600
800
1000
1500
2000
3000
4000 kW
n = 1500 min⁻¹
n = 1000
n = 750
n = 600
n = 500
n = 428,6
n = 375
n = 600
n = 500
dᵢ = 0,03
0,043
0,055
0,07
0,085
0,10
0,885
0,785
0,695
0,62
0,545
0,485
0,435
0,43
0,38
0,34
0,3

Fig.33

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 same dimensions of the buckets.

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

An example is shown in Fig.33 which enables the nozzle, runner sizes and speed to be determined for any values of H_n and Q or P_t.

In our study case 5 standard sizes of nozzles and wheel buckets for IO 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 - 3000 kW.

Table 5 gives the required combinations. It is seen that I7 standard-sizes of turbines are required to cover the entire area of application given above.

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

Table 4

Nº	D \ n	1500	1000	750	600	500	428,6	375	333	300
I	0,3	III,6	49,6							
2	0,34	I43,6	63,7							
3	0,38	I79,0	79,6	44,8						
4	0,43	229,2	IOI,9	57,3						
5	0,485	29[illegible],6	I29,6	72,9	46,7					
6	0,545	368,3	I63,7	92,0	58,9	40,9				
7	0,62	476,6	2II,8	II9,I	76,2	52,9				
8	0,695	598,9	266,2	I49,7	95,8	66,5	48,9			
9	0,785		339,6	I9I,0	I22,2	84,9	62,4	47,8		
I0	0,885		43I,6	242,8	I55,4	I07,9	79,3	60,7	47,9	
II	I,00		55I,0	309,9	I98,4	I37,8	IOI,2	77,4	6I,2	49,6

Table 5

D \ d_j max	0.043	0.055	0.07	0.085	0.IO
0.3	+				
0.034	+				
0.38	+	+			
0.43	+	+			
0.485		+			
0.545		+	+		
0.62			+	+	
0.695			+	+	
0.785				+	+
0.885				+	+

6. SOME EXAMPLES OF STANDARDIZATION

6.I. Normalization of Small Hydraulic Turbines in USSR

For the first time in the world practice the normalization and standardization of small hydraulic turbines were realized in the USSR in the first years after the second world war to meet the demands of the National Economy and Power Industry as well to cope with the construction of small hydroplants on a mass scale.

The mass production of small hydroturbines was preceded by the work on developemnt of respective standards and unification of turbine constructions.

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.

At the second phase, the selected systems for most widely spread standard sizes underwent detailed development. The design development covered construction of not separate turbines but the whole turbine series. There came possibility of unifying 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 constructions are more adaptable to the requirements of mass-production process. Specialised production of turbine equipment can be set up at several factories.

6.1.1. 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 - 1000.

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% it is possible to develop a set of 10 turbine series. Out of this number, 7 will be of the Francis type. The specific speeds of two neighbouring series are related by the approximate ratio:

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

If 3-4% efficiency loss is allowed, a line up of 5 series will do and at 6-7% efficiency drop a line up of 3 series is sufficient.

But selection of the number of series, as indicated above, depends not only on the efficiency drop condition. 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 Francis turbine) H_s = 1 - 2 m (for 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

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

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 I2 series of reaction turbines. Out of this number 8 turbine series are of Francis type and 4 series are of 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 of Francis type and two series are of Kaplan type). But it entails an appreciable effeciency loss to 3-4%.

As indicated above, one of the major objectives of small hydroturbine standardization is tc 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 = I.05 - I.075. To cut down the number of standard sizes, the denominator K_D can be raised to I.II but it entails about 2% loss of average operating efficiency.

At the initial stage of small turbine normilization, a higher value of K_D = I.I9 was adopted. When changing over from one size to the next larger one, the power increases as much as $I.I9^2$ = I.4I6. The following line up of diameters is adopted for Francis turbines D_I = 0.30; 0.35; 0.42; 0.50; 0.59; 0.7I; 0.84; I.00 m.

For propeller and Kaplan turbines:

D_I = 0.35; 0.46; 0.59; 0.80; I.00; I.20; I.40; I.50 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 proceeding from the conditions of application which ensured a high operating efficiency.

It may be noted that the construction of standardized turbines can provide for a minor change (about ± 3%) 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 tangibly restricted for a relatively high ratio K_d.

6.I.2. Reaction Turbine Configurations

Though great experience has been accumulated in designing engineering and manufacture of large-size hydraulic turbine its transfer to the area of small hydraulic turbines without considerable amendments is not possible.

Or to be more exact, the designs which are peculiar to the small hydro turbines 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 tangibly lower than that of large-scale hydro power developments, efforts should be taken to cut the cost of hydroelectric plant including reduction of the equipment cost.

There are two possible ways.

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 process will result in lower cost of equipment without detriment to the energy and cavitation characteristics of the turbines.

Secondly, use of such types and configurations that don't find application in the large-size installations.

Departure from conventional designs and simplification in configuration and manufacturing process would give the turbines somewhat poorer in quality but much cheaper.

From the view point of organization of cheap turbine manufacture, it is practicable to limit the number of turbine designs. The first alternative of nomenclature covered the following turbine configurations.

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

2) Horizontal-shaft turbines in steel case with frontal water supply and shaft passing through draft tube elbow for P_t upto I000 kW.

3) Horizontal shaft spiral turbines for P_t upto 2500 kW.

6.I.3. NOMENCLATURE OF IMPULSE TURBINES.

First let us consider the nomenclature of Pelton turbines. 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 equal to about 7%, the maximum power output of the turbine can be established within 0,25-0,I of its full power to be obtained at the maximum nozzle opening. From this condition it follows that:

$$d_j,(n+I) = 2d_j,n$$

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;0.I0m can be obtained in the area of practical interest.

It is desirable from the view point of manufacture, but not economically viable because it causes the extra consumption of metal. In the limiting case when installing the turbine, the capacity of which four times as much as required, its linear dimensions will be twice and the mass eight times more than those of the turbine designed for the given parameters.

Taking into consideration feasibility analysis, the following line up of water jet diameters is taken:

d_j=0.025;0.036;0.050;0.065;0.082;0.I0m.

The range factor varies from I.44 to I.22.

When selecting the line up of runner diameters the minimum value D_I/d_j = I0 is taken.

For the smallest turbines with d_j=0.025m the greater ratio values are taken, reaching the value of D_I/d_j=20.

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

D_I=0.036;0.050;0.65;0.82; I.00; I.20m

For H_n=40-250m and P_t=I0-500kW resulting in Q=0.0I2-0.4 m^3/sec. and n=250-750min.$^{-I}$

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

For normalized Tourgo turbines d_j=0.025-0.2m, D_I=0.2I-0.84m. These turbines may be applied in the range H_n=50-400m and P_t=I0-4000kW.

The area of application of the normalized Banki-Mitchell turbines lies within H_n=I0-I60m and P_t=5-300kW.

