CHAPTER 5

Performance Tests

Introduction

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

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

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

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

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

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

5.1 Tests on prototype

5.1.1 Principles for test

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

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

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

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

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

5.1.2 Measurement of the turbine power

Power output P

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

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

The available power P_n

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

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

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

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

5.1.3 Methods for determination of discharge

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

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

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

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

5.1.3.1 Current meter method

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

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

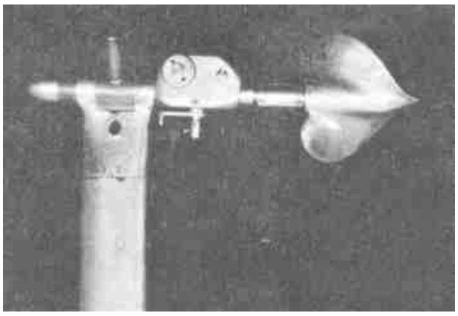

Fig. 5.1 Current meter /3/

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

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

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

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

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

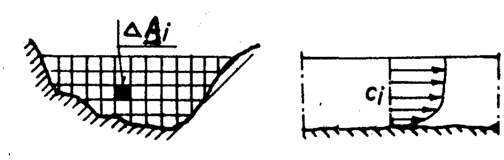

Fig. 5.2 Flow cross section /3/

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

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

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

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

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

5.1.3.2 Pitot tube gauging

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

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

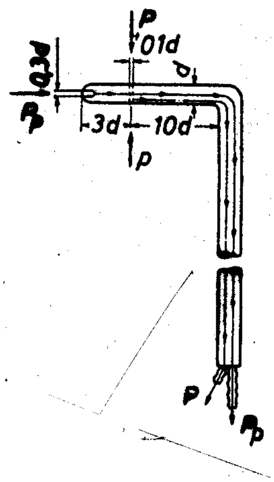

Fig. 5.3 Prandtl tube /3/

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

The local velocity v_i is given by:

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

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

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

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

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

5.1.3.3 The pressure-time method (Gibson method)

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

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

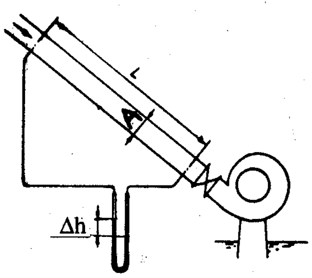

Fig. 5.4 Measurement of differential pressure in a pipeline /3/

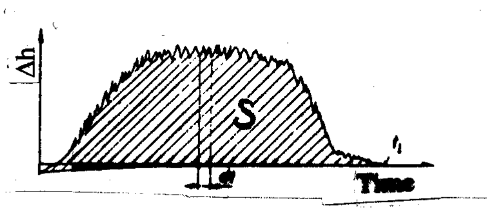

Fig. 5.5 Pressure differential-time-diagram /3/

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

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

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

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

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

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

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

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

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

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

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

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

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

5.2.3.4 Tracer methods

Allen salt velocity method

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

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

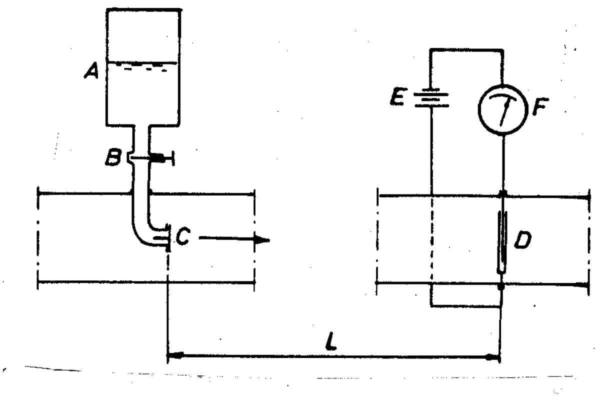

Fig. 5.6 Allen salt velocity method /6/

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

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

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

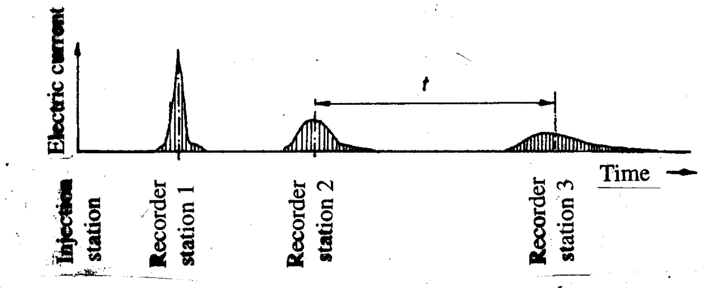

Fig. 5.7 Record of a salt cloud passing the electrode stations in a pipeline /6/

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

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

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

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

Salt dilution method

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

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

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

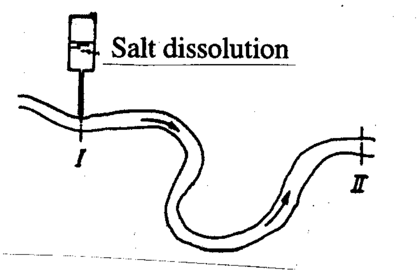

Fig. 5.8 Dilution method /5/

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

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

The discharge Q can be determined from:

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

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

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

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

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

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

5.1.3.5 Ultrasonic method

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

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

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

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

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

where L is the distance between transmitter and receiver.

