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Hydraulic turbines, storage pumps and pump-turbines – Model acceptance tests

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HYDRAULIC TURBINES, STORAGE PUMPS AND PUMP-TURBINES – MODEL ACCEPTANCE TESTS

FOREWORD

The IEC (International Electrotechnical Commission) is a worldwide organization for standardization comprising all national electrotechnical committees (IEC National Committees). The object of the IEC is to promote international co-operation on all questions concerning standardization in the electrical and electronic fields. To this end and in addition to other activities, the IEC publishes International Standards. Their preparation is entrusted to technical committees; any IEC National Committee interested in the subject dealt with may participate in this preparatory work. International, governmental and non-governmental organizations liaising with the IEC also participate in this preparation. The IEC collaborates closely with the International Organization for Standardization (ISO) in accordance with conditions determined by agreement between the two organizations.

The formal decisions or agreements of the IEC on technical matters express, as nearly as possible, an international consensus of opinion on the relevant subjects since each technical committee has representation from all interested National Committees.

The documents produced have the form of recommendations for international use and are published in the form of standards, technical reports or guides and they are accepted by the National Committees in that sense.

In order to promote international unification, IEC National Committees undertake to apply IEC International Standards transparently to the maximum extent possible in their national and regional standards. Any divergence between the IEC Standard and the corresponding national or regional standard shall be clearly indicated in the latter.

The IEC provides no marking procedure to indicate its approval and cannot be rendered responsible for any equipment declared to be in conformity with one of its standards.

Attention is drawn to the possibility that some of the elements of this International Standard may be the subject of patent rights. The IEC shall not be held responsible for identifying any or all such patent rights.

ternational Standard IEC 60193 has been prepared by IEC technical committee 4: Hydraulic rbines.

ıis second edition of IEC 60193 cancels and replaces the first edition of IEC 60193 published 1965, its amendment 1 (1977), IEC 60193A (1972), as well as IEC 60497 (1976) and :C 60995 (1991).

lauses 1 to 3 of this standard cover the scopes dealt with in the above-mentioned ıblications. Additional information is given in clause 4.

he text of this standard is based on the following documents:

FDIS	Report on voting
4/157/FDIS	4/162/RVD

ull information on the voting for the approval of this standard can be found in the report on oting indicated in the above table.

ınnexes B, F, G, K, L and M form an integral part of this standard.

ınnexes A, C, D, E, H, J, N and P are for information only.

'he committee has decided that this publication remains valid until 2004. At this date, in ccordance with the committee's decision, the publication will be

reconfirmed;

withdrawn;

replaced by a revised edition, or

amended.

HYDRAULIC TURBINES, STORAGE PUMPS AND PUMP-TURBINES – MODEL ACCEPTANCE TESTS

1 General rules

1.1 Scope and object

1.1.1 Scope

This International Standard applies to laboratory models of any type of impulse or reaction hydraulic turbine, storage pump or pump-turbine.

This standard applies to models of prototype machines either with unit power greater than 5 MW or with reference diameter greater than 3 m. Full application of the procedures herein prescribed is not generally justified for machines with smaller power and size. Nevertheless, this standard may be used for such machines by agreement between purchaser and supplier.

In this standard, the term "turbine" includes a pump-turbine operating as a turbine and the term "pump" includes a pump-turbine operating as a pump.

This standard excludes all matters of purely commercial interest, except those inextricably bound up with the conduct of the tests.

This standard is concerned with neither the structural details of the machines nor the mechanical properties of their components, so long as these do not affect model performance or the relationship between model and prototype performances.

1.1.2 Object

This International Standard covers the arrangements for model acceptance tests to be performed on hydraulic turbines, storage pumps and pump-turbines to determine if the main hydraulic performance contract guarantees (see 1.4.2) have been satisfied.

It contains the rules governing test conduct and prescribes measures to be taken if any phase of the tests is disputed.

The main objectives of this standard are:

- to define the terms and quantities used;
- to specify methods of testing and of measuring the quantities involved, in order to ascertain the hydraulic performance of the model;
- to specify the methods of computation of results and of comparison with guarantees;
- to determine if the contract guarantees, which fall within the scope of this standard, have been fulfilled;
- to define the extent, content and structure of the final report.

The guarantees can be given in one of the following ways:

- guarantees for prototype hydraulic performance, computed from model test results considering scale effects;
- guarantees for model hydraulic performance.

eover additional performance data (see 1.4.4) can be needed for the design or the ration of the prototype of the hydraulic machine. Contrary to the requirements of clauses 1 ; related to main hydraulic performance the information of these additional data given in ise 4 is considered only as recommendation or guidance to the user (see 4.1).

particularly recommended that model acceptance tests be performed if the expected field ditions for acceptance tests (see IEC 60041) would not allow the verification of guarantees in for the prototype machine.

; standard may also be applied to model tests for other purposes, i.e. comparative tests and earch and development work.

iodel acceptance tests have been performed, field tests can be limited to index tests (see 60041, clause 15).

contradiction is found between this standard and any other standard, this standard shall vail.

Normative references

following normative documents contain provisions which, through reference in this text, stitute provisions of this International Standard. At the time of publication, the editions cated were valid. All normative documents are subject to revision, and parties to eements based on this International Standard are encouraged to investigate the possibility applying the most recent editions of the normative documents indicated below. Members of ; and ISO maintain registers of currently valid International Standards.

; 60041:1991, *Field acceptance test to determine the hydraulic performance of hydraulic bines, storage pumps and pump-turbines*

; 60609:1978, *Cavitation pitting evaluation in hydraulic turbines, storage pumps and pump-bines*

; 60609-2:1997, *Cavitation pitting evaluation in hydraulic turbines, storage pumps and mp-turbines – Part 2: Evaluation in Pelton turbines*

; 60994:1991, *Guide for field measurement of vibrations and pulsations in hydraulic ichines (turbines, storage pumps and pump-turbines)*

; 61364:1999, *Nomenclature of hydraulic machinery*

; 61366 (all parts), *Hydraulic turbines storage pumps and pump-turbines – Tendering cuments*

O 31-3:1992, *Quantities and units – Part 3: Mechanics*

O 31-12:1992, *Quantities and units – Part 12: Characteristic numbers*

O 468:1982, *Surface roughness – Parameters, their values and general rules for specifying quirements*

O 1438-1:1980, *Water flow measurement in open channels using weirs and Venturi flumes – art 1: Thin-plate weirs*

ISO 2186:1973, *Fluid flow in closed conduits – Connections for pressure signal transmissions between primary and secondary elements*

ISO 2533:1975, *Standard atmosphere*
Addendum 1: 1985

ISO 4006:1991, *Measurement of fluid flow in closed conduits – Vocabulary and symbols*

ISO 4185:1980, *Measurement of liquid flow in closed conduits – Weighing method*

ISO 4373:1995, *Measurement of liquid flow in open channels – Water level measuring devices*

ISO 5167-1:1991, *Measurement of fluid flow by means of pressure differential devices – Part 1: Orifice plates, nozzles and Venturi tubes inserted in circular cross-section conduits running full*

ISO 5168:1978, *Measurement of fluid flow – Estimation of uncertainty of a flow-rate measurement*

ISO 6817:1992, *Measurement of conductive liquid flow in closed conduits – Method using electromagnetic flowmeters*

ISO 7066-1:1997, *Assessment of uncertainty in the calibration and use of flow measurement devices – Part 1: Linear calibration relationship*

ISO 7066-2:1988, *Assessment of uncertainty in the calibration and use of flow measurement devices – Part 2: Non-linear calibration relationships*

ISO 8316: 1987, *Measurement of liquid flow in closed conduits – Method by collection of the liquid in a volumetric tank*

ISO 9104:1991, *Measurement of fluid flow in closed conduits – Methods of evaluating the performance of electromagnetic flow-meters for liquids*

VIM:1993, *International vocabulary of basic and general terms in metrology (BIPM-IEC-ISO-OIML)*

1.3 Terms, definitions, symbols and units

1.3.1 General

For the purpose of this International Standard the following common terms, definitions, symbols and units apply. Specialized terms are explained where they appear.

Clarification of any term, definition or unit of measure in question shall be agreed to in writing by the contracting parties in advance of the test.

1.3.1.1 point
A *point* is established by one or more consecutive sets of readings and/or recordings at unchanged operating condition and settings, sufficient to calculate the performance of the machine at this operating condition and these settings

1.3.1.2 test
a *test* comprises a collection of points and results adequate to establish the performance of the machine over a specified range of operating conditions

.1.3 **hydraulic performance**
erformance parameters attributable to the machine due to hydrodynamic effects

.1.4 **main hydraulic performance data**
ubset of the hydraulic performance parameters, i.e. power, discharge and/or specific raulic energy, efficiency, steady-state runaway speed and/or discharge. The influence of itation must be considered.

.1.5 **additional data**
ubset of hydraulic performance data which can be determined for information on the model e 1.4.4). However, the prediction of the corresponding prototype data is less accurate than t achievable for the main hydraulic performance data, due to application of approximate ilarity rules.

.1.6 **guarantees**
cified performance data contractually agreed to

.2 Units

e international System of Units (SI, see ISO 31-3) has been used throughout this standard.

terms are given in SI base units or derived coherent units[1]. The basic equations are valid ng these units. This has to be taken into account if other than coherent SI units are used for tain data (e.g. kilowatt instead of watt for power, kilopascal or bar instead of pascal for ssure, min^{-1} instead of s^{-1} for rotational speed, etc.). Temperatures may be given in degrees lsius since absolute temperatures (in kelvins) are rarely required.

y other system of units may be used but only if agreed in writing by the contracting parties.

$N = kg \cdot m \cdot s^{-2}$ $Pa = kg \cdot m^{-1} \cdot s^{-2}$ $J = kg \cdot m^2 \cdot s^{-2}$ $W = kg \cdot m^2 \cdot s^{-3}$

1.3.3 List of terms, definitions, symbols and units

1.3.3.1 Subscripts and symbols

Subclause	Term	Definition	Subscript or symbol
1.3.3.1.1	High pressure [1] reference section	The high pressure section of the machine to which the performance guarantees refer (see figure 1)	1
1.3.3.1.2	Low pressure [1] reference section	The low pressure section of the machine to which the performance guarantees refer (see figure 1)	2
1.3.3.1.3	High pressure measuring sections	Whenever possible, these sections should coincide with section 1; otherwise the measured values must be adjusted to section 1 (see 3.5.2.1.3)	1',1" ..
1.3.3.1.4	Low pressure measuring sections	Whenever possible, these sections should coincide with section 2; otherwise, the measured values must be adjusted to section 2 (see 3.5.2.1.3)	2', 2" ..
1.3.3.1.5	Specified	Subscript denoting values of quantities such as rotational speed, discharge etc. for which other quantities are guaranteed	sp
1.3.3.1.6	Maximum/minimum	Subscript denoting maximum or minimum values of any term	max min
1.3.3.1.7	Limits	Contractually defined values: – not to be exceeded – to be reached	$\underline{////}$ □ or $\overline{////}$
1.3.3.1.8	Prototype	Subscript denoting values related to the full size machine	P
1.3.3.1.9	Model	Subscript denoting values related to the model	M
1.3.3.1.10	Model at constant Reynolds number	Subscript denoting values related to a model and referred to a constant value of Reynolds number	M*
1.3.3.1.11	Reference	Subscript denoting values related to a specified reference condition	ref
1.3.3.1.12	Optimum	Subscript denoting the best efficiency point	opt
1.3.3.1.13	Ambient	Subscript referring to surrounding atmospheric conditions	amb
1.3.3.1.14	Plant	Subscript denoting values related to the operating conditions of the prototype in the plant	pl
1.3.3.1.15	Runaway	Subscript referring to runaway conditions	R

[1] The terms "high pressure" and "low pressure" define the two sides of the machine irrespective of the flow direction and therefore are independent of the mode of operation of the machine.

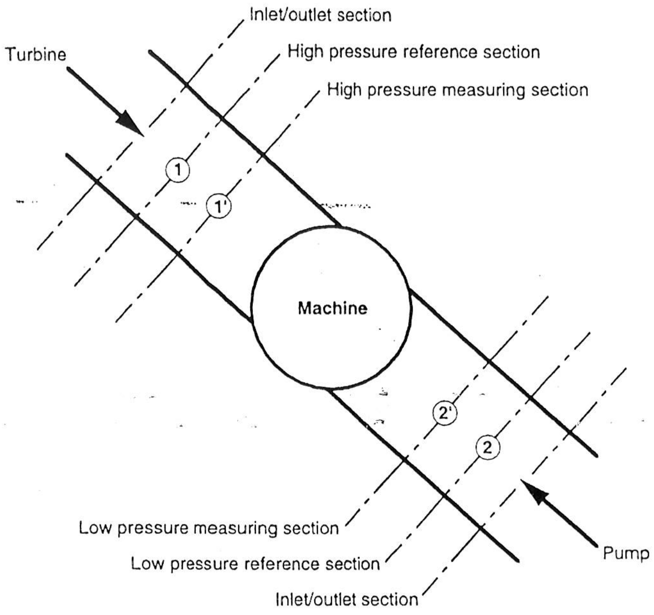
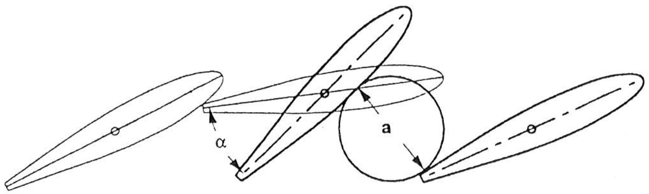


Figure 1 – Schematic representation of a hydraulic machine

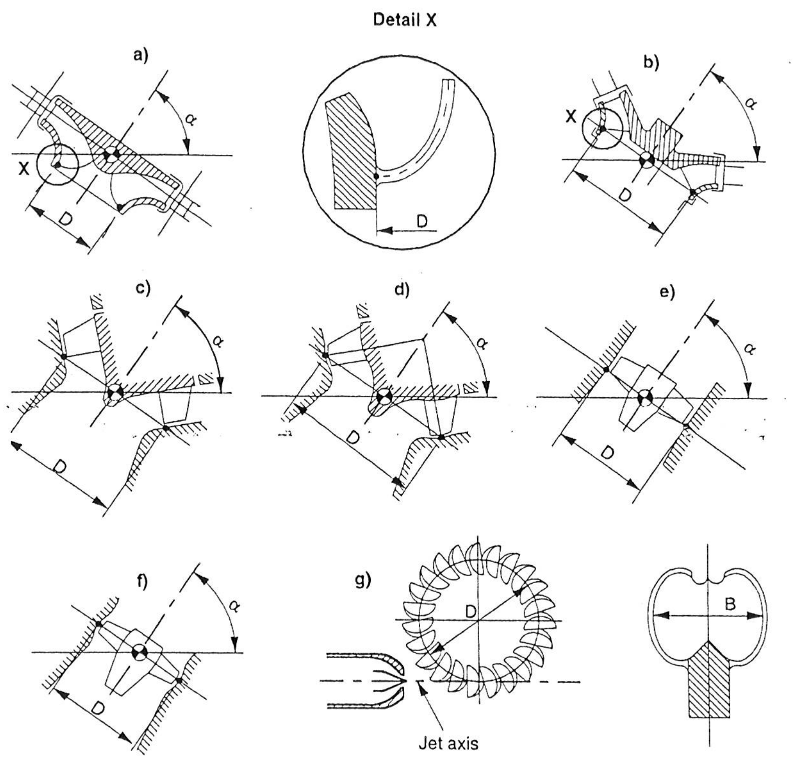


Closed position: $\alpha = 0^\circ$ or a = 0 mm

Figure 2 – Guide vane opening and angle

1.3.3.2 Geometric terms

Subclause	Term	Definition	Symbol	Unit
1.3.3.2.1	Area	Net cross sectional area normal to general flow direction	A	m^2
1.3.3.2.2	Guide vane opening	Average shortest distance between adjacent guide vanes (at a specified section if necessary) (see figure 2)	a	m
1.3.3.2.3	Guide vane angle	Average vane angle measured from closed position (see figure 2)	α	°
1.3.3.2.4	Needle stroke (impulse turbine)	Average needle stroke measured from closed position	s	m
1.3.3.2.5	Runner/impeller blade angle	Average runner/impeller blade setting measured from a reference position	β	°
1.3.3.2.6	Reference diameter	Reference diameter of the hydraulic machine as given in figure 3	D	m
1.3.3.2.7	Runner outlet/impeller inlet width	Average shortest distance between two adjacent blades of runner/impeller (see figures 14 and 15)	a_1, a_2	m
1.3.3.2.8	Bucket width	Inside maximum width of runner bucket of a Pelton turbine (see figure 3)	B	m
1.3.3.2.9	Length scale ratio	The ratio of representative prototype to model lengths; in normal cases this is the reference diameter of the machine. In cases where it is difficult to verify this reference, then another significant length may be taken	λ_L	-
1.3.3.2.10	Level	Elevation of a point in the system above the specified reference datum (usually mean sea level)	z	m



0° ≤ α ≤ 90°

Radial machines, such as Francis turbines, radial (centrifugal) pumps and pump-turbines; for multistage ıchines: low-pressure stage.

Diagonal (mixed-flow, semi-axial) machines with fixed runner/impeller blades and with runner/impeller band.

Diagonal (mixed-flow, semi-axial) machines with fixed runner/impeller blades, without runner/impeller band.

Diagonal (mixed-flow, semi-axial) machines with adjustable runner/impeller blades.

Axial machines, such as propeller turbines, tubular turbines[1], axial pumps and pump-turbines with fixed nner/impeller blades.

Axial machines, such as Kaplan turbines, tubular turbines[1], axial pumps and pump-turbines with adjustable nner/impeller blades.

Pelton turbines.

Figure 3 – Reference diameter and bucket width

The term "tubular turbines" includes bulb, pit, rim generator and S-type units.

1.3.3.3 Physical quantities and properties

Subclause	Term	Definition	Symbol	Unit
1.3.3.3.1	Acceleration due to gravity	Local value of gravitational acceleration at the place of testing (see 2.5.2); theoretical values as a function of altitude and latitude are given in annex B, table B.1	g	m s^{-2}
1.3.3.3.2	Temperature	Thermodynamic Celsius $\theta = \Theta - 273{,}15$	Θ θ	K °C
1.3.3.3.3	Density	Mass per unit volume a) Density of water ρ is commonly used instead of ρ_w. Values for distilled water ρ_{wd} are given in 2.5.3.1.3 and in annex B, table B.2 b) Density of air Values for air are given in 2.5.4.1 and in annex B, table B.5 c) Density of mercury Values for mercury are given in 2.5.5 and in annex B, table B.7	ρ ρ_w ρ_a ρ_{Hg}	kg m^{-3} kg m^{-3} kg m^{-3} kg m^{-3}
1.3.3.3.4	Vapour pressure (absolute)	Absolute partial pressure of saturated vapour in a medium where liquid and gaseous phases of a body are in thermodynamic balance. The vapour pressure depends only on the temperature. Values for distilled water are given in 2.5.3.4 and in annex B, table B.4	p_{va}	Pa
1.3.3.3.5	Dynamic viscosity	A quantity characterizing the mechanical behaviour of a fluid (see ISO 31-3)	μ	Pa s
1.3.3.3.6	Kinematic viscosity	Ratio of the dynamic viscosity to the density of a fluid. Values for distilled water as a function of temperature are given in 2.5.3.3 and in annex B, table B.3	ν	m^2 s^{-1}
1.3.3.3.7	Surface tension	A quantity characterizing the mechanical behaviour of the interface between two fluids (see ISO 31-3)	σ*	J·m^{-2}

.4 Discharge, velocity and speed terms

lause	Term	Definition	Symbol	Unit
4.1	Discharge (volume flow rate)	Volume of water per unit time passing through any section in the system	Q	$m^3 s^{-1}$
4.2	Mass flow rate	Mass of water flowing through any section of the system per unit time. Both ρ and Q shall be determined at the same section and at the conditions existing in that section NOTE – The mass flow rate is constant between two sections if no water is added or removed	(ρQ)	$kg\ s^{-1}$
4.3	Measured discharge	Volume of water per unit time flowing through any measuring section, e.g. 1' (see 1.3.3.1.3 and 1.3.3.1.4)	$Q_{1'}$ or $Q_{2'}$	$m^3 s^{-1}$
4.4	Discharge at reference section	Volume of water per unit time flowing through the reference section 1 or 2	Q_1 or Q_2	$m^3 s^{-1}$
4.5	Corrected discharge at reference section	Volume of water per unit time flowing through the reference section referred to the ambient condition (see 1.3.3.5.2), e.g. $Q_{1c} = (\rho Q)_1/\rho_{amb}$ Given the normal conditions of a model test, Q_{1c} may be assumed equal to Q_1	Q_{1c} or Q_{2c}	$m^3 s^{-1}$
.4.6	Discharge at steady-state runaway speed	Discharge at n_R (see 1.3.3.4.12)	Q_R	$m^3 s^{-1}$
.4.7	No-load turbine discharge	Turbine discharge at zero mechanical power at specified speed (usually synchronous) and specified specific hydraulic energy	Q_o	$m^3 s^{-1}$
.4.8	Leakage flowrate	Volumetric loss as illustrated in figure 6	q	$m^3 s^{-1}$
.4.9	Mean velocity	Discharge Q divided by area A of the cross-section	v	$m\ s^{-1}$
.4.10	Peripheral velocity	Peripheral velocity at the reference diameter (see figure 3): $u = \pi Dn$	u	$m\ s^{-1}$
.4.11	Rotational speed	Number of revolutions per unit time	n	s^{-1}
.4.12	Steady-state runaway speed	The steady-state rotational speed at zero mechanical power at specified hydraulic conditions and specified guide vane/blade/needle opening	n_R	s^{-1}
.4.13	Maximum steady-state runaway speed	The highest value of steady-state runaway speed at specified hydraulic conditions (for the prototype, see detailed definition in IEC 60041)	n_{Rmax}	s^{-1}

1.3.3.5 Pressure terms

Subclause	Term	Definition	Symbol	Unit
1.3.3.5.1	Absolute pressure	The static pressure of a fluid measured with reference to a perfect vacuum	p_{abs}	Pa
1.3.3.5.2	Ambient pressure	The absolute pressure of the ambient air (see 2.5.4.2). Values for standard atmosphere are given as a function of elevation in annex B, table B.6	p_{amb}	Pa
1.3.3.5.3	Gauge pressure	The difference between the absolute static pressure of a fluid at the reference level of the pressure measuring instrument and the ambient pressure at the place and time of measurement, $p = p_{abs} - p_{amb}$	p	Pa

1.3.3.6 Specific energy terms

In the International System of Units, mass (kg) is one of the base quantities. The energy per unit mass, known as specific energy, is used in this standard as a primary term instead of the energy per local unit weight which is called head, and was exclusively used in previous publications.

The latter term (head) has the disadvantage that weight is a force which depends on the local value of acceleration due to gravity g, which changes mainly with latitude but also with altitude. Nevertheless, the term "head" will still remain in use because it is very common. Therefore, both related energy terms are listed, the specific energy terms in this subclause and head terms in 1.3.3.7. They differ only by the factor g, which is the local value of acceleration due to gravity.

Clause	Term	Definition	Symbol	Unit
6.1	Specific energy	The energy per unit mass of water at any section	e	J kg^{-1}
6.2	Specific hydraulic energy of machine	Specific energy of water available between the high and low pressure reference sections 1 and 2 of the machine, taking into account the influence of compressibility [1) 2)] $E = \frac{p_{abs\,1} - p_{abs\,2}}{\overline{\rho}} + \frac{v_1^2 - v_2^2}{2} + (z_1 - z_2)\,g$ with $\overline{\rho} = \frac{\rho_1 + \rho_2}{2}$ and assuming $g = g_1 = g_2$ The value of ρ_1 and ρ_2 can be calculated from p_{abs1} and p_{abs2} respectively, taking into account θ_1 or θ_2 for both values, given the negligible influence of the difference of the temperature on ρ	E	J kg^{-1}
.6.3	Zero-discharge (shut-off) specific hydraulic energy of the pump	Pump specific energy at specified speed and specified guide vane and impeller blade settings with high pressure side shut off	E_o	J kg^{-1}
.6.4	Suction specific potential energy of the machine	Specific potential energy at section 2 corresponding to the difference between the reference level of the machine (see 1.3.3.7.6) and the piezometric level at section 2 $E_s = g\,(z_r - z_{2'}) = g\,(z_r - z_2) - \frac{p_{abs\,2} - p_{amb}}{\rho_2}$ (see figure 45)	E_s	J kg^{-1}
3.6.5	Net positive suction specific energy	Absolute specific energy at section 2 minus the specific energy due to the vapour pressure p_{va} [3)] referred to the reference level of the machine (see figure 45) $NPSE = \frac{p_{abs\,2} - p_{va}}{\rho_2} + \frac{v_2^2}{2} - g\,(z_r - z_2)$ $= -E_s + \frac{p_{abs2} - p_{va}}{\rho_2} + \frac{v_2^2}{2}$	NPSE	J kg^{-1}

Subclause 3.5 illustrates some common cases of application of the basic formula for the specific hydraulic energy.

For derivation of E, see annex C.

See 1.3.3.3.4.

Subclause	Term	Definition	Symbol	Unit
1.3.3.6.6	Thoma number	Dimensionless term indicating the conditions of cavitation under which the machine operates. It is expressed as the ratio of net positive suction specific energy NPSE to a specific hydraulic energy E (see 1.3.3.12.9)	σ	-
1.3.3.6.7	Cavitation coefficient	Dimensionless term indicating the conditions of cavitation under which the machine operates. It is expressed as the ratio of net positive suction specific energy NPSE to n^2D^2 (see 1.3.3.12.10)	σ_{nD}	-
1.3.3.6.8	Thoma number zero [1)]	The lowest value of the Thoma number for which a chosen performance parameter (usually efficiency) remains unchanged as compared to its values at high Thoma number. In some cases, the shape of the cavitation curve η_h (σ) is such that the Thoma number zero is difficult to define (see figure 4)	σ_o	-
1.3.3.6.9	Thoma number one [1)]	The value of the Thoma number for which a drop of one percentage point in efficiency is obtained compared with the efficiency at Thoma number zero. In some cases, the shape of the cavitation curve is such that the Thoma number one is difficult to define (see figure 4)	σ_1	-
1.3.3.6.10	Defined Thoma number [1)]	The value of the Thoma number associated with a defined onset of cavitation, e.g. a specified performance loss	σ_d	-
1.3.3.6.11	Incipient Thoma number[1)]	The value of the Thoma number associated with the beginning of visible runner/impeller cavitation usually detected by observation	σ_i	-
1.3.3.6.12	Plant Thoma number	The value of the Thoma number at the operating conditions of the prototype (see annex M)	σ_{pl}	-
1.3.3.6.13	Specific hydraulic energy loss	The specific hydraulic energy dissipated between any two sections	E_L	J kg^{-1}

1) A similar definition can be given for cavitation coefficient (see figure 4).

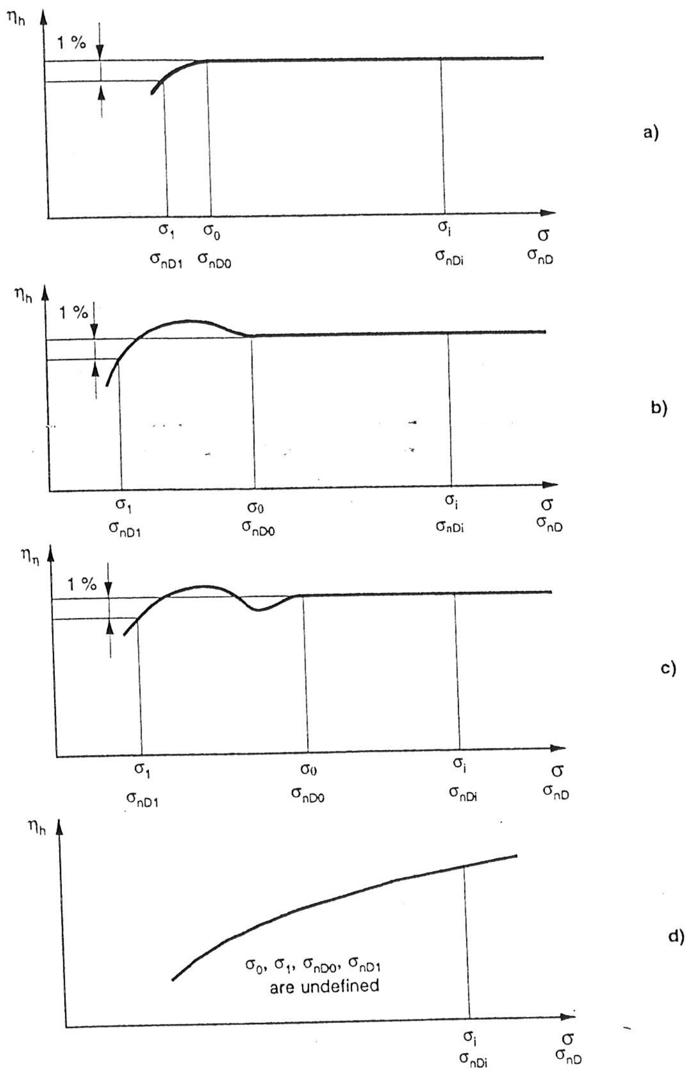
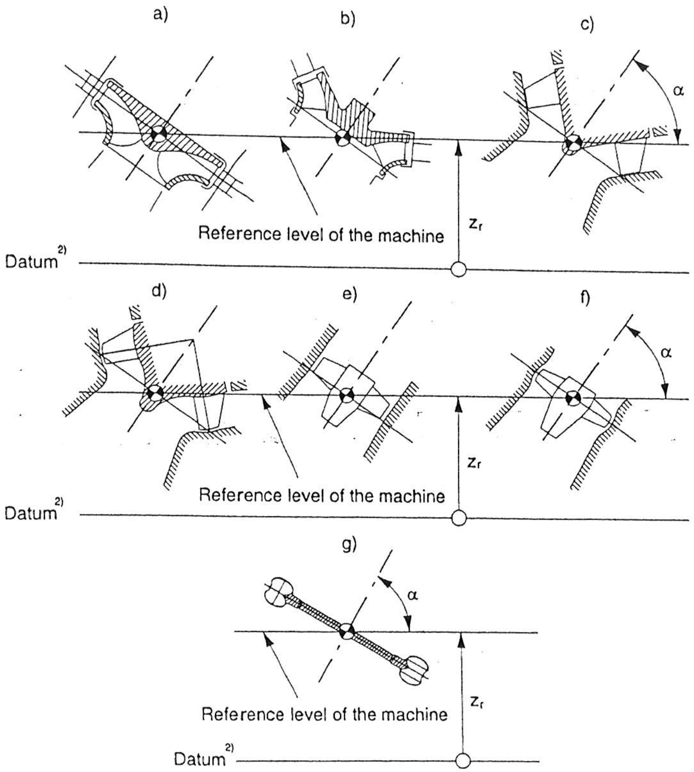


Figure 4 – Definition of σ_0 and σ_1

1.3.3.7 Height and head terms

Subclause	Term	Definition	Symbol	Unit
1.3.3.7.1	Head	Energy per local unit weight of water at any section h = e/g For the definition of e, see 1.3.3.6.1	h	m
1.3.3.7.2	Turbine or pump head	H = E/g For the definition of E, see 1.3.3.6.2	H	m
1.3.3.7.3	Zero discharge (shut-off) head of pump	$H_o = E_o/g$ For the definition of E_o, see 1.3.3.6.3	H_o	m
1.3.3.7.4	Suction height	$Z_s = E_s/g$ For the definition of E_s, see 1.3.3.6.4	Z_s	m
1.3.3.7.5	Net positive suction head	NPSH = NPSE/g For the definition of NPSE, see 1.3.3.6.5	NPSH	m
1.3.3.7.6	Reference level of the machine	Elevation of a point of the machine taken as reference for the setting of the machine (see figure 5)	z_r	m
1.3.3.7.7	Cavitation reference level	Elevation of a point of the machine taken as reference for cavitation evaluation during model tests (see 2.3.1.5.1)	z_c	m
1.3.3.7.8	Reference level of the pressure measuring instrument	Elevation of a pressure measuring device (see figure 38)	z_M	m



$0° \leq \alpha \leq 90°$

Radial machines, such as Francis turbines, radial (centrifugal) pumps and pump-turbines; for multistage :hines: low pressure stage.

Diagonal (mixed-flow, semi-axial) machines with fixed runner/impeller blades and with runner/impeller band.

Diagonal (mixed-flow, semi-axial) machines with fixed runner/impeller blades, without runner/impeller band.

Diagonal (mixed-flow, semi-axial) machines with adjustable runner/impeller blades.

Axial machines, such as propeller turbines, tubular turbines[1)], axial pumps and pump-turbines with fixed ner/impeller blades.

Axial machines, such as Kaplan turbines, tubular turbines[2)], axial pumps and pump-turbines with adjustable ıner/impeller blades.

Pelton turbines.