6.2. SMALL TURBINES OF VOITH COMPANY (Federal Germany,Austria)

The Company developed the standard line up of hydraulic turbines of small hydro application, which includes the following types of units:

a) Pelton turbines for high heads;

b) Francis spiral turbines;

c) Francis open flume turbines;

d), Axial flow turbine of tube,pit and bulb type;

Standard hydraulic turbines cover the following range of operational parameters:

H_n=2-400m; Q=0,05-80 m^3/sec;

P_t= 50-I0000kw.

The Francis turbines with the capacity $P_t >$ 2000kw and axial flow turbines with the capacity $P_t >$ 5000kw are custom-designed.

6.2.I. PELTON TURBINES.

The Pelton turbines are applied in the high head range and relatively low discharges:

H_n=40-400m; Q=0,05-I,3m^3/sec.

The advantages of these turbines are:

I) better efficiency in comparison with the Francis turbines in the said area of application;

2) high efficiency in the wide range of loads;

3) simple in design;

4) relatively low cost of installation.

The standard design provides for the 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=0,05-0,I6m^3/sec., H_n=I00-400m;

P_t=50-500kw;

Double nozzle turbines are applied at:

$Q=0,06-I,3m^3/sec.$; $H_n=40-400m$; $P_t=50-4000kw$.

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

$H_n= 40-I50m$; $Q=0.I-0,8\ m^3/sec$;

The following standard line up of runner diameters is taken:

$D_I=$ 0,305;0,340;0,385;0,430;0,485;0,545;0,6I0; 0,685;0,770;0860;0,970m(K_D=I,I23) when speed n= I500;I000;750;600;500;428 $min.^{-I}$

Highlighting the design peculiarities it should be noted, that the runner is made in two alternatives:

I) wheel buckets are cast integrally with disk and hub;

2) buckets are bolted to the disk.

The runner,cast as a whole piece,is cheaper than that assembled. 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 the 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.

The particular attention is paid to the proper water supply and its distribution among them in the double nozzle turbines.

The Pelton turbines are equipped with the speed governor providing the following methods of regulation:

I) nozzle control;

2) deflector control;

3) 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. Due to this, the 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 fifty percent at the small hydroelectric plant with small-diameter penstocks.

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

6.2.2. FRANCIS SPIRAL TURBINES.

Francis spiral turbines with scroll case water supply are used at:

H_n=I0-I50m; Q=0.I2-I2 m^3/sec; P_t=50-2000kw.

In some cases they are also used at lower heads as well to reduce the cost under some particular conditions. They may be advantageous also for the turbines of particualr small dimensions.

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

To meet the said Q-H_n requirements the Company applies the runners of I0 series, depending on the pressure head and suction head.

For high head turbines in the range of H_n=I00-I50m the allowable suction head varies within H_s=-3,+4m. For low head turbines in the range of H_n=I5-20m the allowable suction head varies within H_s=3,+7m.

A standard line up of runner diameters is established:

D_I= 0.205;0.235;0.265;0.300;0.345;0.390;0.440;
0.470;0.500;0.535;0.570;0.6I0;0.650;0.690;0.740;
0.790;0.840;0.895;0.955;I.0I5;I.085;I.I55;
I.230;I.3I0;I.400 .

Here for small diameters K_D=I.I36 and for greater diameters K_D=I.066.

The turbine shaft speed n=I500;I000;750;600;500;428; (300;250;200) $min.^{-I}$

The last three values are used for low head installalations with H_n=4-IOm.

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

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

The following distinquishing features of the spiral turbine design can be noted.

The turbine has a hydraulically optimum spiral case allowing high water velocity and having the best suited dimensions at minimum losses. Being small in size, the spiral case is made of steel casting. Should the dimensions increase the spiral case is welded of plate steel.The steel welded spiral cases are providedwith 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. When designing, specific attention is paid to sealing and its reliable service in operation. Special holes are drilled for removal of leaks so as to eliminate the external leaks completely.

Depending on the turbine dimensions the shifting ring of the wicket gate assembly may be either less or more than the diameter of stems'location.

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 turbine runners have been worked out in hydraulic laboratory. The company quarantees the operation of runners with the highest pcssible efficiency. The runners are distinquished by the head,discharge and speed.

The runners are one piece casting of steel, bronze, aluminium bronze or chromium steel for the turbines of any high specific speed. The 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 the hydraulic axial thrust on the runner. Due to this the turbine bearing should take up the said thrust as well. The design of bearings is governed by the thrust magnitude and turbine shaft speed.

The Company applies the thrust ring design for ordinary operation conditions. At neglegible hydraulic thrust and speed the self-lubrication oil bearings with fixed position of thrust surface and without water cooling of oil are used. At high magnitudes of axial thrust and speed the cooling system is arranged. The pressure lubrication is also provided for heavily-loaded bearings.

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

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.At a considerable size of the turbines and a small suction head the vertical draft tube becomes very long, which causes the 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 and 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 the rigid shafts couplings are used. In the said case only one bearing in 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 overhang turbine bearing is required, which may simultaneously function as the thrust bearing. When arranging the belt drive, the turbine shaft may have two bearings with 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. To improve the regulation stability flywheels are used.Similar function here is intended for generator rotor.When mounted the flywheel simulteneously may function as a half of coupling at the end of generator shaft.

When using the rigid coupling often the flywheel is 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 the turbine with twin spiral case and two runners is used. A big size turbine is arranged integrally with a small size turbine. The two turbines operate together when the water discharge is high.The big size turbine will operate alone at the average discharge and small size turbine will run alone at the low water discharge. In this installation at the discharge approximating to one six of the full water discharge the use of water energy takes plase with the satisfactory efficiency.

Sometimes, the high speed operation can be specified for the turbine.For such installations the twin turbine is a good solution. This turbine is equipped with the twin runner, 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.

6.2.3. FRANCIS OPEN FLUME TURBINES.

The Company developed the standard line up of open flume turbines (fig.6)and recommends it for application,when

H_n=2-I0m; Q=2.5-30 m^3/sec; P_t=80-2000kw;

These turbines cover the range of axial flow turbines. But there are some positive factors coming out at their application.

I) Vertical open-flume turbines have small dimensions.

2) Generator may be mounted above the maximum water level, which is often difficult in turbines with a horizontal shaft arrangement.

3) Turbines of this type have relatively low cost, require small capital investments and cheap equipment.

4) Application of vertical step up gear allows the use of high speed generators.

The said area Q-H_n is covered by various six series of turbines. Here turbines with I9 normalized dimensions of runners are used.

The following standard line up of diameters is established:

D_I= 0.84;0.895;0.955;I.0I5;I.086;I.I55;I.23;I.3I;
I.40;I.49;I.59;I.695;I.805;I.90;I.99;2.09;2.I9;
2.30,2.4Im.

