Chapter 2
Working Principle of Pelton Turbines

2.1 Conversion of Hydraulic Energy into Mechanical Energy

In hydropower plants with Pelton turbines, the available hydraulic energy exists as potential energy, which is measured in the form of the geodetic height difference between the upper level of water in the reservoir and the turbines in the machine house of a lower altitude. This height difference is denoted as hydraulic head in the terminology of hydropower. The conversion of the potential energy into the usable mechanical energy is completed by first converting the potential energy into kinetic energy in the form of high-speed jets at the altitude of the turbine wheel. For the energy conversion, one or many injectors can be used. By neglecting the friction losses in the injector, the jet speed is calculated according to the Bernoulli equation by

$$C_0 = \sqrt{2gH}, \tag{2.1}$$

with H as the net pressure head at the inlet of the injector. This equation is generally called the Torricelli formula.

As second step, the conversion of the kinetic energy of the jet into the mechanical energy is accomplished by the interaction between the jet and the rotating buckets of the Pelton turbine. As a working principle for simplicity, a straight translating bucket of constant speed U is first considered (Fig. 2.1). This assumption of straight movement means that during the interaction between the jet and the bucket, only the *impulsive force* is effective.

The interaction between the water jet and the bucket is considered directly in the relative moving system. For the flow at the bucket entry (index 1) and with $C_1 = C_0$, the relative velocity between the jet and moving bucket is given by

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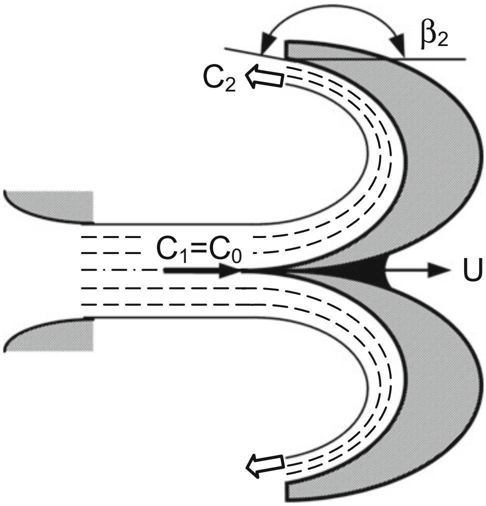
Fig. 2.1 Flow interaction and energy conversion between the jet and a straight-moving bucket



$$W_1 = C_1 - U. \tag{2.2}$$

With this relative velocity, the jet flow spreads in the bucket, forming a water sheet. The change in the direction of the flow along the bucket surface is coupled with a pressure increase below the water sheet as related to the impulsive force and determined by the law of momentum. On the surface of the water sheet where the atmospheric pressure is constant, the flow velocity is equal to W_1 provided that frictionless flow is assumed. The pressure as well as the velocity distribution across the thickness of the water sheet will be considered in more details in Sects. 6.1.2 and 7.3 by considering the relative flow in a rotating Pelton bucket.

Once the water flow reaches and then leaves the bucket exit (index 2) at an angle β_2, it is again subjected to atmospheric pressure. The relative velocity of the total water flow is then reset to its initial value according to Eq. (2.2), i.e., $W_2 = W_1 = W$. The absolute velocity is thus given by

$$C_2^2 = U^2 + W^2 + 2UW\cos\beta_2. \tag{2.3}$$

According to the balance law of momentum, the change of the flow direction is always related to an external impulsive force which acts perpendicular to the flow direction. This force is nothing else than the pressure below the water sheet. For its determination the *momentum flux* difference between the entry and the exit of the moving bucket must be evaluated. The component of the total force in the direction of the bucket motion is calculated by the following momentum balance equation:

$$F_{\mathrm{bucket}} = \dot{m}_{\mathrm{w}}(W_1 - W_2\cos\beta_2) = \dot{m}_{\mathrm{w}}W(1 - \cos\beta_2). \tag{2.4}$$

F_{bucket} denotes the force exerted by the water on the bucket. Moreover, $\dot{m}_{\text{w}} = \rho W A_{\text{jet}}$ is the total mass flow of water in the relative frame of the moving bucket. It is related to the mass flow rate $\dot{m}_{\text{c}} = \rho C_0 A_{\text{jet}}$ in the absolute frame by the relation

$$\dot{m}_{\text{w}} = \frac{W}{C_0} \cdot \dot{m}_{\text{c}}. \tag{2.5}$$

This equation states that because of $\kappa = C_0/W > 1$, a jet piece leaving the injector nozzle within one second will need κ seconds in order to completely reach and enter the moving bucket. The factor κ can therefore also be understood as a time factor. For this reason, the impulsive force exerted on the bucket, as given in Eq. (2.4), may be rewritten as

$$F_{\text{bucket}} = \dot{m}_{\text{c}} \frac{W^2}{C_0} (1 - \cos \beta_2). \tag{2.6}$$

The power, received by the bucket, is thus calculated as

$$P = F_{\text{bucket}} U = \dot{m}_{\text{c}} \frac{W^2}{C_0} (1 - \cos \beta_2) \cdot U. \tag{2.7}$$