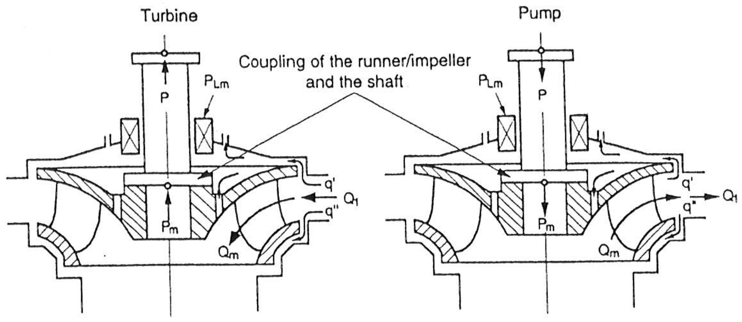
Figure 5 – Reference level of machine

'he term "tubular turbines" includes bulb, pit, rim generator and S-type units.

;ee 1.3.3.2.10.

1.3.3.8 Power and torque terms

Subclause	Term	Definition	Symbol	Unit
1.3.3.8.1	Hydraulic power	The hydraulic power available for producing power (turbine) or imparted to the water (pump) $P_h = E(\rho Q)_1$	P_h	W
1.3.3.8.2	Mechanical power of the machine (power)	The mechanical power delivered by the turbine shaft or to the pump shaft, assigning to the hydraulic machine the mechanical losses of the relevant bearings and shaft seals (see figure 6)	P	W
1.3.3.8.3	Mechanical power of runner(s)/impeller(s)	Mechanical power transmitted through the coupling of the runner(s)/impeller(s) and the shaft (see figure 6)	P_m	W
1.3.3.8.4	Mechanical power losses	Mechanical power dissipated in guide bearings, thrust bearings and shaft seals of the hydraulic machine (see figure 6)	P_{Lm}	W
1.3.3.8.5	Zero discharge (shut-off) power of the pump	Pump power at specified speed and at specified guide vane and impeller settings with high pressure side shut-off	P_o	W
1.3.3.8.6	Shaft torque	Torque applied to the shaft of the hydraulic machine and corresponding to the mechanical power of the machine (see 1.3.3.8.2)	T	N·m
1.3.3.8.7	Runner/impeller torque	Torque transmitted through the coupling of the runner/impeller and the shaft and corresponding to the mechanical power of runner/impeller (see 1.3.3.8.3)	T_m	N·m
1.3.3.8.8	Friction torque	Friction torque in guide bearings, thrust bearings and shaft seals of the hydraulic machine (see 1.3.3.8.4)	T_{Lm}	N·m



	Turbine	Pump
	$q = q' + q''$	$q = q' + q''$
	$Q_1 = Q_m + q$	$Q_1 = Q_m - q$
	$P_h = E \cdot (\rho \cdot Q)_1$	$P_h = E \cdot (\rho \cdot Q)_1$
	$P = P_m - P_{Lm}$	$P = P_m + P_{Lm}$
lumetric efficiency	$\eta_v = \frac{Q_m}{Q_1}$	$\eta_v = \frac{Q_1}{Q_m}$
draulic efficiency (note 3)	$\eta_h = \frac{P_m}{P_h}$	$\eta_h = \frac{P_h}{P_m}$
iciency	$\eta = \frac{P}{P_h}$	$\eta = \frac{P_h}{P}$

)TES

The formulae ignore the compressibility of the water.

For detailed analysis of internal losses, refer to annex N.

The disk friction losses and leakage losses (volumetric losses) are considered as hydraulic losses in this rmula. These "disk friction losses" are the friction losses of the outer surfaces of the runner/impeller not in ›ntact with the flow Q_m passing the runner/impeller blades.

Figure 6 – Flux diagram for power and discharge

1.3.3.9 Efficiency terms

Subclause	Term	Definition	Symbol	Unit
1.3.3.9.1	Hydraulic efficiency [1]	a) Turbine: ratio of mechanical power of runner to the hydraulic power $\eta_h = \frac{P_m}{P_h}$ b) Pump: ratio of hydraulic power to the mechanical power of the impeller $\eta_h = \frac{P_h}{P_m}$ See figure 6	η_h	-
1.3.3.9.2	Mechanical efficiency	a) Turbine $\eta_m = \frac{P}{P_m}$ b) Pump $\eta_m = \frac{P_m}{P}$	η_m	-
1.3.3.9.3	Efficiency	a) Turbine $\eta = \frac{P}{P_h} = \eta_h \eta_m$ b) Pump $\eta = \frac{P_h}{P} = \eta_h \eta_m$	η	-
1.3.3.9.4	Weighted average efficiency	Calculated from the formula $\eta_w = \frac{w_1\eta_1 + w_2\eta_2 + w_3\eta_3 + \dots}{w_1 + w_2 + w_3 + \dots}$ where η_1, η_2, η_3, ... are the values of efficiency at specified operating conditions and w_1, w_2, w_3 ... are their agreed weighting factors respectively	η_w	-
1.3.3.9.5	Arithmetic average efficiency	The weighted average efficiency (1.3.3.9.4) with $w_1 = w_2 = w_3 \dots$	η_a	-

[1] The disk friction losses and leakage losses (volumetric losses) are included and are considered here as hydraulic losses. The disk friction losses are the friction losses of the outer surfaces of the runner/impeller not in contact with the flow passing the blades.

].10 General terms relating to fluctuating quantities

60994 provides a reference for terms relating to these quantities. The following table lists :erms relevant to this standard, some of which are illustrated in figure 7:

Subclause	Term	Definition	Symbol	IEC 60994 Reference
.3.3.10.1	Discrete quantity	Quantity represented by a sequence of its momentary values	X	
.3.3.10.2	Fluctuation of quantity (pulsation of quantity)	Oscillatory variation of a quantity X referred to its mean value during a time interval Δt previously selected	$\tilde{X}$ (t)	2.3.1.5, etc.
.3.3.10.3	Mean value	$\bar{X} = \frac{\sum_1^N X_n}{N}$	$\bar{X}$	2.3.3.1
.3.3.10.4	Maximum value		X_{max}	
.3.3.10.5	Minimum value		X_{min}	
.3.3.10.6	Standard deviation (effective value referred to the mean)	$\tilde{X}_{eff} = \sqrt{\frac{\sum_1^N (X_n - \bar{X})^2}{N}}$	$\tilde{X}_{eff}$	2.3.3.2
1.3.3.10.7	Root-mean-square value	$X_{rms} = \sqrt{\frac{\sum_1^N X_n^2}{N}}$	X_{rms}	2.3.3.3
1.3.3.10.8	Peak-to-peak value	$\Delta X_{pp} = X_{max} - X_{min}$	ΔX_{pp}	2.3.2.11
1.3.3.10.9	Amplitude	Maximum value of a sinusoidal quantity X(t): $A = \frac{1}{2}\Delta X_{pp}$	A	2.3.2.10

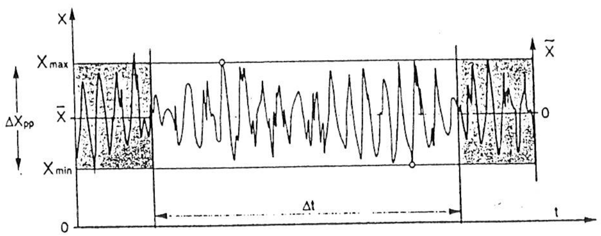


Figure 7 – Illustration of some definitions related to oscillating quantities

1.3.3.11 Fluid dynamic and scaling terms [1)]

Subclause	Term	Definition	Symbol	Unit
1.3.3.11.1	Reynolds number	Ratio of inertia forces to viscous forces $Re = \frac{Du}{\nu}$ or $\frac{Bu}{\nu}$ (see 2.3.1.1)	Re	-
1.3.3.11.2	Froude number [2)]	Square root of the ratio of inertia forces to gravity forces $Fr = \left[\frac{E}{gD}\right]^{1/2}$ or $\left[\frac{E}{gB}\right]^{1/2}$ (see 2.3.1.1)	Fr	-
1.3.3.11.3	Weber number [2)]	Ratio of inertia forces to surface tension forces $We = \left[\frac{\rho L v^2}{\sigma^*}\right]^{1/2}$ where v is the velocity σ* is the surface tension ρ is the density L is a linear dimension	We	-
1.3.3.11.4	Euler number	Ratio of pressure forces to inertia forces $Eu = \frac{\Delta p}{\rho v^2}$ where Δp is the differential pressure	Eu	-
1.3.3.11.5	Loss distribution coefficient	Ratio of relative scalable losses to relative total losses	V	-
1.3.3.11.6	Relative scalable loss	$\delta = (1 - \eta_h)\ V$	δ	-
1.3.3.11.7	Relative non-scalable loss	$\delta_{ns} = (1 - \eta_h) - \delta = (1 - \eta_h) \cdot (1 - V)$	δ_{ns}	-
1.3.3.11.8	Relative total loss	$1 - \eta_h = \delta + \delta_{ns}$		-
1.3.3.11.9	Difference of hydraulic efficiency	Difference between the values of hydraulic efficiency at two hydraulically similar[3)] operating points measured at different Reynolds numbers.	$\Delta\eta_h$	-

[1)] See ISO 31-12.

[2)] Other definitions of these numbers can be found in relevant scientific works.

[3)] See 2.3.1.2.

.12 Dimensionless terms

nine performance may be characterized by dimensionless terms based $E = 1$, $D = 1$ and or on $n = 1$, $D = 1$ and $\rho = 1$[1].

relations of these dimensionless terms to other existing definitions are given in annex A.

clause	Term	Definition	Symbol	Relations
3.12.1	Speed factor	$\frac{nD}{E^{0,5}}$	n_{ED}	$= \frac{1}{E_{nD}^{0,5}}$
3.12.2	Discharge factor	$\frac{Q_1}{D^2E^{0,5}}$	Q_{ED}	$= \frac{Q_{nD}}{E_{nD}^{0,5}}$
3.12.3	Torque factor	$\frac{T_m}{\rho_1 D^3 E}$	T_{ED}	$= \frac{T_{nD}}{E_{nD}} = \frac{P_{ED}}{2\pi n_{ED}}$
3.12.4	Power factor	$\frac{P_m}{\rho_1 D^2 E^{1,5}}$ (see note)	P_{ED}	$= Q_{ED}\,\eta_{hT}$ (turbine) $= \frac{Q_{ED}}{\eta_{hP}}$ (pump) $= \frac{P_{nD}}{E_{nD}^{1,5}} = P_{nD}n_{ED}^3$ $= 2\pi n_{ED}T_{ED}$
3.12.5	Energy coefficient	$\frac{E}{n^2D^2}$	E_{nD}	$= \frac{1}{n_{ED}^2}$
3.12.6	Discharge coefficient	$\frac{Q_1}{nD^3}$	Q_{nD}	$= \frac{Q_{ED}}{n_{ED}} = Q_{ED}E_{nD}^{0,5}$
.3.12.7	Torque coefficient	$\frac{T_m}{\rho_1 n^2 D^5}$	T_{nD}	$= \frac{T_{ED}}{n_{ED}^2} = T_{ED}E_{nD} = \frac{P_{nD}}{2\pi}$
.3.12.8	Power coefficient	$\frac{P_m}{\rho_1 n^3 D^5}$ (see note)	P_{nD}	$= E_{nD}Q_{nD}\eta_{hT}$ (turbine) $= \frac{E_{nD}Q_{nD}}{\eta_{hP}}$ (pump) $= \frac{P_{ED}}{n_{ED}^3} = P_{ED}E_{nD}^{1,5} = 2\pi T_{nD}$
.3.12.9	Thoma number	$\frac{NPSE}{E}$	σ	$= \frac{\sigma_{nD}}{E_{nD}} = \sigma_{nD}n_{ED}^2$
.3.12.10	Cavitation coefficient	$\frac{NPSE}{n^2D^2}$	σ_{nD}	$= \sigma E_{nD} = \frac{\sigma}{n_{ED}^2}$
.3.12.11	Specific speed	$\frac{nQ^{0,5}}{E^{0,75}}$	N_{QE}	$= n_{ED}\,Q_{ED}^{0,5} = \frac{Q_{nD}^{0,5}}{E_{nD}^{0,75}}$
TE – Reference is made to the mechanical power of the runner/impeller, usually measured on the model.				

Units: H (m); D (m); E (J kg^{-1}); n (s^{-1}); ρ (kg m^{-3}); T (N m); P (W); Q (m^3 s^{-1}).

1.3.3.13 Dimensionless terms relating to oscillating quantities

For presentation and analysis of the measured oscillating quantities, it is recommended to use dimensionless terms as defined hereafter and in 4.3 to 4.6. They are identified by the symbol of the measured quantity with subscripts denoting the machine component and the quantities taken as unity; for instance $T_{G,ED}$ will denote the guide vane torque factor, i.e. the torque acting on the guide vanes based on specific hydraulic energy of the machine and reference diameter equal to unity. The symbols used to define the measurements are listed below.

- Measured quantities are:
  - F force;
  - M moment;
  - p pressure;
  - T torque.
- Subscripts for machine components are:
  - B runner blade;
  - D Pelton deflector;
  - G guide vane;
  - N Pelton needle.
- Subscripts for force and torque components are:
  - a axial;
  - r radial;
  - x,y,z cartesian coordinates related to the machine.

The following table lists the dimensionless terms relating to oscillating quantities.

Subclause	Term	Definition	Symbol
1.3.3.13.1	Torque factor	$T_{ED} = \frac{T}{\rho D^3 E}$	T_{ED}
1.3.3.13.2	Force factor	$F_{ED} = \frac{F}{\rho D^2 E}$	F_{ED}
1.3.3.13.3	Torque coefficient	$T_{nD} = \frac{T}{\rho n^2 D^5}$	T_{nD}
1.3.3.13.4	Force coefficient	$F_{nD} = \frac{F}{\rho n^2 D^4}$	F_{nD}
1.3.3.13.5	Factor of pressure fluctuation	$\tilde{p}_E = \frac{\tilde{p}}{\rho E}$	$\tilde{p}_E$

1 Nature and extent of guarantees related to hydraulic performance

1.1 General

1.1.1 Design data and co-ordination

e purchaser shall be responsible for specifying the data on which guarantees are based :luding, for example, reference sections, water levels, specific hydraulic energies ;e 1.3.3.6.2), and specific hydraulic energy losses. The purchaser is also responsible for co-dinating of efforts to determine interactions between the waterways and the electrical and ;chanical parts of the machine.

e purchaser shall arrange for the machine supplier to be provided with data, accurate and fficient in detail, to cover the following:

operating water levels in all reservoirs;

hydraulic losses of each part of the water conduit from intake to outlet;

design drawings of water conduits associated with the hydraulic machine including valves and gates;

any information relevant to the water flow in the conduit, such as any results of model tests of the conduits;

In the case of rehabilitation of existing machines, particular attention is to be paid to the limitations imposed by the existing equipment (for example openings).

:ention shall be paid to flow conditions at the model inlet and outlet (see 2.1.2.4 and 2.1.3.3). most cases, it will be sufficient to extend the model just beyond the high and low pressure 'erence sections. These sections shall be included for the model tests in order to conform to s standard. If the prototype installation is such that there is reason to believe that the total w pattern through the waterways is not fully represented by the model, the contract shall ecify the steps to be taken, which may include specifying the extent of the water passages to modelled. In some other cases, the validity of the approach conditions shall have been rified by the purchaser through the test of a partial model of the plant prior to the test of the ›del machine.

1.1.2 Definition of the hydraulic performance guarantees

contract for both regulated[1] and non-regulated prototype machines shall contain, as a nimum, guarantees covering power, discharge and/or specific hydraulic energy, efficiency, ıximum momentary overspeed and maximum/minimum momentary pressure and maximum ;ady-state runaway speed (reverse runaway speed for a pump), as well as guarantees lated to cavitation.

ır a pump, the guarantee may also cover the maximum specific hydraulic energy (head) and ; power at zero discharge, the latter with the impeller rotating in water and/or air at the ecified speed.

ıe guarantees shall be given for one or more operating points (see 1.4.2.2). These points ;long to performance curves of the machine, which are usually submitted by the supplier. In ıme cases (e.g. small hydro), a table may be sufficient.

ıe current state of the art permits the verification of some of these prototype guarantees by odel test (see 1.4.2), while others cannot be verified by model tests (see 1.4.3). Moreover, lditional data may be obtained from the model as an indication of the expected prototype ›eration (see 1.4.4).

"Regulated" as used in this standard refers to the control of the discharge through variations of guide vane ıening, needle stroke and/or runner/impeller blade angle.

1.4.1.3 Guarantees of correlated quantities

It is recommended that the contract does not fix more than one guarantee for correlated quantities; for example in the case of a regulated turbine, efficiency shall be guaranteed versus either discharge or power, but not both.

1.4.1.4 Form of guarantees

Either of the following two forms of guarantees may be applied when the performance of a prototype machine is to be accepted on the basis of a model test:

a) Guarantees for the hydraulic performance of the prototype, computed from model test results with allowance for scale effect. For reaction machines, the scale effect is to be taken into account in accordance with 3.8. For impulse turbines, when it is agreed in the contract, the scale effect should be taken into account in accordance with annex K.

b) Guarantees for the hydraulic performance of the model referred to a Reynolds number (Reynolds, Froude and Weber numbers for impulse turbines) to be specified in the contract.

In each case, the similitude requirements of 2.3.1 shall be met.

1.4.2 Main hydraulic performance guarantees verifiable by model test

1.4.2.1 Guaranteed quantities for any machine

The main hydraulic performance guarantees of the prototype or model that can be verified by model tests are described in detail in 1.4.2.1.1 to 1.4.2.1.5 below.

1.4.2.1.1 Power

The term "power" usually refers to the mechanical power of the runner/impeller (see 1.3.3.8.3). When the mechanical power of the machine (see 1.3.3.8.2) is guaranteed for the prototype, the mechanical power losses (see 1.3.3.8.4) are to be taken into account.

1.4.2.1.2 Discharge and/or specific hydraulic energy

This refers either to the discharge at the reference section (see 1.3.3.4.4), obtained when operating under specified specific hydraulic energies, or to the specific hydraulic energy obtained when operating with specified discharges.

1.4.2.1.3 Efficiency

The term "efficiency" refers to the hydraulic efficiency (see 1.3.3.9.1) unless otherwise specified. When the prototype efficiency (see 1.3.3.9.3) is guaranteed, the mechanical power losses (see 1.3.3.8.4) or the mechanical efficiency (see 1.3.3.9.2) is to be taken into account.

In reaction machines, the relationship between model and prototype hydraulic efficiencies due to the Reynolds number scale effect is well documented at and around the point of best efficiency, as explained in 3.8. The currently accepted convention is to apply this relationship to the entire range of guaranteed efficiencies (see annex F), however recognizing the decreased reliability the further away the point to be checked is from the point of best efficiency.

.2.1.4 Steady-state runaway speed and/or discharge

guarantee for maximum steady-state runaway speed (see 1.3.3.4.12) or for the reverse ıaway speed in case of pumps is required. An additional guarantee for maximum discharge der runaway conditions may be required.

e no-load discharge for turbine operation shall be determined as part of the characteristic ıaway-curve (see figure 55).

ı.2.1.5 Influence of cavitation on hydraulic performance

e contract shall clearly state the hydraulic conditions (specific hydraulic energies and net sitive suction specific energies) for which the hydraulic performances are to be guaranteed.

e guarantees of the hydraulic performance of the prototype shall include the influence of vitation. According to current engineering practice, this influence is determined by the model ıts performed in accordance with 2.3.1.5.1, 2.3.3.3.6, 3.8.2.3.7 and 3.8.3.2.

e application of efficiency scale-up due to Reynolds number is limited to the range of values specified in 3.8.2.4.2. The contract shall specify the procedure to be used if σ_{pl} falls tside this range.

some cases, the contract may contain an additional clause on cavitation specifying that the easured value of the Thoma number chosen as criterion (σ_d) shall be lower than the plant ıoma number σ_{pl} by a safety margin (see 3.10.5). The criterion and the safety margin shall be ecified in the contract.

4.2.2 Specific application

4.2.2.1 Regulated turbine (see 3.10.3.1)

ower: to be reached at one or more specified specific hydraulic energies.

scharge: to be reached at one or more specified specific hydraulic energies.

ficiency: guarantees may be required for one or more specified specific hydraulic energy as follows:

- at one or more specified power or discharge,

or

- as weighted average efficiencies over a specified range of power or discharge,

or

- as arithmetic average efficiencies[1] over a specified range of power or discharge.

unaway speed: guaranteed steady-state runaway speed not to be exceeded when operating under maximum or any other specified specific hydraulic energy.

In the case of double regulated machines, it shall be stated if the guarantee refers to the condition in which the optimum relationship between guide vane opening and runner blade angle is maintained (on-cam condition) or/and to the maximum runaway speed occurring in the worst possible off-cam condition.

[1] Weighted or arithmetic average efficiencies and a series of individual efficiencies shall not be guaranteed imultaneously.

Cavitation: guarantees as outlined in 1.4.2.1.5 may be required for one or more specified specific hydraulic energy, discharge or power, usually for the corresponding minimum σ_{pl} value.

1.4.2.2.2 Non-regulated turbine (see 3.10.3.2)

Power: power to be reached and power not to be exceeded over a specified range of specific hydraulic energies[1].

Discharge: discharge to be reached and/or not to be exceeded over a specified range of specific hydraulic energies. This guarantee is usually replaced by the corresponding power guarantee.

Efficiency: guarantees may be required as follows

- at one or more specified specific hydraulic energy,

or

- as a weighted average efficiency over a specified range of specific hydraulic energy,

or

- as an arithmetic average efficiency[2] over a specified range of specific hydraulic energy.

Runaway speed: guaranteed steady-state runaway speed not to be exceeded when operating under maximum specific hydraulic energy.

Cavitation: guarantees as outlined in 1.4.2.1.5 may be required for one or more specified specific hydraulic energy, usually for the corresponding minimum σ_{pl} value.

1.4.2.2.3 Non-regulated/regulated pump (see 3.10.3.3)

Power: power not to be exceeded over a specified range of specific hydraulic energy or discharge.

Discharge and/or specific hydraulic energy: discharge over a specified range of specific hydraulic energy or specific energy over a specified range of discharge, including values to be reached and/or not to be exceeded.

Efficiency: guarantees may be required for one or more specified specific hydraulic energy or discharge as follows:

- at one or more specified specific hydraulic energy or discharge,

or

- as a weighted average efficiency over a specified range of specific hydraulic energy or discharge,

or

- as an arithmetic average efficiency[2] over a specified range of specific hydraulic energy or discharge.

Runaway speed: guaranteed maximum steady-state reverse runaway speed not to be exceeded when operating under maximum specified specific hydraulic energy.

In the case of double regulated machines, it shall be stated if the guarantee is referred to the condition in which the optimum relationship between guide vane opening and runner blade angle is maintained (on-cam condition) or/and to the maximum runaway speed occurring in the worst possible off-cam condition.

[1] For the contractual limits of the power corresponding to the specified specific hydraulic energies, see 3.10.3.2.

[2] Weighted or arithmetic average efficiencies and a series of individual efficiencies shall not be guaranteed simultaneously.

ıvitation: guarantees as outlined in 1.4.2.1.5 may be required for one or more specified specific hydraulic energy, discharge or power, usually for the corresponding minimum σ_{pl} value.

ł.3 Guarantees not verifiable by model test

ere are certain guarantees that cannot be checked by model tests. Amongst them are:

ł.3.1 Guarantees on cavitation erosion

e amount of cavitation pitting is to be guaranteed for the prototype only. Evaluation of this arantee on the prototype shall be carried out in accordance with the recommendations of Ꞇ 60609.

e model test may reveal some of the potential areas of cavitation erosion by visual ;pection during the tests (see 2.3.3.3.6).

ł.3.2 Guarantees on maximum momentary overspeed and maximum momentary pressure rise

ɪmentary overspeed (including momentary runaway speed) and pressure depend primarily on water conduit geometry (penstock length, surge tank, etc.), the inertia of the rotating part the unit and the operating law of the guide vanes. They can therefore not be determined ectly by a dynamic test on the model which does not reproduce either the full extent of the plicable waterways, the inertia of the unit or the characteristics of the speed governor. ·vertheless, the steady-state model test data, transferred to the prototype, provide values abling calculation of the transient phenomena with sufficient accuracy.

ł.3.3 Guarantees covering noise and vibration

:termination of prototype noise and vibration by model tests lies outside the scope of this ındard. This standard should be used only as a guide for the modelling of the hydraulic urces of these phenomena (i.e. through determination of the pressure fluctuations or other namic loads).

ł.4 Additional performance data

lditional data may be obtained from the model as an indication of expected prototype eration:

- pressure fluctuations (see 4.3);
- shaft torque fluctuations (see 4.4);
- hydraulic thrust, both radial and axial (see 4.5);
- hydraulic torque on guide vanes and adjustable runner/impeller blades or hydraulic forces on needles and deflectors over the full operating range (see 4.6);
- characteristics in four quadrant operation including power and specific hydraulic energy at zero discharge (shut-off conditions) of a pump with impeller rotating in water or air (see 4.7);
- differential pressure measurements in view of prototype index tests (see 4.8);
- cam relationship for optimum performance in the case of double regulated machines (relation between guide vane and runner/impeller blade openings) (see 3.8).

may also be specified to determine other additional data, such as velocity or pressure stribution in various components of the machine, etc.

2 Execution of tests

2.1 Requirements of test installation and model

2.1.1 Choice of laboratory

Any laboratory satisfying the criteria set out in this standard concerning general layout, capacity and quality of instrumentation should be deemed suitable. An independent laboratory is sometimes preferable, particularly when comparative tests are required with models from different manufacturers.

2.1.2 Test installation

2.1.2.1 General characteristics of the test circuit

The test circuit shall be such that when cavitation appears in the model, it shall not occur elsewhere to an extent whereby it affects the stability or the satisfactory operation of the installation or of the model performance measurement.

Any entrained air bubbles produced by cavitation in the model shall not affect the functioning of instrumentation, particularly the flow measuring device and the pressure measurement lines.

The circuit shall be designed in such a manner that no leakage or addition of water may occur between the flow measuring instrument and the model. This criteria should be easily verifiable.

2.1.2.2 Capacity of the test installation

The capacity of the test installation (i.e. power, pressure, specific hydraulic energy, discharge and NPSE) shall be such that the minimum values for model size and the required test conditions as listed in 2.3.2.2 can be met.

The operation shall be stable and steady without surging or fluctuating effects (see 2.3.2.3).

2.1.2.3 Condition of the water

The test water shall be clean, clear and free of any solid material in suspension and any chemical impurities which may have an influence on the water properties such as viscosity and vapour pressure. Free gas and air bubbles should be removed as far as possible before testing.

The gas content (see 2.3.1.6.2), including both entrained and dissolved gas, of the water used in the test rig, should be recorded for the test, especially with respect to repeatability of cavitation test results and for comparison purposes. It shall be measured at the inlet, close to the model (see 2.5.3.2).

Experience in closed circuits indicates that dissolved air may produce nuclei in the water and the nuclei content plays a major role in travelling cavitation[1]). It may strongly influence the cavitation pattern and the resulting cavitation characteristics (see 2.3.1.5, 2.3.1.6.2 and figures 71 and 72).

Cavitation similitude should be respected by performing the test using an appropriate specific hydraulic energy or/and nuclei injection.

[1]) Nuclei are small air or gas bubbles with diameter less than 50 μm. Travelling cavitation has the aspect of bubbles moving with the flow. Travelling cavitation is typical at the outlet of Francis turbines (see [1], annex P, Bibliography).

The water temperature should in principle not exceed 35 °C and should not vary significantly during the tests (e.g. 5 °C per day). Large differences between the water temperature and the ambient temperature of instruments should be avoided, as they could influence the accuracy of the measurements.

2.1.2.4 Flow conditions given by the installation

At the model inlet, the test installation shall ensure favourable hydraulic conditions free from vortices, undue turbulence and unsteadiness.

At the model outlet, the flow pattern shall not be influenced by the layout or construction of the test facility.

2.1.2.5 Measuring instruments

The measuring equipment used to determine the main parameters shall satisfy the specified conditions of this standard.

The traceability of each instrument to a recognized national or international standard shall be ensured. All instruments should be calibrated *in situ*, especially discharge and torque measuring instruments.

The measuring instruments should be such that a direct reading, independent of the data acquisition system, is possible in order to permit an easy verification.

2.1.3 Model requirements

The model shall meet the following conditions:

2.1.3.1 Model size

The minimum values for model size are prescribed in 2.3.2.2.

Normally, models shall be as large as practical but never less than the values stated. The same model should be used for all tests related to main hydraulic performance guarantees and influence of cavitation (see 1.4.2). Comparative model tests shall be performed with models of the same size.

2.1.3.2 Layout and mechanical design of the model

The layout and mechanical design of the model shall comply with the specified items of testing. The following points shall be carefully considered:

- deformations due to loading under the chosen test head shall be minimized by appropriate design and choice of material;
- elements used to vary machine geometry (runner/impeller blades, guide vanes, nozzles) shall be capable of repeating and maintaining a set position;
- bearing system, shaft and stationary parts shall have sufficient rigidity to avoid contact in the labyrinth during normal operation. Blade tip and seal labyrinth clearances of less than 0,15 mm are not recommended for mechanical reasons;
- hydraulically smooth surfaces of the water passages are recommended;
- in addition to general requirements for surface conditions, great care should be given to the proper matching of joints in order to avoid any local flow separation;
- materials should be chosen to avoid oxidation and electrochemical corrosion. The water passage surfaces should remain in good condition for the test duration;

- provision shall be made for easy cleaning or repairing of water passages;
- a transparent cone or windows should be provided for observation of the flow in the runner/impeller and in the adjacent portion of the draft tube (at the low pressure stage in case of a multistage machine);
- wherever shaft seals and shaft bearings are required to be a part of the model (e.g. multistage machines) when comparative model tests are performed, these parts are to be of identical design;
- for comparative tests of the same model, but with different impeller/runner, provisions shall be made that all runners/impellers can be tested with the same clearances;
- where additional data have to be measured (such as pressure fluctuations, guide vane torque, velocity distribution) the model shall permit easy installation and checks of the corresponding measuring equipment;
- if any non-homology exists (e.g. differences in runner band/crown thickness, seal design, etc.), any analytical procedures adopted to account for these differences shall be mutually agreed.

2.1.3.3 Extent of model

The position of reference sections and the extent of the water passages from inlet to outlet to be included in the model (at least the part between the high pressure and low pressure reference sections) shall be clearly defined in the contract (see figure 1 and as an example figure 44).

All the water passages influencing the performance of the prototype, i.e. inflow and outflow conditions, should as far as possible be included in the model.

Especially for low specific hydraulic energy turbines, it is recommended that the model extends from the prototype inlet to the outlet section of the draft tube.

Upstream or downstream gate slots are not required, unless they are located between the measuring or reference sections of the machine. Information on the influence of these parts on the hydraulic behaviour of the machine may be obtained from tests in addition to the model acceptance tests.

2.1.3.4 Geometric similarity of model and prototype

2.1.3.4.1 General requirements

A basic requirement for determining prototype performance from model tests is to have geometrical similarity (homology) between model and prototype. Therefore it is necessary to compare on both machines the geometrical dimensions and the surface finish of all components in contact with the flow.

The geometric similarity between model and prototype is to be checked in accordance with 2.2.

Unless otherwise specified, the model shall be geometrically similar (homologous) to the prototype in all wetted parts within the limits defined in 2.1.3.3. This also includes details that may have a measurable influence on the performance. However, in particular cases where some minor deviations from the similarity cannot be avoided, an agreement shall be reached whether the results are to be corrected or not.

In the case of acceptance tests on both model and prototype, the same measuring sections should be used if possible.

For comparative model tests, all models shall rotate in the same direction.

.1.3.4.2 Multistage machines

ormally the model test should be performed with the same number of stages as the prototype.

or a prototype with four or more stages, in exceptional cases, the model may be tested with a ›duced number of stages, for instance a three-stage model for a four-stage prototype.

.1.3.4.3 Labyrinth seals and thrust balancing provisions

or mechanical reasons, it may not be possible or desirable (especially for large scale ratios) ır the shaft and runner/impeller seal clearances and the thrust balancing provision to be either əometrically similar or hydraulically equivalent between the model and the prototype. In such ıses, seal leakage losses and thrust coefficient will differ between model and prototype. The fferences shall either be negligible or be accurately estimated so that the hydraulic efficiency ıd the thrust of the prototype can be computed.

n agreement should be reached before the tests concerning the monitoring and measuring of akage flow rate q' (see figure 6) and whether or not it should be taken into account in ətermining machine performance.

.2 Dimensional check of model and prototype

.2.1 General

s stated in 2.1.3.4, the geometrical similarity between model and prototype shall be checked. he checking procedure and admissible tolerances are described below.

his applies even in the case where the model is manufactured according to an existing achine and hence the prototype is the starting point of the process.

.2.1.1 Explanation of terms used for model and prototype

ı individual value

The value resulting from:

the measurement of the **same** dimension of the **same** component taken at **different** locations (e.g. reference diameter of a runner/impeller);

r:

the measurement of the **same** dimension of **different recurrent** components taken at the **same** location (e.g. maximum thickness of guide vanes).

) average value

The value is the arithmetical mean value calculated from several individual values.

) theoretical value

The design value indicated on a drawing. Corresponding model and prototype dimensions L_M and L_P are linked by the scale length ratio λ_L (see 1.3.3.2.9).