D_I up to I.805, K_D=I.0658 at bigger sizes K_D=I.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.

Turbine series	H_n, m	Q,m^3/sec.	n,$min.^{-I}$	P_t, kw
F I60	2-I0	I.6-I6	40-200	80-I000
F I90	2-I0	I.6-I6	45-225	80-I000
F 225	2-I0	2-20	40-200	80-I000
F 260	2-I0	2.4-30	45-225	80-I000
F 295	2-I0	2.4-30	50-250	80-2000
F 330	2-I0	2.4-30	55-275	80-2000

The efficiencies of adopted series of turbines are various.The F I60 series turbine is the best in terms of efficiency.With the growth of turbine specific speed the efficiency tends to decrease both under optimum operation conditions and especially at partial loadings. The operation of turbines is recommended at the loads not lower than fifty percent of the maximum value.

The effeciency of turbine at full gate is about 82-85%. The magnitude of efficiency depends also on the size of turbine and series. For example: Under the optimum operation conditions the F 330 series has the efficiency by two percent less and at fifty percent load it is less by six and a half percent as compared with the F I60 series.

The efficiency of turbines may be lower at modernized old hydroelectric plants, where some separate sections of water passage remain non optimum.

The effifiency of single stage step up gear is evaluated by the Company at about 96.5-98.5% at full load.

The developed turbine series have satisfactory cavitation characteristics.The turbines of the F I60,F I90, F 225 series at the recommended range of heads may have the suction head up to 7m and turbines of the F 260, F 296, F 330 up to 6m.

The Company developed standard design of turbine installation and the main dimensions are given depending on turbine types and sizes of runners.

The scope of supply includes: turbine, step up gear, generator and speed governor.

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 resistance is used for the corrosive medium with suspensions. The runners have been developed experimentally and the Company quarantees 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 (fig.6).

Application of the Francis turbines for such low heads results in low speeds of the turbine shaft.Use of an economic high speed generator (500-I000min.$^{-I}$) requires step up gear. The Company applies for the said purpose, the planetary coaxial step-up single-stage gears with the similar direction of the shaft rotation. The transmission ratio is within the following ranges:

P_t = up to 200kw; i= 5-3I.5;
P_t = up to I000kw; i= 7-28;
$P_t \geqslant$ I000kw; i= 2.7-7.3

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

As indicated the generators with speed range 500-I000min.$^{-I}$ are used in hydraulic units.Depending on the type of turbine and head the runaway speed is ranging from 200 percent up to 25o percent that of nominal.

Elastic coupling is installed between generator and step-up gear at the application of synchroneous unit, the flywheel is 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 last case the horizontal shaft generator is installed.

6.2.4. HORIZONTAL AXIAL-FLOW TURBINES.

The Voith Company, as well as most of other hydraulic turbine manufacturers developed the standards for horizontal shaft axial hydraulic turbines with the upstream bulb arrangement and S-shaped draft tube (fig.12).

Wide propogation of this type of turbines stems from certain advantages in comparison with axial vertical-shaft turbines:

I) simpliest water passage from hydraulic point of view,

2) small dimensions of the power house and convinient arrangement in question.

The turbines mostly have fixed wicket gates and adjustable blade runner.

The Company developed nomographs for preliminary selection of main parameters of turbine equipment, depending on water head and discharge:

H_n= 2-I5m, Q =3-70m^3/sec, P_t= I00-7000kw, n=80-300min^{-I}.

The three series of turbines are recommended distinquished by the Z number of blades:

where Z=5 H_n=5-I5m, Q=7-70m^3/sec, P_t=200-7000kw,
where Z=4 H_n=3-I0m, Q=3.5-60m^3/sec., P_t= I00-5000kw,
where Z=3 H_n=2-6m, Q=3-50m^3/sec., P_t=50-2800kw

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

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

D_I=0.96;I.090;I.I6;I.23;I.3I;I.40;I.49;I.59;I.70;

I.8I;I.90;I.99;2.09;2.I9;2.30;2.4I;2.53;2.65;
2.78;2.92m.

Here K_D= I.066 at small D_I and K_D = I.049 at big D_I.

The supply of hydraulic turbine includes the step-up gear, generator and governor.

Water conveying and outlet features of turbine may be adopted to the 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.

As mentioned before, the runner has 3-5 blades made of steel or bronze casting, which rotate in bronze bearing housed in the runner hub.

The runner blade mechanism, consisting of cross head, links, levers and journals, 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 the equal clearance among the runner blades and chamber in any position of blades. To install and dismantle 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 outer cone with flanges, gates, welded to it and hub with removable casing.

The draft tube is very important especially in turbines featuring high specific speed turbines in question.

. Due to the said, the shape of water passage plays a very important role.

To improve hydraulic properties it is welded of many separate segments. To increase the rigidity of the tube 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 shaft the cylinder of oil servomotor is forged. The shaft is flanged to the runner and step-up gear shaft.

The quide bearing with oil lubrication is inside the wicket gate assembly hub. To prevent penetration of water special sealing with the pipe to divert the leaks is used. The labyrinth seal is used as well.

The second bearing of the shaft and thrust bearing are installed in the gear tramsmission unit.

At the end of shaft, where it passes through the wall of 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 500min^{-1}. or 750min^{-1}. The transmission ratio may be easily selected by the ratio of speeds of turbine and generator shafts and by the number of teeth in gears.

It should be noted,that the minimum turbine speed is within the range of 95-I40min.$^{-I}$ for the maximum size of the runner. For the turbines with minimum diameter the speed amounts up to 420-750min.$^{-I}$

The low speed shaft of the gear is 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 at the step-up gear. The oil pump to lubricate the gear and the oil pump of the governor is 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.$^{-I}$ or 750 min.$^{-I}$ The maximum runaway speed is higher than the normal one by 250-290 percent irrespective of the head.

An elastic coupling is installed between the generator and step-up gear.

In synchronious units the required flywheel is installed on the generator shaft in combination with the elastic coupling.

6.2.5. VERTICAL-SHAFT KAPLAN TURBINES.

The Voith Company offers also axial vertical turbines of a conventianall design for low head installation.

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

H_n=I.5-8m, Q=2.5-45m^3/sec., P_t=50-25000kW, n=80-500min^{-I}.

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

I) Z=3, H_n=I.5-5m, Q=2.5-40m^3/sec., P_t=50-I500kW

2) Z=4, H_n=3-8m, Q=3.0-45m^3/sec., P_t=I00-2500kW.

The suction head in the operational area is within the range of 1=5-7m for turbines with 3 blades and within the range of 0-6m for turbines with 4 blades.

The same line up of diameters, as for the horizontal turbines in the range of I.02-2.92m, is offered for vertical-shaft turbines. A spiral supply passage with a trapezoadal asymetric 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 2I0°. The draft tube is characterized by the height of h=2.ID_I and the length of ℓ=4.85D_I.