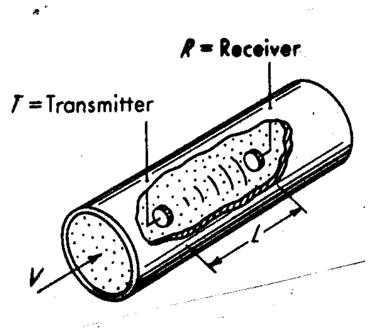

Fig. 5.9 Principle of ultrasonic signalling /2/

The velocity

a is the acoustic (sound) velocity in the fluid

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

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

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

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

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

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

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

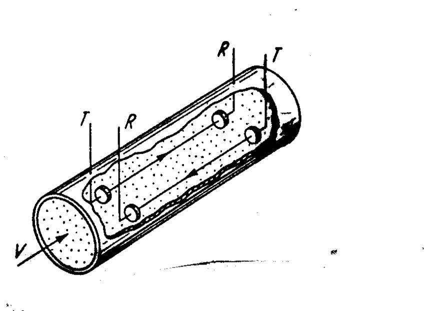

Fig. 5.10a Sound signal sent with and against the flow direction /2/

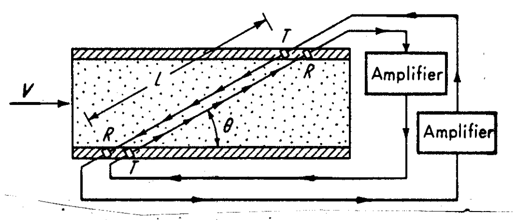

Fig. 5.10b Two self-excited oscillated systems /2/

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

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

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

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

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

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

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

5.1.3.6 Weirs

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

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

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

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

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

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

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

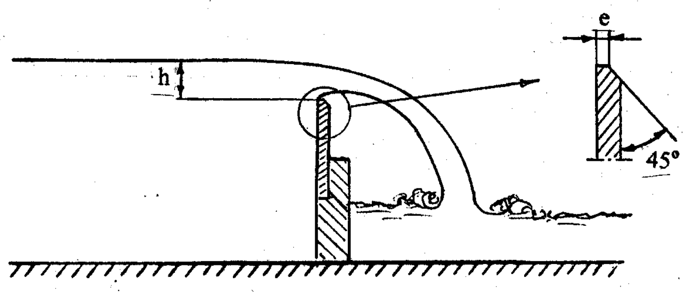

Fig. 5.11 Sketch of a sharp-crested rectangular weir

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

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

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

5.1.3.7 Standardised differential pressure devices

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

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

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

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

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

5.1.3.8 Volumetric gauging method

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

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

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

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

5.1.3.9 Relative discharge measurement

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

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

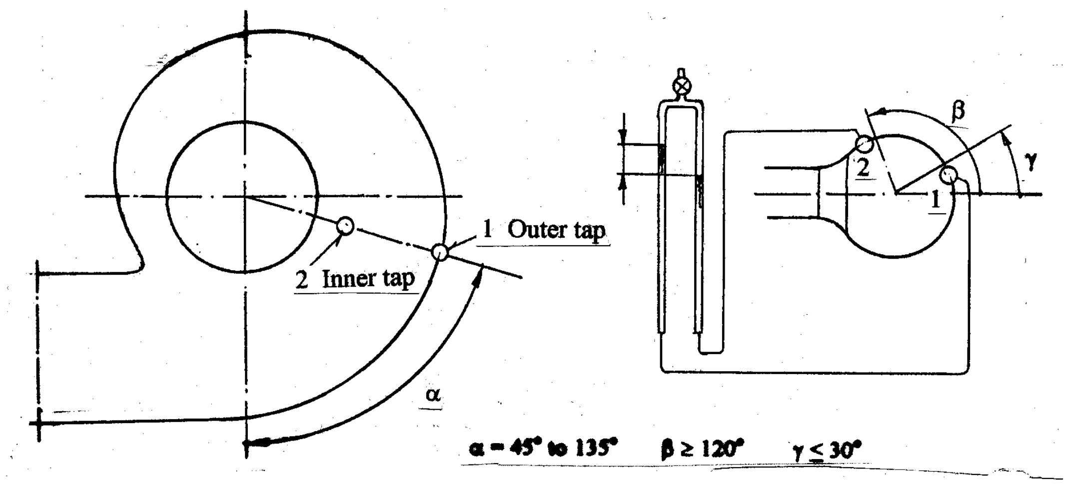

Fig. 5.12 Winter-Kennedy measurement

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

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

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

5.1.4 Thermodynamic measurement of flow losses

5.1.4.1 Measurement of power losses

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

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

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

Measuring equipment

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

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

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

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

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

5.1.4.2 Efficiency and specific energies

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

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

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

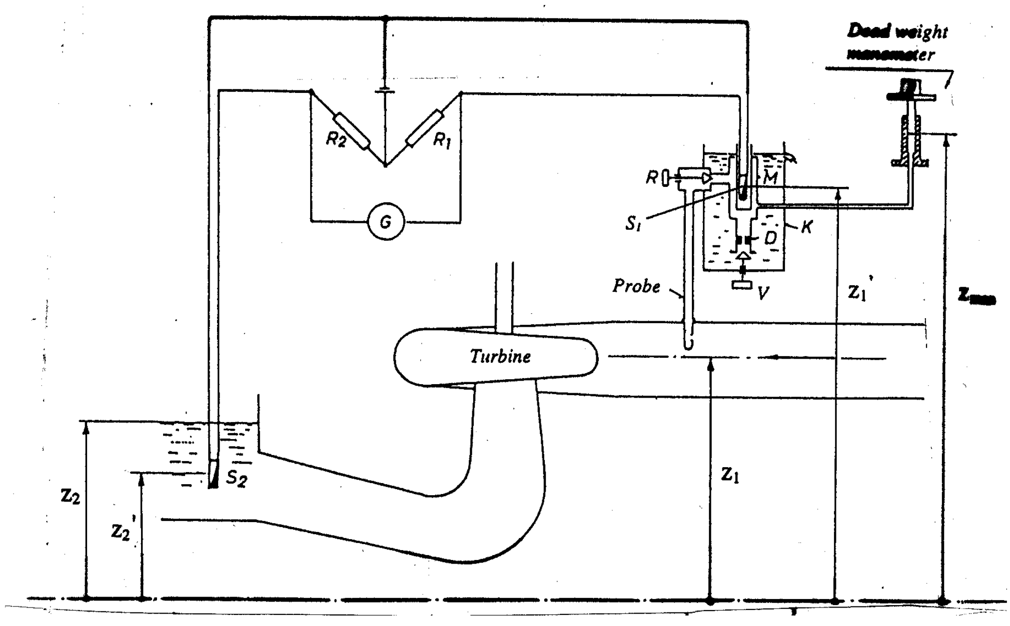

Fig. 5.13 Arrangement of thermodynamic measurements on a turbine /3/

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

On this basis and with reference to Fig. 5.13:

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

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

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

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

The specific mechanical energy at the runner

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

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

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

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

or

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

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

From the thermodynamics

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

and the specific heat at constant pressure

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

Therefore the differential of the enthalpy can be converted in

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

and the isothermal factor of water is then:

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

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

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

5.1.4.3 Measuring technique

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

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

Determination of the specific available hydraulic energy E_n

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