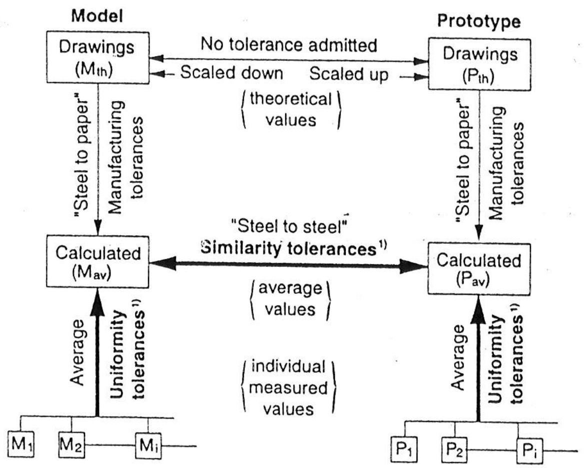
.2.1.2 Purpose of dimensional checks

)imensional checking:

) ascertains the main dimensions of model and prototype;

b) checks the uniformity of model and prototype in their recurrent components, i.e. compares the relative deviation between the individual values and the corresponding average value. The permissible maximum deviations are given in the "uniformity tolerance" columns of 2.2.2.1.7 and 2.2.2.2.5;

c) checks the geometrical similarity between prototype and model, i.e. compares the relative deviation between the average prototype value and the corresponding average model value scaled up by λ_L. The permissible maximum deviations are given in the "similarity tolerance" columns of 2.2.2.1.7 and 2.2.2.2.5.



1) Numerical values of similarity tolerances and uniformity tolerances are given in 2.2.2.1.7 and 2.2.2.2.5.

Figure 8 – Procedure for dimensional checks, comparison of results and application of tolerances for model and prototype

2.2.1.3 Procedure

Figure 8 illustrates schematically the application of the geometrical tolerances given by this standard, on individual and average values of model and prototype.

a) Checking the uniformity of model and prototype components

By comparing the measurement of individual values with the corresponding average value, it is possible to determine if the uniformity requirement has been met.

If the uniformity requirement is not fulfilled, agreement shall be reached on what components are to be corrected or remanufactured.

In the case where the difference between the average value and the theoretical value is outside the uniformity tolerance, it shall be agreed whether the theoretical value or the geometry of the component shall be corrected.

Where spot-checks are performed, i.e. when not all recurrent components or dimensions are checked, it may be agreed that the individual values are compared directly with the theoretical value.

b) Checking the geometric similarity between prototype and model

By comparing the corresponding average values of model and prototype and considering the similarity tolerances as given by this standard, it can be determined whether the geometric similarity requirement has been met. If agreed, the corresponding theoretical values may be substituted for the average values.

If the deviations are greater than the similarity tolerances, further steps shall be agreed which could include a new test with the corrected model.

For some dimensions (e.g. overall dimensions) which are difficult to measure on the model and/or prototype, it may be agreed that, instead of average values of model and/or prototype, theoretical values can be used for comparison, provided that the sum of manufacturing and installation tolerances is less than the similarity tolerance.

.2.1.4 Application for different types of machines

or turbines, dimensional tolerances are listed for Francis, Kaplan and Pelton turbines.

he tolerances for Francis turbines apply also to diagonal flow machines with fixed blades.

he tolerances for Kaplan turbines apply also to diagonal flow machines with adjustable blades)eriaz turbines), and to axial flow machines with fixed or adjustable blades (propeller, tubular, tc.).

he tolerances for Pelton turbines can be adapted to inclined jet impulse turbines.

or pumps and pump-turbines, dimensional tolerances are given for centrifugal, mixed flow and xial machines.

.2.1.5 Methods

o measure the shape of runner/impeller blades, guide vanes and stay vanes, several methods re suitable, including three dimensional co-ordinate measuring machines, optical measuring ystems, templates, etc.

he inspection for homology of the hydraulic profiles using three dimensional co-ordinate ıeasuring machines or optical systems can be made either by measuring points along curves r by measuring points on the surface:

curves are essentially the equivalent of mechanical templates and their use and interpretation are similar to the currently prevailing practice using mechanical templates;

surfaces are represented by a number of measuring points sufficient to define the complete prof e positioned either in geometrically similar locations on the prototype and the model or at random. Computer algorithms must be available to adjust the "surface" so that the resulting measured "error" is minimized for the total surface. Such adjustment is however limited by the permissible tolerances shown in 2.2.2.1.7 and 2.2.2.2.5.

:onsidering the manufacturing and measuring methods of the runner/impeller, the ıanufacturer shall propose the most appropriate method in agreement with the customer to emonstrate the geometric similarity between prototype and model.

:igures 9 to 19 give examples showing schematically the location and extent of the geometric hecking.

n some cases it is not possible to measure the relevant dimension directly, e.g. if a point of ıtersection is covered by a fillet. In such cases the measurement shall be made in an agreed ocation.

n order to protect the confidentiality of the manufacturer's hydraulic design, the manufacturer ıeed only submit the differences between the measured and the theoretical profile and not the ıctual values of the co-ordinates of the profile. For checking purposes, the measured absolute 'alues of profiles are available from the manufacturer.

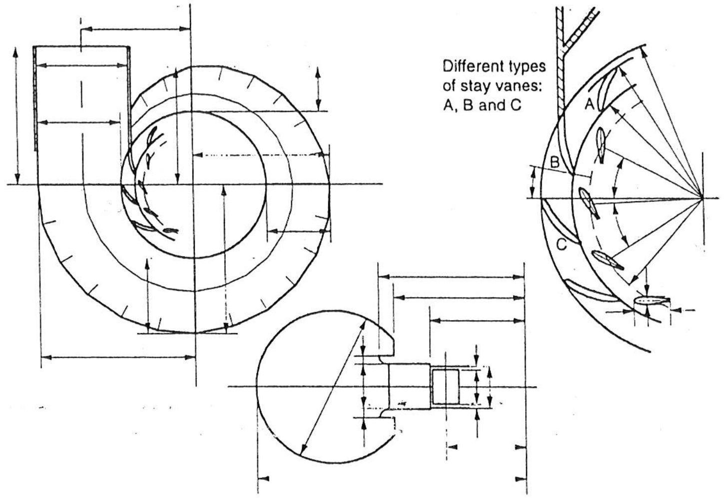


Figure 9 – Example of spiral case and distributor dimensions to be checked

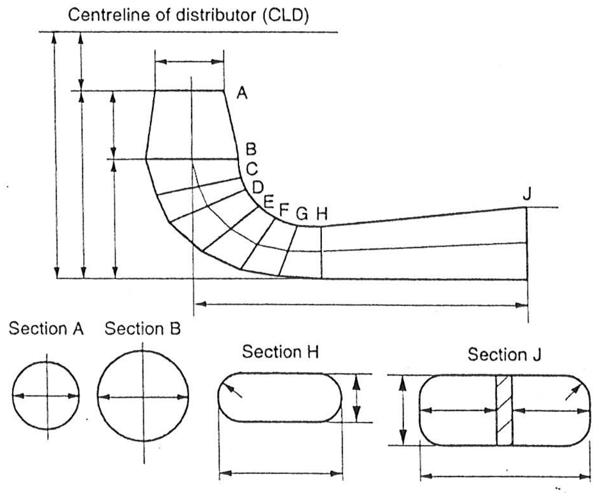


Figure 10 – Example of draft tube dimensions to be checked

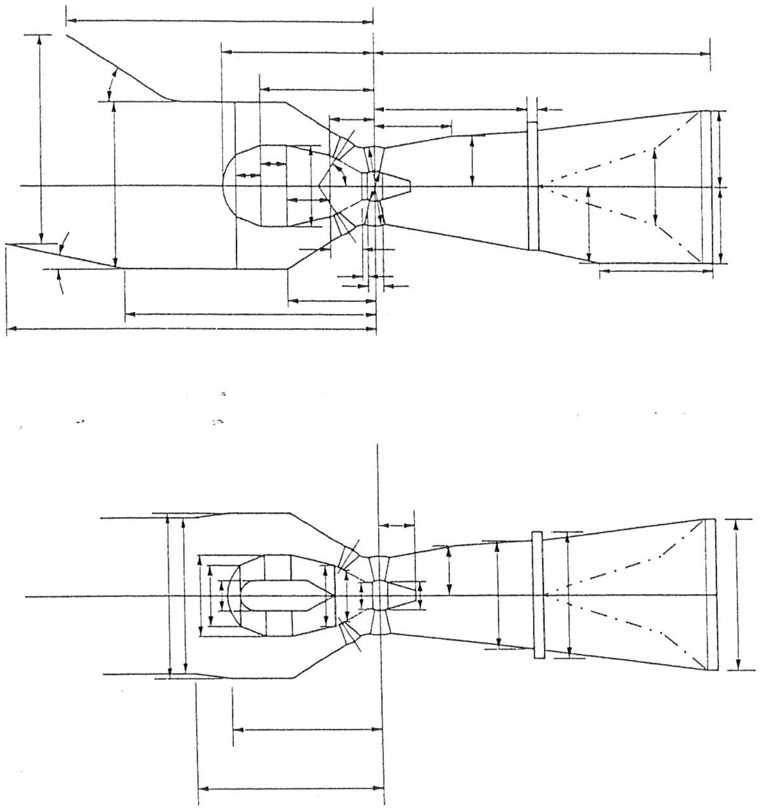


Figure 11 – Example of the dimensions to be checked on a bulb unit

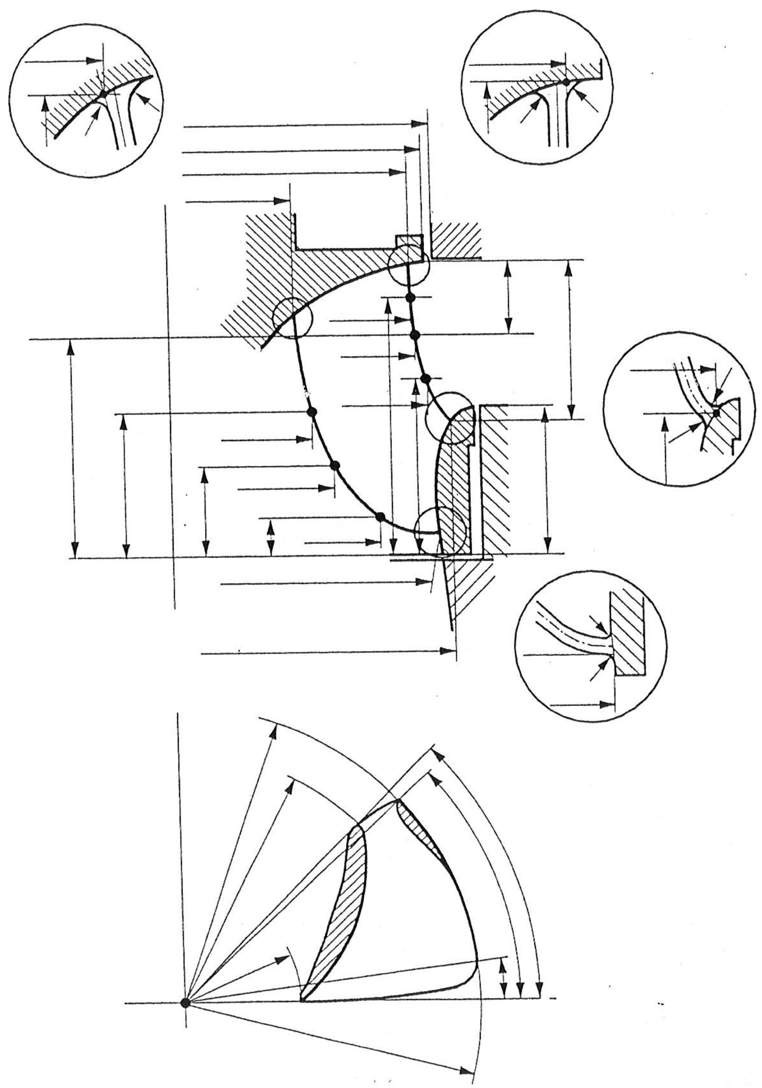
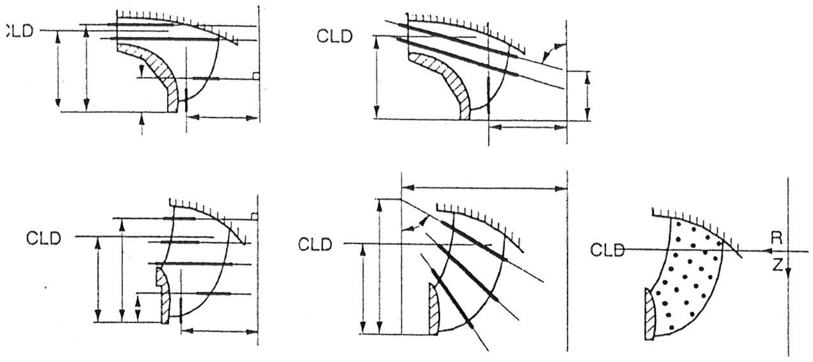


Figure 12 – Example of the dimensions to be checked on the runner/impeller of a radial flow machine



(CLD: Centre line of distributor)

Figure 13 – Runner/impeller of radial flow machine. Examples of locations for blade profile measuring sections or measuring points

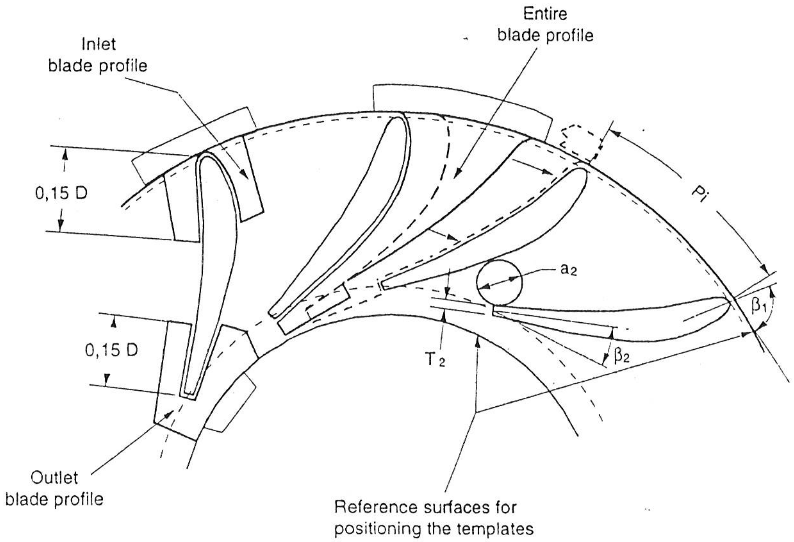


Figure 14 – Runner/impeller of radial flow machine. Check of outlet width and blade profiles by means of templates or co-ordinate system as illustrated on a Francis runner

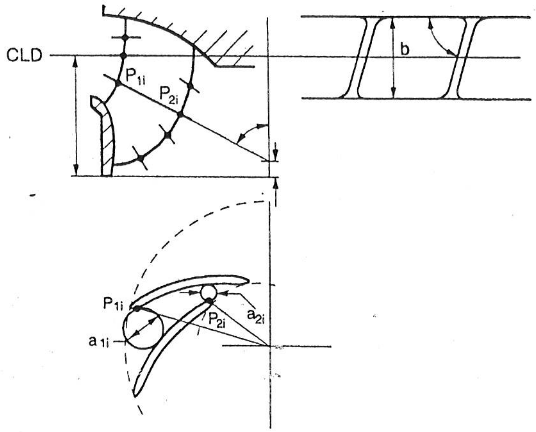


Figure 15 – Runner/impeller of radial flow machine. Check of inlet and outlet widths between blades (example of a pump-turbine runner)

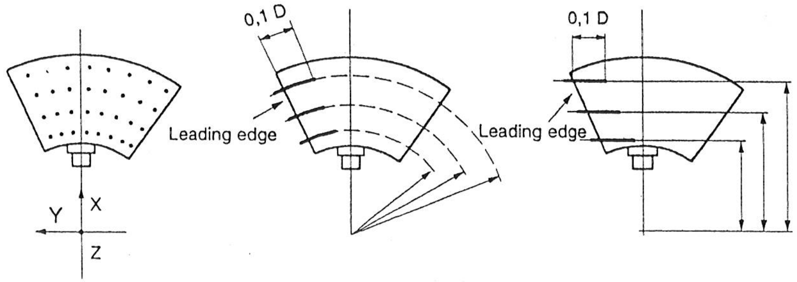


Figure 16 – Runner/impeller of axial flow machine. Example of locations for blade profile measuring sections or measuring points

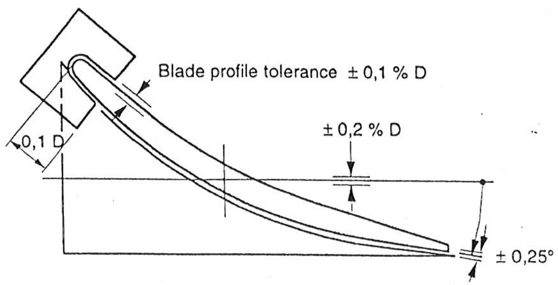


Figure 17 – Runner/impeller of axial flow machine. Definition of blade adjustment and of blade profile tolerances

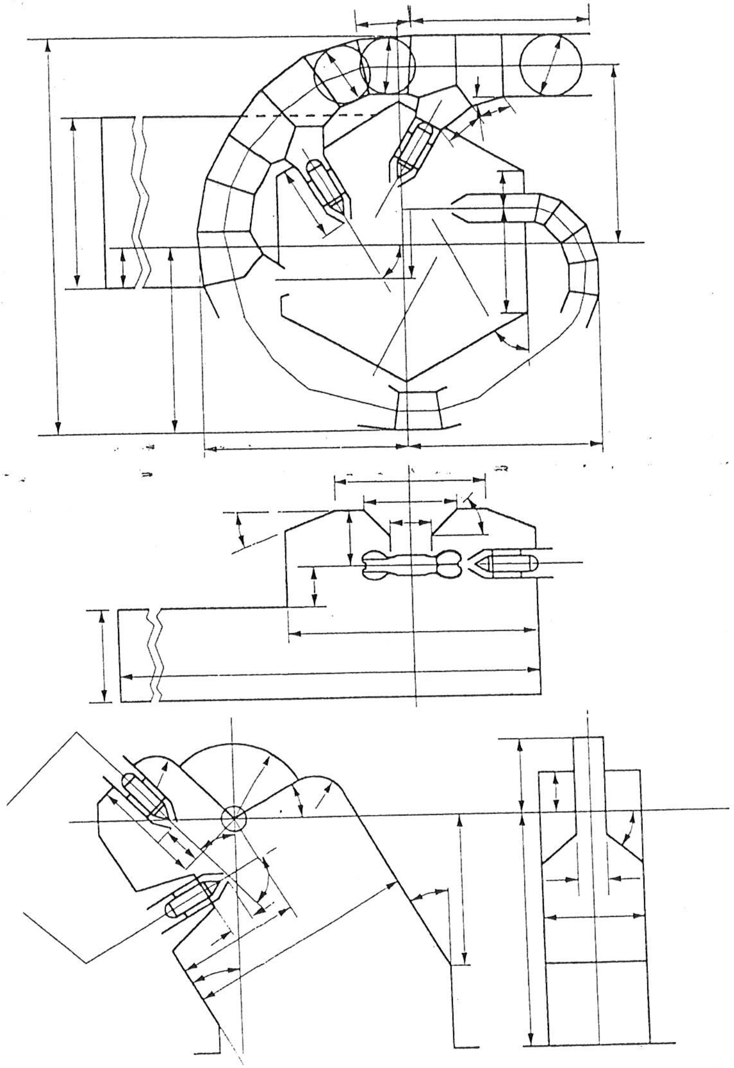


Figure 18 – Pelton turbine: Example of dimensions to be checked on the distributor and the housing of vertical and horizontal shaft machines

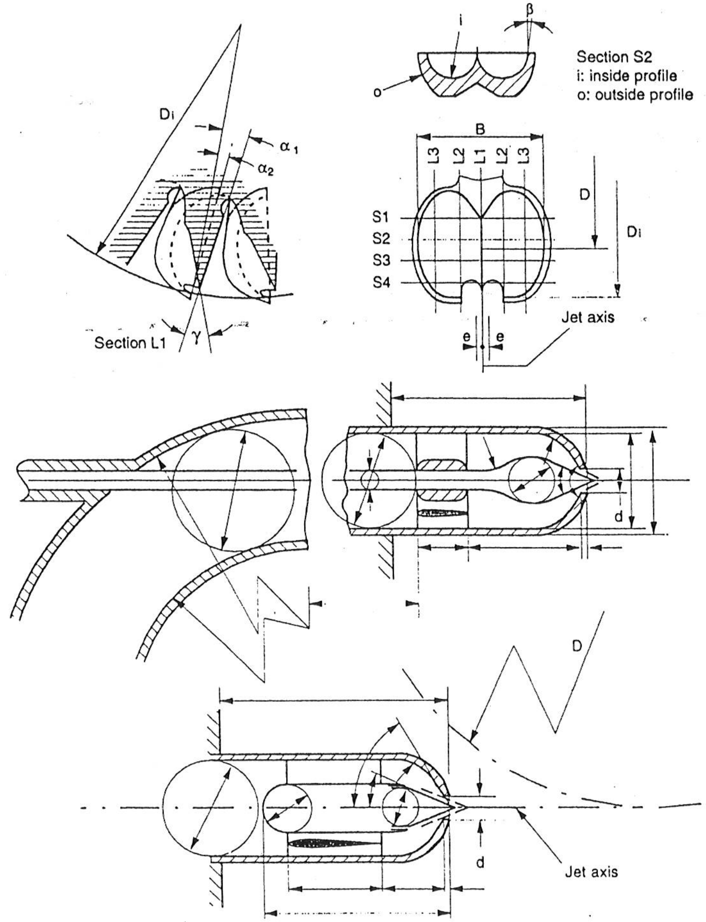


Figure 19 – Pelton turbine: Example of dimensions to be checked on the buckets and nozzles

.2 Dimensions of model and prototype to be checked

ɜ dimensions of all significant components defining the water passages shall be checked to extent agreeable to the parties concerned.

ɜ relevant dimensions and their permitted tolerances for turbines, pumps and pump-turbines defined in 2.2.2.1 and 2.2.2.2. When a location is defined by more than one dimension, the ɪrance applies to only one dimension.

heck of geometric similarity comprises both:

a check on the homology of the individual components before assembly. For practical reasons the dimensions shall be referred to the component itself;

a check on the homology of the whole machine, i.e. with the components assembled. The relevant overall dimensions are usually referred to the runner axis and/or to the centre line of the distributor. Special attention shall be paid to water passages formed by the transition between adjacent components, stationary or rotating, for which no general value of tolerance can be prescribed in this standard.

ɜ meridional contour of the runner/impeller shall be checked with templates or by other ɔrop:iate measuring techniques, such as three-dimensional co-ordinate measuring machines ɔptical measuring systems.

appropriate checking of the runner/impeller blade profile and its geometric location, the ɪfile angles are properly determined.

the following subclauses 2.2.2.1 and 2.2.2.2 the permissible maximum deviations are either erred to the corresponding scaled model average value or to the following reference ɪensions:

maximum thickness of guide vanes, stay vanes or runner blades for each measuring section;

reference diameter (see figure 3);

blade inlet pitch for Francis runners (see figure 14);

nozzle orifice diameter (see figure 19);

maximum inside width of runner bucket (see figure 19);

a_2 opening between two adjacent blades at high pressure and low pressure side of the runner/impeller (see figures 14 and 15). The opening can be defined either on a blade profile measuring section or as the distance from a given point of the blade edge to the adjacent blade surface.

ɪe terms used in the following subclauses to indicate the various components of hydraulic ɜchines are defined in IEC 61364.

2.2.1 Turbines

2.2.1.1 Main components

Reaction turbines:

- the principal dimensions of the spiral casing (or intake in the case of a tubular turbine), the stay ring, the distributor, the draft tube and if necessary the space between runner and head cover (see for example figures 9 and 10);
- the principal dimensions of the runner including inlet and outlet diameters, inlet height, runner band and crown, and hub in case of Kaplan turbine runners;
- the number of runner blades, guide vanes and stay vanes;

- the form of the runner water passages and of the guide and stay vanes, including the maximum thickness of stay vanes, guide vanes and, if applicable, of the runner blades;
- the seal and blade tip clearances of runner and end clearances of guide vanes;
- the roughness of all components of the machine (see 2.2.3.3);
- the waviness of runner blades, guide vanes and stay vanes (usually only relevant for prototype) (see 2.2.3.2).

b) Impulse turbines:

- the principal dimensions of the runner, manifold, housing and nozzles (see figure 18);
- the number of buckets;
- the form of buckets, nozzles and needles (see figure 19);
- the alignment of the jets to the runner;
- the roughness and waviness of the buckets, nozzles and needles (usually only relevant for the prototype).

2.2.2.1 2 Francis turbine runners

- The blade inlet profile shall be checked at least at two sections for low specific speed turbines and at three sections for high specific speed turbines. The inlet section shall extend from the nose of the blade to a distance of 0,15[1] of the reference diameter along both pressure and suction sides of the blade (see figures 13 and 14 for examples).
- The entire blade profile (from inlet edge to outlet edge) shall be measured, if possible, depending on the runner specific speed, at least for one section or randomly on the whole surface (see figures 13 and 14).
- The blade inlet angles shall be checked at the same sections as for inlet profiles. The inlet angle can be measured, for example, by using a template extending from the nose of the blade to a distance of 0,15[1] of the reference diameter along both sides of the blade and located to give the best fit to both sides of the blade simultaneously[2] (see figure 14).
- The blade outlet profiles shall be measured at least at three sections. The outlet section shall extend from the trailing edge of the blade to a distance of 0,15[1] of the reference diameter back along both sides of the blade (see figure 14).
- The location of the runner leading and trailing edges shall be checked at least at two or three points depending on the specific speed (see figure 12).
- Outlet width between blades shall be checked at least at four points on each blade (see figures 14 and 15).

For fabricated prototype runners (assembled by welding of prefabricated components) it may be agreed that the check of blade profiles can be made prior to assembly, i.e. after having completed machining of the individual blades. After assembly, it is sufficient to check the blade position (e.g. as outlined in figure 12) the fillets and the outlet width.

[1] This value can be reduced to 0,1 depending on specific speed and profile length.

[2] Where the blade surface is sufficiently well represented, the independant determination of angles can be omitted (see 2.2.2).

.2.1.3 Kaplan turbine runners

Blade profiles shall be measured at least at three sections along the entire profile (either along cylindrical or plane sections), on both the pressure and suction sides of the blade or randomly on the whole surface (see figure 16).

The nose profile of each of the measured sections shall be checked. The nose profile shall extend from the nose of the blade to a distance 0,1 D along the blade. In making these checks, the best fit between blade and each reference profile may be obtained by rotating the reference profile, provided the angular and axial adjustment do not exceed the values defined in 2.2.2.1.7 compared with the correct location of the reference profile. These adjustments shall be allowed only once for each section and all check measurements shall be made in this position on both the pressure side and the suction side of the section considered (see figure 17).

With the blades mounted in the hub, the inclination of the blades with respect to each other shall be checked. For this purpose, the inclination of the outer measured profiles shall be compared. The difference of recorded blade angles shall not exceed ±0,25°.

!.2.1.4 Pelton turbine runners

The profile of each bucket shall be checked at least at four transverse and four longitudinal sections or randomly on the whole surface (see figure 19).

The discharge angle β of each bucket shall be checked at four points on each side (see figure 19).

The form of the cutout, the angle γ of the back of the cutout, the splitter edge and the bucket inclination α shall be checked on each bucket (see figure 19).

The bucket inside width and profile shall be checked.

The bucket outside profile shall also be checked to ensure that the interference between the jet and the bucket on the prototype is always less than that on the model, the tolerances of the outside profile (see 2.2.2.1.7) shall be applied to the bucket discharge area which can influence this interference.

2.2.1.5 Guide vanes

›r a cylindrical distributor arrangement, the profile shall be measured at least at one section ıd, for conical arrangements, at least at two sections.

2.2.1.6 Clearances

ıe runner/impeller seal clearances, blade tip clearances and guide vane end clearances on action turbines shall be checked.

ıe prototype clearances shall not exceed the scaled model clearances.

he possible influence of the pressure on the guide vane end clearance should be considered ›r model and prototype.

he extension (length) of the prototype runner/impeller seals shall not be smaller than that of ıe scaled model.

2.2.2.1.7 Permissible maximum deviations in geometrical similarity between prototype and model turbines

	Permissible maximum deviation		
	Uniformity tolerance		Similarity tolerance
	Model	Prototype	Prototype/Model
	Individual value to average value	Individual value to average value	Prototype average value to scaled model average value $(L_P-\lambda_L L_M)$/(reference value) [1]
Principal dimensions of hydraulic passages			
Metallic or concrete passages (casing, draft tube, etc.) [2]	±2 %	±2 %	±1 %
Stay ring diameters	±1 %	±1 %	±1 %
Length of stay vanes	±2 %	±2 %	±2 %
Maximum thickness of guide vanes T'	±5 %	±5 %	±5 %
Maximum thickness of stay vanes T"	±5 %	±8 %	±5 %
Stay ring height	±2 %	±2 %	±2 %
Distributor height	±0,3 %	±0,3 %	±0,2 %
Guide vane pitch circle diameter	±0,2 %	±0,2 %	±0,2 %
Relative position between stay vanes and guide vanes (for example expressed as an angle)	±1°	±1°	±1°
Guide vane profile	±3 % T'	±5 % T'	±3 % T'
Stay vane profile	±3 % T"	±8 % T"	±5 % T"
Maximum guide vane opening	±1,5 %	±2 %	≥0
Clearances			
Seal and blade tip clearance	± 50 %	± 50 %	≤ 0
Seal clearance length	-	-	≥0
Guide vane end clearances	± 50 %	± 50 %	≤0
Francis runners			
Blade profile:			
inlet and outlet edges	±0,1 % D	±0,1 % D	±0,1 % D
remaining part of the surface	±0,2 % D	±0,2 % D	±0,2 % D
Inlet pitch P_i	±0,2 % D	±0,5 % D	-
Inlet angle β_1 [3]	±1,5°	±2°	±1,5°
Outlet angle β_2 [3]	±1°	±1,5°	±1°
Outlet opening a	± 3 %	$^{+5}_{-3}$ %	$^{+3}_{-1}$ %
Maximum blade thickness T [4]	$^{+3}_{-6}$ %	$^{+5}_{-8}$ %	$^{+3}_{-6}$ %
Blade thickness near the outlet edge	±15 %	±15 %	±15 %
Inlet and outlet diameter and other runner dimensions [5]	±0,25 % D	±0,5 % D	±0,25 % D

1) The reference value is taken to be the scaled model average value ($\lambda_L L_M$) unless otherwise indicated. Angular tolerance is the difference between prototype and model angles.

2) For concrete surfaces, uniformity tolerances should be progressively changed from ±2 % to ±1 % on machines with prototype runner diameter between 3 m and 1 m. Also for concrete surfaces, abrupt changes resulting from shifting formwork, interface between concrete and metallic surfaces, etc., should be limited to 6 mm on prototype machines greater than 3 m runner diameter and progressively decreasing to 3 mm for machines between 3 m and 1 m runner diameter.

3) Where the blade surface is sufficiently well represented, the independant determination of angles can be omitted (see 2.2.2).

4) Not required if complete profile data are provided.

5) In certain cases, it may be appropriate to agree upon increased tolerances for the trailing edge dimensions (e.g. some radii shown in figure 12) while maintaining the required tolerances for outlet width.

	Permissible maximum deviation		
	Uniformity tolerance		Similarity tolerance
	Model	Prototype	Prototype/Model
	Individual value to average value	Individual value to average value	Prototype average value to scaled model average value $(L_P-\lambda_L L_M)$/(reference value) 1)
aplan and diagonal flow runners			
lade profile	±0,1 % D	±0,1 % D	±0,1 % D
ose profile	±0,1 % D	±0,1 % D	±0,1 % D
laximum blade thickness 2) T	$^{+3}_{-6}$ %	$^{+5}_{-8}$ %	$^{+3}_{-6}$ %
lade thickness near the outlet edge	±15 %	±15 %	±15 %
ischarge ring diameter D	±0,1 %	±0,1 %	±0,2 %
ther runner dimensions	±0,25 % D	±0,5 % D	±0,25 % D
ngular difference of the profile	±0,25°	±0,25°	±0,25°
xial adjustment of the profile	±0,2 % D	±0,2 %D	±0,2 % D
laximum blade angle	±0,25°	±0,25°	≥0°
elton turbines			
leedle and nozzle diameter	±0,3 % d	±0,3 % d	±0,3 % d
leedle and nozzle profile	±0,1 % d	±0,1 % d	±0,1 % d
lozzle angle	±0,5°	±1°	±1°
leedle angle	±1°	±2°	±1°
ucket inside width B	±0,3 %	$^{+0,8}_{-0,5}$ %	$^{+0,8}_{-0,5}$ %
ucket outside profile	±0,5 % B	$^{+1}_{-0,8}$ %	$^{+1}_{-0,8}$ %
ucket inside profile	±0,5 % B	±0,5 % B	±0,5 % B
ucket inclination α	±1°	±1°	±1°
ucket discharge angle β	±1°	±1°	±1°
ut-out profile	± 1 % B	± 1 % B	±1 % B
ngle γ of face at the back of cut-out	±1°	$^{0}_{-3}$ °	$^{0}_{-3}$ °
let circle diameter D	±0,2 %	±0,2 %	±0,2 %
Offset of jet to runner e	±0,5 % B	±0,5 % B	±0,25 % B
lignment of jet to runner δ	±0,5°	±0,5°	±0,5°
Bucket pitch at outer diameter	±1 % B	±1,5 % B	..
Diameter D_i (figure 18)	±0,3 % D	±0,3 % D	±0,2 % D

1) The reference value is taken to be the scaled model average value $(\lambda_L L_M)$ unless otherwise indicated. Angular tolerance is the difference between and model angles.