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

The draft tube, symmetrical in plan, has a slight rise of the invert of the straight diffusor (0-8)°. The pier is installed along the axis of the diffuser. The width of outlet section of the draft tube is 3.2D_I, the height is -I.I4D_I.

6.3. SMALL TURBINES OF VOEST-ALPINE COMPANY (Austria)

The Company developed the standards for small hydraulic turbines, including the following well known types:

I) Axial low head hydraulic turbines;

2) Francis turbines for medium heads;

3) Pelton turbines for high heads.

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

H_n=I-I000m, Q=0.0I-75m3/sec, P_t up to I5000kW.

The custom-made design cunting in the specific requirements for the structure and the cost is made for axial-flow turbines at more than 5000kW capacity and for Francis turbines operating under the head $H_n >$ I20m.

6.3.I. 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 use six various series of turbines, distinguished by the specific speed at I5 various diameters for every type. The standard provides for the overage of the whole of the range of application at:

H_n=I-30m, Q=I-60m3/sec.

The standard diametersof runners are characterized by the following values:

D_I = 0.56;0.63;0.7I;0.80;0.90;I.00;I.I2;I.25;I.40;
I.60;I.80;2.00;2.24;2.50;3.00m.

Where K_D=I.I27.

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

H_n=I-5m, Q=I-50m3/sec, P_t=I2-2000kW.

The second series:

H_n=4-I2m, Q=1.6-75 m3/sec,P_t=50-8000kW.

The third series:

H_n=8-I6m,Q=I.9-70m3/sec. P_t=200-I0000kW.

The fourth series:

H_n=I2-22m, Q=I.9-60m3/sec. P_t=200-I2000kW.

The fifth series:

H_n=I6-26m, Q=I.8-60m3/sec. P_t=250-I2000kW,

The sixth series:

H_n=20-30m, Q=I.8-55m^3/sec, P_t=250-I3000kW.

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 distinquishing feature of these turbines is the approximately straight flow of the water through the turbine, including the draft tube. This offers the possibility to simplify the design of the Power House, to reduce the distance between the units and to reduce the cost accordingly. 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.

Due to the high level of efficiency the possibility of economic operation of turbine is improved with variation of the head and the discharge, resulting in an increase of power output. The Company states, that activities in research and standartization of the straught-flow bulb turbines allowed to 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 causes the growth of reduced discharges and the speed.

The optimum matching of turbine features and local conditions is reached by respective 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 and flat curve characteristics with higher efficiency, than that in the propeller turbines with fixed blades and adjustable gates (see fig.24). For example, the reduction of the efficiency by four per cent in the first case will take place at Q_I,/Qopt=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 distinquishing features have been realized in turbine designs:

a) stainless steel is used for the main components,

b) minimum machining of welded elements,

c) free accessand interchangability 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 the head the gates are provided with one or two stems but in any case the stems do not require special servicing.

The hub of the runner is usually made of steel castings. The blades are made either of 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 librication to avoid penetration of the oil into the water. The main quide bearing at the end of the shaft is provided with oil lubrication.

For various local conditions the developed standard allows to use the following alternative arrangements of axial-flow turbines:

I) horizontal; 2) inclined; 3)vertical;

4) with an open headrace;5) 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.

The free access and the low cost of structural elements are reached 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:

I) straight flow turbines with a straight draft tube and a high speed asynchronous generator,which is connected to the turbine shaft through the angle step-up gear;

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

3) straight flow- with a double water supply line.

The diameter of the runner does not exceed 3m.

The economic efficiency of the first type is defined by the range:

H_n=I.5-6m and P_t=50-I000kW.

The second type of turbines is used at H_n up to I0m and P_t up to 5000kW.

For the heads H_n=10-30m in most cases vertical or inclined units are used. In these cases the conveying of the water is effected by means of a bend at an angle of 90-120^o. The water outlet is realized by the respective 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. The 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 desings is the stability of the water passage in the area of 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 prefarable for low and medium capacities a 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 either in horizontal or vertical plane. The turbine shaft is brough out to the free space between the conveying pipes. The step-up gear transferred the energy to the shaft of either vertical or horizontal generator.The units of this type are manufactured either with horizontal or vertical turbine shafts.

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

6.3.2. FRANCIS TURBINES.

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

H_n=I5-I20m, P_t=250-I5000kW; Q=2-30m^3/sec.

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

For installations at H_n more than I20m and up to 300m the hydraulic units may be manufactured by the custom-made design.

The standard employs the following diameters:

D_I= 0.56;0.63;0.7I;0.80;0.90;I.00;I.I2;I.I4;I.60; I.80m.

Table 7 shows the ranges of application of each of eight series of turbines with n_s =I20-450.

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 nomograph has been 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 shall be taken into account.

Table 7

Turbine series	H_nm	Q m^3/sec.	P_t,kW
I	I5-30	2-27	250-8000
2	25-40	2.5-30	500-I0000
3	30-50	2.5-30	600-I2000
4	40-60	2.5-30	800-I5000
5	50-70	2.2-25	I000-I5000
6	60-80	2.0-22	I200-I5000
7	75-I00	2.0-20	I400-I5000
8	80-I20	2.0-I8	I500-I5000

The Company developed some basic typical standartized 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 bearings of wicket gates are arranged.

The wicket gate assembly usually made cast or forged of the stainless materials.The bearings of gates and couplings of levers and links, as well,do not require servicing.

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

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

The Reinfenstein turbines with horizontal or vertical shafts are recommended for very small capacities P_t=I0-200kW and heads H_n=5-40m.

For heads H_n=I0-40m and capacity P_t=I20-5000kW at the positive suction head open flume turbines are used.

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

The spiral turbines with classic 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 as well.

6.3.3.PELTON TURBINES.

The Company developed Pelton turbines for further application.The turbine is used for high heads and low water discharges.This type of turbines is used for low heads only under specific conditions, for example,at considerable errosion, expected for the water with sand suspensions or at an increase of danger of the hydraulic hammer.

Turbines are simple and economically efficient.The efficiency of these turbines is favourable over a wide range of operation.

The standards recommended the use of Pelton turbines when:

H_n= 80-1000m, P_t= 250-15000kW, Q= 0.3-5m^3/sec.and low heads: H_n=20-80m, P_t=10-150kW, Q=0.02-0.2 m^3/sec.

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 the high efficiency.

The 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 features, 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.

The mass-produced generators are used for small capacities.There are two alternatives existing for large capacities: mass-produced generators with elastic coupling between the turbine and the generator (4 supports design) or the generator of specific design with overhang arrangement of the runner and the flywheel(2 supports design)

6.4.SMALL TURBINES OF THE KESSLER COMPANY(Austria)

The Kessler Company, specialized in manufacturing of turbines for small hydroelectric stations, as judged by available information may make small turbines of capacity I0-5000kW for hydroelectric stations,operating at heads from 2 up to 500m.