Although the condition for maximum power output can be calculated from $\mathrm{d}P/\mathrm{d}U = 0$ leading to $U/C_0 = 1/3$, this condition, however, does not represent the condition for the maximum conversion of the kinetic energy stored in the jet into mechanical energy of the moving bucket. To reveal the energy conversion process, the specific energy (J/kg) of the jet must be taken into consideration. Therefore, a unit mass of water (1 kg) is assumed to flow out of the injector nozzle within the time $t_{\text{c}} = 1/\dot{m}_{\text{c}}$. This mass of fluid will then need a time of $t_{\text{c}}\kappa$ to completely reach and enter the moving bucket. The specific work, done by its interaction with the moving bucket, is given by

$$e = P \cdot t_{\text{c}} \kappa. \tag{2.8}$$

With Eq. (2.7) as well as $\kappa = C_0/W$ and $\dot{m}_{\text{c}} t_{\text{c}} = 1$ one thus obtains

$$e = UW(1 - \cos \beta_2). \tag{2.9}$$

The maximum specific work done by a unit mass of the fluid is then obtained by setting $\mathrm{d}e/\mathrm{d}U = 0$. With $W = C_0 - U$ it follows

$$\frac{U}{C_0} = 0.5. \tag{2.10}$$

Such a speed ratio represents the condition under which Pelton turbine operations theoretically should always be configured. The specific work done by the unit mass of the jet flow is then obtained from Eq. (2.9) to

$$e = \frac{1}{4}C_0^2(1 - \cos\beta_2). \tag{2.11}$$

The exit velocity of water out of the bucket results from Eq. (2.3) as

$$C_2^2 = \frac{1}{2}C_0^2(1 + \cos\beta_2). \tag{2.12}$$

From Eq. (2.11) it is evident that the maximum specific work is obtained when the flow angle at the bucket exit is configured to $\beta_2 = 180°$. It then follows

$$e = \frac{1}{2}C_0^2. \tag{2.13}$$

It is obviously equal to the specific kinetic energy which is available in the jet. The exit velocity is obtained from Eq. (2.12) as

$$C_2 = 0. \tag{2.14}$$

This means that the total energy stored in the jet is entirely transferred to the moving bucket.

In practical design of Pelton turbines, the exit velocity C_2 cannot be zero because water, after leaving the bucket, has to fly away from the bucket to make the way free for the following buckets. As a consequence, the flow angle for the exit flow has usually been configured to be $\beta_2 \approx 170°$. The kinetic energy corresponding to the exit velocity $C_2 \neq 0$ and thus remains unexploited and must be regarded as a loss. In practice, it is often referred to as the exit or the *swirling loss*.

The model shown in Fig. 2.1 is a hydraulic model at which for inviscid fluids the exit loss represents the only loss in the model system. The hydraulic efficiency is then defined as the ratio of the specific work to the specific kinetic energy in the jet. From Eq. (2.9), with $W = C_0 - U$, this is calculated to be

$$\eta_h = \frac{e}{C_0^2/2} = 2 \cdot \left(1 - \frac{U}{C_0}\right)\frac{U}{C_0}(1 - \cos\beta_2), \tag{2.15}$$

or, with $k = U/C_0$,

$$\eta_h = 2k(1 - k)(1 - \cos\beta_2). \tag{2.16}$$

In Fig. 2.2, this hydraulic efficiency is illustrated as a function of the velocity ratio k and for the exit angle $\beta_2 = 180°$. The maximum hydraulic efficiency is obviously obtained at $k = 0.5$,

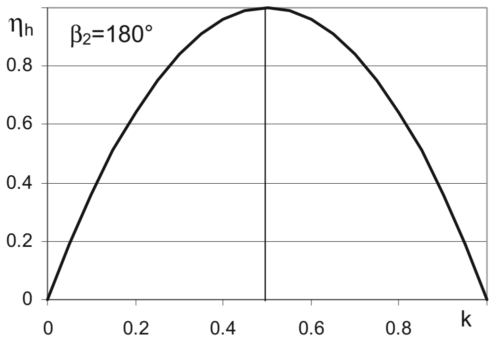


Fig. 2.2 Efficiency characteristic of the flow interaction system according to Fig. 2.1 with $\beta_2 = 180°$

$$\eta_{\mathrm{h,\,max}} = 0.5(1 - \cos \beta_2). \tag{2.17}$$

When the viscous friction loss in the flow of the water sheet cannot be neglected, the calculation of the hydraulic efficiency then has to be modified accordingly. This will be described in detail in Chaps. 10, 11, and 15.

2.2 Pelton Turbines and Specifications

A Pelton turbine essentially consists of one or more injectors for generating the high-speed jet and a wheel with buckets for receiving the jet energy (Fig. 2.3). An injector must primarily perform two tasks. Firstly, the injector nozzle converts the pressure energy of the water into the kinetic energy of the high-speed jet. Secondly, the injector regulates the flow rate via a built-in needle which is driven by a servomotor. The power exchange is finally achieved by the interaction between the jet and the Pelton buckets. Because of the rotation of the Pelton wheel, both the centrifugal and Coriolis forces are influencing the flow. The form of the flow and its distribution within the bucket therefore differ fundamentally from those in the straight-moving bucket. The basic principle of energy conversion, as described in Sect. 2.1, also applies to the Pelton turbines. For Pelton turbines, however, both the design and the flow parameters need to be specially specified as described in the following sections.