2) Not required if complete profile data are provided.

2.2.2.2 Pumps and pump-turbines

2.2.2.2.1 Main components

As a minimum, the following parts shall be checked to show compliance with tolerances stated:

- the principal dimensions of the casing, diffuser, suction pipe and space between impeller/runner and head cover when necessary (figures 9 and 10);
- the principal dimensions of the impeller/runner including inlet and outlet diameters[1], outlet height, impeller/runner band and crown (figures 12 and 15);
- the number of impeller/runner blades, diffuser vanes/guide vanes and stay vanes;
- the form of the impeller/runner water passages and of the diffuser vanes/guide vanes and stay vanes, including the maximum thickness of stay vanes, diffuser vanes and impeller/runner blades;
- the seal and blade tip clearances of the impeller/runner and end clearances of guide vanes, if any;
- the roughness of all components of the machine (see 2.2.3.3);
- the waviness of diffuser vanes, guide vanes if any, and impeller/runner blades (usually only relevant for the prototype).

2.2.2.2.2 Centrifugal and mixed-flow impellers/runners

The requirements for centrifugal and mixed-flow impellers/runners are, for convenience, expressed in terms of a single-flow single-stage machine. For double-flow and multi-stage machines, additional measurements shall be taken of all inlets and stages.

- Blade inlet profiles shall be measured at least at three sections. The inlet section shall extend from the nose of the blade to a distance of 0,15[2] of the reference diameter D for both sides along the blade (see figure 14).
- Blade outlet profiles shall be checked at least at two sections for low specific speed machines and at three sections for high specific speed machines. The outlet section shall extend from the trailing edge of the blade to a distance of 0,15[2] of the reference diameter back along both pressure and suction sides of the blade (see figures 13 and 14).
- Entire blade profile (from inlet edge to outlet edge) shall be measured at least for one section, if possible, depending on the impeller/runner specific speed or randomly on the whole surface (see figures 13 and 14).
- Blade inlet angles shall be checked at the same sections as for inlet profiles. The inlet angle can be measured, for example, by using a template extending from the leading edge of the blade to a distance of 0,15[2] of the reference diameter along the blade and located to give the best fit to both sides of the blade simultaneously[3].
- The location of the impeller/runner inlet and outlet edges shall be checked at two or three points depending on the specific speed (see figure 12).
- Inlet width between blades shall be checked at least at four points on each blade (see figure 15).
- For the outlet section of the impeller/runner, the following procedure is recommended (see figures 13 and 15):

1) "Inlet/Outlet" for pump-turbines refers to pump mode.

2) This value can be reduced to 0,1 depending on specific speed and profile length.

3) Where the blade surface is sufficiently well-represented, the independant determination of angles can be omitted (see 2.2.2).

1) measurement of blade outlet profiles at two or more sections depending on specific speed;

2) measurement of the heights (b) between shrouds of the individual blade passages for impeller/runner;

3) determination of the outlet width (a) where the maximum inscribed circle is tangential to the pressure and suction faces at the outlet.

r fabricated prototype runners (assembled by welding of prefabricated components) it may agreed that the check of blade profiles can be made prior to assembly, i.e. after having mpleted machining of the individual blades. After assembly it is sufficient to check the blade sition, (e.g. as outlined in figure 12), the fillets and inlet width.

2.2.2.3 Axial flow impellers /runners

e blade profiles shall be checked using the same procedure given in 2.2.2.1.3 for Kaplan bine runners.

2.2.2.4 Guide vanes and clearances

r guide vane profiles and clearances, see 2.2.2.1.5 and 2.2.2.1.6.

2.2.2.2.5 Permissible maximum deviations in geometrical similarity between prototype and model pumps/pump-turbines

	Permissible maximum deviation		
	Uniformity tolerance		Similarity tolerance
	Model	Prototype	Prototype/Model
	Individual value to average value	Individual value to average value	Prototype average value to scaled model average value $(L_P - \lambda_L L_M)$/(reference value) [1)]
Principal dimensions of hydraulic passages			
Metallic or concrete passages (casing, draft tube, etc.)	±2 %	±2 %	±1 %
Stay ring diameters	±1 %	±1 %	±1 %
Length of diffuser vanes/stay vanes	±2 %	±2 %	±2 %
Maximum thickness of guide vanes T'	±5 %	±5 %	±5 %
Maximum thickness of stay vanes T"	±5 %	±8 %	±5 %
Stay ring/diffuser height	±2 %	±2 %	±2 %
Distributor height	±0,3 %	±0,3 %	±0,2 %
Guide vane pitch circle diameter	±0,2 %	±0,2 %	±0,2 %
Relative position between stay vanes and guide vanes(for example expressed as an angle)	±1°	±1°	±1°
Guide vane/stay vane/diffuser profile	±3 % T	±5 % T	±3 % T
Maximum guide vane opening	±1,5 %	±2 %	≥0
Clearances			
Seal and blade tip clearance	±50 %	±50 %	≤0
Seal clearance length	-	-	≥0
Guide vane end clearances	±50 %	±50 %	≤0
Radial impellers/runners [2)]			
Blade profile: inlet and outlet edges remaining part of the surface	 ±0,1 % D ±0,2 % D	 ±0,1 % D ±0,2 % D	 ±0,1 % D ±0,2 % D
Inlet pitch P_i	±0,2 % D	±0,5 % D	-
Inlet angle β_2 [3)]	±1,5°	±1,5°	±1°
Outlet angle β_1	±1°	±1,5°	±1°
Inlet and outlet width a	±3 %	$^{+5}_{-3}$ %	$^{+3}_{-1}$ %
Maximum blade thickness T [4)]	$^{+3}_{-6}$ %	$^{+5}_{-8}$ %	$^{+3}_{-6}$ %
Blade thickness near the outlet edge	±15 %	±15 %	±15 %
Inlet and outlet diameters and other impeller/runner dimensions	±0,25 % D	±0,5 % D	±0,25 % D

1) The reference value is taken to be the scaled model average value ($\lambda_L L_M$) unless otherwise indicated. Angular tolerance is the difference between prototype and model angles.

2) "Inlet/outlet" for pump-turbines refers to pump mode.

3) Where the blade surface is sufficiently well represented, the independant determination of angles can be omitted (see 2.2.2).

4) Not required if complete profile data are provided.

	Permissible maximum deviation		
	Uniformity tolerance		Similarity tolerance
	Model	Prototype	Prototype/Model
	Individual value to average value	Individual value to average value	Prototype average value to scaled model average value to reference value $(L_P-\lambda_L L_M)$/(reference value) [1]
:ial flow impellers/runners			
ade profile	±0,1 % D	±0,1 % D	±0,1 % D
›se profile	±0,1 % D	±0,1 % D	±0,1 % D
ıximum blade thickness T [2]	$^{+3}_{-6}$ %	$^{+5}_{-8}$ %	$^{+3}_{-6}$ %
ade thickness near the outlet edge	±15 %	±15 %	± 15 %
scharge ring diameter D	±0,1 % D	±0,1 % D	±0,2 % D
her runner dimensions	±0,25 % D	±0,5 % D	±0,25 % D
ıgular difference of the profile	±0,25°	±0,25°	±0,25°
:ial adjustment of the profile	±0,2 % D	±0,2 % D	±0,2 % D
ıximum blade angle	±0,25°	±0,25°	≥0°

1) The reference value is taken to be the scaled model average value $(\lambda_L L_M)$ unless otherwise indicated. Angular tolerances are the difference between prototype and model angles.

2) Not required if complete profile data are provided.

?.3 Surface waviness and roughness

?.3.1 Definitions

?.3.1.1 Waviness

aviness is the deviation of a surface profile from a smooth curve to which a flexible stick ›uld readily conform. Waviness is expressed as the ratio of maximum deviation to distance er which the deviation from the smooth curve occurs. This is the ratio X/U of figure 20. In der to distinguish waviness from surface roughness, U should be not less than 50 mm. The ›int of maximum deviation X should be in the middle third of U.

should be noted that bumps on the surface are sometimes more difficult to assess than ›llows. However, bumps are relatively easier to correct.

2.3.1.2 Surface roughness

ırface roughness is the characteristic quality of the surface due to small departures from its eneral form such as those produced by the cutting action of tool edges, abrasive grains, feed the machine, coating and painting or originally produced by the fabrication (welding) ocess.

is characterized by the roughness criterion R_a (arithmetical mean deviation from the mean ıe of the profile) as defined in ISO 468.

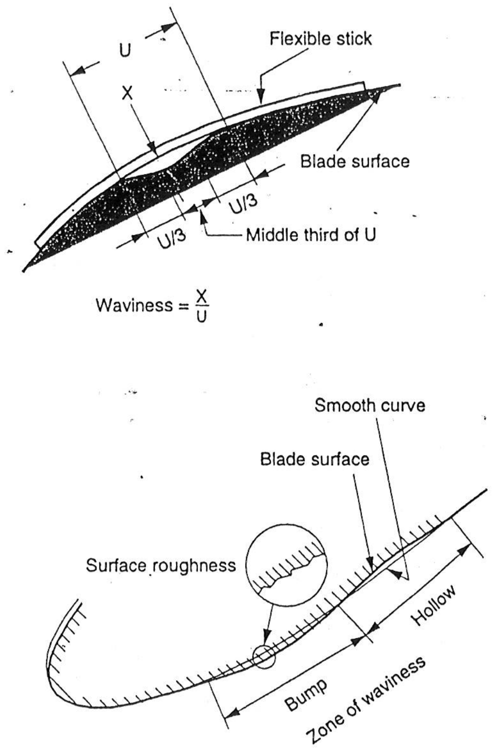


Figure 20 – Definition of waviness and surface roughness

2.2.3.2 Waviness requirements

The whole surface of the blades, including the defined profile sections and the runner/impeller crown and band shall be checked to establish that profiles are smooth continuous curves with a waviness of less than ±0,02. A flexible stick may be used for this purpose.

On surfaces subject to cavitation, the waviness shall be less than ±0,01.

2.2.3.3 Prototype roughness requirements

Rough surfaces in the water passages of both model and prototype machines reduce the efficiency below the value which is potentially obtainable.

The model should be such that a hydraulically smooth flow (see [2], annex P) is achieved, even if it is not normally the case in the most significant prototype components. It is conventionally agreed to apply the efficiency scale-up formulae described in 3.8 despite the fact that these formulae are valid only for hydraulically smooth flows.

ıe prototype roughness shall be specified taking into account the economic value of ·iciency, the cost of manufacture, the size of the machine, the likelihood of quick damage of ɘ initial surface finish due to erosion or corrosion during on site operation, etc.

ıble 1 indicates guidance values of prototype surface roughness.

ıbject to the above considerations, the recommendations for surface roughness values are lit into two groups. The boundary of these groups, in terms of specific hydraulic energy E, is ,en in table 1 for various types of machines, corresponding approximately to machines with ver to medium specific speed (medium to high head machines) and to machines with higher ·ecific speed (low head machines).

ıe values of prototype surface roughness are valid for the finished condition of the surface. ıis means that if the hydraulic wetted components have painted or coated surfaces, this ındition shall be checked.

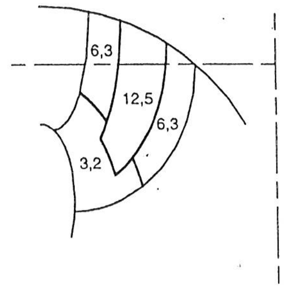
ɔreover, the finish required may be chosen at some locations to improve cavitation resistance for fatigue reasons; then the choice of finish is not related to model-to-prototype dimensional nilarity.

determining the surface roughness of fabricated components in the inlet and outlet structure, ıiral case, draft tube cone and inlet chamber of tubular turbines, the welds, due to their small ırtion of the whole surface area, are not to be included in the surface checking. However, this ;sumes that the welds are clean without contour edges. This also means that for machines th lower heads (left hand column in table 1), those welds need not be ground.

Table 1 – Recommended prototype surface roughness R_a (finished surfaces, eventual painting included)

Type of machine		Component	R_a µm	
Reaction machine	Axial		E < 300 J·kg^{-1}	E > 300 J·kg^{-1}
		Runner/impeller blades Guide and diffuser vanes Spiral case, stay ring, discharge ring, draft tube cone and tubular machine intake	≤6,3 ≤12,5 ≤25,0	≤3,2 ≤6,3 ≤12,5
	Radial or diagonal		E < 2 000 J·kg^{-1}	E > 2 000 J·kg^{-1}
		Runner/impeller blades Guide and diffuser vanes Spiral case, stay ring (including return vanes for multi-stage machines), facing plates and draft tube cone	≤6,3 ≤12,5 ≤25,0	≤3,2 ≤6,3 ≤12,5
Pelton turbine			E < 5 000 J·kg^{-1}	E > 5 000 J·kg^{-1}
		Inside of buckets and exit of nozzle Nozzle Manifold	≤3,2 ≤12,5 ≤25,0	≤1,6 ≤6,3 ≤12,5

NOTE – The values given are average values for the total surface of the component involved. Due to local hydraulic conditions, deviations up to ± 1 class of R_a may be accepted in certain areas (for instance, from R_a = 3,2 µm to R_a = 12,5 µm instead of the recommended mean value R_a = 6,3 µm for the runner blades of a radial turbine, see sketch).



Hydraulic similitude, test conditions and test procedures

.1 Hydraulic similitude

.1.1 Theoretical basic requirements and similitude numbers

heory, to achieve hydrodynamic similitude between two hydraulic turbomachines A and B ere A can stand for model and B for prototype) the following conditions should be met:

geometrical similitude between machines A and B;

dentical ratios of the various forces, acting between the fluid and the components of each nachine.

ese ratios are defined by dimensionless terms and are identified by similitude numbers.

he context of this standard, the major similitude numbers are summarized in table 2:

Table 2 – Similitude numbers

Similitude number (symbol)	Ratio of forces	General definition	Definition used in this standard
Reynolds (Re)	$\frac{\text{inertia}}{\text{viscosity}}$	$\frac{v_c \cdot L_c}{\nu}$	See 1.3.3.11.1
Euler (Eu)	$\frac{\text{pressure}}{\text{inertia}}$	$\frac{\Delta p_c}{\rho \cdot v_c^2}$	See 1.3.3.11.4
Thoma (σ)		$\frac{NPSE}{E}$	See 1.3.3.6.6
Froude (Fr)	$\frac{\text{inertia}}{\text{gravity}}$	$\frac{v_c}{(g \cdot L_c)^{1/2}}$	See 1.3.3.11.2
Weber (We)	$\frac{\text{inertia}}{\text{surface tension}}$	$\frac{\rho \cdot L_c \cdot v_c^2}{\sigma^*}$	See 1.3.3.11.3 (identical to general definition)

; is the characteristic length;

; is the characteristic velocity;

p_c is the characteristic differential pressure;

* is the surface tension of fluid.

ually, it is impossible to choose the test conditions to satisfy the various similitude numbers nultaneously. Therefore, the similitude condition to be considered should be the one with the eatest influence on the results.

most model tests, it is not possible to achieve the corresponding prototype similitude mber. Therefore, corrections have to be applied to the model results when they are nsformed to prototype conditions. Such corrections will also be required if the Reynolds mber for model performance data is different from a specified Reynolds number.

2.3.1.2 Conditions for hydraulic similitude as used in this standard

From the above basic requirements, it can be derived that two machines A and B are operated under hydraulically similar operating conditions if the following conditions are met:

a) the requirements of geometric similarity between A and B stipulated in 2.1.3.4 and in 2.2 are fulfilled;

b) the ratios of corresponding flow velocity components at any homologous point of both machines are identical, making the corresponding velocity triangles at the runner/impeller (defined by the absolute, circumferential and relative velocity components) geometrically similar.

As a consequence, both machines have, at corresponding operating points, identical discharge, energy and cavitation coefficients (see 1.3.3.12):

same discharge coefficient $(Q_{nD})_A = (Q_{nD})_B$

and same energy coefficient $(E_{nD})_A = (E_{nD})_B$

and same cavitation coefficient $(\sigma_{nD})_A = (\sigma_{nD})_B$

or identical discharge and speed factors and Thoma number (see 1.3.3.12):

same discharge factor $(Q_{ED})_A = (Q_{ED})_B$

and same speed factor $(n_{ED})_A = (n_{ED})_B$

and same Thoma number $\sigma_A = \sigma_B$

The equality of these coefficients and factors characterizes the hydraulic similitude of both machines. This is important with respect to "hydraulic" characteristics and/or data which are guaranteed or specified according to 1.4.

Other similitude conditions which are important for "mechanical" aspects (e.g. hydroelasticity etc.) are not covered by this standard.

2.3.1.3 Similitude requirements for various types of model tests

Table 3 gives an overview of similitude conditions which should be observed when performing tests on a model whose results are related to a prototype machine.

Independent of the type of test, as a minimum, the similitude conditions for discharge, specific hydraulic energy or speed and cavitation (if a cavitation influence is expected) shall be fulfilled according to 2.3.1.2 b), in order to achieve hydraulic similitude between model and prototype.

The following subclauses 2.3.1.4, 2.3.1.5 and 2.3.1.6 give detailed information on the influence of the various similitude conditions covered by this standard.

Table 3 – Similitude requirements for various types of model tests

Type of test	Similitude conditions to be observed and comments
Reaction machines	
Performance test	The possible influence of cavitation on discharge, specific hydraulic energy, efficiency and power shall be checked by cavitation tests (σ-variations) at selected operating points, independent of the agreement to perform these tests at $\sigma_M = \sigma_{pl}$ or $\sigma_M > \sigma_{pl}$.
	Efficiency, power: The influence of Re shall be considered within the guarantee range (see 2.3.1.4.1). Because usually $Re_M < Re_P$, corrections have to be applied for efficiency and power: see 3.8.2.4.
	Discharge, specific hydraulic energy: It is assumed that Re and Fr have no influence.
Cavitation test	The influence of Fr, Re and water quality (see 2.3.1.6) shall be considered. If $Fr_M \neq Fr_P$ then at least $\sigma_M = \sigma_{pl}$ shall be observed (see 2.3.1.5). See note.
Runaway test	It is assumed that Re and Fr have no influence. The influence of cavitation shall be considered
Four-quadrant and additional tests	It is assumed that Re and Fr have no influence. In some cases cavitation influence shall be checked in certain ranges of operation
Impulse turbines	
Type of test	Similitude conditions to be observed and comments
Performance test	It is recommended that Fr similitude be respected for performance tests (see 2.3.1.5.2).
	Efficiency, power: The influences of Fr, We and Re are considered according to 3.8.2.4 and annex K
	Discharge, specific hydraulic energy: It is assumed that Fr, We and Re have no influence
Cavitation test	Usually not performed
Runaway test	It is assumed that only Fr similitude shall be respected
Additional tests	It is assumed that Fr, We, Re and cavitation have no influence

NOTE – It is assumed that there is a lower limit for σ below which the influence of the Reynolds number on performance need no longer be considered, because the influence of two-phase flow is dominant (see 3.8.2.4.2).

2.3.1.4 Reynolds similitude

2.3.1.4.1 Reaction machines

Friction losses are mainly dependent on the Reynolds number provided that flow conditions are hydraulically smooth. Because the Reynolds number of the model, referred to the reference diameter of the machine (or to a characteristic length of a component) is usually smaller than that of the prototype, the ratio of friction losses to total losses for the model becomes larger than the corresponding ratio for the prototype. Therefore, in most cases, model efficiency is somewhat lower than prototype efficiency.

As a consequence, within the guarantee range, where the ratio of friction losses to total losses is important, model efficiencies and power factors or coefficients shall be corrected when they are referred to a Reynolds number different from that experienced during testing, for example if model results are scaled up to prototype conditions (see 3.8).

The influence of the Reynolds number is not considered:

– in the range of guaranteed efficiencies, if the influence of cavitation effects a drop of more than 0,5 % of efficiency (see 3.8.2.4.2);
– outside the range of guarantee, i.e. at extreme off-design operation, where the ratio of friction losses to total losses becomes small. This is, for example, the case for
  - runaway conditions;
  - shut off conditions of a pump;
  - four-quadrant operation of a pump-turbine (except within the guaranteed operating range).

2.3.1.4.2 Impulse turbines

The influence of the Reynolds number on hydraulic efficiency is considered according to annex K and 3.8.2.4.

2.3.1.5 Froude similitude and cavitation tests

Froude similitude should be respected mainly for model tests conducted under the following operating conditions:

– two-phase flow (e.g. large zones of cavitation on runner impeller blades, draft tube vortices, or in the housing of a Pelton turbine with water discharging and splashing in air);
– flow with a free surface (e.g. pump inlets with the possibility of free vortices).

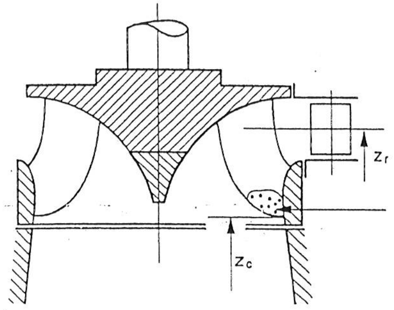
The influence of the Froude number could be especially important for impulse turbines and reaction machines of low specific hydraulic energy, when performance at plant conditions becomes influenced by cavitating flow attached to runner/impeller blades or which appears as vortex cavities in a turbine draft tube.

For cavitation tests, the application of Froude similitude should be considered only when the vertical distance between the highest and lowest points of the full size runner/impeller blades becomes significant in relation to the plant turbine/pump head. This can be the case for large horizontal axis machines operating at low specific hydraulic energy.

1.5.1 Cavitation tests on reaction machines

:avitation reference level z_c

'he cavitation reference level z_c shall be chosen to correspond to the location where the elevant cavitation occurs. This may result in the cavitation reference level z_c (see 1.3.3.7.7) leviating from the machine reference level z_r, as defined in 1.3.3.7.6, because the location /ith maximum cavitation is not necessarily at elevation z_r. The geometrical relation between $_r$, which defines the setting of the machine, and z_c is illustrated by figure 21, and the elation between the corresponding σ-values is given by the following formula which applies o both model and prototype.



$$\sigma_c = \sigma_r - \frac{g \cdot (z_c - z_r)}{E}$$

Figure 21 – Relation between the setting level z_r of a Francis turbine and the cavitation reference level z_c

Cavitation tests with application of Froude similitude

The cavitation test may be performed, unless otherwise agreed in the contract, on a large-size model, installed with the axis in the same position as that of the prototype (e.g. horizontal or vertical) and under a specific hydraulic energy as required by Froude similitude.

As a consequence, for any corresponding elevation of model and prototype, the same Thoma number or cavitation coefficient results, provided that homologous cavitation reference levels z_c are used as illustrated by figures 21 and 22.

Cavitation tests without full application of Froude similitude

If the prototype machine dimensions are not significant in relation to the plant turbine/pump head, for the model test, it is sufficient to use σ_{pl} to achieve a sufficient similitude for the cavitation pattern between model and prototype. However, it is essential that on model and prototype, homologous cavitation reference levels z_c are used (see figures 21 and 22).

The Froude condition cannot be applied, when it would result in excessively large models and/or very low test specific hydraulic energies compared with the minimum values indicated in 2.3.2.2.

In all cases where Froude similitude cannot be respected, the equality of Thoma number σ cannot be attained simultaneously for all homologous elevations on model and prototype. It is recommended that the homologous reference levels z_{cP} and z_{cM} in the prototype and model (for which the equality of Thoma numbers $\sigma_{cP} = \sigma_{cM}$ shall be observed) be selected by mutual agreement before the test (see figure 22).

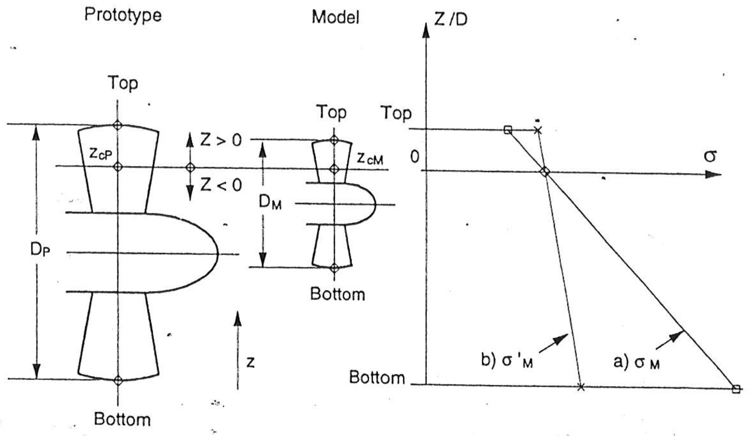


Figure 22 – Dependance of σ-values on level z for model and prototype, if:

a) Froude similitude is respected: $Fr_M = Fr_P$ then $\sigma_M = \sigma_P$

b) Froude similitude is not respected: $Fr_M > Fr_P$ then $\sigma'_M \neq \sigma_P$

The variation of σ for model and prototype between the lowest and highest points for a tubular turbine with horizontal shaft as illustrated by figure 22 is as follows:

$$\sigma_{bottom} = \sigma_c - \frac{g \cdot (z_{bottom} - z_c)}{E} = \sigma_c - \frac{g \cdot Z_{bottom}}{E}$$

$$\sigma_{top} = \sigma_c - \frac{g \cdot (z_{top} - z_c)}{E} = \sigma_c - \frac{g \cdot Z_{top}}{E}$$

If the Froude condition between model and prototype is respected, i.e. $Fr_M = Fr_P$ then on all homologous elevations the ratios:

$$\frac{g \cdot (z - z_c)}{E} \text{ or } \frac{g \cdot Z}{E}$$

are identical for both model and prototype, and consequently, identical σ-values result.

If the Froude condition between model and prototype is not respected, i.e. $Fr_M \neq Fr_P$ then on homologous elevations $z \neq z_c$ the difference $\sigma_P - \sigma'_M$ results[1]):

$$\sigma_P - \sigma'_M = g \cdot \left(\frac{(z - z_c)_M}{E_M} - \frac{(z - z_c)_P}{E_P} \right)$$

In cavitation tests where Froude similitude is not fulfilled, the net positive suction specific energy or the Thoma number should be adjusted to cover all conditions specified in the contract (variation of tailwater level), and to establish the safety margin between the minimum σ_{pl} and a contractually specified value such as σ_d (see 1.3.3.6.10), taking into account the difference $\sigma_P - \sigma'_M$ as shown in figure 22.

In some cases, more than one cavitation reference level can be used, for example in the case of a large tubular turbine with horizontal shaft. Then performance characteristics can be measured at a plant Thoma number for example referred to an elevation located 0,2 D below the top of the runner, or referred to the top of the runner, or referred to the top of the hub. Agreement between the parties will be necessary to formulate the procedure to be followed and the subsequent interpretation of results.

Cavitation tests on pump-turbines

For pump-turbines, principally the same Thoma number or cavitation coefficient as defined in 1.3.3.6.6 or 1.3.3.6.7 shall be used for turbine and pump operation.

3.1.5.2 Performance tests on impulse turbines

ıcause of two-phase flow in the turbine housing, the efficiency of impulse turbines (e.g. ılton type) may be strongly influenced by the Froude number. Therefore, it is recommended, · impulse turbine model tests, to choose a specific hydraulic energy which satisfies the oude similitude.

3.1.6 Other similitude conditions

3.1.6.1 Weber number

though this standard does not consider any influence of the Weber number (except in ınex K), it can be stated that the Weber number becomes important for two-phase flow as it :curs in the housing of impulse turbines. The degree of atomization of droplets, which luences the windage losses and/or the jet disturbance, is dependent on the Weber number.

›r model tests of impulse turbines it is usually not possible to respect Froude and Weber nilitude simultaneously. Froude similitude is maintained in most cases, because the effects ›pending on Froude generally dominate over Weber similitude effects.

When the specific hydraulic energy E_M of the test is bigger than the one required by the Froude condition, then $-z_c)_M/E_M < (z - z_c)_P/E_P$. and it follows that $\sigma'_M > \sigma_P$ for all homologous elevations above the cavitation reference vel (more favourable cavitation conditions occur on the model than on the prototype). The opposite occurs for all ›mologous elevations below the selected cavitation reference level.

2.3.1.6.2 Influence of nuclei content on cavitation pattern and performance

According to research work (see [1] and [3]) the visible extent of appearance of cavitation bubbles and the resulting cavitation characteristics can be significantly influenced by the content of nuclei in the water (non visible air or gas bubbles with a radius less than 50 µm).

At present, it is not possible to define the required minimum values for nuclei content and dissolved gas content which are linked in some way, because the impact of influencing parameters such as: type of machine, specific hydraulic energy, etc. has not yet been sufficiently established.

The measurement of nuclei content is described in 2.5.3.2. The appearance of cavitation and the possibility of observing it depend on the type of cavitation, which is correlated with the type of machine. Especially for cavitation tests (σ-variation) with Francis runners of medium or high specific speed, where cavitation usually occurs at the runner outlet, it is important that the water contains sufficient nuclei, which can be activated to grow in zones where the local pressure is equal to the vapour pressure.

Prototype measurements show that normally the number of nuclei is sufficient to produce cavitation in all zones of the runner/impeller, where vapour pressure prevails.

However, in test rigs with closed circuits, the number and the size of nuclei are reduced by degassing of the water occurring during cavitation tests. The result is that, at a defined σ-value (e.g. at σ-plant), an insufficient number of nuclei are activated to grow in the low pressure zones, which reduces the extent of visible cavitation.

Therefore, with respect to cavitation, model water quality is similar to prototype conditions if the nuclei content in the model is sufficient to ensure that cavitation development occurs in all the zones where the local pressure is equal to or less than the vapour pressure. This means that, in this condition, the extent of the zones with visible cavitation and the resulting drop in efficiency due to cavitation influence are no longer affected by different test conditions. This can be checked by varying the number of nuclei injected upstream of the runner/impeller or by increasing the test specific hydraulic energy, which could however violate Froude similitude.

2.3.2 Test conditions

2.3.2.1 Determination of test conditions

The test conditions to be chosen for the different types of tests are dependent upon:

- capacity of test installation and its instrumentation;
- size and mechanical design of the model machine;
- guaranteed and/or specified operating range.

The same model shall be used for performance and cavitation tests.

If guarantees or specified data are given for the prototype, the required model test parameters can be calculated from hydraulic similitude, defined in 2.3.1.2 b), applying the formulae in 3.8.2.5. It is essential that the following aspects are taken into account:

- the permitted minimum values of table 4 shall be satisfied;
- the mechanical limitations due to model design shall be checked;

the limitations due to design and operation of the test rig and available instrumentation shall be considered.

dels used for comparative tests shall be the same size and tested at approximatively the ne Reynolds number (see 2.1.3.2)

performance tests, it is sometimes recommended to choose a test specific hydraulic ergy higher than the minimum value to achieve a higher Reynolds number (Froude number impulse turbines), which is closer to that of the prototype, in order to reduce the relative ccuracies in measurement.

s unnecessary to require that cavitation model tests at a specific hydraulic energy be equal that of the prototype. However, it is generally not desirable to test the model at very low cific hydraulic energies because of the reduced measurement accuracy and the risk of gassing in the low-pressure region.

m the above considerations, the absolute range of values of the following model quantities, the test conditions for each type of test, can be defined:

speed or specific hydraulic energy;

discharge;

power and/or torque;

net positive suction energy (for reaction machines).

.2.2 Minimum values for model size and test conditions to be fulfilled

order to achieve good hydraulic similarity between model and prototype, it is necessary to fine minimum values for model size, Reynolds number, and test specific hydraulic energy as en in table 4, in addition to the requirements for geometrical similarity and surface ghness of model and prototype stipulated in 2.1.3.4 and 2.2.

ese minimum values shall be satisfied in order to:

ensure the required dimensional accuracy, with normal manufacturing techniques;

obtain test results with sufficient measurement accuracy, independent of inadmissible test conditions (e.g. due to air or gas separation during cavitation tests);

reduce scale effects between model and prototype, by testing at appropriate Reynolds and Froude numbers.

e various minima are independent of each other and shall all be satisfied. Normally, models ould be as large as practicable, but never less than the values stated.

Table 4 – Minimum values for model size and test parameters

Parameter	Type of machine			
	Radial (Francis)	Diagonal (Mixed-flow)	Axial (Kaplan, bulb)	Impulse (Pelton)
Reynolds number Re (-)	4×10^6	4×10^6	4×10^6	2×10^6
Specific hydraulic energy (per stage) E ($J \cdot kg^{-1}$) (note 1)	100	50	30 (note 2)	500
Reference diameter D (m)	0,25 (note 3)	0,30	0,30	------
Bucket width B (m)	------	------	------	0,08

NOTES

1 With respect to the Froude similitude condition, the test specific hydraulic energy for cavitation tests may be chosen so that the resulting Re number is lower than the value given.

2 $E_{min} = 20\ J \cdot kg^{-1}$ if $D \geq 0,4$ m

3 For pumps and pump-turbines with low specific speed, a reference diameter such as $0,20\ m \leq D \leq 0,25$ m may be allowed if the outer diameter is equal to or greater than 0,5 m.