Typical designs are available for turbines of up to I500kW capacity and for some types of turbines only up to I000kW.More powerfull turbines are made by custom-made designs.

The nomenclature of the offered equipment is rather extensive.

6.4.I AXIAL-FLOW HYDRAULIC TURBINES.

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

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_I= I.06;I.I2;I.I8;I.25;I.32;I.4;I.5;I.6m

(preferable diameters are I.06;I.I8;I.32;I.5).

The speed is n=I50-333 min^{-I} the capacity range is from 50 to 300kW .

The Company made 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 allows 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.The designs with direct connection of turbine and generator shafts are available. The flywheel is on the generator shaft.The turbine shaft is provided with two supports, one of them is 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 the Company is ready to supply the equipment with the inclined arrangement of the axis of straght -flow turbine with the S-shaped draft-tube as well as bulb turbines with the vertical axis and the curved draft tube.

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

D_I= 0.5;0.6;0.72;0.85;I.0;I.I5;I.3;I.45;I.6m.

The speed of the turbine shaft:

n=750;600;500;428;375;333;300;250;200min^{-I}.

Capacity range P_t=25-I200kW at Q=I.4-I5m^3/sec and H_n=2-I2m.

6.4.2. 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.

The Reifenstein turbines with horizontal shafts and straignt 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_I=0.225;0.25;0.3;0.35;,40;0.45;0.50;0.55;0.60m

The turbine shaft speed n=I500;I000;750;600;500;428; 333;300min.$^{-I}$

Capacity range P_t=5-200kW at water discharge Q=0.08-I.5m^3/sec. and head H_n=5-40m.

The straight-flow turbines in case are also used for medium heads .These turbines are characterized by a simple and unique design.From the stand point of 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 gives the possibility to use high speed generators.

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

For reduction temporary nonuniform rotation of the shaft at variations of loads the heavy flywheel is installed for turbines of capacity up to I50kW.

The turbine is provided with the adjustable wicket gate assembly, driven by the hydraulic servomotor.

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

D_I=0.5;0.55;0.60;0.65;0.7;0.75;0.8;0.85;
0.9;0.95;I.0;I,I5;I.3m.

The speeds of turbine shaft:

n=650;600;500;450;400;350;300;250;200;I50min^{-I}.

Capacity range: P_t=50-2000kW at water discharge
Q=0.8-I0m^3/sec. and head H_n=8-30m.

The design of spiral turbines is distinquished 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 the flywheel and reinforced supports.The design provides for use of low voltage generators of a specific design. The flywheel may be supplied also together with conventional high voltage generators.

The Company indicated the low cost of the unit because of a louble support arrangement and absence of a intermediate gear.

The turbines of this design are characterized by
P_t=50-500kW at Q=0.I45-I3m^3/sec.

6.5 BANKI-MITCHELL TURBINES OF THE OSSBERGER COMPANY (Federal Republic of Germany)

The Ossberger Company manufactures small hydraulic turbines,quite simple in design-Banki-Mitchell or divided type turbines. For the first time this turbine was proposed and studied by a Hungarian D.Banki and Australian Mitchell. It's design has been modified by F.Ossberger.

The Banki-Mitchell divided type turbines due to their design and hydraulic features are not effective for the hydraulic stations with units of medium and large capacities. Therefore application of these turbines is limited by small capacities.

At present the Ossberger Company,perhaps,is the only Company, making the given turbines.This company is specialized in these turbines only .

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

6.5.I. RANGE OF APPLICATION.

The Banki-Mitchell (divided type) turbine is related to the system of impulse turbines.During operation the turbine runner is partialy submerged. From the conveying structures the water enters the runner and flowing from the periphery to the center giving up some 70-80% 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 strikes the blade from inside again and in a centrifugal flow gives up the remaining 20-30% 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.

The Osoberger Company has developed the nomenclature of such turbines to be used at:

H_n=I-200m, Q=0.02-9m^3/sec, P_t=5-I000kW, n=50-2000min^{-I}.

The comparison with the nomenclature of standard turbines made by the Voith Company, shows, that the range Q-H_n, covers the range, recommended by the Voith Company 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 runners diameters:

D_I=0.3;0.4;0.5;0.6;0.8;I.0;I.25m.

Thus, the eight standard sizes of turbines offered by the Company, cover rather wide range of application by Q-H_n.

6.5.2 DESIGN FEATURES

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

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

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

Due to a cylinder configaration of blades the runner is not subjected to axial thrust during operation.

The runner is all-welded.After the finish machining it is balanced.

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 up 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 the two sections in width by a special partition in ratio I:2. The quide vane in each section may turn independently.

Therefore the flow area may be full open by 2/3 or I/3 and the runner may take either full discharge or two thirds or one third of it. During partial discharge only the respective part of the whole width of the runner operates.

Thus a step-like variation of water discharge is reached.

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

Both quide vanes may turn independently with the help of control levers, connected either with the system of automatic or manual control.

The main turbine bearings are fit 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 do not require any servicing.

The lubrication material is changed in bearings once a year.

Despite the fact, that the Banki-Mitchell turbine is an impulse turbine at medium and low heads ($H_n < 35m$) the draft tube is installed for more efficient utilization of the head. It is considered necessary to have the possibility to control 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 solve the said task in the way, that a small head of one meter only, may be used with the optimum efficiency. The draft tube, made completely of steel, with a bend reduces the cost of construction for low-head installations in particular.

Due to the above noted design distinquishing features the efficiency of the Banki-Mitchell turbines remains high in a wide range of discharge variations . The maximum efficiency up to 84-88% is observed in medium and large units, which is lower than in modern specific speed turbines.

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

The Banki-Mitchell turbines of the Ossberger Company are supplied together with all necessary accessories.If the turbine is used to drive synchronous or asynchronous generator a step-up gear is applied. In units with synchronous generators the flywheel is installed to reduce temporary nonuniformity.The governor is belt driven.

In low capacity installations all supplying equipment is installed on a single frame and supplied as a complete set.

The Banki-Mitchell turbines are also used for direct connection with other units. . The said turbines find application to drive the high head centrifugal pumps.

In conclusion it may be noted, that in conformity with information ,supplied by the Company,the said turbines have a number of advantages:

I) a simple design and manufacture procedure resulting in relatively low cost;

2) high efficiencies (more than 80%) are observed in a wide range of discharges (0.I67-I);

3) complete automation and simple servicing;

4) quaranteed period of reliable operation is about

30-40 years.

All the abovementioned allows to consider the given turbine competitive in relation to other modern economic turbines.