2.2.1 Geometric Specification of the Pelton Wheel

According to Fig. 2.4, a Pelton wheel is mainly configured by the following parameters:

Jet circle (also called pitch circle) diameter $D_{\mathrm{m}} = 2R_{\mathrm{m}}$
Wheel bucket inner diameter $D_{\mathrm{b}} = 2R_{\mathrm{b}}$

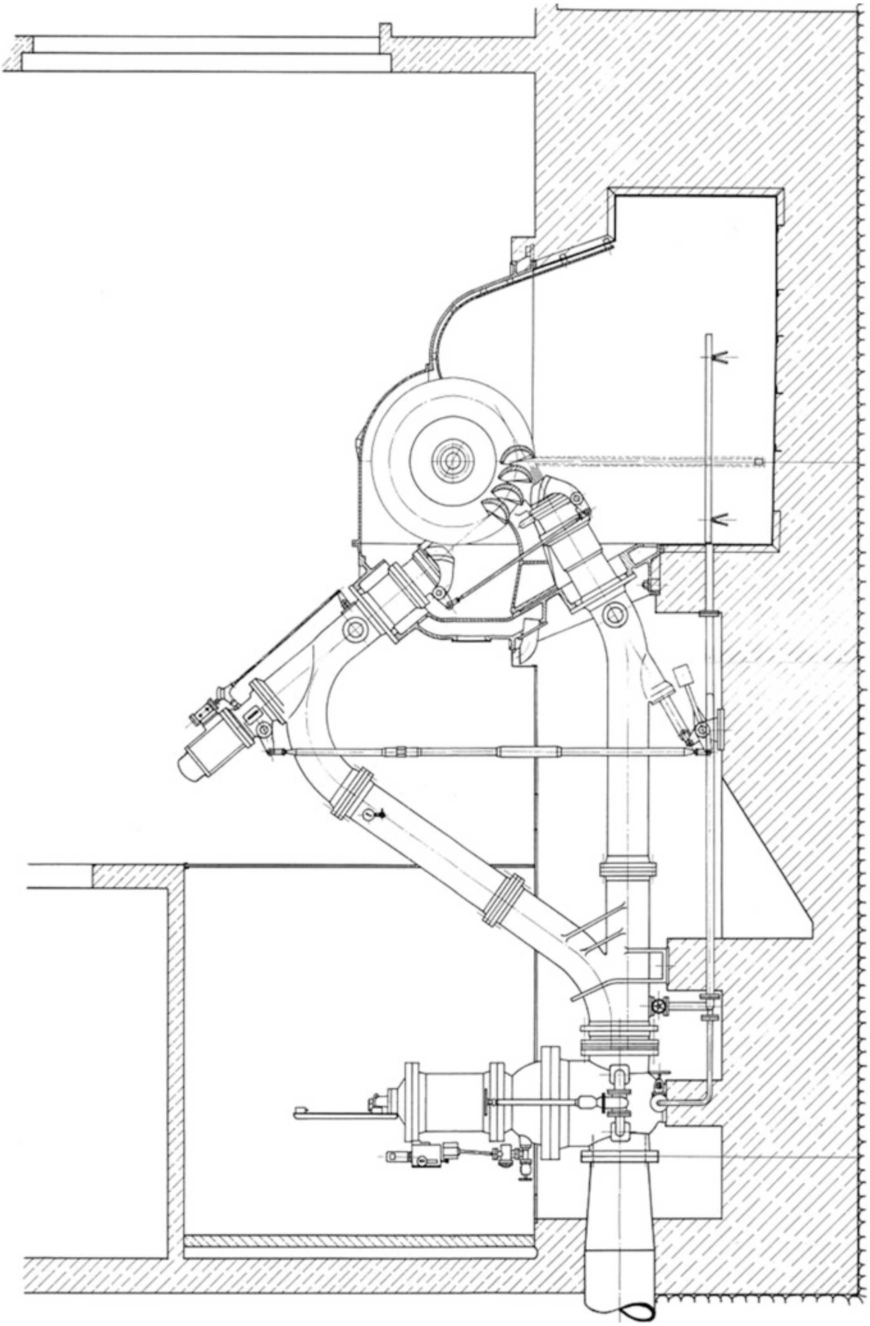


Fig. 2.3 Pelton turbine with two injectors at a hydropower plant in Kleintal

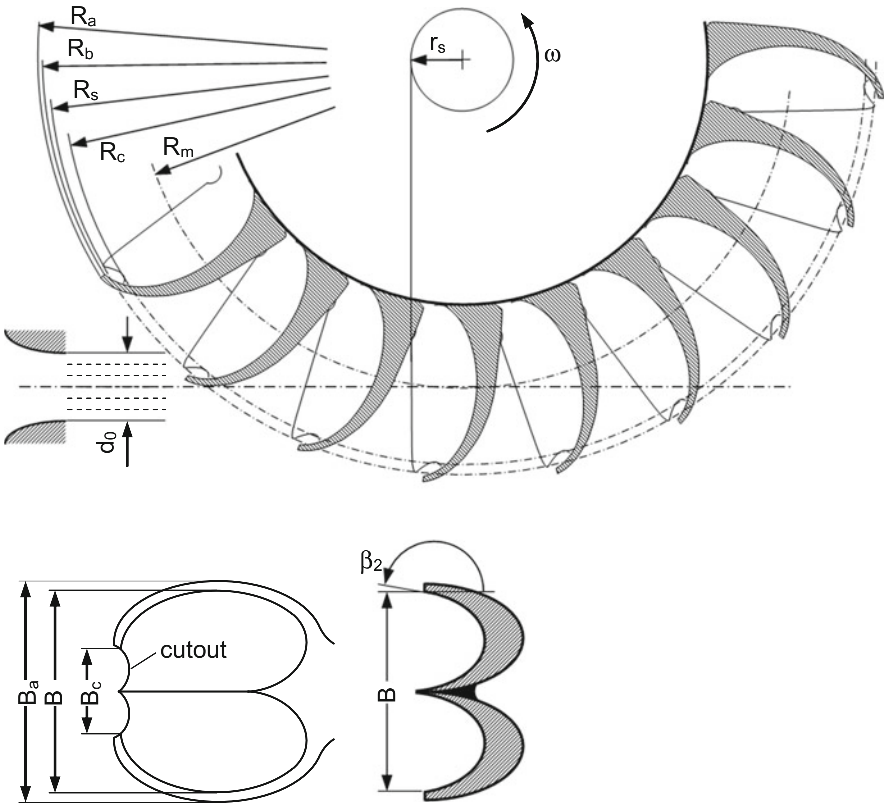


Fig. 2.4 Geometric parameter specification of a Pelton wheel

Wheel diameter $D_\mathrm{a} = 2R_\mathrm{a}$
Tip circle diameter of the main splitter $D_\mathrm{s} = 2R_\mathrm{s}$
Circle diameter of the bucket cutout edge $D_\mathrm{c} = 2R_\mathrm{c}$
Number of buckets N
Bucket inner width B
Bucket exit angle β_2
Base circle radius of the main splitter r_s

The basic design of a Pelton wheel relies on the flow specification in the turbine system. The dimensioning of the Pelton wheel additionally depends on the permissible rotational speed of the generator. Additional details can be found in Chap. 18. The bucket exit angle β_2 is determined under certain circumstances by the required exit flow conditions (Chap. 8), while the optimum bucket number will be derived from *coincidence* and *symmetry conditions* (Chap. 5).

2.2.2 *Characteristic Hydromechanical Parameters*

In the field of turbomachinery, diverse dimensionless numbers function as parametric quantities which are often used to specify the size of the machines and to quantify the flow performances. For Pelton turbines, however, only few of them are relevant for the geometric and hydraulic design as well as for the turbine operations. The most important parametric quantities are summarized in this section.

2.2.2.1 Peripheral Speed Coefficient k_{m}