2.3.2.3 Stability of test conditions

Fluctuations in the various measured quantities cannot be avoided. Such fluctuations can be of periodic or random nature within a large frequency range. They can be generated by elements of the test rig (e.g. booster pump, throttling devices, control system for low head pressure, etc.) and/or by the model machine (e.g. vortices in the draft tube, rotating stall, etc.). During model tests it is essential that at all operating points, repeatable steady-state operating conditions are achieved, especially within the range of guaranteed operation of the prototype.

2.3.2.3.1 Stability and fluctuations during measurements

Before and during the measurement of a point, the operating stability of the test rig shall be such that repeated measurements of the same operating point are within the agreed random uncertainties of the various measured quantities (e.g. within ±0,3 % for efficiency close to best efficiency conditions). This also means that drifting effects of discharge, specific hydraulic energy, speed and net positive suction energy shall remain small (e.g. within ±0,3 %). Drifting effects usually introduce systematic errors, and shall therefore be eliminated.

Fluctuations during the measurement of a point can be checked by means of the standard deviation of a measured quantity or the derived random uncertainty if the test rig is equipped with a suitable data acquisition system. If the results are affected by fluctuations, it may be necessary to apply linear hydraulic damping (see 3.3.3.4), to adjust the filtering of electric signals (see 3.1.4.3.2), to adjust the test conditions or to apply other means to eliminate these effects. However, such means shall not bias the measured quantity.

2.3.2.3.2 Adjustment of the operating point

When the operating points to be measured are defined by specified dimensionless values (speed factor, discharge factor, Thoma number or other dimensionless terms) corresponding to the contractually specified ones, it is necessary to adjust the test conditions so that they are as close as possible to the specified data. Deviations between measured and specified dimensionless values should not exceed ±0,5 % for speed, discharge or power coefficient or factor and ±3 % for the cavitation coefficient or Thoma number.

.3 Test procedures

.3.1 Organization of tests

en planning model tests, the following items covering general aspects of preparation, anization and realization shall be clarified and agreed on by the parties involved.

.3.1.1 Specification for model tests

pecification for model tests is the basis for planning and preparing a model acceptance test I for drafting the detailed technical programme of tests (see 2.3.3.3.2). This model test ›cification is often part of the general technical and/or commercial specification for a whole ject (e.g. supply and manufacture of a turbine or replacement runner/impeller) issued by the chaser or its engineer.

ong other items, the specification for model tests shall define:

object, scope and extent of model tests;

reference to guaranteed and specified plant data;

reference to test standards;

model scale and/or size;

place(s) where the model(s) or certain components are manufactured;

place(s) where the model(s) is (are) tested;

documentation of results (test reports);

time schedule indicating, at a minimum, start and end dates of work for model testing.

.3.1.2 Time schedule

ime schedule should be agreed on, which indicates as a minimum, the different stages, adlines and/or duration for the following items:

submission of model drawings (especially those showing model construction and principal dimensions of model high and low pressure side at the transition to the test rig);

description of testing equipment (including methods of calibration, calculation and representation of results and expected uncertainties of measurement);

calibration of instruments;

check of model geometry;

preliminary and acceptance (or witnessed) tests and sequence of models to be tested in the case of comparative model tests;

test reports.

e manufacturer shall always be given sufficient time to manufacture the model, making all cessary dimensional checks and carrying out his own preparatory tests, whether in his own ›oratory or elsewhere. In case of test equipment defects or model machine defects arising ring preliminary or acceptance tests, a mutual agreement is necessary to modify the time hedule and/or the test programme.

3.3.1.3 Personnel and responsibilities

is important that the responsible persons of the purchaser and/or his engineer, of the pplier and of the independent laboratory if any are designated in ample time before test ›mmencement. Their responsibilities and authority shall be clarified, so that any problems ising during the preparation for and conducting of tests can be quickly resolved.

Both purchaser and supplier shall be entitled to have authorized representatives present at all contractual tests, inspections and dimensional checks of the model and prototype, in order to verify that they are performed in accordance with both this standard and any prior agreement.

For the carrying out of tests (as defined in 2.3.3.3.1), a chief of tests shall be designated. The chief of tests can be selected from the technical staff of the laboratory where the model is tested or, by special agreement between the parties, an independent expert can act as chief of tests. The chief of tests will assume full responsibility for the correct calibration of instruments, execution of tests and computation of results, including determination of uncertainties of measurement and documentation of results in the final test report. The chief of tests shall consider any remarks or suggestions by any of the officially authorized representatives of the parties attending the tests.

When model tests are conducted in an independent or external laboratory, the supplier can be present during all activities for the test of his model, including the installation and any preparatory tests.

The tests shall be carried out by personnel having experience with the test equipment.

2.3.3.1.4 Design of the model and preparation of the test installation

In order to prepare the model and the testing equipment, the technical specification for model tests (see 2.3.3.1.1) shall provide all relevant details for design and manufacturing of the model machine such as:

- extent of the water passages to be modelled on the high and low pressure sides;
- location of the pressure measuring sections;
- number and location of observation windows;
- provisions for exchange and/or adjustment of model components;
- provisions for special tests (video recording of cavitation pattern, pressure fluctuations, guide vane torque, axial or radial thrust, velocity profiles, etc.);
- test conditions for all types of specified tests;
- adjusting range for geometric parameters (e.g. maximum distributor opening, etc.).

In addition, sufficient data defining the extent and conditions for all the tests shall be given to enable the model to be constructed and prepared according to the state of the art. It is then possible to select the appropriate test facility, and to check the general arrangement of the model in the test rig, modifying or rearranging some of the existing components of the test rig if necessary.

It is recommended that inflow and outflow parts as well as the definitive location of the pressure measuring sections and the number of pressure tappings are agreed upon at an early stage of model design.

2.3.3.1.5 Instrumentation and processing of test data

The person responsible for the selected laboratory or chief of tests shall provide the purchaser, or his engineer, with a description of the standard instrumentation installed on the test rig, including details on the measuring and calibration methods, on the processing of calibration and test data and on documentation of results.

In some laboratories, there are two or more instruments permanently installed for measuring a particular quantity. One shall be selected as the instrument whose measuring signals or indications are to be used in calculating the results, and the other(s) shall be used only for reference and functional control purposes.

netimes the calculation of test results occurs in several steps using more than one ıputer system. In this case, the evaluation procedure for the quantities concerned shall be ned in detail. It shall be agreed which quantities are processed on-line and immediately sented on records and/or diagrams, and which test results are calculated and documented r.

.3.1.6 Dimensional checks

› components which are to be checked before, during or after the acceptance tests shall be cified. The methods to be used for dimensional checks and the approximate extent and ıber of dimensions to be checked (see 2.2) shall be agreed in sufficient time to enable ıvant documents and measuring equipment to be prepared accordingly. If accepted by the ties, it is possible to make spot-checks during the acceptance tests.

.3.2 Inspections and calibrations

ırtly before the start of the preliminary and/or acceptance tests, the model, the test rig, the rumentation and the data acquisition and processing system shall be thoroughly inspected the representatives of the parties and the chief of tests, to ensure that the test results are affected by any mechanical, structural or other defects of the model or the testing ıipment.

measuring instruments should be calibrated against primary methods (as explained in 3.2 to) prior to the tests and also after completion of tests if one of the parties so desires. calibrations during tests may be necessary if serious problems with the standard measuring ıipment occur, such as a defect of an instrument or a measuring chain or a significant and tematic deviation of a measuring signal at zero or reference conditions.

› methods to be used, the extent of various calibrations (spot-check or complete calibration) l the procedure shall be agreed on by the parties before beginning the preliminary or eptance tests. Based on certified documents produced by the responsible of the laboratory chief of tests, it may be mutually agreed, for certain instruments, that the latest existing ibration data be used without further calibrations or checks.

.3.2.1 Inspection of the model machine

ns to be checked and/or recorded are:

identification of model components by means of the corresponding model drawings (especially if it is planned to exchange or modify certain components during testing);

geometric dimensions as listed in 2.2;

characteristic dimensions used for calculation of test results (e.g. areas of high pressure and low pressure measuring sections);

seal clearances and/or blade tip clearances;

individual and average values of guide vane angle or needle openings and/or runner/impeller blade openings;

surface quality, impurities and local faults in the different components;

proper matching at the joints of components;

conformity with reference dimensions on the model drawings.

ıecessary, these checks shall be repeated if mechanical faults or defects occur during the its (see 2.3.3.4), or at the end of witnessed tests, in the presence of the authorized presentatives of all parties concerned. These checks are part of the official acceptance test.

2.3.3.2.2 Inspection of the test circuit

As a minimum, the following checks shall be carried out:

- no water leakage or supply shall occur between the model and the discharge measuring section;
- no water leakage shall occur at pressure tappings and measuring pipes (on the low pressure side, no air leakage into the water circuit shall occur if the internal pressure is below ambient pressure);
- flow conditions close to the model inlet and outlet and close to the discharge measuring section shall be regular. The water passages shall have no disturbances and the wetted surfaces shall be of good quality;
- the booster pumps and regulating devices (valves, supply of water and pressurized air, etc.) shall work properly;
- the water quality and temperature shall be stable (see 2.1.2.3).

If necessary, some checks shall be repeated at the end of the acceptance tests, in the presence of all parties concerned.

2.3.3.2.3 Calibration of instruments and check of the data acquisition system

It shall be agreed by the parties concerned:

- to what extent the various instruments shall be calibrated or spot-checked;
- if all instruments require to be spot-checked or calibrated before and after acceptance tests;
- the conditions for the recalibration of an instrument during or after acceptance tests;
- range of calibration and number of calibration or check points for the various instruments;
- the basic data and the procedure for evaluating the systematic uncertainties related to the calibration data and to each measured quantity.

Before and after the acceptance tests, as a minimum, the following checks or measurements shall be made:

- identification of instruments and/or measuring devices used;
- "zero readings", i.e. readings in well defined conditions for the different instruments, to detect if any drifting effects occurred during the tests;
- check of the data acquisition system by repeated measurements in well defined operating conditions. Manual sample calculations shall demonstrate that acquisition, transmission and processing of test data (usually automatic) work properly;
- measurement of mechanical friction torque in bearings and shaft seals which are not built into or connected to a stator of the swinging type (automatically yielding the torque on the runner or impeller), in order to decide if any further corrections should be applied.

The "zero readings" are usually checked during preliminary and acceptance tests. All the other checks and calibrations will be repeated, if necessary, at the end of the acceptance tests in the presence of all parties concerned.

the differences between the two checks or calibrations compared are less than the stematic uncertainty evaluated at the beginning of the tests, the test data are valid and no rection needs to be applied. If, between the two calibrations or checks, differences greater in the anticipated systematic uncertainty occur in several points, the test shall be cancelled d repeated.

order to ensure the validity of calibration results, it is essential to maintain the influence antities (such as ambient temperature and humidity, power supply, electromagnetic field, :.) within a reasonable range during calibration and during tests (see 3.9.2.1.2)

y of the interested parties may, for declared and substantiated reasons, call for a calibration of any instrument during the course of a test.

3.3.3 Execution of tests

3.3.3.1 Type of tests

conduct an acceptance test successfully and efficiently requires good preparation by means adequate preparatory tests. Depending on the results of preliminary and acceptance tests, it sometimes necessary to perform additional tests.

r easier identification of the status of tests and for improved communication between the rties involved, this standard defines the following types of tests:

Preparatory tests[1)]

Often these tests are not specified in detail, but they are essential with respect to the quality of all further tests. They comprise:

- checks and tests to ascertain that model performance and cavitation pattern are not affected by any mechanical defects of model or test rig and measuring equipment;
- a check on the correct operation of the measuring equipment and data acquisition system. Often, it is helpful to perform a systematic variation of the test speed (i.e. a variation of E) at constant σ and a variation of NPSE (i.e. a variation of σ) at constant E, for at least one operating point.

se of results:

normally, for internal use of the supplier or the laboratory only.

Preliminary tests

These are tests covered by the technical specification (see 2.3.3.1.1) or technical programme (see 2.3.3.3.2), and the extent is mainly dependent on the future use of the results. Therefore, the parties involved have to agree before beginning the tests, whether:

- the results are to be used for information only and have no contractual value. In that case, the preliminary tests are used only to explore the general behaviour or some limits of the model with respect to specified items;
- or whether the results will be used officially and have contractual value. In that case, these results are part of the official test data to be completed or spot-checked during the acceptance tests. For example, the results could be used to establish the efficiency step-up, as explained in 3.8.2.4, and the systematic and random uncertainties as explained in 3.9.2. However, such data shall be confirmed during the acceptance tests.

[1)] If mutually agreed, sufficient time should be allowed to the supplier during this period for the selection of lefinitive components or any last-minute adjustments, but these final development tests are not included in the scope of this standard.

c) Acceptance tests (or witnessed tests)

These tests measure, establish or check all relevant model data which are defined by the technical specification (see 2.3.3.1.1) or technical programme (see 2.3.3.3.2), and which form the basis for comparison with guarantees or other contractually specified data.

Use of results:

all results are of contractual value and shall be summarized in the final test report (see 2.3.3.5).

d) Additional tests

These are tests which supplement the results of preliminary and acceptance tests and which may include tests for additional data explained in 1.4.4. Such tests may or may not be witnessed.

Use of results:

these results are also summarized in the final test report (see 2.3.3.5). However, their possible contractual value shall be agreed on.

2.3.3.3.2 Technical programme

A technical programme, referring to the technical specification (see 2.3.3.1.1) and the relevant contractual guarantees and data, shall be established in advance and agreed upon by the parties involved. If not covered by the contractual documents, the programme shall define the purpose and the extent of the various types of tests to be performed during preliminary and/or acceptance tests. To allow efficient progress in testing and documentation, the following shall be specified for each type of test, such as main hydraulic performance tests, cavitation tests, guide vane torque tests, etc.:

- ranges of hydraulic parameters to be varied and corresponding increments, to define the number and distribution of test points;
- test conditions to be maintained constant, i.e. usually the test speed (or specific hydraulic energy) and the Thoma number (tests at σ_{pl} or at higher σ-values);
- number and type of test data to be recorded and the method of recording;
- definitions, formulae and procedures (see 3.8.1 and figure 62) to be used for calculation of model and/or prototype results;
- guidelines for graphical representation of test results.

If necessary, for the acceptance tests, the technical programme can be supplemented, in order to clarify and define, for example:

- extent and sequence of checks, calibrations and tests to be witnessed;
- recording and/or graphical representation of test data to be signed by the parties to the tests;
- number and definition of the test point(s) to be measured for a sample calculation and for calculation of systematic, random and total uncertainties;
- preparation of a daily log and of a final protocol (see 2.3.3.3.9).

2.3.3.3.3 Data to be used for calculation of results

Before the beginning of preliminary and/or acceptance tests, the data and formulae to be used for calculation of model and prototype results shall be checked and agreed on. Such data are:

- areas of high- and low-pressure measuring sections;

ȝeometric and hydraulic references and data of model and prototype;

ɔhysical constants and properties for model and prototype;

ɱodel friction torque (if not automatically compensated);

ɔalibration data for all relevant instruments;

leakage flow-rate through labyrinth seals, if considered (see 2.1.3.4.3);

efficiency step-up procedure.

sample calculation shall demonstrate the correct use of these data and explain the culation procedure.

.3.3.4 Signing and handling of test records

mplete records of measured model data and corresponding data at zero or reference ɪditions, as well as notes from inspections and calibrations, readings and observations ·ing the acceptance tests and/or preliminary tests shall be agreed on and signed by the nessing parties and by the chief of tests immediately on completion of each phase, and kept a complete set by each party.

..3.3.5 Performance tests

s recommended to determine first the best efficiency point in non cavitating conditions, i.e. $_{\text{plM}}$ (for a pump-turbine, in both modes of operation) as the basis for calculation of δ_{ref} (see ..2.2) and efficiency increase $\Delta\eta_h$.

least within the range of guarantee, the performance tests shall be carried out at constant eed or at constant test head. For computation of test results (see 3.8), it is preferable to rform the tests at constant speed and water temperature (if possible), so that the resulting ·ynolds number is approximately constant. If, due to limitations of the model and/or the test ;tallation, constant test conditions cannot be maintained, 3.8.2.2 explains the further ɔcessing required of such performance test results. In the same subclause, some typical aphical presentations of results of performance tests for various types of machines are own (see figures 55 to 61).

Reaction machines: influence of cavitation

Performance tests can be carried out in one of the following two test conditions:

– at plant Thoma number $\sigma_M = \sigma_{pl}$

  In this case, especially for machines of higher specific speed, cavitation which can affect efficiency could occur within the range of guaranteed efficiency. This has to be checked by means of σ-variations in the affected zones as explained and illustrated in 2.3.3.3.6.

– at non-cavitating condition, with $\sigma_M > \sigma_{pl}$

  This means that the Thoma number σ_M is sufficiently high to avoid any cavitation. As a consequence, the possible influence of cavitation on the performance curves at plant conditions is to be checked by σ-variations as explained in 2.3.3.3.6. If these tests reveal that within the range of guarantee an influence exists, figure 70 (see 3.8.2.3.7) explains the correction procedure for the efficiency curve measured with $\sigma_M > \sigma_{pl}$.

ı Impulse turbines: influence of tailwater level

For impulse turbines, it is recommended to determine the tailwater elevation at which performance is affected. This check is to be done at selected full load operating points, by varying the tailwater level.

2.3.3.3.6 Cavitation tests on reaction machines

Cavitation tests, i.e. systematic variations of the Thoma number σ, combined with recording of the cavitation pattern, are performed at selected operating points. The results are usually represented as illustrated in figures 71 and 72. At specified values of σ (e.g. at σ_{pl}) it can be demonstrated how cavitation influences performance (efficiency, discharge or specific hydraulic energy, power). It is the only way to check the safety margin between the σ-values related to plant conditions and the σ_d-value at which performance is or begins to be affected by cavitation (see figure 70).

At each σ-variation it is important that the chosen geometric model parameters remain constant. It shall be agreed, whether the energy coefficient (or speed factor), or the discharge coefficient (or discharge factor) shall be kept constant.

If the model is equipped with suitable windows or transparent parts, it is possible, at specified values of σ, to observe the cavitation pattern at the runner/impeller and/or in the draft tube (where, for example, the vortex rope at part load operation of Francis turbines becomes visible), by the use of stroboscopic light. It is also possible to observe the cavitation pattern by introducing an endoscope into the model. Usually, this cavitation pattern is recorded by means of manual sketches, photos or video.

The results of cavitation tests may be used to indicate other phenomena associated with cavitation such as noise, vibration and pressure fluctuation.

For the consequences of not respecting the Froude similarity for large low specific hydraulic energy machines, see 2.3.1.5.1 c).

Contractual parties shall agree on the different reference levels and the corresponding σ_{pl} values to be used (see 2.3.1.5.1 and annex M), as well as on the method of how the influence of cavitation on performance, if any, should be considered when transposing the results to prototype conditions (see 3.8.2.4.2).

2.3.3.3.7 Runaway tests

The runaway test method depends on the design of the test rig, the instrumentation and the model design. If the test rig is, or can be, equipped with a driving motor to compensate friction torque due to shaft bearings and seals, it is usually possible to maintain $P_{mM} = 0$, and thus to establish directly the points of the runaway characteristics. If this procedure is not possible, the runaway conditions can be determined by extrapolation (as an example see figure 76) or by interpolation.

In most cases, the specific hydraulic energy is reduced, so that the highest speed which can be endured by the model and/or the testing facility is not exceeded. However, the minimum runaway speed of the model should not be lower than the speed used for performance testing. The influence of Reynolds and Froude numbers is assumed to be negligible in the range near to runaway.

The runaway tests shall be carried out with a sufficient variation of geometric parameters of the model machine, so that the most unfavourable combination of parameters and all specified conditions are covered. For multijet impulse turbines (Pelton turbines), the maximum steady-state runaway speed shall be measured taking into account the most unfavourable combination of operating nozzles.

Subclause 3.8.3.1 explains how the model runaway characteristics are determined for different types of machines and figures 74 to 76 give some examples of graphical representation of the runaway speed factor $n_{ED,R}$.

reaction machines, the influence of cavitation on model runaway data is to be checked (see 3.2). The relevant σ_{pl}-value shall first be agreed on; it shall relate to the most unfavourable ıt conditions which can occur during runaway (often at maximum specific hydraulic energy). ıre are then two possible methods for checking the influence of cavitation:

ɔerform the model runaway tests at a sufficiently high Thoma number, and then check at ɜelected, critical operating points the influence of cavitation by means of σ-variations, i.e. ɜstablish curves $n_{ED,R}(\sigma)$ and $Q_{ED,R}(\sigma)$ for each of these points;

ɔarry out the model runaway tests at $\sigma_M = \sigma_{pl}$ and at $\sigma_M > \sigma_{pl}$.

.3.3.8 Tests to check additional data

ı procedures applied for tests, such as:

ɔressure fluctuations;

ɟuide vane torque;

our-quadrant performance characteristics;

ɑxial/radial thrust;

ɜtc.

described in clause 4 which details the relevant aspects of these additional tests.

.3.3.9 Daily log and final protocol on acceptance tests

aily log shall summarize, for each day:

ıames of the persons participating in the tests;

ɑctivities such as checks, calibrations, test series, discussions;

ɑgreements, decisions and unresolved issues with respect to test results;

ɱodifications of the technical programme and/or test programme.

he end of the acceptance tests, a final protocol shall be established covering at least:

ɔurpose of acceptance tests;

location and date of tests;

names of persons participating;

identification of the model and/or the model components;

comments and/or conclusions on:

- inspection of test rig, instrumentation and installation of the model;
- calibration of instruments;
- data acquisition system (sample calculation);
- check of model geometry;

discussion of results and comparison with guaranteed and/or specified data, covering at least:

- performance tests;
- cavitation tests;
- runaway tests;

- conclusions regarding whether:
  - guarantees and specifications have been met;
  - test results are complete with respect to the technical programme and technical specification for the model tests;
- agreements on:
  - additional tests (if any);
  - documentation of test results;
  - shipment or storage of the model.

2.3.3.4 Faults and repetition of tests

2.3.3.4.1 Types of faults and consequences

During the acceptance tests, faults in the model, test rig, instrumentation or data processing may arise, such as:

a) mechanical faults in the model:
   - bearing or seals of the model fail and effect a change of mechanical friction losses. Replacement bearings and/or seals need to be fitted, but the mechanical friction losses may have changed;
   - running clearances and/or throttling conditions in piping for leakage flow-rate may have changed, effecting changes in hydraulic performance;
   - guide vanes and/or runner/impeller blades become misaligned affecting performance and/or cavitation pattern;
   - mechanical defect on runner/impeller or at other model components occurs.

b) faults in the test rig or instrumentation:
   - unusual variations of speed, specific hydraulic energy or discharge are introduced due to faults in the auxiliary control systems of the test rig or due to malfunctioning of components;
   - additional leakage due to an open valve or a defect in the measuring leads is detected;
   - excessive shift of instrument indications at zero conditions before and after a series of tests is detected.

c) faults in data processing
   - malfunctioning of the data acquisition system may produce faulty test results;
   - faulty test results are produced due to incorrect reference or calibration data.

The correction of such faults shall be closely supervised by the chief of tests and all parties concerned.

In comparative tests, particular care shall be taken to ensure that none of the parties is advantaged from the results obtained up to that point. The tests shall be completed in the manner they began and without modification of the hydraulic design.

After correction of the fault, several preparatory tests and/or preliminary tests should be made to ensure that the model is behaving exactly as it was before the faults occurred. If its performance is proven to have changed, then by mutual agreement, the previous series of tests shall either:

be allowed to stand, and no further tests be deemed necessary;

or be declared invalid and the whole series of tests repeated.

.3.4.2 Procedure for repetition of tests

y of the parties shall have the right to require interruption and/or repetition of the tests, ›vided the chief of tests deems valid the reasons put forward, for example:

unsatisfactory agreement between calibrations before and after tests;

test rig, instrumentation or data processing malfunction;

mechanical faults in the model;

non-compliance, to a significant degree, with this standard, except as otherwise agreed beforehand.

such a situation, the other party(ies) and/or the chief of tests may also demand a repetition the tests. It is recommended that the subject of repeated tests, repeated calibrations and the sponsibility for the associated costs be subject to written agreement beforehand.

no agreement can be reached on how to conduct such a repetition of tests or who is sponsible for the additional costs, the matter shall be referred to an independent arbitrator ceptable to all parties.

.3.5 Final test report

ter completion of all tests according to the technical specification for model tests (see .3.1.1) and the technical programme (see 2.3.3.3.2), a final test report shall be prepared cording to the rules set out in this standard and signed by the chief of tests. The parties shall ree whether a draft of the whole report or only selected chapters have to be approved before stributing the final version.

e final report shall cover primarily the following items:

object and purpose of tests, with reference to technical specification of model tests including relevant guarantees and other contractual data;

records of all agreements, and other essential documents pertinent to the tests;

personnel taking part in the tests;

description of the model machine together with drawings showing, as a minimum, the main section of the model and its general arrangement in the test rig;

description of the test rig and the measuring equipment, including calibration methods and data processing;

calculation of model test results and transposition to contractual model and/or prototype conditions (including consideration of scale effects, if any);

calibration data and inspection reports;

test procedures for the different types of tests;

log covering activities related to the specified tests and the sequence of different tests;

)) relevant test records and data sheets from measurements and observations of the various tests, together with the test results graphically represented;

1) calculation of uncertainties of measurement with reference to the calibration data, results and further observations;

12) discussion and interpretation of test results and comparison with guaranteed and other contractual data;

13) conclusions as to whether or not the guaranteed and contractual requirements have been met, and the tests are complete with respect to the technical specification.

2.4 Introduction to the methods of measurement

It is recognized that no physical quantity can be measured without error. Consequently, the result of any measurement is worthless if not accompanied with the associated uncertainty estimated at a given confidence level. General prescriptions on analysis and combination of uncertainties are given in 3.9.

The quantities to be measured during a model test to verify the main hydraulic performance guarantees defined in 1.4.2 require a high level of accuracy. For this reason, the prescriptions given in clause 3 of this standard are mandatory in order that the test may be deemed to be in accordance with this standard. Subclause 2.4.1 gives prescriptions on how to calculate the quantities (power, efficiency) which are derived from the basic quantities (discharge, specific hydraulic energy, shaft torque, speed of rotation). Hydraulic quantities such as NPSE and σ related to the influence of cavitation on the hydraulic performance are determined from the same physical measurements, and thus with the same accuracy, as those used for the above basic quantities.

Additional quantities are determined mainly for guidance, even if some of them may be of contractual interest (see clause 4).

2.4.1 Measurements related to the main hydraulic performance guarantees

2.4.1.1 Hydraulic efficiency

The purpose of a model acceptance test on a hydraulic machine in accordance with this standard is to compare the achieved hydraulic performance, expressed either as the measured model performance or transposed to prototype performance, with guarantees given by the supplier.

Since there is no correlation between the mechanical losses (power dissipated in the guide bearings, thrust bearings and shaft seals) of the model and those of the prototype, the comparison shall be based on the mechanical power of runner/impeller P_m (see 1.3.3.8.3) and hydraulic efficiency η_h (see 1.3.3.9.1) and not the mechanical power of the machine P (see 1.3.3.8.2) and efficiency η (see 1.3.3.9.3).

Hydraulic efficiency is calculated from the mechanical power P_m transmitted through the coupling of the runner/impeller and the shaft and the hydraulic power P_h exchanged with the water. It is given by:

$$\eta_h = \frac{P_m}{P_h} \text{ for a turbine,}$$

and

$$\eta_h = \frac{P_h}{P_m} \text{ for a pump.}$$

It shall be noted that according to these definitions the disc friction losses and leakage losses (volumetric losses) are considered in this standard as hydraulic losses, and therefore no correction is made.

ı principle of the method involves the measurement of discharge Q, specific hydraulic ırgy E, torque T and rotational speed n.

ı direct determination of hydraulic efficiency by the thermodynamic method is not ommended for model acceptance tests.

.1.2 Hydraulic power

ɜ determination of hydraulic power requires the knowledge of the specific hydraulic energy of machine and of the mass flow rate through the high pressure reference section of the del. The formula is:

$$P_h = E\,(\rho\,Q)_1$$

/ transfer of water to or from the system between the reference section and the discharge asuring section shall be taken into account. Where the volume flow rate is measured, the ue obtained shall be associated with the value of water density in the conditions of pressure ɟ temperature prevailing in the discharge measuring section.

ɔclause 2.5 explains how to determine the values of physical quantities such as the local ɔeleration due to gravity, the water density, etc. either by direct measurement or from ɜrnationally recognized formulae or tables. Methods of discharge measurement are ɜcribed in 3.2. Methods of specific hydraulic energy determination, from pressure asurement in accordance with 3.3 (or possibly from water level measurement in accordance h 3.4), are described in 3.5.

.1.3 Mechanical power

lculation of the mechanical power from the measurement of electrical power at the ıtor/generator terminals and from the efficiency of the latter is not recommended for model ɔeptance tests. Thus the determination of mechanical power at the runner/impeller requires ɔwledge of the torque supplied by/applied to the runner/impeller and of the rotational speed:

$$P_m = 2\,\pi\,n\,T_m$$

ɜthods of torque measurement are described in 3.6. Methods of rotational speed ɜasurement are described in 3.7.

ɬ.1.4 Computation of efficiency

om the definitions given in the previous subclauses, the hydraulic efficiency of a model in the ıerating conditions prevailing during a point may be calculated by:

$$\eta_h = \frac{2\pi n T_m}{E(\rho Q)_1} \text{ for a turbine,}$$

ıd

$$\eta_h = \frac{E(\rho Q)_1}{2\pi n T_m} \text{ for a pump.}$$

When the guarantees are referred to the prototype performances, the hydraulic efficiency and mechanical power of runner/impeller of the prototype shall be determined from the corresponding quantities of the model by applying a scale-up formula, as explained in 3.8.2.4. The mechanical losses of the prototype shall then be taken into account to determine the mechanical power of the machine P (power delivered by the turbine shaft or to the pump shaft) and the total efficiency of the hydraulic machine $\eta = \eta_h \cdot \eta_m$, more simply called "efficiency" in this standard (see 1.3.3.9.3).

2.4.2 Measurements related to additional data

In addition to the verification of the main hydraulic performance guarantees, model tests may be used to determine some additional data (see clause 4). This implies the measurement of the stationary and/or fluctuating components of various hydraulic or mechanical quantities.

2.4.3 Acquisition and processing of data

Whatever the quantity to be measured, particular care shall be given to the method of averaging a fluctuating signal to obtain the true mean value of the physical quantity and to the analysis of this signal to characterize the frequency and amplitude of the fluctuations. Subclauses 3.1 and 4.2 give guidance on the requirements of the system of measurement and on data processing for obtaining average and fluctuating quantities respectively.

2.5 Physical properties

2.5.1 General

This subclause defines the main physical properties needed to characterize the hydraulic behaviour of hydraulic machines. The terms and definitions of most of these quantities are listed in 1.3.3.3 together with their symbols and units.

The formulae which can be used in data processing to calculate these quantities are listed in the following subclauses. For convenience, tables of numerical values derived from these formulae are given in annex B.

2.5.2 Acceleration due to gravity

Acceleration due to gravity g (see 1.3.3.3.1) is given as a function of latitude and altitude:

$$g = 9{,}7803\,(1+0{,}0053 \sin^2 \varphi) - 3 \cdot 10^{-6} \cdot z$$

where

φ is the latitude in degrees, and

z is the altitude in metres.

The computed values of g are given in table B.1 and are represented in figure 23.

The international standard value of g is 9,806 65 $m \cdot s^{-2}$.

If measured values of g are available, they shall be used. The local value of g can be measured for example by a pendulum or free fall (in vacuum).

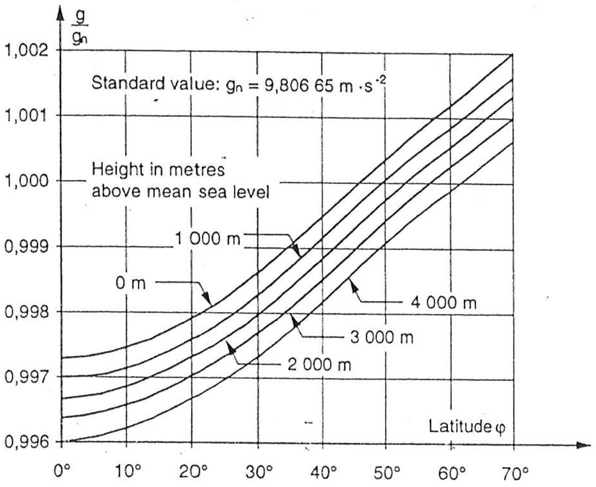


Figure 23 – Acceleration due to gravity g (m · s^{-2})

.3 Physical properties of water

.3.1 Density of water

.3.1.1 Application of the density of water

testing hydraulic machines, the density of water ρ (see 1.3.3.3.3) shall be known in order to:

determine the specific hydraulic energy of machine E from pressure measurements (see 1.3.3.6.2);

determine the mass flow rate (ρQ) needed for the calculation of the hydraulic power (see 1.3.3.8.1);

d, if necessary, to:

calculate the pressure from measurements by using a water column manometer (see 3.3.4.2);

determine the discharge when the weighing method is used for the measurement itself or for the calibration (see 3.2.2.1 and 3.2.2.2).

5.3.1.2 Density of actual water

ıe water used for model testing in the laboratory contains slight quantities of dissolved bstances, depending on the local hydrological conditions. Therefore its actual density ρ_{wa} is gher than that of distilled water ρ_{wd} (see 2.5.3.1.3). However, the value of ρ_{wa} in the model st equipment generally differs by less than 0,05 % from the value of distilled water ρ_{wd}.