6.6. SMALL TURBINES OF THE BELL COMPANY. (Switzerland)

The Bell Company is a branch of Escher Wyss Company, related to the Sulzer group. In the field of hydraulic turbines this Company is specialized in designing and manufacturing of hydraulic turbines for small hydro plants. Bearing in mind particular interest to small hydraulic turbines, which has been created recently, the Company has developed the standard line-up of hydraulic turbines, covering the operating range H_n=2-800m, Q = 0.06 -86m^3/sec. P_t=100-2000kW. and in this case the axial flow turbines are intended for application at H_n=2-25m, Q=5-86m^3/sec. Francis turbines at H_n=6-150m, Q=0.3-6 m^3/sec. Pelton turbines at H_n=50-800m, Q=0.06-0.8m^3/sec.

The following conditions were taken into consideration when developing the nomenclature:

1) optimum utilization of the existing scientific researches and designs;

2) supply of complete electromechanical equipment ready for operation,

3) application of simple hydraulic solutions for standard basic designs to reduce the cost and speed up the supply.

4) guaranteed service by the resident offices of the

Company throughout the world.

The unit cost per one kW of hydraulic turbines, of capacity below I00kW is relatively high.

The turbines with capacities higher 2000kW are manufactured by custom-made designs.

6.6.I. RANGE OF APPLICATION OF AXIAL TURBINES.

Standardized axial hydraulic turbines are used at H_n=2-I5m, Q=2-38 m^3/sec. P_t=I00-2000kW.

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

H_n=2-25m, Q=4.8-86m^3/sec, P_t=I00 - I0000 kW,

i.e the turbines supplied by custom-made designs are also included here.

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

D_I=I.0; I.2; I.4; I.65; I.9;2.2; 2.5;2.8;3.2;3.6m

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

n'_I =I50-200min^{-I} , Q'_I=2.4-30 m^3/sec. 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 ÷ +2m.

For example, at H_n=25m the range of design conditions with respect to dischange at variation of H_s in the given limmts, is at

Q_I= 0.8-I m^3/sec .

From this it is inferred, that the Company utilized several types of water passages in axial hydraulic turbines to cover the range of application.

6.6.2. DESIGN FEATURES.

The Bell Company uses only axial-flow horizontal-shaft turbines with a 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 is the design with the adjustable wicket gate assembly and blades. This is the double control with 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. It simplifies the turbine and the governor.The simpliest alternative is the turbine with the fixed gates and blades. Its application is possible at constant water discharges and loading. But the start-up of such turbine by all means 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 given 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 turbine bearing takes up the axial thrust as well. There is no second turbine bearing.The turbine shaft is rigidly connected with the shaft of the step-up gear. Therefore the gear supports act as the second turbine bearing.

The S-shaped draft tube is provided with the straight diffusor with an inclined axis at the outlet.The alternative arrangements of the turbine in which the whole of the draft tube is 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 servomotor.

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% takes place in turbines:

- propeller type at 0.85 Qmax;
- with fixed gates at 0.35 Q max;
- with dual control at 0.2 Q 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 sixteen.

Turbines with fixed wicket gate, assembly require the additional stop device.

Depending on the value of the head two types of installations of horizontal turbines are recommended.

At low heads the open headrace $2.4D_I$ wide is applied. The power House is integrated with the dam.

At high heads the penstock of $I.4D_I$diameter is used. Here the Power House is separated from the dam.

The width of tailrace - $2D_I$.

The width of Power House with one unit - $3.6D_I$.

The length of Power House along the axis of the unit -$8D_I$.

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

- with a horizontal shaft and with a draft tube, having a vertical diffusor at the outlet;

- with an inclined shaft of the unit with the generator installed above the turbine and the bottom of the headrace and the tailrace are approximately at the same level;

- vertical unit with an open flume,with a bulb and conical wicket gate assembly and a bent draft tube;

a) with the generator installed above the upstream water level;

b) with the generator,installed in the pit under the bend of the draft tube.

The small hydraulic turbines, supplied by the Company, are provided with automatic control.It includes:

I) electric system of the automatic start-up;

2) hand control;

3) remote control of automatic operation;

4) local or remote control of the governor.

The following devices ensure the reliable operation:

I) The devices ensuring the closing of the gates at any time under the action of the special weight suspended to the shifting ring;

2) A mechanical centrifugal pendulum which eliminates the possibility the speed rize;

3) Automatic devices preventing any undersirable operating conditions.

Apart from the hydraulic turbine the Company also supplied the governor.

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, the oil supply system).

6.7.SMALL TURBINES OF THE SANDEN COMPANY (Norway)

The Sanden Company, which is a part of Kverner amalgamation is a leading enterprise in Norway in designing and manufacturing of turbines for small hydroelectric stations.

The range of standard hydraulic turbines of the Company covers:

$$H_n=3\text{-}1000m,\ Q=0.05\text{-}30m^3/sec.\ P_t=100\text{-}10000kW.$$

The range of high heads is covered by Pelton turbines of standard design, adjusted to local conditions.The same is true to Francis turbines with the capacity, exceeding I000kW.

For up to I000kW capacity there are standard turbines with the predeveloped design.

The axial turbines are recommended for the range:

H_n=3-I8, Q=3-27m^3/sec. P_t=200-2800kW.

Francis turbines are used at H_n=4-400m, Q=0.4-25m^3/sec. P_t=I00-I0000kW.

Pelton turbines cover the range :

H_n=80-I000m, Q=0.05-2m^3/sec., P_t=I00-I0000kW.

6.7.1 SPIRAL FRANCIS TURBINES

The spiral turbines are manufactured by the basic design.To cover a wide range of application by the head and the discharge with the high efficiency the Company uses twelve series of preliminary worked out and tested 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 serious.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; I.0; I.2; I.4; I.6; I.8; 2.0m.

The Company adopted the diameter D_2 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 ÷ +2.5m at the upper boundary in respect to the head and H_s= 6.5÷ 7.5m at the lower boundary in respect to the head.

By information presented by the Company the efficiency of turbines at D_2=Im is as follows:

I) type C-9I.7%; 2) type H-92.3%; 3) type N-92%.

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

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

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

In low speed and low capacity turbines less than 5000kW a step-up gear between the turbine and generator is allowed.

At very low heads the vertical turbines with an open flume are applied to reduce the cost.

Some design and manufacture 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 labirynth seals.

In turbines with the fixed wicket gate assembly the

Table 8

Turbine series

	B	C	E	F	G	H	J	K	M	N	O	P
H_n, m	II0-400	I00-350	85-300	70-250	60-230	50-200	45-I70	35-I30	25-95	I8-65	I4-55	8-40
Q, m^3/sec.$^{-I}$	0.5-2.8	0.5-3.2	0.5-4.0	0.5-5.5	0.5-6.5	0.7-9.0	0.8-I0.0	0.9-I5	I.2-I8	I.6-23	2.5-26	2.5-33
n_s	75	80	I00	I20	I30	I40	I70	200	250	370	400	475

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. It allows a simple assembling and dismantling of the given connection.