The *peripheral speed coefficient* is defined as the ratio of the peripheral speed U_{m} of the Pelton wheel on the jet circle (R_{m} in Fig. 2.4) to the jet speed C_0

$$k_{\mathrm{m}} = \frac{U_{\mathrm{m}}}{C_0} = \frac{U_{\mathrm{m}}}{\sqrt{2gH}}. \tag{2.18}$$

The peripheral speed coefficient is one of the most important parameters in the design of Pelton turbines. It has the same meaning as the parameter k in Eq. (2.16) and represents the hydraulic efficiency in a similar form as that in Fig. 2.2. In practice, the peripheral speed coefficient has so far always been set in the range 0.45–0.48, at which the maximum possible hydraulic efficiency can be achieved.

In general considerations of turbomachinery, the pressure head coefficient, defined as $\psi = Y/\left(U_{\mathrm{m}}^2/2\right)$, is often used, where $Y = gH$ is the specific work of water sources per unit mass. It can be shown that there exists the following relation between the peripheral speed coefficient and the pressure head coefficient

$$k_{\mathrm{m}} = \sqrt{\frac{1}{\psi}}. \tag{2.19}$$

For this reason, the head coefficient in Pelton turbine flows will not be used.

2.2.2.2 Bucket Volumetric Load φ_{B}

The jet thickness relative to the bucket width represents the dimensionless *bucket volumetric load*.[1] Because the flow rate is proportional to the square of the jet thickness, the bucket volumetric load can also be represented by the flow rate of a single injector ($\dot{Q}_{\mathrm{jet}}$). It is thus defined by

[1] In Chap. 23, the bucket mechanical load will be evaluated with respect to the jet impact force and the material strength of the bucket.

$$\varphi_{\mathrm{B}} = \frac{\dot{Q}_{\mathrm{jet}}}{\pi/4 \cdot B^2 \sqrt{2gH}}, \tag{2.20}$$

with B as the bucket inner width.