For calculation of the hydraulic efficiency, this deviation is negligible if the determination of the specific hydraulic energy of the machine E is predominantly obtained by pressure measurement (see annex D). Therefore, in most cases, it is sufficient and suitable to apply the values for distilled water.

However, if it is necessary to determine the density ρ_{wa} of the water actually used, several methods can be applied:

- an indirect method using a calibrated pressure gauge connected to a static free water level, as described in 3.3.5.2;
- direct methods such as a precision hydrometer (e.g. a pycnometer, or so called "density bottle") or a buoyancy method.

It may be assumed that the ratio of the actual and distilled water densities is constant whatever the pressure and temperature. So, if the density of the actual water has been measured at certain conditions c of pressure and temperature ($\rho_{wa,c}$), its value for any other condition can be calculated by:

$$\rho_{wa} = \frac{\rho_{wa,c}}{\rho_{wd,c}} \cdot \rho_{wd}$$

where ρ_{wd} and $\rho_{wd,c}$ are calculated according to 2.5.3.1.3.

2.5.3.1.3 Density of distilled water

The formula for the density ρ_{wd} of distilled water as a function of temperature and pressure is derived by Herbst and Roegener [4] from the empirical state equation of the free enthalpy of distilled water. In determining the coefficients below, all the test results of Kell and Whalley [5] and of Kell, McLaurin and Whalley [6] were used.

$$\rho_{wd} = 10^2 \left[\sum_{i=0}^{3} \sum_{j=0}^{3} R_{ij} \cdot \alpha^j \cdot \beta^{(i-1)} \right]^{-1}$$

where

$$\beta = \frac{1}{p^*}\left(p_{abs} + 200 \cdot 10^5\right) \qquad \left(p^* = 10^5 \text{Pa}\right)$$

$$\alpha = \frac{1}{\theta^*}(\theta - \theta_1) \qquad (\theta^* = 1\ °\text{C})$$

From 0 °C to 20 °C: θ_1 = 0 °C From 20 °C to 50 °C: θ_1 = 20 °C

The formula is valid in the used pressure range from p_{abs} = 0 to 150 · 10^5 Pa

Table 5 furnishes the coefficients R_{ij} (m^3·kg^{-1}).

Table 5 – Coefficients of the Herbst and Roegener formula

R(i,j) in the temperature range 0,0 °C to 20,0 °C			
j = 0	j = 1	j = 2	j = 3
$0{,}4466741557 \cdot 10^{-4}$	$-0{,}5594500697 \cdot 10^{-4}$	$0{,}3402591955 \cdot 10^{-5}$	$-0{,}4136345187 \cdot 10^{-7}$
$0{,}1010693802$	$-0{,}1513709263 \cdot 10^{-4}$	$0{,}1063798744 \cdot 10^{-5}$	$-0{,}8146078995 \cdot 10^{-8}$
$-0{,}5398392119 \cdot 10^{-5}$	$0{,}4672756685 \cdot 10^{-7}$	$-0{,}1194765361 \cdot 10^{-8}$	$0{,}1366322053 \cdot 10^{-10}$
$0{,}7780118121 \cdot 10^{-9}$	$-0{,}1619391322 \cdot 10^{-10}$	$0{,}5883547485 \cdot 10^{-12}$	$-0{,}8754014287 \cdot 10^{-14}$

R(i,j) in the temperature range 20,0 °C to 50,0 °C			
j = 0	j = 1	j = 2	j = 3
$-0{,}4410355650 \cdot 10^{-4}$	$0{,}3052252898 \cdot 10^{-4}$	$0{,}9207848427 \cdot 10^{-6}$	$-0{,}2590431198 \cdot 10^{-7}$
$0{,}1011269892$	$0{,}1763956234 \cdot 10^{-4}$	$0{,}5750340044 \cdot 10^{-6}$	$-0{,}1923769978 \cdot 10^{-8}$
$-0{,}4832441163 \cdot 10^{-5}$	$0{,}1533281704 \cdot 10^{-7}$	$-0{,}3749721294 . 10^{-9}$	$0{,}1322804180 \cdot 10^{-11}$
$0{,}6194433327 \cdot 10^{-9}$	$-0{,}3164540431 \cdot 10^{-11}$	$0{,}6311389123 \cdot 10^{-13}$	$0{,}2469249342 \cdot 10^{-15}$

:ead of the formulae of Herbst and Roegener [4], the formulae of Borel and Lan [7] or of ır, Gallagher and Kell [8] can also be used for calculation by computer.

these authors have taken the experimental values [5, 6] as their basis. The values of these ɜrences are within the same range of accuracy (± 0,01 %) in the range of temperature and ssure as mentioned above.

numerical application, the simpler empirical equation of Weber [9], somewhat transformed, y be used. The values calculated for temperatures up to 35 °C and pressures up to ı·10^5 Pa are within the same range of accuracy as mentioned above:

v is the specific volume in $m^3 \cdot kg^{-1}$:

$$v = 1/\rho = v_0\,[(1 - A \cdot p) + 8 \cdot 10^{-6} \cdot (\theta - B + C \cdot p)^2 - 6 \cdot 10^{-8} \cdot (\theta - B + C \cdot p)^3];$$

h $v_0 = 1 \cdot 10^{-3}\ m^3 \cdot kg^{-1}$

$A = 4{,}6699 \cdot 10^{-10}$ ($p = p_{abs}$ in Pa)

$B = 4{,}0$ (θ = temperature in °C)

$C = 2{,}1318913 \cdot 10^{-7}$

lues for distilled water on the basis of the formula of Herbst and Roegener [4] are given in ıle B.2 and are represented in figure 24.

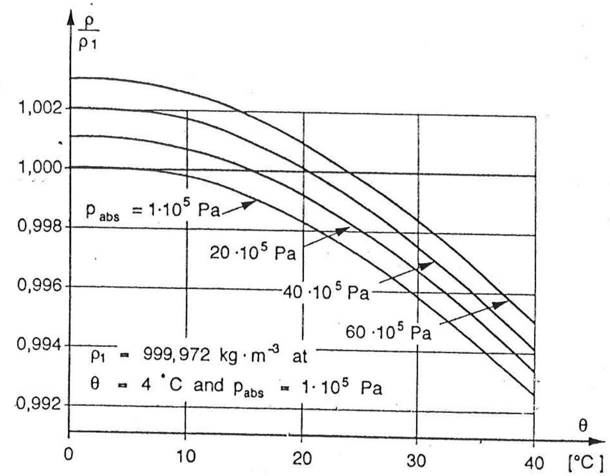


Figure 24 – Density of distilled water ρ_{wd} ($kg \cdot m^{-3}$)

2.5.3.2 Condition of water during test

2.5.3.2.1 Definition of nuclei and gas content in water

As stated in 2.1.2.3 and 2.3.1.6.2, to determine the influence of cavitation on the performance of the machine it is useful to know, in addition to pressure and temperature, other conditions of the water passing through the machine. These other conditions are:

- **nuclei content** (see 2.1.2.3) in terms of number of nuclei per unit volume of water. The nuclei content corresponds approximately to the undissolved gas (air) content in terms of volume per unit volume of water;
- **dissolved gas content** in terms of volume per unit volume of water.

Possible extreme conditions for water are:

- completely degassed water (water with no nuclei and no gas content);
- completely saturated water (water saturated with dissolved gas, however with low content of nuclei);
- high nuclei content of water due to an artificial injection, independently of the content of dissolved gas.

It is currently not possible to specify in advance a value for the required nuclei content, which depends on model scale, test specific hydraulic energy and other factors.

For a discussion of the influence of water conditions (nuclei content, gas content) in model testing, see 2.3.1.6.2 and references [1] and [3].

.3.2.2 Determination of nuclei and gas (air) content in water

.3.2.2.1 Nuclei content in water

e determination of the nuclei content of water requires the measurement of the number and he critical pressure of nuclei.

special cavitation nuclei counter is described in [1] and [3]. In this counter, the flow is celerated through a restricted section to promote the explosive growth of nuclei.

e injection of nuclei can be performed by the injectors described in [1] and [3]. These ectors are based on the rapid expansion of saturated water through an orifice.

.3.2.2.2 Gas content (air content) in water

ater normally contains gases in dissolved or undissolved form. These gases may be air or ier substances such as carbon dioxide (CO_2).

Dissolved gas content

The maximum possible amount of dissolved gases depends on the pressure and temperature of the water and the actual gas content shall be measured. Two basic methods of measurement can be applied[1]:

- measurement of dissolved oxygen content using an electrical analyzer based on the oxygen diffusion through a PTFE membrane (e.g. Beckman apparatus [10]);
- physical separation: Van Slyke method [11]. This method allows the extraction of the totality of the air content, whether in dissolved or occluded form, by cascading the sampling under vacuum in a packed column. The method is relatively rapid, but necessitates working on small volume samplings.

Undissolved gas content

A gas completely dissolved on the high pressure side of a hydraulic machine may become free while moving from the high pressure side to the low pressure side of the circuit, thus changing the behaviour of the machine. Therefore, the gas content not only for dissolved but also for undissolved gases should be determined. Such a method using an extraction vessel is described in [13].

[12] a comparison of the Van Slyke, Merl and Brand [13] apparatus and a dissolved oxygen eter is made, including measurement of dissolved and undissolved gases.

5.3.3 Kinematic viscosity

ne kinematic viscosity ν (see 1.3.3.3.6) of water depends on its temperature θ and absolute essure p_{abs} and is derived from the basic physical property dynamic viscosity μ using $\nu = \mu/\rho$.

formula for μ is given in [14].

owever, for practical reasons in hydraulic machinery, an approximate value of ν can be omputed using the following formula:

$$\nu = e^{[-16{,}921 + 396{,}13/(107{,}41+\theta)]}$$

[1] The Winkler method, using iodometry, may also be used. It is accurate, but its application is difficult.

Using this equation, the average deviation from the standard values given in [15] is ±0,05 %, the maximum deviation is ±0,09 %.

The influence of pressure is negligible. The deviation for $p = 10 \cdot 10^5$ Pa against a reference pressure of $p = 10^5$ Pa is about –0,05 %.

Values for ν are given in table B.3.

2.5.3.4 Vapour pressure

The vapour pressure p_{va} (see 1.3.3.3.4) of water between the water temperatures of $\theta = 0$ °C and $\theta = 40$ °C can be calculated using the following empirical formula:

$$p_{va} = 10^{(2{,}7862 + 0{,}0312\,\theta - 0{,}000104\,\theta^2)}$$

The resulting error is less than ±7 Pa.

Numerical values of p_{va} are given in table B.4, see [8].

Attention shall be paid to dissolved chemical substances in water which may influence the vapour pressure.

2.5.4 Physical conditions of atmosphere

2.5.4.1 Density of dry air

The density of air ρ_a (see 1.3.3.3.3) as a function of p_{abs} and temperature of air θ can be calculated using the following formula according to ISO 2533:

$$\rho_a = (p_{abs} \cdot 3{,}4837 \cdot 10^{-3})/(273{,}15 + \theta)$$

Values of ρ_a are given in table B.5.

The influence of humidity on the density of air is negligible for the determination of E.

2.5.4.2 Ambient pressure

Normally, the ambient pressure p_{amb} (see 1.3.3.5.2) is the barometric pressure measured in the laboratory by a barometer. For conversion of test results to the conditions of prototype at site (e.g. determination of NPSE or σ of the site) the ambient pressure shall be calculated using the standard atmosphere defined in ISO 2533.

Assuming a linear change of temperature with elevation equal to $-6{,}5 \cdot 10^{-3}$ K/m, the ambient pressure may be calculated by the following approximate formula derived from ISO 2533:

$$p_{amb} = 101\,325\,(1 - 2{,}2558 \cdot 10^{-5} \cdot z)^{5{,}255}$$

where z is the elevation, in metres.

The resulting error is less than ±15 Pa.

Values for p_{amb} from ISO 2533 are given in table B.6.

.5 Density of mercury

ɩ density of mercury ρ_{Hg} (see 1.3.3.3.3) used in liquid manometers is calculated for pure rcury by the following equation with p_o = 101 325 Pa (standard ambient pressure at sea əl):

$$\rho_{Hg} = (13{,}595 - 2{,}46\ \theta)\left[1 + 3{,}85 \cdot 10^{-11}\ (p - p_o)\right]$$

ues of ρ_{Hg} are given in table B.7, see [16].

rcury used in practical applications may be contaminated by dissolved metal or by other terials. To ensure valid measurements the mercury shall be pure and clean.

3 Main hydraulic performances: methods of measurement and results

3.1 Data acquisition and data processing

3.1.1 Introduction and definitions

Data acquisition and data processing involve the conversion of measured signals into appropriate engineering units through a measuring chain of several components such as transducers, multiplexers, signal converters or conditioners, data storage equipment and computers. The final output is a presentation of parameters as meaningful performance data.

The measurands are fluctuating quantities. However, their average values are of main interest for the determination of the main hydraulic performances of a model.

Definitions:

measurand
quantity subjected to measurement

transducer
measuring device, which provides an output quantity with a given relationship to the input quantity

transducer with digitized output
measuring device with built-in electronics to give a digitized output (e.g. serial port RS 232)

multiplexer (MUX)
device for switching two or more signals in order to share the same analog-to-digital converter, frequency counter or cabling facilities

analog-to-digital converter (A/D converter)
device that translates continuous analog signals into discrete digital signals

counter
device that measures frequency, time period or number of pulses

voltage-to-frequency converter (V/F converter)
device that translates with a given relationship, voltage levels into frequency

aliasing
when sampling analog signals with a sampling rate less than twice that of the frequency of the highest-frequency signal or noise component ("Nyquist rate"), the sampling process will produce spurious low-frequency signals (aliases) that cannot be distinguished from the original signal

computer interface
communication port that enables the computer to control and communicate with other compatible devices

3.1.2 General requirements

The output from the data acquisition and data processing system shall be a true reflection of the measurand.

Documented calibration procedures shall exist for all instruments in use. Maintained records of all measurement standards and measuring equipment used to establish compliance to specified requirements shall also exist.

Vhere applicable, it should be made possible via parallel connections to witness *in situ* the :alibration of all instruments in the measuring chain by a primary method to verify that the com- ılete data acquisition system represents the measurand within the specified limits. This usually neans that the same signal path, hardware and software configuration shall be used during ıoth calibration and performance testing.

)uring a performance test, the averages of each of the measured quantities shall be obtained ıy measurements performed during the same time interval.

\ possible check of the complete measuring chain would be to have parallel instrumentation. It ; also preferable to have the capability of comparing the results from the data acquisition ;ystem and the reference instrument(s) even under running conditions of the machinery being ested.

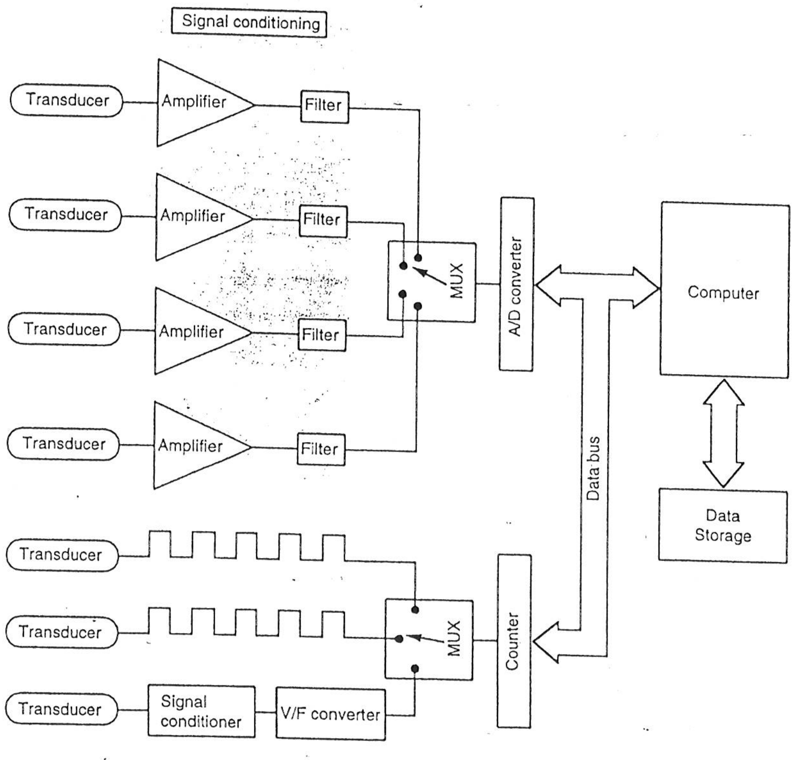


Figure 25 – Time multiplexing data acquisition system

3.1.3 Data acquisition

The data acquisition system may be arranged in different ways (including manual methods) depending on the hardware available and the chosen averaging method.

Possible arrangements and examples of different data acquisition systems are given. Usually a combination of different systems is used.

3.1.3.1 Time multiplexing system

In a time multiplexing system (figure 25), the measurands are measured via multiplexers that scan the channels sequentially a number of times during a given time period.

The calculated average value of the measurand is taken for further processing.

3.1.3.2 Parallel measuring system

In a parallel operated system (figure 26), the measurands are measured by a computer collecting the data directly from each channel. This arrangement makes possible high speed data logging and simultaneous sampling of all the channels (see 3.1.4.4).

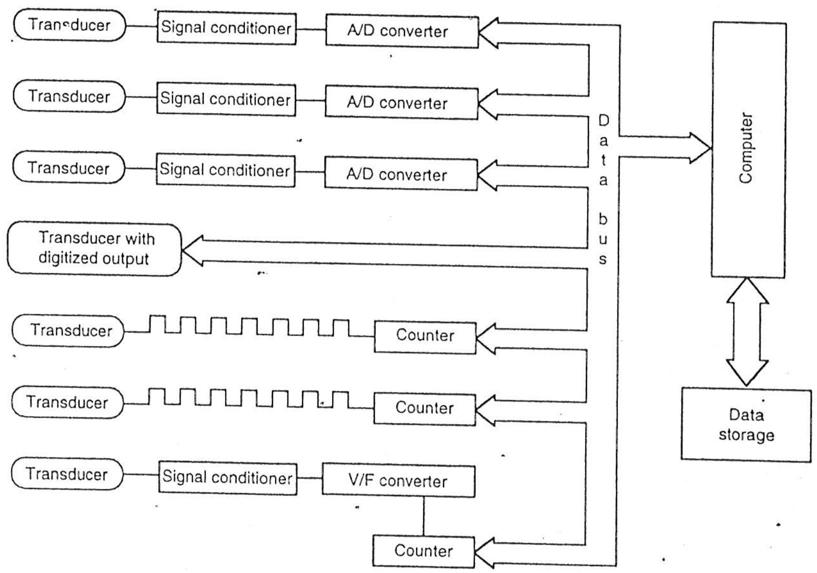


Figure 26 – Bus operated data acquisition system

1.4 Component requirements

ıe components in the measuring chain shall be able to manage the frequency range of terest.

ɔmponents transferring information about the measurand to the transducer, for example essure piping, can cause spurious effects and cause errors in the measurements.

ɔr all the components in the measuring chain, note that temperature variations, in their ıvironment, can cause errors in the measurements outside the specified limits.

·operties such as linearity and hysteresis shall be documented during calibration.

1.4.1 Transducer

·ansducers used for the measurement of performance parameters should operate in a stable mperature environment. They should be located where they are not influenced by mperature variations, for example from direct sunlight, heating panels, ventilation channels, c.

ıe dynamic behaviour of the measurands shall be known, as the transducers shall only be ;ed in the frequency domain they are designed for.

ɜre should be taken when using transducers with special inherent damping property or ljustable response time, and transducers with extremely high deflection of the sensing ɜment. Such transducers can cause erroneous measurements in both averaging and ;cillating measurements.

1.4.2 Cables and termination

ıe signal path between transducers and amplifier shall be designed in such a way that :ternal influences on the signals (e.g. from power lines or temperature variations) are inimized. Proper shielding and grounding shall be observed. Connectors and terminations ıall have stable, reliable mechanical and electrical properties.

/en if all the above precautions have been taken, additionally be aware of spurious influences ɔm the power network on the measurement results.

1.4.3 Signal conditioning

ıe output from transducers with analog output are often amplified and filtered in a signal ɔnditioning unit.

1.4.3.1 Amplifier

ɔ exploit the resolution of the A/D converter, the output range of the amplifier shall match the ınge of the converter.

he amplifier shall be located as close as possible to the transducer in order to minimize the fluence of noise pick-up in the cabling.

3.1.4.3.2 Filter

When choosing filters, special attention should be paid to properties such as:

– a.c. signals: cut-off frequency, attenuation (order) and time delay;
– d.c. signals: offset, temperature drift and linearity.

In analyses where simultaneous measurement of two or more measurands is important, be aware of delays in the conditioning and data acquisition systems. Filters cause delays (phase shift) that are a function of the filter type and the cut-off frequency (figure 27).

The cut-off frequency of a low-pass filter shall be a maximum of half the sampling frequency in order to avoid aliasing effects. This is illustrated in figure 28. In practice, however, a cut-off frequency of one-third or less of the sampling rate is used.

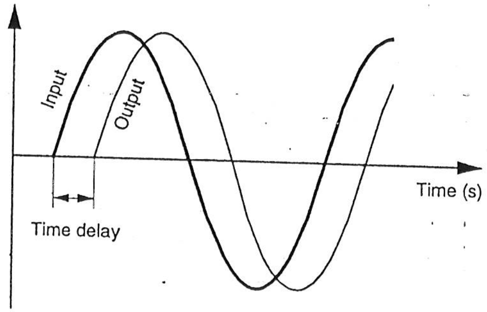
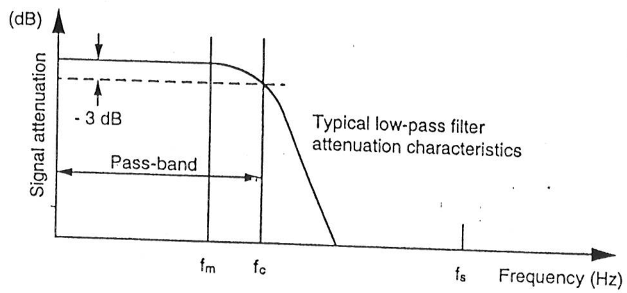


Figure 27 – Time delay



f_m is the maximum frequency component of interest;
f_c is the cut-off frequency of low-pass filter;
f_s is the sampling rate.
To obtain desired frequency content: $f_c > f_m$.
To avoid aliasing in the pass-band: $f_s \geq 2 f_c$.

Figure 28 – Filtering and sampling frequencies

1.4.4 **Multiplexer**

ıe effective switching rate for the multiplexer shall be compared to the requirements for each easurand. Because the A/D converter is sampling several channels sequentially, the .mpling rate for each channel is reduced in proportion to the number of channels.

ıe switching system is mostly either of a relay or a solid state switching type. Relay switching usually more accurate than the solid state switching, but has a lower switching rate. When /itching between different voltage levels, be aware of interference effects between ·ighbouring channels. Generally, these errors increase with the switching rate.

1.4.5 **Analog-to-digital converter**

ıfore the continuous analog signal can be read by the computer, the signals must be nverted into digital numbers.

ıportant parameters for analog-to-digital converters are the conversion time, resolution, curacy, input range, temperature drift and linearity.

ıe resolution of an A/D converter is defined as the number of bits the converter uses to de- ribe the analog signal. A 3-bit converter divides the range into $2^3 - 1 = 7$ divisions.

ır performance tests, a minimum requirement would be a 14-bit resolution. For dynamic əasurements, a lower resolution can be accepted.

ı obtain simultaneous measurements during A/D conversion, one A/D converter per channel a simultaneous sample-and-hold equipment can be used.

1.4.6 **Computer**

ıe controller in the data acquisition system is the computer. It shall have the following ıction to configurate and synchronize the data logger, handle data transfer, communicate th peripheral equipment, perform calculations and present results.

ıe computer interface should have a selectable data transfer rate (baud rate in bits/s) ıabling it to communicate with and control different devices on the bus.

1.4.7 **Data processing**

ıpical software tasks are:

- control of the data acquisition system;
- calculation of calibration coefficients;
- conversion of electrical values into engineering units;
- calculation of average values and other statistics;
- calculation of data;
- evaluation of random uncertainty;
- presentation of results;
- data storage.

ıe raw data for each parameter in an acceptance performance test shall be available after the /aluation of a test point in order to perform a manual calculation and verify the computer ıde.

If possible, essential performance data should be continuously displayed during the test to give an overview of model performance together with the hydraulic system to which it is connected.

The number of samples and the sampling rate shall reflect the characteristics of the complete measuring chain to give:

- an accurate mean value for performance measurements;
- a satisfactory determination of the necessary characteristics of oscillating measurements.

3.1.5 **Check of the data acquisition system**

Each measuring chain shall have a complete schematic diagram showing its main components. This will help the parties to decide where checks should be made when particular problems occur, or when oscillating signals need to be more closely investigated. Figure 29 shows some typical measurement chains with suggested test and checkpoints.

3.1.5.1 **Transducer with analog output**

In figure 29, point 1 is a test point for determining the dynamic behaviour of the measurand.

Proper operation of the signal conditioning system can be confirmed by correlating signals at input point A to test point 3.

Proper amplifier operation can be confirmed by correlating signals at input point A to test point 2.

Proper filter operation can be confirmed by correlating signals at input point B to test point 3

To check the proper operation of the multiplexer and A/D converter, a reference signal can be applied to point C, and compared with the output at point 5.

3.1.5.2 **Transducer with frequency or scaled pulse output**

The signal quality shall be controlled at point 4 to ensure proper triggering of the counter. A reference signal can be applied at D to check the timebase of the counter.

3.1.5.3 **Transducer with digitized output**

The transducer and a measurement chain of this kind can best be checked during calibration.

3.1.5.4 **Check for bias effects**

To check that the signal conditioning system does not give any bias effects, the output signal from the system may be correlated with the input signal: The latter could be a reference signal from a separate electrical source. The checkpoints are indicated in figure 29 at points A and 3.

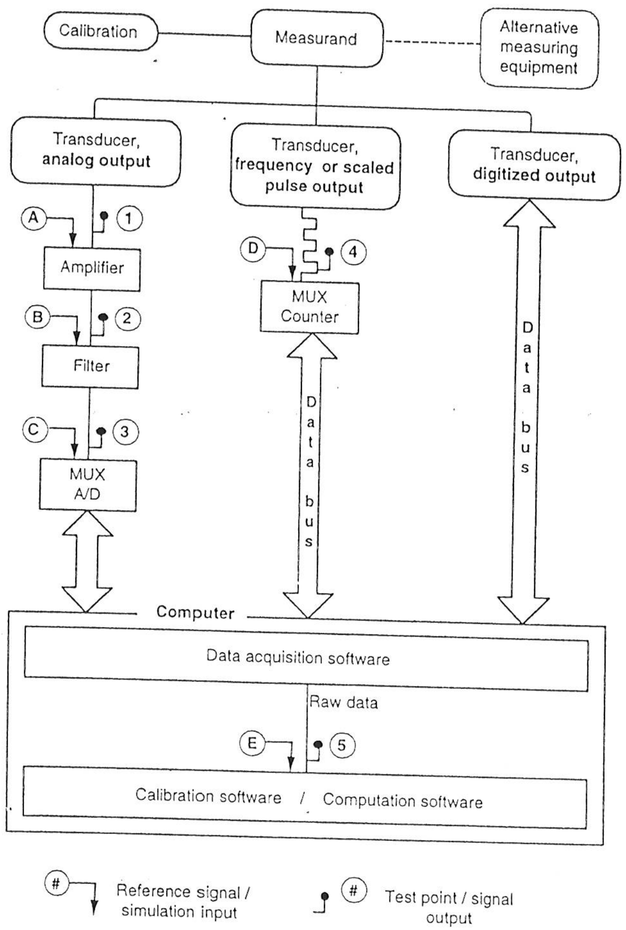


Figure 29 – Different measurement chains and their recommended checkpoints

3.1.5.5 Software

The software code can be verified by an alternative computation using the raw data read at control point 5, and comparing it with the result from the computer.

The performance algorithm can be verified by entering numerical values at E giving a known performance result.

It should be documented that the algorithms used for the calibration of a measuring chain are equivalent to those used in the performance computation.

3.2 Discharge measurement

3.2.1 General

As far as possible, there should be no loss or gain of water between the machine and the discharge measuring device. Nevertheless, if such auxiliary discharges exist, they shall be measured independently.

3.2.1.1 Choice of the method of measurement

The methods which may be used for measuring the discharge during a model acceptance test can be classified into primary and secondary methods.

3.2.1.1.1 Primary methods

Primary methods are those which need only measurements of fundamental quantities: length, mass and time. In the field of application of this standard, the primary methods which may be used are:

- the weighing method;
- the volumetric method;
- the moving screen method.

These methods are the most accurate. For this reason, and notwithstanding some inherent disadvantages (cumbersome equipment, duration of measurement, etc.), any model test facility shall necessarily include the possibility of using one of these methods; nevertheless, for convenience of use, it is generally supplemented by a secondary method.

3.2.1.1.2 Secondary methods

Many other methods, founded on various principles and thus considered as secondary methods, may be used for discharge measurements in model tests. Although some of them are standardized, the high accuracy required for the purpose of this standard makes it mandatory to calibrate the measuring equipment *in situ* under the prevailing test conditions by one of the above primary methods. Consequently, and since repeatability is the most important quality required from the secondary method measuring device, it is not necessary for it to comply with all the requirements of the relevant standards.

The arrangements necessary to carry out such a calibration periodically, without dismantling or modifying the measuring line, shall be provided at the design stage of the test facility.

e main secondary methods of discharge measurement are:

velocity-area method by means of current-meters or Pitot tubes and tracer methods are very seldom used for model tests, and will not be mentioned in the following subclauses;

thin-plate weirs and differential pressure devices (orifice plates, nozzles and Venturi tubes), even when designed, installed and used in accordance with the relevant ISO standards, do not achieve the accuracy required for model tests, when using standardized discharge coefficients, therefore periodic calibration *in situ* is mandatory;

various types of flow meters, such as turbine, electromagnetic, acoustic or vortex flow meters; they are particularly convenient to use for they allow quick measurements, their output signal can easily be introduced into a data acquisition system and most of them generate little disturbance in the flow pattern. For the time being, the effect of installation conditions on their response is not established with sufficient accuracy for model tests. Thus, their calibration *in situ* is mandatory and their repeatability over the whole range of flow conditions to be encountered shall be checked regularly.

2.1.2 Accuracy of measurement

2.1.2.1 Reference to ISO standards

method of measurement is described in detail in the following subclauses only when no andardized procedure exists elsewhere. Whenever possible, reference has been made to isting standards, especially to those published by ISO, which are particularly suited to the ecise requirements of this standard.

2.1.2.2 Evaluation of uncertainty

e numerical values of systematic uncertainty indicated in the following clauses are to be ed only as guidance. They are valid only:

with the best conditions for measurement;

if all requirements specified in this standard and in the relevant standards are satisfied; and

if the testing and analysis are carried out by qualified and experienced personnel.

these conditions are not satisfied, there may be an unpredictable increase in both the stematic and random errors of the discharge measurement.

each particular case, the actual values of systematic and random uncertainties shall be aluated by the user, taking into account the whole measuring system and the operating nditions of the test facility.

e method for combining the random and systematic uncertainties associated with the dividual sources of error is explained in 3.9.2.2.4. The final result is then expressed as the certainty at a confidence level of approximately 95 %.

2.1.2.3 Steadiness of the flow

hatever method is used, a discharge measurement for a model acceptance test is valid only the flow is steady or nearly steady during each point.

most situations, the primary methods require a rather long time of measurement and oduce only a mean value of the discharge during this time. Thus it is possible to ascertain a ariation of the flow occurring between two runs, but not the possible fluctuations (see 2.3.2.3).

Most secondary methods quoted in 3.2.1.1.2. produce quasi-instantaneous readings which have to be averaged to obtain the mean value of the discharge during the point and which can be treated graphically and statistically to assess the nature and the extent of the fluctuations of the flow (see 3.1 and 4.2). This is another reason why a test facility should have access to a primary and a secondary method.

3.2.2 Primary methods

3.2.2.1 Weighing method

3.2.2.1.1 Principle of the method

ISO 4185 gives all necessary requirements concerning the measuring apparatus, the procedure, the method for calculating the discharge and the uncertainties associated with the measurement. Although ISO 4185 specifies two alternative methods, the "static" and the "dynamic" method, only the static weighing methods, which consists of diverting the flow to the weighing tank for a measured time and then weighing the diverted quantity, is recommended for the purpose of this standard.

The weighing method, which gives only the average discharge value during the time taken to collect a suitable quantity of water, may be considered the most accurate method of discharge measurement.

As stated in ISO 4185, the calibration of the weighing device shall be periodically checked at least every two years for a mechanical weigh-beam and every year for a load cell. These intervals may be extended if the calibration history shows stable results.

3.2.2.1.2 Uncertainty of measurement

The weighing method is affected by errors relating to weighing, measuring of filling time, determination of density, taking into account the temperature of the fluid and motion of the diverter. Furthermore, a buoyancy correction must be made to the readings of the weighing machine to account for the difference between the upthrust exerted by the atmosphere on the liquid being weighed and on the reference mass used during the calibration of the weighing device.