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

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

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

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

Determination of the specific mechanical energy E_R

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

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

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

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

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

By these measurements and observations the specific mechanical energy becomes

$$\begin{aligned} E_R &= a(p_{abs.z_1'} - p_{abs.z_2'}) + c_p(\theta_1 - \theta_2) - \frac{c_2^2}{2} + g(z_1' - z_2') = \\ &= a\left[p_{abs.man} + \rho g(z_{man} - z_2)\right] + g(z'_1 - z'_2)(1 - a\rho) + c_p(\theta_1 - \theta_2) - \frac{c_2^2}{2} \end{aligned}$$

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

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

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

5.1.4.4 Corrections for leakage and friction

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

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

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

where the turbine discharge is

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

Finally the turbine efficiency becomes

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

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

5.1.5 Dynamic properties of the turbines

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

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

5.1.6 Cavitation behaviour of prototype

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

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

5.1.7 Governor test – Rejection tests

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

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

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

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

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

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

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

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

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

5.2.1 Laboratory qualifications

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

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

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

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

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

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

5.2.2 Model tests

Required size and surface roughness of the models

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

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

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

The definitions of the Reynolds numbers are:

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

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

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

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

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

Testing procedure

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

Accuracy of model tests

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

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

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

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

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

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

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

Cavitation tests on model

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

According to these outlines the Thoma parameter for the model:

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

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

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

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

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

Runaway test

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

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

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

5.2.3 Scale effect on efficiency from model to prototype

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

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

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

where $\Delta\eta$ is the difference in efficiency of the prototype and the model

η_m is the efficiency of the model turbine

R_{em} is the Reynolds number of the model

R_{ep} is the Reynolds number of the prototype

V is the scaleable part of the losses. $V \approx 0.7$ according to IEC code[/10/]. However, it is proven that $V = f({}^{*}\underline{\Omega})$

α is exponent, estimated $\alpha = 0.16$

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

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

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