The flow rate of a single injector is given by $\dot{Q}_{\mathrm{jet}} = \pi/4 \cdot d_0^2 \sqrt{2gH}$, with d_0 as the jet diameter. Consequently, the *bucket volumetric load* is given by

$$\varphi_{\mathrm{B}} = \left(\frac{d_0}{B}\right)^2. \tag{2.21}$$

It is simply expressed by the ratio of the jet thickness to the bucket width. Because of its geometric feature, the bucket volumetric load expressed by Eq. (2.21) is much more comprehensible than that of Eq. (2.20). For this reason, Eq. (2.21) will be preferably used in this book.

The bucket volumetric load is used on the one hand to represent the flow rate in a dimensionless form and on the other hand to determine the necessary width of the Pelton buckets. The bucket width is usually designed so that at nominal or maximum flow rate, the jet diameter d_0 does not exceed one-third of the bucket width B. This yields $\varphi_{\mathrm{B}} = 0.09$ to 0.11 as design criterion for the bucket width.

2.2.2.3 Specific Speed n_{q}

The *specific speed* is a parametric quantity which has been widely used in all types of rotating fluid machinery. In the specification of Pelton wheels, the specific speed has a special explicit meaning. As directly taken over from the technical literature, for instance Pfleiderer and Petermann (1986), the specific speed is defined by

$$n_{\mathrm{q}} = n \frac{\sqrt{\dot{Q}_{\mathrm{jet}}}}{H^{3/4}}. \tag{2.22}$$

It is obviously not dimensionless, nor does it have the same dimension as the rotational speed n. To avoid confusion in applications, $\dot{Q}_{\mathrm{jet}}$ and H in the above equation can be considered to be normalized by the unit flow rate $\dot{Q}_{\mathrm{jet}} = 1\,\mathrm{m}^3/\mathrm{s}$ and the unit pressure head $H = 1\,\mathrm{m}$, respectively. Then n and n_{q} have the same unit, either 1/s or 1/min. In the present work, the specific speed n_{q} is mainly used with the unit 1/s.

As an alternative to the specific speed defined in Eq. (2.22), the following definition can also be found in the literature:

$$n_{\mathrm{y}} = n\frac{\sqrt{\dot{Q}_{\mathrm{jet}}}}{(gH)^{3/4}} = n\frac{\sqrt{\dot{Q}_{\mathrm{jet}}}}{Y^{3/4}}. \tag{2.23}$$

In this definition, the unit of the rotational speed is 1/s.

Between the two definitions of the specific speed, the following relation exists:

$$n_{\mathrm{q}} = g^{3/4} n_{\mathrm{y}} = 5.54 n_{\mathrm{y}} (1/\mathrm{s}). \tag{2.24}$$

One obtains further by using the unit 1/min for the speed,

$$n_{\mathrm{q}} = 333 n_{\mathrm{y}} (1/\mathrm{min}). \tag{2.25}$$

The specific speed is primarily used when for a given flow rate and a given pressure head, a Pelton turbine should be defined by specifying the injector number, the rotational speed, and the wheel dimension. The exact computational algorithm for the design of a Pelton turbine by using the specific speed is presented in detail in Chap. 18. For practical engineering applications, only the specific speed according to Eq. (2.22) is used in this book.

It should also be noted that the specific speed indeed represents the diameter ratio $\delta = D_{\mathrm{m}}/d_0$ which is a dimensionless quantity and called the diameter number (Sigloch 2006). This can be confirmed using Eq. (2.22) by considering the flow rate $\dot{Q}_{\mathrm{jet}} = \frac{1}{4}\pi d_0^2 \sqrt{2gH}$ and the peripheral speed coefficient according to Eq. (2.18). One obtains

$$n_{\mathrm{q}} = g^{3/4} \frac{k_{\mathrm{m}}}{2^{1/4}\sqrt{\pi}} \frac{d_0}{D_{\mathrm{m}}} = 2.63 k_{\mathrm{m}} \frac{d_0}{D_{\mathrm{m}}} (1/\mathrm{s}). \tag{2.26}$$

Here, the significance of the specific speed is clearly explained. Since the peripheral speed coefficient k_{m} of Pelton turbines is practically a constant, the specific speed represents exclusively the diameter ratio d_0/D_{m} and is therefore essentially a geometric parameter. With respect to the bucket volumetric load according to Eq. (2.21), the specific speed is also interpreted as

$$n_{\mathrm{q}} = \frac{(2g)^{3/4}}{2\sqrt{\pi}} k_{\mathrm{m}} \sqrt{\varphi_{\mathrm{B}}} \frac{B}{D_{\mathrm{m}}} = 2.63 k_{\mathrm{m}} \sqrt{\varphi_{\mathrm{B}}} \frac{B}{D_{\mathrm{m}}}. \tag{2.27}$$

According to this equation and under nominal operations, at which $\varphi_{\mathrm{B}} \approx 0.11$, the specific speed also specifies the geometric design parameter (B/D_{m}) of the Pelton wheel.

As the specific speed is, according to its definition in Eq. (2.22), directly determinable from the flow rate and the pressure head, it is a particularly convenient parameter for the design of a Pelton turbine from the given $\dot{Q}_{\mathrm{jet}}$ and H. For this reason, the diameter number $\delta = D_{\mathrm{m}}/d_0$ will not be used in this book.