If the installation is carefully constructed, maintained and used, a systematic uncertainty on the discharge measurement (at a confidence level of 95 %) within ± 0,1 % to ± 0,2 % should be achieved.

3.2.2.2 Volumetric method

3.2.2.2.1 Principle of the method

ISO 8316 gives all necessary requirements concerning the measuring apparatus, the procedure, the method for calculating the discharge and the uncertainties associated with the measurements.

Although ISO 8316 specifies two alternative methods, the "static" and the "dynamic" methods, only the static gauging method is recommended in this standard.

The volumetric method is approximately of the same accuracy as the weighing method and similarly supplies only the average discharge value during the time taken to collect a suitable quantity of water.

As stated in ISO 8316, the calibration of the volumetric tank shall be periodically checked; at least every five years for concrete tanks and every three years for metallic tanks. These intervals may be extended if the calibration history shows stable results.

.2.2.2.2 Uncertainty of measurement

'he volumetric method is affected by errors relating to the calibration of the reservoir, the ıeasurement of levels, the measurement of filling time and the motion of the diverter. The ıater tightness of the reservoir shall be checked and a leakage correction applied if necessary.

the installation is carefully constructed, maintained and operated, a systematic uncertainty (at confidence level of 95 %) within ±0,1 % to ±0,2 % can be achieved.

.2.2.3 Moving screen method

.2.2.3.1 Principle of the method

he principle of the method is, to a certain extent, similar to the volumetric method, for it is ased on the determination of the volume of water displaced in a channel between the cross-ections A and B, by means of a screen moving with the water (see figure 30). The discharge ; calculated by the following formula:

$$V = b \cdot d \cdot L$$

$$Q = \frac{V}{t} = b \cdot d \cdot \frac{L}{t} = b \cdot d \cdot v$$

'here

is the displaced volume of water between cross-sections A and B;

is the distance between the cross-sections A and B (length of measuring section);

is the mean width of the channel within the measuring section;

is the mean depth of water in the channel within the measuring section;

is the travel time of the screen between cross-sections A and B;

is the mean discharge during travel time t;

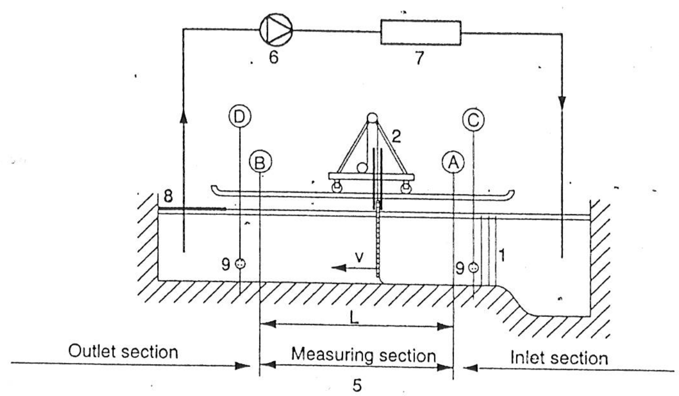
is the mean flow velocity within the measuring section of the channel.

.2.2.3.2 Measuring equipment

.2.2.3.2.1 Channel

he measuring channel shall be a straight horizontal section, generally rectangular, precisely alibrated over the entire path of the screen travel. The width and depth of the channel shall be uch that, for the range of discharge to be measured, the velocity lies within 0,05 m/s to 1 m/s.

he water supply of the channel shall ensure a regular velocity distribution, free of swirl, symmetry and excessive turbulence. This may be achieved by means of straightening devices ɔerforated plates, honeycomb, etc.).



1 Straightening devices
2 Moving screen and carriage
3 Measurement of water level
4 Measuring wells
5 Measurement of travel time
6 Booster pump of test circuit
7 Flowmeter of the test rig to be calibrated
8 Plate cover to reduce the downstream free water surface
9 Perforated plate, flush with the wall

Cross-sections Ⓒ and Ⓓ

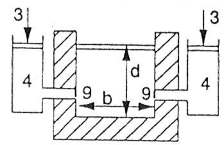


Figure 30 – Moving screen method

The total length of the channel includes:

- an inlet section, where the screen is introduced in the water and reaches uniform motion;
- a measuring section, the length of which is very accurately determined;
- an outlet section, where the screen is extracted from the flow.

The cross-sectional area of the channel corresponding to a given water level can be determined by geometric measurements. All the geometric dimensions involved shall be periodically checked, recommended every five years. Dimensional changes due to thermal expansion and strain due to the weight of the water may need to be taken into account in extreme conditions.

3.2.2.3.2.2 Screen

The screen is generally suspended from a carriage rolling on rails installed along the length of the channel or supported by fluidic guide-blocks. The use of a floating screen should preferably be avoided.

The screen is often constructed of a light rigid material mounted on a lightly built frame. The carriage and screen assembly shall be as light as possible and friction shall be reduced to a minimum or compensated for by means of a driving motor such that the screen velocity will rapidly equal the mean water velocity and will move smoothly even at the lowest velocity.

s essential that the introduction of the screen into the channel shall disturb the flow as little possible and not initiate ripples or waves resulting in serious errors. One possible way to sure this is to accelerate the screen carriage by means of an electric drive unit in order to e it approximately the same speed as the water velocity before the screen is lowered into water.

e clearance between the screen and the walls and bottom shall be as small as possible in der to minimize the leakage. It is good practice to fit flexible lip seals to the screen provided it their friction is negligible or is compensated by a driving motor.

!.2.3.2.3 Measurement of travel time

reen travel time shall be measured between two fixed points installed at the beginning and end of the measuring section. When passing these points, the screen actuates an ectronic timer through electro-mechanical, optical or magnetic switches.

!.2.3.2.4 Measurement of water level

e level shall be measured before, during and after the travel time of the screen by means of asuring wells located in the walls on each side upstream and downstream of the measuring ction. They shall be fitted with point or hook gauges or high accuracy pressure transducers r measuring apparatus, see 3.4).

s essential that the level remains constant (e.g. within 0,5 mm) between the front and rear of screen. This confirms that the moving screen velocity is equal to the water velocity.

order to obtain high accuracy it is important to reduce slow mass oscillations (e.g. from justment of discharge) in the calibration channel. In order to observe such oscillations the e water surface downstream of the measuring section shall be reduced as far as possible. e mass oscillations may then be observed as variation of the water level, which may thus be ed to indicate stable conditions.

?.2.3.2.5 Controls before and during the run

fore starting a run, it is essential to check that the indication of the water level is constant in der to ensure that no oscillations occur within the channel.

iform velocity of the screen is essential to avoid ripples and waves which might substantially crease the uncertainty of measurement. This can be checked by means of a few pplementary switches evenly distributed along the measuring reach in order to determine ermediate travel times.

is also important that any leakages from one side of the screen to the other are as small as ssible. This can be checked by injecting a dye liquid near the wall and bottom seal.

evertheless, slight disturbances in the front and the rear of the screen or very small leakages, pecially in the vicinity of the free surface, are often observed and in no way indicate faulty eration of the apparatus.

e time interval between two consecutive runs shall be of adequate duration to dampen the rturbation in the channel caused by the previous run.

3.2.2.3.3 Uncertainty of measurement

If the installation is carefully constructed, maintained and used, and if the above requirements are satisfied, a systematic uncertainty (at the 95 % confidence level) on the discharge measurement within ±0,2 % to ±0,3 % can be achieved.

3.2.3 Secondary methods

3.2.3.1 General requirements

Various types of flowmeter may be used by agreement under the following conditions:

– the device chosen shall be of the best quality available, particularly with respect to its repeatability and its sensitivity to influence quantities (ambient temperature, frequency and voltage of the power supply, etc.);
– the flowmeter and the associated measuring system shall be calibrated by a primary method in the actual operating conditions (see 2.3.3.2.3 and 3.2.3.8);
– the repeatability of the measurement shall be checked over the whole range of discharge to be measured.

Although their application is not mandatory, the relevant standards and manufacturer's instructions give useful advice concerning the best installation and measurement conditions.

The types of flowmeter most often used are described in 3.2.3.2 to 3.2.3.7.

3.2.3.2 Weirs

Only rectangular or triangular sharp-edged thin-plate weirs may be used within the scope of this standard.

For the design of the weir, its installation and the conditions for measuring the head over the weir, ISO 1438-1 should be referred to; however, the standardized discharge coefficients will not achieve the required accuracy (see 3.2.1.1.2). In addition, weirs are very sensitive to any change in the distribution of approach velocity and in the condition of the plate (roughness of the upstream face, cleanliness and sharpness of the edge, etc.).

The weir is commonly located on the low pressure side of the machine, and care shall be taken to ensure that smooth flow (free from eddies, surface disturbances or significant amounts of entrained air) exists in the approach channel.

When the weir is located on the outlet side of the machine being tested, it shall be far enough from the machine or the discharge conduit outlet to enable the water to release its air bubbles before reaching the weir. Stilling screens and baffles shall be used when necessary to give a uniform velocity distribution over the whole cross-section. Disturbed surface or undercurrents, or asymmetry of any kind, shall be corrected by suitable screens.

3.2.3.3 Differential pressure devices

Orifice plates, nozzles or Venturi tubes may be used for discharge measurement in model test facilities, particularly operating on closed circuit without a free water surface.

For the design of the differential pressure device, including its pressure tappings, its installation and operating conditions, refer to ISO 5167-1; however the standardized discharge coefficients will not achieve the accuracy required in this standard (see 3.2.1.1.2). Other types of differential pressure devices than those described in ISO 5167-1 may also be used.

ifferential pressure devices offer high reliability, but are very sensitive to the flow pattern and ·eate a high pressure loss, particularly orifice plates and nozzles.

ıe differential pressure generated by the device shall be measured in accordance with 3.3.4.

ıe connecting pipes between the primary device and the pressure gauge shall conform with ;O 2186.

are should be taken to avoid cavitation.

2.3.4 Turbine flowmeters

ırbine flowmeters, which generally include a flow straightener, require a minimum upstream ıd downstream straight length and generate only very slight disturbance of the flow, but eate a somewhat high pressure loss. The output signal, which is a frequency measurement, simple to measure without loss of accuracy. Care shall be taken to maintain the bearings in ›od condition and to maintain the blades of the turbine clean. The calibration shall be ıecked at least after each maintenance.

nce cavitation on the runner blades can occur in low pressure conditions, calibration shall be ıecked at the lowest test pressure (see 3.2.3.8).

2.3.5 Electromagnetic flowmeters

ectromagnetic flowmeters are the subject of ISO 6817 and ISO 9104.

ıe main advantages of electromagnetic flowmeters are that they generate neither disturbance the flow, nor pressure loss, and are not very sensitive to wear. They produce an stantaneous reading of discharge and thus are particularly convenient for observing scharge fluctuations. Care shall be taken to detect any drift in the electronic circuitry output ıd in the surface condition of the electrodes. The calibration shall be checked at least after ıch maintenance.

2.3.6 Acoustic flowmeters

›veral methods of acoustic discharge measurement exist. Currently available knowledge is ch that the method preferred for the purpose of this standard is that based on the easurement of the transit time of acoustic pulses travelling upstream or downstream, eferably including several parallel paths.

ıe data acquisition and processing system shall demonstrate that the equipment is operating ›rrectly (separate measurement of the average velocity along each individual path, verification the speed of sound, checking of the proportion of lost pulses, etc.).

ɔre details on this method are given in IEC 60041.

:her kinds of acoustic flowmeters, based, for instance, on the measurement of the refraction an acoustic beam by fluid velocity, or on the cross-correlation of acoustic signals emitted in ›o cross-sections, may also be used.

ɔoustic flowmeters have the advantage of not introducing any flow disturbance or pressure ss, but are somewhat sensitive to the velocity distribution and to the presence of gas bubbles ıd acoustic noise. Their sensitivity to turbulence and limited sampling of local instantaneous ›locities do not allow the use of successive readings to assess the level of discharge ıctuations.

3.2.3.7 Vortex flowmeters

The principle of vortex flowmeters is based on measuring the frequency of vortex-shedding generated by a bluff body inserted in the flow, the frequency of which is proportional to the mean velocity in a given range of Reynolds numbers.

Although many devices of this type are available, experience with this method of discharge measurement is still limited and the method shall only be used with caution. For instance, any vibration of the pipe is liable to alter the measured frequency and shall thus be avoided.

Due to the risk of cavitation on the bluff body of the vortex flowmeter, the calibration shall be checked at the lowest test pressure (see 3.2.3.8).

3.2.3.8 Calibration procedure

As stated above, any device used for discharge measurement by a secondary method shall be calibrated against one of the primary methods described in 3.2.2. The calibration shall be made without dismantling the flowmeter from the test circuit or modifying the flow conditions at the inlet of the flowmeter.

Calibration shall include the whole of the flowmeter and the associated measuring system: for instance an orifice plate, the connecting pipes, the pressure transducer, its power supply and the data acquisition system.

Calibration normally should be carried out in the actual operating conditions (pressure, temperature, water quality, etc.) prevailing during the tests. Should the pressure encountered during the tests be lower than the minimum pressure attainable in the open circuit used for calibration, it shall then be demonstrated that the flowmeter calibration is not affected by cavitation at that reduced pressure. This may be achieved by using two secondary flowmeters in series, one of which is not sensitive to cavitation effects. It is not permitted to make a discharge measurement with a flowmeter affected by cavitation, even if calibrated in the same operating conditions, as the phenomena involved are not sufficiently reproducible.

Any calibration shall include sufficient measuring points evenly distributed over the whole range of discharge to be measured during tests, to allow an accurate evaluation of their scattering.

In most cases, the result of a calibration can be written, at least in the range of use, as $Q = CR^{\alpha}$, where

R is the output signal delivered by the secondary flowmeter;

α is an exponent which is known by theoretical considerations ($\alpha = 1$ when R is the frequency of a turbine meter, $\alpha = 1/2$ when R is the differential pressure of a Venturi tube, $\alpha = 3/2$ when R is the head over a rectangular weir, etc.);

C is a discharge coefficient which may be constant or variable over the range of the meter.

It is thus possible to plot the discharge coefficient versus the discharge indicated by the primary method or preferably versus the corresponding non-dimensional coefficient appropriate to the type of flowmeter (Reynolds number for flowmeters in closed conduit, Froude number for weirs).

In any case, the best fitted curve (often a straight line) through the measured points should be determined by a regression technique such as the least squares method. Guidance for deriving this calibration curve and evaluating the associated uncertainty may be found in ISO 7066 (see also annex H).

ıe secondary flowmeter should normally be calibrated before and after the test (see 3.3.1.5). If a significant difference appears between both calibrations[1]), the flowmeter and the ssociated measuring system shall be carefully inspected to find the reason for this deviation ıd the test may possibly be rejected. A historical record of the flowmeter calibration should be /ailable for inspection and analysis. If no systematic trends occur, an average of all obtained ılues may be a better approximation of the true value than the mean of the two values ›tained before and after the test.

3 Pressure measurement

3.1 General

ıis clause deals only with the measurement of the time-averaged value of the pressure, ıereas the measurement of pressure fluctuatinos is described in 4.3. Pressure measure-ents in hydraulic machinery are made in order to determine:

quantities of the hydraulic performance such as

- specific hydraulic energy E (see 3.5.2 and 3.5.3) and
- net positive suction specific energy NPSE (see 3.5.4);

gauge pressures or differential pressures at special locations of the water passage in the model for various purposes: for example to measure the discharge utilizing differential pressure devices (see 3.2.3.3) or to obtain information on

- local pressure;
- pressure distribution;
- index testing (values to be converted to site conditions).

ıe pressure p is measured as single gauge pressure or differential pressure at steady-state ›nditions.

3.2 Choice of pressure-measuring section

›ecial attention shall be paid to the location of the measuring sections; normally they are entical to the reference sections. There should be minimum disturbance of the flow. The high 'essure and low pressure reference sections 1 and 2, specified by the contract, should ›rmally fulfil these conditions. However, under exceptional circumstances where the velocity stribution of the reference section is considerably distorted this section should be replaced, if ›ssible, by other measuring sections as close as possible to the reference section and fering better flow conditions.

ıe plane of the measuring section should preferably be normal to the average direction of ›w. Its cross-sectional area, which is required for computing the mean water velocity, shall be ·adily measurable.

he measuring section should preferably be arranged in a straight conduit section but could be ightly convergent or divergent.

It may be agreed that the relative deviation $2(Q_1 - Q_2) / (Q_1 + Q_2)$ shall be less than a permissible value ›ecified prior to the test (for instance 0,1 %), Q_1 and Q_2 being the readings of the secondary flowmeter delivered ›fore and after the test for the same discharge given by the primary method.

3.3.3 Pressure taps and connecting lines

3.3.3.1 Number and location of pressure taps

Generally, for any form of section at least two pairs of opposed pressure taps shall be used (four pressure taps). In the case of circular sections the four pressure taps shall be arranged on two diameters at right angles to each other. The taps should neither be located at or near the highest point of the measuring section in order to avoid air pockets, nor near the lowest point because of the risk of dirt obstructing the taps.

In the case of non-circular sections (in most cases rectangular sections) the taps should not be located near the corners. If taps are arranged at the top or bottom of a section, special care has to be taken to avoid disturbances due to air or dirt.

If the flow conditions are disturbed or asymmetric, more than four pressure taps shall be used.

Individual mean pressure measurements at the same measuring section shall not differ from one another by more than 0,5 % of the specific hydraulic energy of the machine or, for low head machines, by more than 20 % of the specific kinetic energy calculated from the mean velocity in the measuring section (see 3.5.2.4), both referring to operation close to the best efficiency point.

If this requirement is not fulfilled, mutual agreement should be reached either

- to select another location; or
- to make an evaluation of the specific kinetic energy distribution in the measuring section according to 3.5.2.4; or
- to accept this deviation, and add arithmetically an additional uncertainty to the measurement uncertainty in the specific hydraulic energy E (see 3.5.2.5 and J.2.3).

ı.3.2 Design of pressure taps

əssure taps shall be located in inserts of non-corroding material. Figure 31 shows typical ıerts which shall be installed flush with the wall of the conduit.

e cylindrical bore of the pressure tap shall be 2 mm to 4 mm in diameter and have a nimum length ℓ of at least twice the diameter. It shall be perpendicular to the conduit wall d free of all burrs or irregularities which could cause local disturbances. The edges of the enings shall be sharp or rounded with a radius $r \leq d/4$ smoothly joining the flow passage. The rpose of this rounding is to eliminate possible burrs.

e surface of the conduit shall be smooth and not be curved in the flow direction in the vicinity the bore for at least 100 mm upstream and downstream.

d = 2 mm to 4 mm; $\ell \geq 2d$; $r \leq d/4$

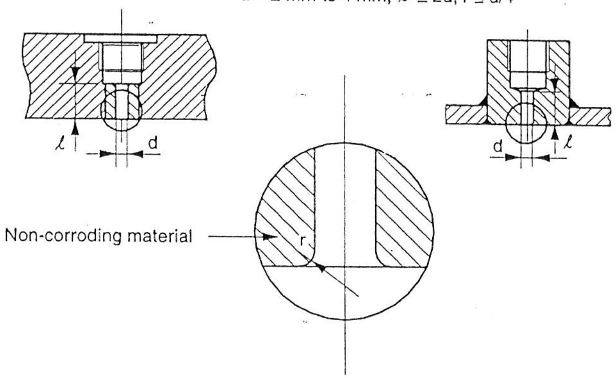


Figure 31 – Examples of pressure taps

3.3.3.3 Gauge piping

Pressure taps may be manifolded (figure 32), but each tap shall be separately valved so that it can be read individually. The diameter of the connecting piping shall be at least twice that of the tap and not less than 6 mm. The diameter of the manifold (or of the ring manifold, figure 32b) shall be at least three times the diameter of the tap. Connection pipes should, if possible, be of equal length, slope upward to the gauge or manometer with no intermediate high spots where air may be trapped. Valves with a gas-collecting chamber shall be provided at all high points for flushing out air. Transparent plastic piping, available for a wide pressure range, is recommended as it is useful in disclosing the presence of air bubbles. However, in the case of pressure fluctuations, the damping effect existing in plastic pipes shall be considered (see 3.3.3.4). No leaks shall be permitted in the gauge connections.

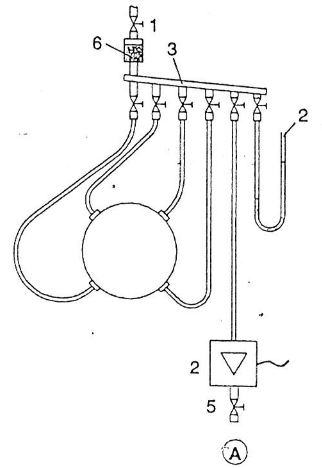


Figure 32a – Separate connecting pipe to manifold

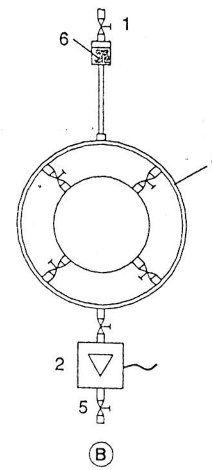


Figure 32b – Ring manifold

1 Vent
2 Pressure measuring instrument
3 Manifold
4 Ring manifold
5 Drain
6 Gas-collecting chamber

Figure 32 – Types of pressure manifolds

3.3.4 Damping devices

henever possible, all measurements shall be made under steady-state conditions and essure fluctuations eliminated at their source. In particular, no fluctuations requiring a damp-) arrangement for a manometer shall occur in the range of normal operation.

certain ranges of operation (low values of Q and σ, etc.), fluctuations cannot be avoided. In der to obtain correct readings on a pressure measuring instrument under such conditions, a itable damping device may be installed, provided the flow through it is laminar and equal in sistance in both directions, thus ensuring a linear viscous resistance. This may be secured using a capillary tube of about 1 mm bore and suitable length or a specially designed valve. imping device can also be obtained by using long plastic tubes. Additional damping may be tained from an air or surge chamber connected to the pressure line ahead of the gauge. The e of an orifice plate is not recommended because it may introduce error due to non-linear mping. A valved bypass around any throttling device shall be provided and kept open except the short time during which readings are taken. Bending or pinching the connecting pipes or serting any non-symmetrical throttling device (e.g. a valve) is not permitted.

3.4 Apparatus for pressure measurement

3.4.1 Types of apparatus

paratus for pressure measurements fall into two classifications:

primary methods (or instruments), such as liquid column manometers (see 3.3.4.2), dead weight manometers (see 3.3.4.3) and pressure weighbeams (see 3.3.4.4); these methods use only measurements of fundamental quantities (length, mass) and thus do not need any calibration;

secondary methods (or instruments), such as pressure transducers (see 3.3.4.5) and other apparatus such as spring gauges (see 3.3.4.6), which need to be calibrated against a primary method used as a standard.

e choice of the measuring apparatus shall take into account the requirements of an tomatic data acquisition system. Often a combination of primary and secondary methods is plied.

e physical principles and typical examples of possible experimental set-ups are described in e following subclauses for various instruments.

3.3.4.2 Liquid column manometers (primary method)

Liquid column manometers are used to measure low pressures or small pressure differences (up to about 5×10^5 Pa, when mercury is used as the manometer liquid). Mostly water or mercury column manometers are used (see figures 33a, 33b and 33c). In some cases other liquids of known density may be used.

The pressure measured with a liquid column manometer is determined by the following basic relation:

$$p = \rho \cdot g \cdot h$$

where

h is the height of the liquid column;

ρ is the density of the liquid used in the manometer and taken for the temperature of this liquid.

The tube of a water column manometer shall have a minimum inside diameter of 12 mm in the measuring range to minimize capillary effects. For mercury manometers, this diameter shall be at least 8 mm.

Common types of liquid manometers for gauge pressure or differential pressure are:

a) single-limb manometer (standpipe)

 – mercury pot with standpipe (figure 33a)

If the manometer is calibrated or corrected for the change in level h_1 in the pot, only the height h_2 in the single limb need be read.

 – water column (standpipe) (figure 33b)

b) U-tube

 – normal U-tube (figure 33c)

 The heights of the liquid columns in the two legs shall be read simultaneously, which can be achieved by optical reading. Whatever combination of liquid is used, the correct densities of both manometric liquids shall be used.

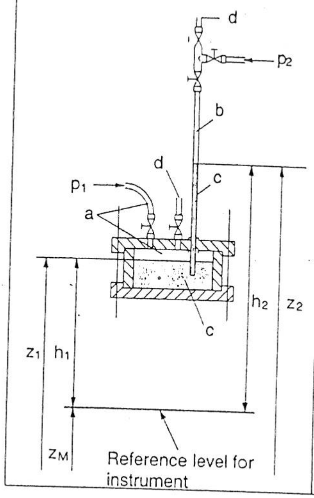
Manometer	Gauge pressure $p_2 = p_{amb}$ $p = p_{abs} - p_{amb}$	Differential pressure $p_2 \neq p_{amb}$ $\Delta p = p_1 - p_2$
Figure 33a - Pot with standpipe 	p_M = pressure at the reference level of instrument $p_M = g\,[\rho_{Hg}\,(h_2 - h_1) + \rho h_1]$ $h_1 = z_1 - z_M$ $h_2 = z_2 - z_M$ a = water b = air c = mercury d = vent	$\Delta p = g\,(\rho_{Hg} - \rho)\,(h_2 - h_1)$ $\Delta p = g\,(\rho_{Hg} - \rho)\,(z_2 - z_1)$ a = water b = water c = mercury d = vent

Figure 33 – Liquid column manometer (example of experimental setup) (values of ρ, ρ_{Hg} and ρ_a are given in annex B) *(continued)*

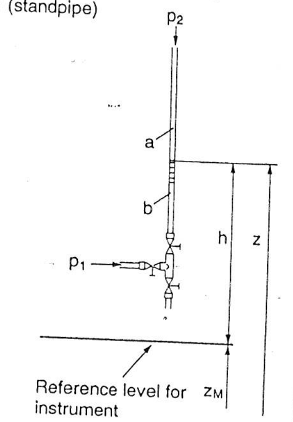
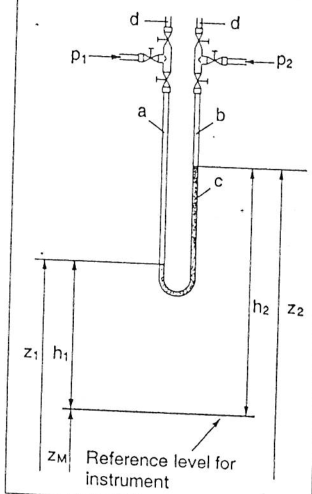
Manometer	Gauge pressure $p_2 = p_{amb}$ $p = p_{abs} - p_{amb}$	Differential pressure $p_2 \neq p_{amb}$ $\Delta p = p_1 - p_2$
Figure 33b - Water column (standpipe) 	$p_M = g\ \rho\ h$ $h = z - z_M$ a = air b = water	Not applicable
Figure 33c - U - tube 	p_M = pressure at the reference level of instrument $p_M = g\,[\rho_{Hg}\,(h_2 - h_1) + \rho h_1]$ $h_1 = z_1 - z_M$ $h_2 = z_2 - z_M$ a = water b = air c = mercury d =vent	$\Delta p = g\,(\rho_{Hg} - \rho)\,(h_2 - h_1)$ $\Delta p = g\,(\rho_{Hg} - \rho)\,(z_2 - z_1)$ a = water b = water c = mercury d =vent

Figure 33 *(concluded)*

3.3.4.3 Dead weight manometers (primary method)

Dead weight manometers (also called piston manometers) may be of the simple or differential type. Their application range depends on the effective piston area A_e and on the sensitivity of the mechanical piston system related to the pressure to be measured. For low pressures or low pressure differences, large effective areas A_e are used (e.g. $A_e \approx 0{,}0005\ m^2$ for pressures down to about 3×10^4 Pa) and vice versa (e.g. $A_e \approx 0{,}0001\ m^2$ for pressures $> 2 \times 10^5$ Pa).

The effective piston diameter d_e may be determined as the arithmetic mean value of the piston diameter d_p and bore diameter d_b: $d_e = (d_b + d_p)/2$.

This apparatus may be used for pressure calculation without further calibration if:

$$(d_b - d_p)/(d_b + d_p) \leq 0{,}001.$$

The pressure p measured at the lower end of the piston of a dead weight manometer loaded with the mass m is:

$$p = (gm)/A_e = (4gm)/(\pi d_e^2)$$

Dead weight manometers shall fulfil the following main conditions:

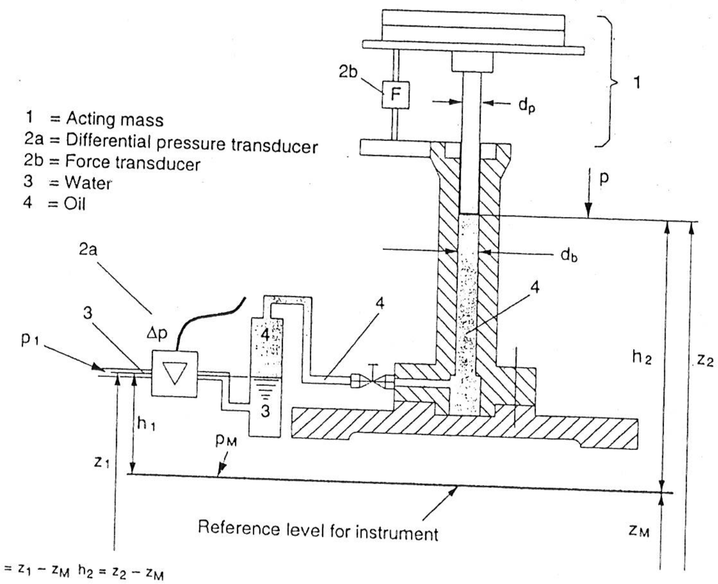
- the effective piston diameter d_e shall be determined within a relative uncertainty $f_{de} < 5 \times 10^{-4}$;
- the friction between piston and bore shall be eliminated by rotating the piston slowly ($0{,}25\ s^{-1} \leq n \leq 2\ s^{-1}$), and the cylinder shall be filled with a suitable fluid, usually by oil of low viscosity ($\nu \approx 10^{-5}\ m^2\ s^{-1}$);
- the axis of the piston shall be vertical. All the acting masses (weights, piston, weight plate etc.) shall be calibrated.

When using a data acquisition system, it is recommended to use a set-up combining a dead weight manometer with a pressure or force transducer (see figure 34).

The correction curve for these set-ups shall be determined either by checking them against a calibrated dead weight manometer without compensating devices or by loading the weight plate at constant pressure with additional small weights of calibrated appropriate mass, so that the indicator of the compensator indicates zero.

Dead weight manometers of the above type connected to transducers or load cells are preferred for use with automatic data acquisition.

The sensitivity of a dead weight manometer in good condition is less than 0,002 kg, i.e. less than $(0{,}02/A_e)$ Pa (e.g. $A_e = 0{,}0002\ m^2$, sensitivity: 100 Pa).



$h_1 = z_1 - z_M$ $h_2 = z_2 - z_M$

$p = (4mg)/(\pi d_e^2)$

$d_e = (d_b + d_p)/2$

Case a: compensation by differential pressure transducer

$p_M = p_1 + \rho g h_1 = p + \rho_{oil}\, g\, (h_2 - h_1) + \rho g h_1 + \Delta p$

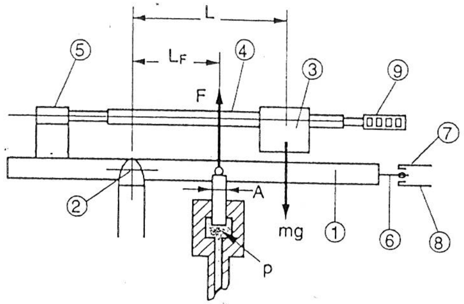
Case b: compensation by force transducer

$p_M = p_1 + \rho g h_1 = p + \rho_{oil}\, g \left(h_2 - h_1\right) + \rho g h_1 + (4F)/(\pi d_e^2)$

Figure 34 – Dead weight manometer with compensation by pressure or force transducer (example of experimental set-up)

3.3.4.4 Pressure weighbeam (primary method)

An extension of the dead weight manometer is the pressure weighbeam which comprises a weighbeam mounted on frictionless pivots, and bearing on one or more weight manometers or a differential type of weight manometer. The force exerted by the piston of the weight manometer is balanced by a jockey weight moving along the weighbeam (figure 35). The operation of the weighbeam and jockey weight may be by hand or by an automatic servobalance system. The pressure weighbeam is in principle a primary method but in some cases needs to be calibrated.



1 = Weighbeam
2 = Frictionless pivot
3 = Jockey weight (mass m)
4 = Measuring screw
5 = Servo motor
6, 7 and 8 = Electric contacts
9 = Counter of revolutions to determine L
$p = (mgL)/(AL_F)$
A = Section area of the piston
p = Pressure to be measured
F = pA = Force on the weighbeam

Figure 35 – Pressure weighbeam (example of experimental set-up)

3.3.4.5 Pressure transducers (secondary method)

Pressure transducers are electromechanical devices in which mechanical effects produced by pressure are converted into electrical signals.

Although pressure transducers are also used for measuring pressure fluctuations, this subclause deals only with static pressure measurements in steady state conditions, in order to obtain the mean value.

Depending on the pressure to be measured, the appropriate range for the pressure transducer shall be selected.