Therefore this method eliminates the use of the key or some other device loosening the shaft and entailing fatique of the metal.

The runner has removable rings of a slit seal on the drive and driven disks.

The case of the shaft seal is of a welded construction. Within the limits of the seal the shaft is plated with white metal. The seal does not actually require servicing. There is drainage pipe running from the seal.

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 diffusor. To prevent cavitation failure the stainless steel is used at the inlet of the bend.

The turbine installations, which always operate in the big low voltage system and which are not designed for frequency control may be fitted with the control system. These control devices are aimed at 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 the isolated line. But it ensures adequate connections with output loads and with water level regulator.

The installations, which operate to the isolated line, require the application of the speed governor.The speed governor 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 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 160m.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 160m. The opening and closing are effected by the water control system or the oil pressure plant.

At low heads, at the hydroelectric plants with short penstocks,the shut off valve in front of the turbine may be omitted.

6.7.2 FRANCIS TURBINES IN DRUM.

This type of the turbine has no definite designation. It differs substantially from the spiral hydraulic turbines mainly by water conveying features.

Eight standard sizes were developed for these turbines.

Table 9 shows the main operating parameters.

Table 9.

	I	II	III	IV	V	VI	VII	VIII
H,m	8-25	8.2-34	8.5-40	I0-55	I3-55	I6-82	22-I05	28-II0
Q,m^3s^{-I}	2.2-4.5	I.4-3.6	I.2-3.0	0.9-2.5	0.7-I.8	0.55-I.55	0.4-I.2	0.3I-0.8
H_s,m	I.4-6.5	4.0-6.5	2.8-6.5	3.6-6.5	2.3-6.5	4.5 6.5	2.8-6.5	3.9-6.5
n_s	380	280	280	235	235	I55	I55	I20

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_s - at the maximum head.

The maximum efficiency of the given turbines is within the range from 85% to 90%,which is considerably less, than in turbines of the spiral type.

The water is conveyed to the turbine through the tube, which is connected to the cylindric drum.The axis of drum coincides with the axis of the turbine and the water conveying is normal to the axis.

The cylindrical drum is welded of ordinary carbon steel and provided with the convex bottom.The hatch and 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.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.It is provided with the replaceble sealing rings.

The box of the shaft sealing is of a welded construction.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 the hand, automatic and remote control of the start-up, loading and shut down is used.

The hydroelectric stations, operating for the 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 the hydromechanical 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 oil servomotor.

6.7.3. AXIAL TURBINES.

The axial hydraulic turbines of the Sanden Company 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 the 0.2m pitch is established:

$$D_I = 1.0; 1.2; 1.4; 1.6; 1.8; 2.0; 2.2; 2.4m$$

It is recommended to use the normalized turbines at H_n=3.5-18m; Q=3.5-27m^3/sec. P_t=400-3000kW.

The two types of axial turbines are produced:

1) with the fixed wicket gate assembly and runner blades;

2) with the fixed wicket gate assembly and adjustable runner blades.

The turbine is not provided with the speed governor if it operates for the interconnected power grid. The turbines operate without servicing.

The first type of turbines is used at the hydraulic stations with relatively constant discharge and head, when it is not necessary to control power.

The second type of them is used at the station with variable discharge and power control. These turbines are characterized by a good level of efficiency in the range of discharges from 40% up to 100% of the full maximum discharge.

The following are the main components of the turbine.

1) a butterfly valve;

2) an inlet element with the quide bearing and the wicket gate assembly;

3) a runner;

4) a draft tube with the quide bearing;

5) 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,the oil to which is supplied from the governor.The piston shifts the valve for the opening.The throttle is self-closing,i.e. it is closed automatically when the oil is drained from the cylinder.

The inlet section of turbine is of a welded construction,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 insensible 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. The inlet section of the tube 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 during operation the high efficiency and the minimum vibration are observed .

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 spherical roller type with oil lubrication and the temperature monitoring.

The draft tube is provided with a hatch to inspect the runner and draft 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 the two hydraulic cylinders, housed in the thrust bearing case through the pipe,wrapping the turbine shaft and connected to the runner hub.This pipe rotates together with the shaft and is driven by the servocylinder.The hydreulic 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 hand and automatic start-up putting the unit on line shut down and the remote control of the loading.It is not intended for speed control, but well fitted for the load and head control.

The governing system of turbine consists of the standard parts.

The standard high speed generator is usually applied to reduce the cost. Therefore the gear is installed between the turbine and generator.

6.8 SMALL TURBINES OF THE HITACHI COMPANY (Japan)

Under the conditions of energy crisis the Hitachi Company began to pay much attention to deaigning and manufacture of small electric stations.The main obsticle in wide use of energy of small rivers is the relative high cost of small hydroelectric stations and equipment for them. For profitable construction it is necessary to reduce the cost of equipment. It is possible only under the conditions of the wide standartization of the main and auxiliaty equipment.

The Hitachi Company developed the standardized ranges of application of various types of hydraulic turbines under conditions of small hydroelectric stations with the range of capacities from 50 to I0000kW. Table I0 shows the ranges of application of various types of turbines.

Table I0

Type	H_n, m	n_s
Pelton turbine	200m and higher	9-35
Francis turbine	I0-600	55-400
Kaplan turbine	5-I00	230-I050
Horizontal axial turbines	3-20	450-II90

Below is the detailed information of separate types of turbines.

6.8.1 FRANCIS TURBINES

The nomographs for preliminary selection of turbines were developed for the countries, using frequencies 50 or 60c.p.s. in electric lines.

It is proposed to use the normalized turbines at:

H_n=20-I80m; Q=0.8-25m^3/$sec.^{-I}$; P_t=300-I0000kW;

n=375-I000$min.^{-I}$ (for frequency of 50 cps).

The following line-up of the diameters of runners for these turbines was taken:

D_I=0.375;0.40;0.425;0.450;0.475;0.50;0.530;

0.560;0.60;0.63;0.67;0.7I0;0.750;0.80;0.85;

0.90;0.95;I.00;I.060;I.I20;I,I80;I.250;I.320m

23 normalized sizes were adopted altogether.It should be noted that the minimum diameter at the leading edge of the runner blade was expressed through D_I.Fourteen different models of the runner are used for the considered range of H_n and Q.

Table II shows the range of application of each type of turbine (for frequency of 50 cps.)

All in all I5I standard sizes of turbines are recommended for use.

Similar information is available and for the frequency of 60 cps. Here I37 standard sizes of turbines are used. The difference of the used models may be observed.

The distinquishing feature of the vertical turbines design is the distinet difference from conventional designs for the 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 the two supports.One of them is the radial support in the area of the top spider of the generator and the other is just on the turbine cover,where the radial and axial thrusts are taken up.