2.2.2.4 Characteristic Bucket Position Angle α_o

The bucket position angle α_o as shown in Fig. 2.5 represents an angle at which the cutout of the bucket just intersects the jet layer on the jet axis. This angle has a special meaning because it directly determines the so-called runaway speed of the Pelton turbine. Whereas the computation of the runaway speed will be treated in Chap. 17, the properties of this special bucket position angle and its connection with the specific speed should be explained here.

The Pelton bucket geometries are usually similar from turbine to turbine. Especially, the ratio of the bucket length to the bucket width is between 0.8 and 0.9. If, on this basis, the difference $D_c - D_m = 0.85B$ is applied, it follows from Fig. 2.5 that

$$\cos\alpha_o = \frac{R_m}{R_c} = \frac{1}{1 + 0.85B/D_m}. \tag{2.28}$$

To replace the term B/D_m, Eq. (2.27) is used. This results in

$$\cos\alpha_o = \frac{k_m\sqrt{\varphi_B}}{k_m\sqrt{\varphi_B} + 0.32n_q}. \tag{2.29}$$

For nominal operations (index N) on average with $k_{m,N} = 0.47$ and $\varphi_{B,N} = 0.11$, it follows further that

$$\cos\alpha_o = \frac{1}{1 + 2n_q}. \tag{2.30}$$

The application of this *characteristic bucket position angle* α_o will be demonstrated in detail in Chap. 16 for calculating the real efficiency curve and in Chap. 17 for determining the runaway speed of the Pelton wheel.

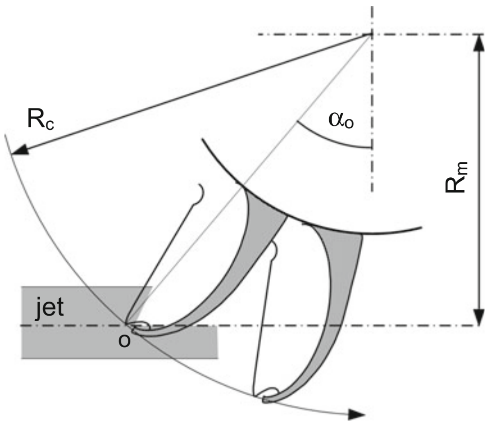


Fig. 2.5 Geometric relation of the bucket position angle α_o

2.2.2.5 Peripheral Speed of the Bucket Cutout Edge

Another frequently used reference speed in flow calculations is the *peripheral speed* of the cutting edge of the bucket cutout. From Eqs. (2.28) and (2.30), one first obtains the diameter ratio

$$\frac{D_\mathrm{c}}{D_\mathrm{m}} = 1 + 2n_\mathrm{q}. \tag{2.31}$$

The ratio of the corresponding peripheral speed to the jet speed is, in view of Eq. (2.18), given by

$$\frac{U_\mathrm{c}}{C_0} = \frac{U_\mathrm{m}}{C_0}\frac{U_\mathrm{c}}{U_\mathrm{m}} = k_\mathrm{m}\frac{D_\mathrm{c}}{D_\mathrm{m}}. \tag{2.32}$$

With Eq. (2.31) it follows further that

$$\frac{U_\mathrm{c}}{C_0} = k_\mathrm{m}\left(1 + 2n_\mathrm{q}\right). \tag{2.33}$$

2.2.3 *Hydromechanical Specification of the Pelton Turbine*

The main operating parameters of a Pelton turbine are the *peripheral speed coefficient* k_m and the *bucket volumetric load* φ_B, while the specific speed only determines the shape of the Pelton wheel and is thus related to the nominal operation. On the one hand, both parameters, k_m and φ_B, describe the hydraulic similarity between two Pelton wheels with similar geometric designs and therefore the same specific speed (see Chap. 20). On the other hand, they together determine the flow mechanical efficiency of a Pelton turbine. With regard to the maximum efficiency, the operation point of a Pelton turbine is commonly configured by the peripheral speed coefficient $k_\mathrm{m} = 0.45{\sim}0.48$ and the bucket volumetric load $\varphi_\mathrm{B} = 0.09{\sim}0.11$.

To describe the flow mechanical interaction between the jet and the rotating buckets of a Pelton turbine, basically the same calculation is used as described in Sect. 2.1 for the straight-moving bucket. The relative flow velocity at the bucket entry is assumed to be $W_1 = C_0 - U_\mathrm{m}$. Analogous to Eq. (2.6), the interaction, i.e., the impulsive force exerted on a bucket, is obtained as

$$F_\mathrm{bucket} = \dot{m}_\mathrm{c} C_0 (1 - k_\mathrm{m})^2 (1 - \cos\beta_2). \tag{2.34}$$

Contrary to the case of the straight-moving bucket, each jet in a Pelton turbine with rotating buckets simultaneously interacts with about two buckets. More accurately, the number of buckets interacting with one jet is given by

$$2\lambda = \frac{\dot{m}_{\mathrm{c}}}{\dot{m}_{\mathrm{w}}} = \frac{C_0}{W_1}. \tag{2.35}$$

Here, λ denotes the *multi-bucket factor*. Its application will be explained in detail in Chap. 5. Accordingly, the total impulsive force resulting from one jet is given by

$$F_{\mathrm{jet}} = 2\lambda F_{\mathrm{bucket}} = \dot{m}_{\mathrm{c}} \frac{C_0^2}{W_1} (1 - k_{\mathrm{m}})^2 (1 - \cos\beta_2). \tag{2.36}$$

Because $W_1 = (C_0 - U_{\mathrm{m}}) = C_0(1 - k_{\mathrm{m}})$, there follows

$$F_{\mathrm{jet}} = \dot{m}_{\mathrm{c}} C_0 (1 - k_{\mathrm{m}})(1 - \cos\beta_2). \tag{2.37}$$

The power exchange achieved by one jet flow is thus

$$P = F_{\mathrm{jet}} U_{\mathrm{m}} = \dot{m}_{\mathrm{c}} C_0^2 k_{\mathrm{m}} (1 - k_{\mathrm{m}})(1 - \cos\beta_2). \tag{2.38}$$

The maximum power achieved is given at $k_{\mathrm{m,max}}$ which is obtained by the condition $\mathrm{d}P/\mathrm{d}k_{\mathrm{m}} = 0$ and obtains

$$k_{\mathrm{m,max}} = 0.5. \tag{2.39}$$

From Eq. (2.38) the hydraulic efficiency thus results as

$$\eta_{\mathrm{h}} = \frac{P}{\frac{1}{2}\dot{m}_{\mathrm{c}} C_0^2} = 2k_{\mathrm{m}} (1 - k_{\mathrm{m}})(1 - \cos\beta_2). \tag{2.40}$$

Formally this expression agrees with Eq. (2.16). The reason for this agreement is that perpendicular entry of the jet into the bucket was assumed and thus the relation $W_1 = C_0 - U_{\mathrm{m}}$ was used. Therefore, Eq. (2.40) can be considered to be directly taken over from Eq. (2.16). Because of the assumption $W_1 = C_0 - U_{\mathrm{m}}$, it, therefore, only applies to illustrate the working principle of a Pelton turbine and to roughly estimate the hydraulic efficiency. In particular, the speed ratio $\kappa = C_0/W_1$, which in Eq. (2.5) was referred to as a time factor, stands here for the number of buckets that simultaneously interact with a jet. In Chaps. 5 and 7, this factor will be replaced by the *multi-bucket factor* $\lambda = \kappa/2$ that is computed using other, different reasoning.

In fact, all equations derived so far for F_{jet}, P, and η_{h} only represent the operational principle of a Pelton turbine. Both the jet impact force and the power exchange in a Pelton turbine with rotating buckets behave somewhat differently than those in a nonrotating, i.e., linearly translating bucket. The interaction between

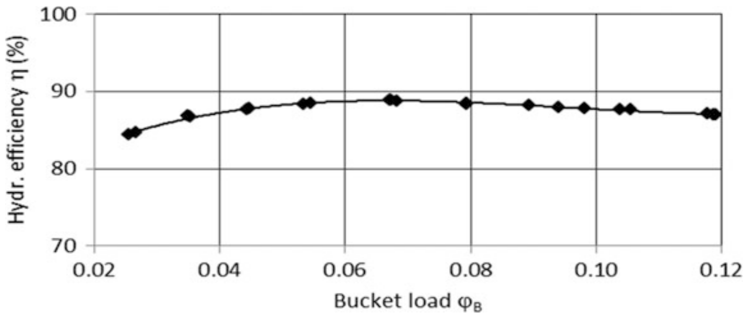


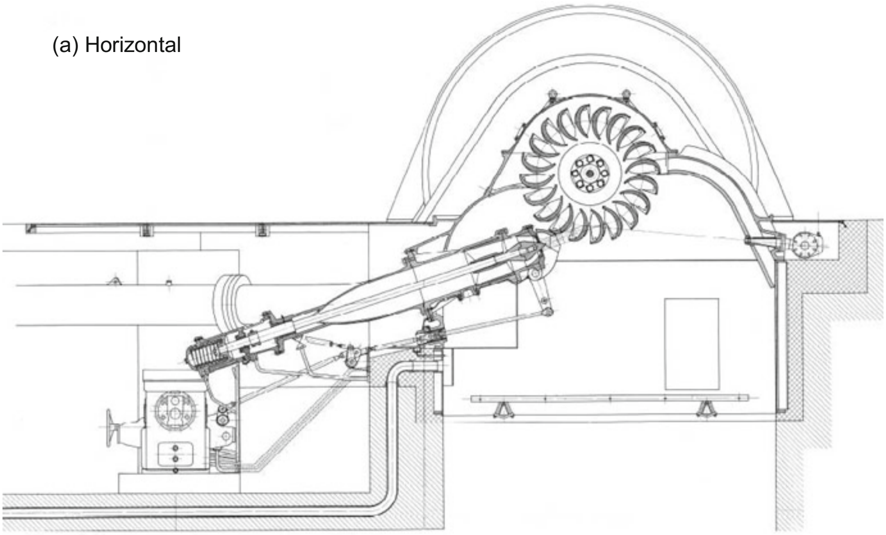
Fig. 2.6 Hydraulic efficiency of a Pelton turbine (KWO) plotted against the bucket volumetric load