The stator rests just on the flange of the turbine stay ring of the specific design.The stay ring is cast together with supports.The shells of the spiral case are welded.The design of the hydraulic part of the unit is conventional.The adjustable wicket gate assembly is provided with the outer shifting ring.

Model	A	B	C	D	D_o	E
n_s	I05	I35	I50	I65	I70	I92
H_n, M	95- I20	68- I60	60- I50	48- I60	38- II5	35- 62
Q, m^3/c	I.8- 2.7	I.0- 9.0	0.9- 9.0	0.8- I2	II- I3	0.8- I.5
P_t, kW	I500- 3000	750- I0000	500- I0000	300- I0000	900- I000	300- 750
n, min^{-I}	750	I000- 600	I000- 600	I000- 500	500	I000
D_I, m	0.8- 0.85	0.6- I.32	0.5- I.I8	0.475 I.32	I.32	0.425- 0.475

Table II.

F	G	H	I	J	L	M
205	235	245	255	290	325	375
29- II5	30- 45	22- 88	65- 85	2I- 76	I8- 55	I9- 48
I.0- I4	0.85- I.2	0.9- I4	I4- I8	I.2- 22	I.3- 25	2.0- 26
300- I0000	300- 450	250- I0000	I000	300- I0000	300- I0000	300- I0000
750- 500	I000	750- 500	428	750- 428	750- 375	750- 375
0.5- I.25	0.4- 0.425	0.45- I.I8	I.32	0.425- I.25	0.375- I.I8	0.4- I.I2

The design of the unit allows to reduce the period of assembly considerably.For example, for turbines of capacity 3.5 MW ,of the conventional design the period of assembly usually took 75 days.This period has been reduced to 35 days for the turbine of the normalized design,thus by nearly one half.

The assembling of such a unit may be done by a mobile crane.

The hydraulic turbine may be connected with the generator of one or two types:

I) synchronous generator;

2) induction generator.

The induction generator has no exciter and governor, if the electric line is sufficiently extensive and of high power.Thus,the induction generator allows to use the simplified equipment,which results in cost reduction. The largest generator of such type in Japan was supplied by the Hitachi Company at the beginning of I979.Its capacity is 7850kW.

The turbine governor is of mechanical or electrical type. The electric governorsare less in size and simple in servicing.

The mechanical governors are standardized for the turbines of small hydraulic stations.If the induction generator is used,then the speed is not controlled by the governor.In this case its function consists in control of the unit, depending on variations of the upstream water level and the load.

The properly designed control and protective equipment gives the possibility of control and protection with the minimum quality of neccssary functions.

Thus the electrical part of 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 5000kW or lower capacity.

Apart from the vertical turbines the standardized horizontal spiral turbines are used at:

H_n=20-I00m and P_t=50-I000kW.

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 quide turbine bearing and two generator bearings are installed.The turbine and generator shafts are rigidly flanged.The flywheel is installed between the flanges.

6.8.2. AXIAL TURBINES.

The Company recommends the two types of horizontal axial turbines for practical use at small hydroelectric stations:

I) straight flow bulb type;

2) bulb type with an S-shaped draft tube.

There are nomographs for preliminary selection of hydraulic turbines for the frequencies of 50 and 60 cps.

The normalized hydraulic turbines cover the following range in relation to the head and discharge for the frequency of 50 cps.

H_n=3-20m; Q=2-28m^3/sec^{-I}; P_t=I00-5000kW,

n=2I4-750min.$^{-I}$

The unified line-up of runner diameters includes the following sizes:

D_I= 0.70; 0.75;0.80;0.90;I.00;I.I2;I.250;

I.40;I.60;I.80;2.00m.

Altogether there ar: II normalized sizes.

Fourty eight standard sizes of axial hydraulic turbines are proposed for use to cover the said ranges in relation to the head and discharge.

6.9. SMALL TURBINES OF THE GILBERT GILKES & GORDON COMPANY

(Great Britain)

The Company manufactures various hydraulic equipment. The turbine equipment intended for small hydroelectric stations is one of the main products.The turbines of all types are made with the exception of axial-flow turbines.

The approximate range by the head and capacity is within the limits:

H_n=3-240m, P_t=8-6000kW.

Seven types of water passages of Francis turbines and five types of impulse turbines are available with the Company to cover the said range of application.

Table I2 shows the values of specific speed for all types of turbines.

Table I2.

Type	C	2	R	III	β_1	IV	V	SJ	TJ	TI	HCTI	PELT
n_s	3II	302	244	I87	I85	I42	I09	65	65	5I,5	65	22,6

The runners with the highest speed are used for the low heads within the limits H_n=2.8-4.2m. Only the Francis turbines are recommended for the heads up to H_n=3-I3m.

The Francis and impulse Tourgo turbines are used for the range of heads H_n=I5-35m. The Francis turbines are not used for the heads higher than I50m.The impulse inclined-jet hydraulic turbines are recommended for the heads H_n=I5-400m and the impulse bucket ones for the heads H_n=40-I200m.

The hydraulic turbines with an open flume for low heads are of a 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 also used to drive the generator. The spiral radial-axial hydraulic turbines are also used for low heads.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. generator (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. The Company,perhaps, is the only one in the world, manufacturing such turbines. There are two types - with one or two nozzles, the specific speed of which is about 65.The turbine is simple in design and manufacturing.

The efficiency of such turbine is higher than that of the low speed radial-axial hydraulic turbine.In particular it is observed in the turbines with the small seal clearances of the runner,which wear considerably at the presence of abrasive suspended matters in the water. The inclined-jet turbine is always characterized by higher efficiency at loads less than 5/8 or 3/4.

The double control: by the nozzle,needle and the 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 with the exception of small capacities.The bucket hydraulic turbines are not used at the heads less than 100m.The bucket hydraulic turbines are made for the heads up to 800 meters.For high heads these turbines may be supplied by custom-made designs.

The impulse turbines are mainly made with one nozzle. The turbines with two runners are also used. For the most part the turbines with the 22.6 specific speed are applied.

The runner is installed at its own bearings at the capacity of turbines less than 750kW. The runner is arranged at the generator shaft end at the higher capacity of the unit. In this case the supports and the generator shaft are of the special reinforced design. It allows to reduce the cost.

The generators are of custom-made designs; they are designed for the increase of the speed by 70-200% in runaway conditions.

The synchronous generator, operating for the line or the isolated load is driven by the turbine at conventional hydroelectric stations.The generator and the governor should be designed in the way as to ensure the required frequency in parallel and isolated mode of operation .

The asynchronous or induction generators are also used on a wider scale. These generators are of a simple design and cheaper in comparison with the synchronous ones. The speed is controlled by the line frequency, for which the generator operates. The efficiency of the asynchronous generator is often less than that of the synchronous one.

The Company does not manufacture its own governors. The units are sub-supplied by the Woodward Company specialized in making the governors during more than 100 years. The low capacity turbines are controlled by the electric governors.