the jet and the rotating buckets is no longer constant but varies with time. The consideration of a linearly translating bucket in Sect. 2.1 has led to the speed ratio $U/C_0 = 0.5$ at which the maximum hydromechanical performance is achieved. In practical operations of Pelton turbines, however, the nominal peripheral speed coefficient $k_{m,N}$ for maximum efficiency is found to be between 0.45 and 0.48. Here, particular attention should be focused on the fact that the water, after the energy exchange with the rotating buckets, still possesses sufficient kinetic energy to be able to leave the buckets in time. The associated loss is called the *exit* or *swirling loss*. The full term of the hydraulic efficiency will be presented in Chap. 15 once all individual hydromechanical losses, including the viscous friction effect, have been treated.

The hydraulic efficiency according to Eq. (2.40) has been shown to be a function of the peripheral speed coefficient and theoretically reaches its maximum at $k_{m,\max} = 0.5$. On the other hand, the hydraulic efficiency of a Pelton turbine also depends on the bucket volumetric load φ_B. Figure 2.6 shows such a dependence of a Pelton turbine obtained by measurements. Usually, the nominal flow rate of a Pelton turbine is designed with a bucket load of about $\varphi_{B,N} = 0.1$. Most Pelton turbines, however, show maximum efficiencies at the flow rate below that value, as this is also confirmed in Fig. 2.6. The basic reason for this intention is that nearly all Pelton turbines also operate at partial loads. As can be recognized, the hydraulic efficiency of a Pelton turbine only insignificantly changes with bucket load when compared with other types of turbines. This is the reason why the Pelton turbines are often used to balance the load of the network. The perceptible decrease of the hydraulic efficiency at a partial load is ascribed to friction effects; see Sect. 10.4 as well as Sect. 11.4.

2.2.4 Installation Form of Pelton Turbines

The practical installation forms of Pelton turbines have been categorized by the orientation of the turbine axis. The turbines with horizontal axes are denoted horizontal turbines (Fig. 2.7a) and those with vertical axes are called vertical



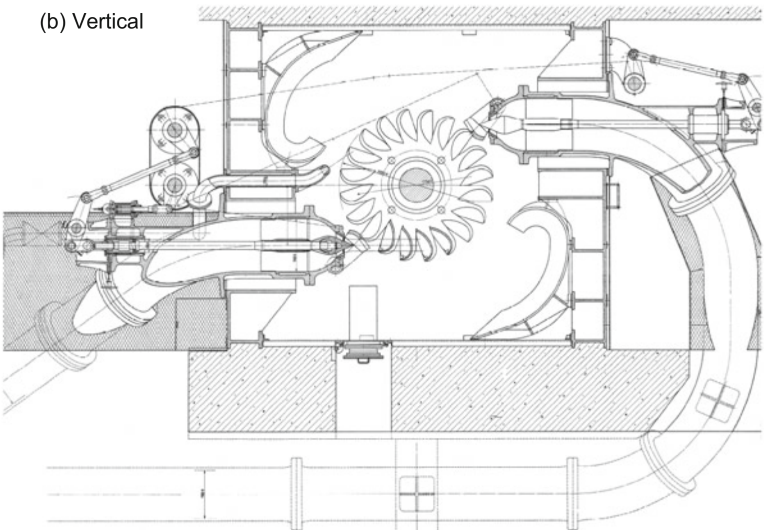


Fig. 2.7 Different installation forms of Pelton turbines at the Oberhasli Hydroelectric Power Company (KWO). (**a**) Horizontal, (**b**) vertical

turbines (Fig. 2.7b), respectively. The horizontal installation is only suitable for turbines with at most two injectors. Vertical turbines can be designed with up to six injectors. The significant advantage of the vertical installation is that the injectors can be and are distributed symmetrically around the wheel. For Pelton turbines with one injector or in all cases of turbines of horizontal installation, the destructive one-sided bearing load is inevitable. For turbines with two or more injectors, care should be taken that no collisions between two jets could occur in the same bucket. The offset angle between two adjacent injectors must be sufficiently large to ensure trouble-free interaction between the jet and the rotating buckets, as well as trouble-free exit flow of water out of the buckets. The relevant criterion will be elaborated in Chap. 19. In the design of vertical turbines, one has to ensure that water after leaving the upper bucket halves should not fall back on the wheel. The relevant criterion is explained in Chap. 8.

2.2.5 Parameter Notations

In addition to the geometric parameters of a Pelton wheel that have already been shown in Fig. 2.4, complete geometric and hydraulic parameters of both the injector and the Pelton wheel are summarized in Appendix A. Other derived parametric and dimensionless quantities are summarized in Appendix B.

References

Pfleiderer, C., & Petermann, H. (1986). *Strömungsmaschinen* (5. Auflage). Springer.
Sigloch, H. (2006). *Strömungsmaschinen*. (3. Auflage). Hanser Verlag.

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