Some advantages in using pressure transducers are:

- easy integration into electronic data acquisition systems;
- they usually require negligible fluid flow through pressure taps thus providing rapid and accurate response;
- average values of fluctuating pressure or pressure differences, as well as records of transient phenomena, are easily obtained using readily available electronic equipment.

The pressure transducers should have the following characteristics:

- sufficient calibration stability;
- high repeatability, negligible hysteresis;
- low zero shift and low temperature sensitivity;
- no influence by bias effect when charged by pressure.

Operation with and without filters on the electronic equipment should be conducted to ascertain the absence of bias when filters are operating.

The complete pressure transducer system shall be calibrated under the test pressure conditions. The accuracy of a transducer will mainly be determined by the accuracy of the calibration. The calibration shall be carried out using a primary method, for example a dead weight manometer, which allows checking of the measurements of the transducer system at any time during the test.

To reduce the systematic uncertainty, it is also recommended to install two similar transducer systems in parallel and to take simultaneous readings during the test. The transducer systems are to be checked before and after the tests and if the two systems show readings that differ more than their systematic uncertainty, a comparison with a primary method shall be effected.

3.3.4.6 Other apparatus such as a spring pressure gauge (secondary method)

This type of gauge uses the mechanical deflection of a loop of tubing (plan or spiral) or of a diaphragm to indicate pressure. Depending on the pressure to be measured, the corresponding range for a spring pressure gauge shall be selected. It may be used by mutual agreement provided the gauge is of suitable accuracy, is used within its optimum measuring range (usually between 60 % and 100 % of full scale) and is suitably calibrated against a primary method before and after the test.

3.3.5 Calibration of pressure measurement apparatus

3.3.5.1 General procedure of calibration

As already mentioned, the pressure measured with secondary methods (spring gauges or transducers) shall be checked or calibrated. This can be done by comparison against primary methods (see 3.3.4.2 and 3.3.4.3) or by comparison with the static pressure obtained by a free water level as described in 3.3.5.2, or with an authorized standard.

It might also be useful to check the influence of the measuring and data acquisition system by means of a dynamic calibration, utilizing a pressure fluctuation generator of variable frequency and known average value, to ensure that no bias exists to affect the average value of the static pressure measurement.

3.3.5.2 Comparison of gauge pressures with a well defined static pressure obtained from free water level

Before and after the acceptance test and, if necessary, during the test, the readings of the pressure gauge p_M may be compared with the static pressure for zero discharge, obtained from a free water level taking into account the buoyancy of water in air:

$$p = (\rho - \rho_a)\, g \cdot \Delta z$$

3.3.6 Vacuum measurements

3.3.6.1 General requirements

For vacuum measurements, 3.3.2 to 3.3.4 also apply, except as indicated in 3.3.6.2.

3.3.6.2 Gauge piping for vacuum measurements

The gauge piping shall either be completely filled with water or, if air is used, shall be transparent to permit observation of the water level, if present. Such pipes, when filled with water, shall be flushed carefully and frequently between runs to remove any air coming out of solution or entering through the pressure tap and to maintain the water in the gauge piping at the same temperature as in the conduit. All piping and connections shall be airtight (free from leaks). Flexible pipes may be used as gauge pipes only if they are sufficiently rigid to avoid distortion or collapse by ambient pressure. Transparent plastic tubing is very convenient for observing air bubbles.

3.3.7 Uncertainty in pressure measurements

Estimations for the absolute systematic uncertainties e_p (at 95 % confidence level) that could be expected[1]:

– liquid column manometers	
mercury / water	±50 Pa to ±300 Pa
water / air	±10 Pa to ±50 Pa
– dead weight manometers	$\pm(1 \text{ to } 3) \times 10^{-3}\, p$
– pressure weighbeams	$\pm(2 \text{ to } 5) \times 10^{-3}\, p$
– spring pressure gauges	$\pm(3 \text{ to } 10) \cdot 10^{-3}\, p_{max}$ [2]
– pressure transducers	$\pm(1 \text{ to } 5) \cdot 10^{-3}\, p_{max}$ [2]

3.4 Free water level measurement [3]

3.4.1 General

In general, the determination of the specific hydraulic energy of the model machine should be based on pressure measurement within the water passages in accordance with 3.5.1.2.

In test stands with free and stable water levels, the specific hydraulic energy can be determined on the basis of measurements of the free water levels (see 3.5.3.3).

The measurement of free water levels is also necessary for some methods of discharge measurements (see, for example, 3.2.2.2, 3.2.2.3 and 3.2.3.2).

3.4.2 Choice of water level measuring sections

The measuring section for the determination of a free water level shall be chosen to satisfy the following requirements:

a) if no special similarity requirements are imposed, the model shall be arranged so that the flow is steady and free of disturbances. Especially, the free water surfaces at the measuring sections shall be stable. Sufficient submergence should therefore be provided;

b) the area used to determine the mean water velocity shall be accurately defined and readily measurable.

3.4.3 Number of measuring points in a measuring section

Measurement of free water levels shall be obtained where possible for at least two points in every measuring section or in each passage of a multiple passage measuring section and the average of the readings taken as the free water level.

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[1] These values are valid for stable pressure conditions. It should be noted that pressure fluctuations at a pump high pressure side can be important and more or less unsymmetrical, so that when they are not correctly damped (see 3.3.3.4), the uncertainties may be increased.

[2] p_{max} is the full-scale reading of the instrument.

[3] See also ISO 4373.

3.4.4 Measuring apparatus

Commonly, a free water level is measured from a reference level z_M of the instrument, determined by means of a high precision instrument in relation to other reference levels.

The free water level is generally not measured directly in the section but in a stilling well connected to the measuring section as shown in figure 36.

3.4.4.1 Point or hook gauge

Point or hook gauges (see figure 37) may be used to determine the level of calm water, preferably in stilling wells or directly in the flow if the free level is particularly undisturbed. In place of the normal visual indication of contact with the water, electrical, optical or other indicators may be used, provided they are calibrated against the direct visual method.

3.4.4.2 Float gauge

Float gauges may be used where the water level varies a great deal. The float diameter shal be at least 150 mm. The minimum dimension of a stilling well shall be 200 mm. Gauges shall be sensitive within 1 mm when manually displaced from the true reading (resolution of ±0,001 m).

3.4.4.3 Pressure measuring device

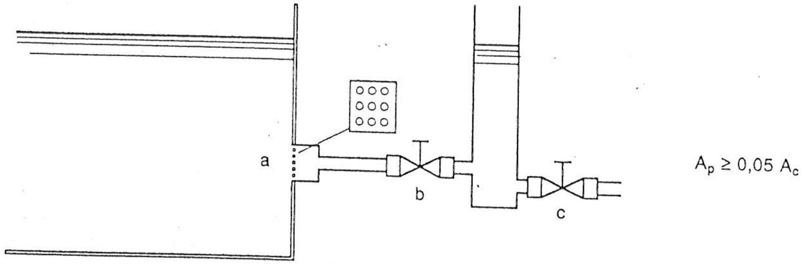
Immersible pressure transducers or any other pressure measuring device, including liquic manometers (standpipe) may be used (see 3.3) to determine the free water level. The pressure indication shall be checked with no water flowing.

3.4.4.4 Bubbler with compressed air

The free water level may also be determined by means of the pressure inside a tube fillec with compressed air, the so-called gas purge "bubbler" technique (for detailed informatior see IEC 60041).

3.4.4.5 Other methods

Other methods may be used, for example ultrasonic devices and capacitive methods, as lonç as they meet the required accuracy (see 3.4.5).



a Perforated plate, flush with the wall

b Disconnecting valve

c Flushing valve

A_p Total cross-sectional area of perforation

A_c Cross-sectional area of the well

Figure 36 – Stilling well

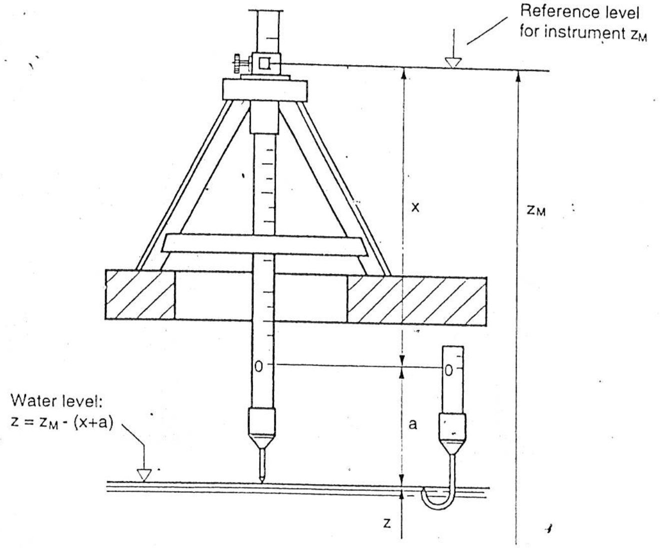


Figure 37 – Point and hook gauges

3.4.5 Uncertainty in free water level measurement

Estimations for the absolute systematic uncertainties e_z (at 95 % confidence level) that could be expected in the case of calm water conditions and velocity smaller than or equal to 1,0 m/s (the lower value referring to a velocity close to zero):

- point or hook gauges ±0,001 m to ±0,003 m
- float gauges ±0,001 m to ±0,003 m
- immersible pressure transducers ±(0,5 to 5) × 10^{-3} z_{max} [1)]
- bubbler with compressed air ±0,001 m to ±0,003 m
- ultrasonic device ±0,002 m to ±0,010 m

In the case of very turbulent flow and v > 1,0 m/s, for example near the outlet of a turbine draft tube, the uncertainties may be considerably higher.

3.5 Determination of E and NPSE

3.5.1 General

3.5.1.1 Object

The specific hydraulic energy E of the machine shall be determined in any test on a hydraulic model machine and the net positive suction specific energy NPSE determined when required. The quantities for the determination of E and NPSE are measured at steady-state conditions as an average over time. The formulae for their evaluation are given in 1.3.3.6.2 and 1.3.3.6.5 respectively.

Annex C provides the derivation of the formula for E.

3.5.1.2 Method of determination

To determine the specific hydraulic energy acting on the model machine, it is necessary to evaluate the specific energy of water in the high pressure and low pressure reference sections. For the net positive suction specific energy, the specific energy of water is evaluated in the low pressure reference section with reference to a specified level. Whenever possible, the absolute pressure, the mean velocity and the elevation should be directly determined in the reference sections, in particular at the low pressure side where the pressure shall be measured within the draft tube. In some cases, due to special model test equipment, it can be mutually agreed to choose a measuring section located as near as possible to the corresponding reference section or even to replace pressure measurements by free water level measurements. The measurement of the pressure is described in 3.3, whereas the measurement of free water level, although seldom used for model tests, is described in 3.4.

3.5.1.3 Steady-state conditions and number of readings

Readings required to determine the specific hydraulic energy shall be taken at regular intervals and when steady-state conditions prevail as defined in 2.3.2.3.1. The number of readings and the intervals between them shall provide for a sufficiently good approximation of the mean value, taking into account the performance of the data acquisition system (see 2.3.2.3 and 3.1).

1) z_{max} is the full scale reading of the instrument.

·.5.2 Determination of the specific hydraulic energy E

·.5.2.1 Measuring sections

·.5.2.1.1 General

'he basic conditions to achieve an accurate determination of the specific hydraulic energy are lescribed in 3.5.1.2. Requirements for the choice of a pressure measuring section are given in ·.3.2.

·.5.2.1.2 Shifted measuring sections

/lodel acceptance tests will generally be conducted with measurements at the reference ;ections of the machine which are specified in the contract. Only in exceptional cases may the neasuring sections be different from the reference sections. This may become necessary if low disturbances occur at the reference section, caused by the machine or by the inflow :onditions; such a shifting shall be agreed by the parties. In such cases, the components ıfluencing the flow pattern may also be modeled, if contractually agreed.

:xamples

:or pumps, the high pressure measuring section should be moved if the pressure and velocity listributions are such that the calculation of specific hydraulic energy from the mean values vould result in significant errors. A measuring section located some conduit diameters from the ›ump will generally increase the reliability of the measurement.

:or turbines, a butterfly valve close to the high pressure reference section may necessitate elocating the measuring section since it is difficult to assess the effect of the valve on the neasurement.

·.5.2.1.3 Specific hydraulic energy correction for shifted measuring sections

Vhen the measuring section is not the reference section, the loss of specific hydraulic energy ›etween the measuring section and the reference section shall be taken into account, due :onsideration being given to the flow direction and distribution and the relative position of the wo sections. Evaluation of this loss may be based on theoretical knowledge and/or practical ›xperience.

3efore a decision is made to use a different measuring section, due consideration shall be jiven to the uncertainty introduced by the loss calculation compared to that arising from ınsatisfactory measuring conditions at the reference section.

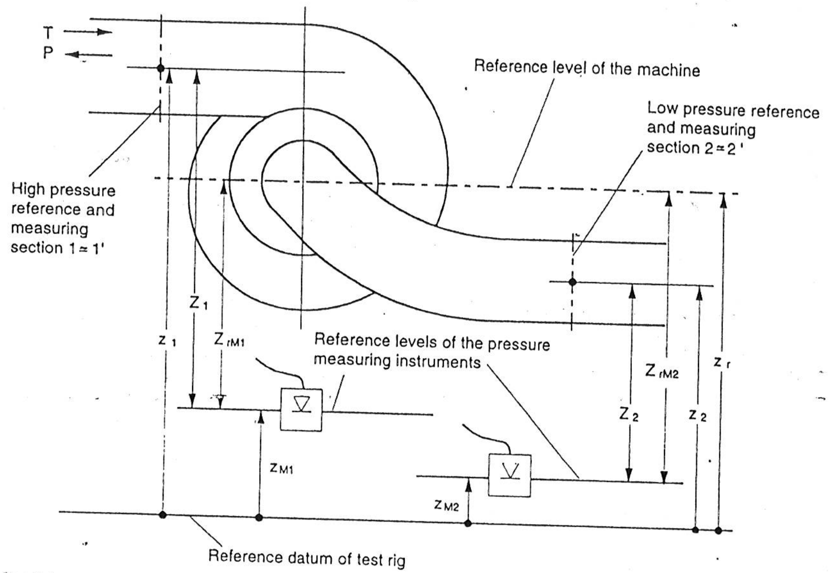
3.5.2.2 Reference levels

3.5.2.2.1 Reference datum

All elevations shall be referred to a reference datum such as a reference level of the test rig or :he reference level of the machine (see 1.3.3.7.6). An example of main elevations and heights :s shown in figure 38.

3.5.2.2.2 Difference of elevations

It is only important to establish accurate differences of elevations. The most important difference of elevations for model tests is the difference between reference level of machine z_r and reference level of pressure measuring instrument z_M: $Z_{rM} = z_r - z_M$. If the reference level of all the pressure measuring instruments is the same and if it is taken as reference datum, then $Z_{rM} = z_r$ (see figure 38).



Special case: $z_{M1} = 0$ and $z_{M2} = 0$, then $Z_{rM1} = Z_{rM2} = z_r$

Figure 38 – Example showing main elevations, heights and reference levels of the test rig and model machine

3.5.2.3 Water density (see 2.5.3)

From the definition of specific hydraulic energy of the machine in 1.3.3.6.2 the mean water density $\overline{\rho}$ shall be calculated as the mean of densities at the two reference sections. As the temperature difference between inlet and outlet of the machine is small, the temperature of the water at the low pressure reference section may be used for calculating the densities involved in the evaluation of $\overline{\rho}$.

The values of distilled water density (see 2.5.3.1.3 and table B.2) can normally be used as water density ρ_w for the determination of E or H = E/g because:

a) the value of the density for the actual water ρ_{wa} in the model test equipment differs rather little from the value of the density for distilled water ρ_{wd}: in general the deviation is smaller than 0,05 % (see 2.5.3.1.2);

b) the hydraulic power $P_h = E\,(\rho Q)_1$, (see 1.3.3.8.1), as the main hydraulic quantity for the determination of the efficiency, depends only to second order accuracy on density, if the specific hydraulic energy of the model machine is predominantly obtained by pressure measurements (see simplified formulae in 3.5.3.2.1 and 3.5.3.2.2 and the explanation in annex D).

In special cases, it may be necessary to determine the density ρ_{wa} of actual water used (see 2.5.3.1.2).

.2.4 Specific kinetic energy

convention the specific kinetic energy term in a reference section is determined from the an velocity of the water perpendicular to that section and is taken as $e_c = v^2 / 2$.

e mean velocity v is the actual discharge passing through the reference section divided by area of the same reference section[1]. This area shall be measured when model similarity is ecked. The same convention is applied when the measuring section differs from the erence section within the limits of the machine.

.2.5 Uncertainty in the determination of specific hydraulic energy E

e determination of E shall be adapted for each method and arrangement according to the amples described in 3.5.3.

J.2.3 an example for the determination of the relative systematic uncertainty f_E is shown rresponding to figure 39).

order to account for the effect of non-uniformity of the pressure distribution in the measuring ctions (see 3.3.3.1), an additional uncertainty $f_{\Delta E}$ is added arithmetically to the total relative certainty f_E: $f_{E,corr.} = f_E + f_{\Delta E}$

.3 Simplified formulae for E

.3.1 General

indicated in annex C, the general formula given in 1.3.3.6.2 is a convenient approximation the exact value of the specific hydraulic energy of the machine.

rther simplifications are possible in each specific case and approximations may be roduced, for example when the water compressibility or the difference in ambient pressure tween sections 1 and 2 is negligible.

For the local velocity v_i in the streamline of a fluid, the specific kinetic energy is $e_{c,i} = v_i^2 / 2$. The mean value of specific kinetic energy of the flow passing a cross-section A (with the mean axial velocity v) can be expressed $e_c = \alpha v^2 / 2$, the kinetic energy coefficient α being defined (see ISO 4006) by:

$$= \int_A v_i^2 v_{zi} \cdot dA \Big/ v^3 A$$, v_{zi} being the meridional component of v_i.

e coefficient α would be equal to 1 for a uniform velocity distribution (rectangular velocity profile) and is always eater than 1 in industrial flow.

testing hydraulic machines, the actual flow configuration in a measuring section shows a non-uniform velocity stribution arising from the layout of the plant and the operation of the machine. It is usually assumed that the flow nfiguration in the model and in the prototype is approximately the same. However, it is impracticable and time-nsuming to determine α during model tests from a detailed measurement of the velocity distribution. Therefore, it conventionally agreed to assume $\alpha = 1$ and thus $e_c = v^2 / 2$.

though the difference between the conventional and actual values of the specific kinetic energy can, for low-head achines, reach 1 % to 2 % of the specific hydraulic energy of the machine, it is agreed to disregard this difference hen evaluating the uncertainty in the measurements (see for example the calculation of the uncertainty in E in ause J.2, where the uncertainty in the specific kinetic energy only accounts for uncertainties in the determination the discharge Q and area A).

It can be assumed that the acceleration due to gravity and the ambient pressure are constant throughout the test rig:

$$\overline{g} = g_1 = g_2 = g \qquad \text{and} \qquad p_{amb1} = p_{amb2} = p_{amb.}$$

The simplified formulae established in this subclause are typical for the described measuring installations. Only the most common installations are reviewed. A simplified formula shall not be used for a different installation without carefully examining its adequacy.

3.5.3.2 Determination of E from pressure measurement (see 3.3)

3.5.3.2.1 Measurement of differential pressure

Figure 39 shows schematically the measuring installations for the determination of specific hydraulic energy of the machine whenever a differential pressure measuring instrument is used. This solution is especially suitable for low model test heads where instruments of sufficient accuracy are available.

3.5.3.2.2 Separate measurement of pressure

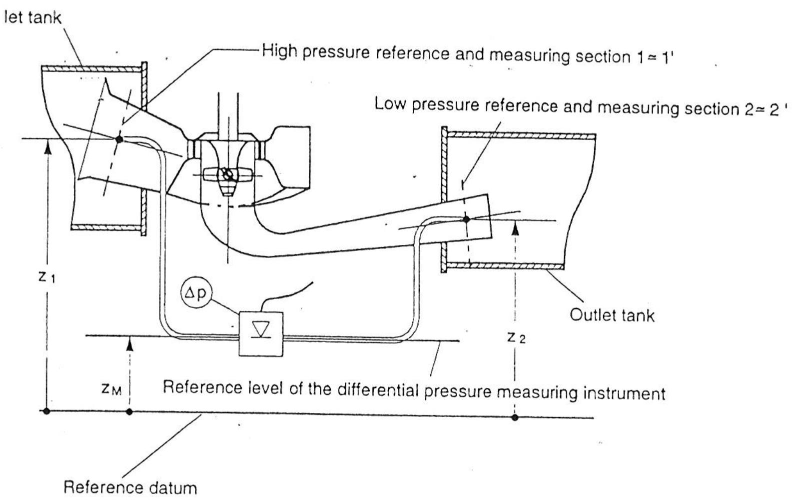
a) Reaction machines

The pressures are measured separately in each section. For pressure differences less than about 400 000 Pa (about 40 m of water column), the compressibility of water can be neglected.

If the pressure measuring instrument provides absolute pressure measurements (e.g. pressure transducers), the ambient pressure does not need to be considered.

If the pressure measuring instrument provides gauge pressure measurements (e.g. spring-pressure gauge or liquid column manometer), it shall be verified whether the difference in the ambient pressure on the instruments shall be included (see figure 40 [influence negligible] and figure 41 [influence taken into account]).

Further simplifications may be introduced by setting the pressure measuring instruments at the same reference level. This is generally easy to achieve (see figure 40, last simplification).



$$E = gH = \left(p_{abs1} - p_{abs2}\right)/\bar{\rho} + \left(v_1^2 - v_2^2\right)/2 + g\ \left(z_1 - z_2\right)$$

sing a differential pressure measurement, the following is obtained:

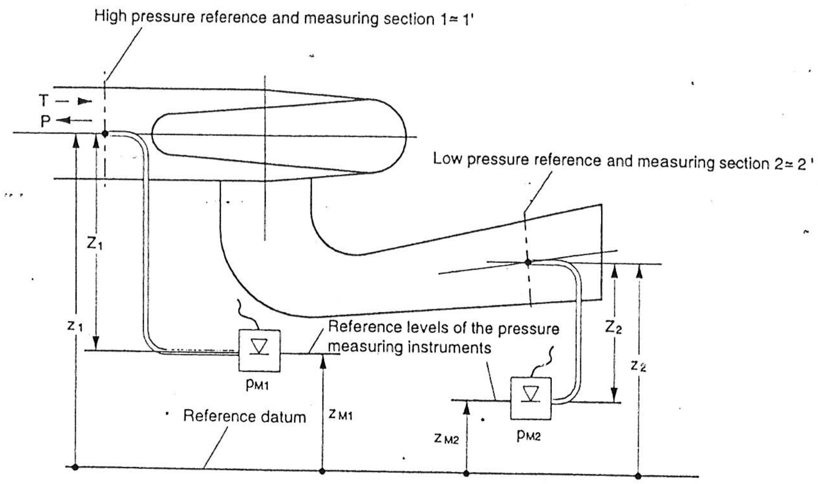
$$\left(p_{abs1} - p_{abs2}\right)/\bar{\rho} = \Delta p/\bar{\rho} + g\left[\left(z_2 - z_M\right)\cdot \rho_2/\bar{\rho} - \left(z_1 - z_M\right)\cdot \rho_1/\bar{\rho}\right]$$

hen applied to low model test heads ($\Delta p \leq 400\,000$ Pa, i.e. $H \leq 40$ m), the compressibility of ater is negligible and it is assumed that $\bar{\rho} = \rho_1 = \rho_2$.

herefore the simplified formula is:

$$E = \Delta p/\rho_2 + \left(v_1^2 - v_2^2\right)/2$$

Figure 39 – Determination of specific hydraulic energy through differential pressure measuring instrument



$$E = gH = \left(p_{abs1} - p_{abs2}\right)/\bar{\rho} + \left(v_1^2 - v_2^2\right)/2 + g\ \left(z_1 - z_2\right)$$

Gauge manometers are applied at points 1 and 2. The difference in ambient pressure between z_{M1} and z_{M2} is negligible, because $(z_{M1} - z_{M2})$ is small compared to H; therefore:

$$p_{ambM1} = p_{ambM2} = p_{amb}$$

$$p_{abs1} = p_{M1} + \rho_1 g\ \left(z_{M1} - z_1\right) + p_{amb}$$

$$p_{abs2} = p_{M2} + \rho_2 g\ \left(z_{M2} - z_2\right) + p_{amb}$$

If the compressibility of water can be neglected, then $\rho_1 = \rho_2 = \bar{\rho}$

hence $\left(p_{abs1} - p_{abs2}\right)/\bar{\rho} = \left(p_{M1} - p_{M2}\right)\ /\bar{\rho} + g\left(z_{M1} - z_1 - z_{M2} + z_2\right)$,

therefore, the simplified formula is:

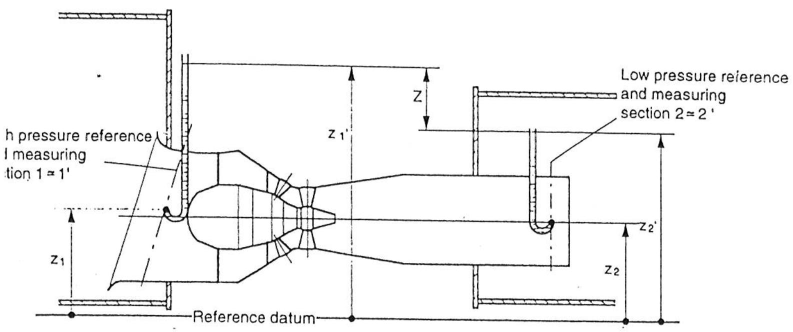
$$E = gH = \left(p_{M1} - p_{M2}\right)/\bar{\rho} + \left(v_1^2 - v_2^2\right)/2 + g\left(z_{M1} - z_{M2}\right)$$

Further simplification:

If the pressure measuring instruments are on the same elevation, $z_{M1} = z_{M2}$, then

$$E = gH = \left(p_{M1} - p_{M2}\right)/\bar{\rho} + \left(v_1^2 - v_2^2\right)/2$$

Figure 40 – Determination of specific hydraulic energy of the machine through separate measurement of gauge pressures



: $z_1' - z_2'$

$$E = gH = \left(p_{abs1} - p_{abs2}\right)/\overline{\rho} + \left(v_1^2 - v_2^2\right)/2 + g\left(z_1 - z_2\right)$$

ıter column manometers are applied at sections 1 and 2:

e compressibility of water is negligible, because the difference of pressure between sections ınd 2 is small.

erefore: $\rho_1 = \rho_2 = \overline{\rho} = \rho$

nce: $p_{abs1} = \rho \cdot g\ (z_{1'} - z_1) + p_{amb1'}$

$p_{abs2} = \rho \cdot g\ (z_{2'} - z_2) + p_{amb2'}$

$p_{amb1'} - p_{amb2'} = -\rho_a \cdot g\ (z_{1'} - z_{2'})$

e simplified formula is

$$E = g\left(z_{1'} - z_{2'}\right)\left(1 - \rho_a/\rho\right) + \left(v_1^2 - v_2^2\right)/2$$
$$= g \cdot Z\left(1 - \rho_a/\rho\right) + \left(v_1^2 - v_2^2\right)/2$$

Figure 41 – Determination of specific hydraulic energy of the machine through separate measurement of pressures by water column manometers

b) **Pelton turbines (impulse turbines)**

For Pelton turbines, when the housing is under atmospheric pressure, only the measurement of the pressure p_1 at the high pressure reference section is required. Thus further simplifications can be introduced if the general formula is applied to Pelton turbines (see figures 42 and 43).

By convention, v_2 is taken as zero, elevation z_2 of the low pressure réference section is the mean elevation of all contact points of the jet axis with the Pelton jet circle and the pressure inside the housing is assumed equal to the ambient pressure, provided the housing is not pressurized and is supplied with sufficient air.

If the housing is pressurized, the ambient pressure in the housing shall be measured and accounted for when determining E ($p_{amb2} \neq p_{amb1}$).

3.5.3.3 Determination of E from water level measurements

Whenever possible, water level measurements (see 3.4) should be avoided for model acceptance tests. Nevertheless, if it is necessary or it is agreed to determine E by measurements of free water levels, especially at the low pressure side, the methods described in 3.4 shall be applied.

The requirements for the flow condition in the surrounding area of the measuring sections are described in 3.4.2.

Figure 44, referring to a low head machine, shows the evaluation of the specific hydraulic energy from measurements of the water levels. The low pressure measuring section 2' shall be as close as possible to the draft tube outlet. For such measurement the water level should be measured directly above 2'. To evaluate the mean velocity, the walls of the draft tube are assumed to extend out to section 2', delineating the fictional area of the section.

3.5.4 Determination of the net positive suction specific energy NPSE

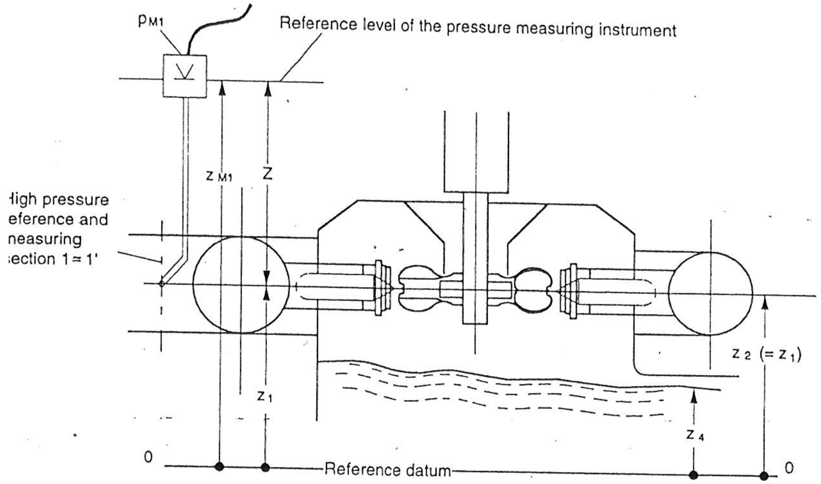
3.5.4.1 Definition

The net positive suction specific energy NPSE is referred to the low pressure side of the machine. Its definition and the general formula for its determination are given in 1.3.3.6.5.

Measurement may be affected by practical circumstances, in the same way as the specific hydraulic energy E of the machine. Subclause 3.5.2 shall also be considered for the determination of the net positive suction specific energy.

3.5.4.2 Simplified formulae

As long as the pressure can be measured in the low pressure reference section, the general formula is directly applicable and valid for both operating modes, pump and turbine. In figure 45, three cases for the determination of NPSE are described.



$Z = z_{M1} - z_2$

is conventionally assumed that the low pressure reference section is the equatorial plane of ıe runner at elevation z_2. For non-pressurized housing, the pressure inside the housing is onventionally assumed to be equal to the ambient pressure.

$$E = gH = (p_{abs1} - p_{abs2})/\overline{\rho} + (v_1^2 - v_2^2)/2 + g(z_1 - z_2)$$

'he difference in ambient pressure between z_{M1} and z_2 is neglected because Z is small ompared to H.

'herefore: $p_{ambM1} = p_{amb2} = p_{amb}$

:urther it is assumed $Z \cdot \rho_1/\overline{\rho} = Z$

lence: $p_{abs1} = p_{M1} + Z \cdot \rho_1 \cdot g + p_{amb}$

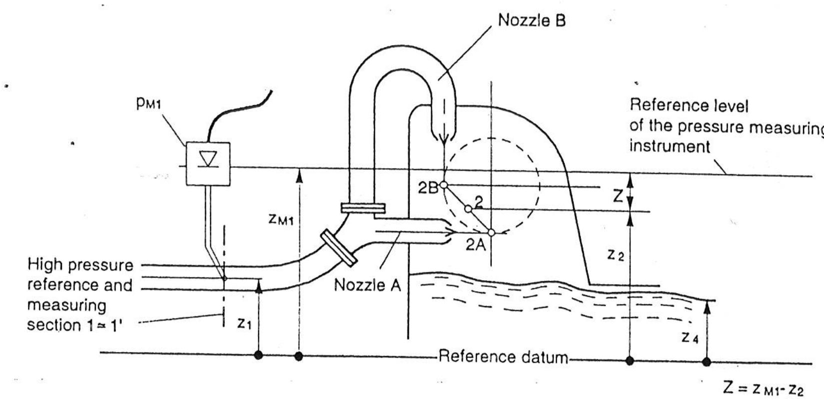
vhere p_{M1} is the gauge pressure measured at z_{M1}

$$p_{abs2} = p_{amb}$$

\s $z_1 = z_2$ and assuming $v_2 = 0$, the simplified formula is:

$$E = p_{M1}/\overline{\rho} + g \cdot (z_{M1} - z_2) + v_1^2/2 = p_{M1}/\overline{\rho} + g \cdot Z + v_1^2/2$$

Figure 42 – Pelton turbines with vertical axis. Determination of specific hydraulic energy of the machine



NOTE – In the case of multiple nozzles, the elevation z_2 of the low pressure reference section is defined as the average of the elevations of the points of contact (2A and 2B in the sketch).

Case of non-pressurized housing: the pressure inside the housing is conventionally assumed as equal to the ambient pressure.

$$E = \;' ' = (\; _{abs1} - \; _{abs2})/\overline{\rho} + (\;^2_1 - \;^2_2)/2 + \;(\;_1 - z_2)$$

The difference in ambient pressure between z_{M1} and z_2 is neglected because Z is small compared to H.

Therefore: $p_{ambM1} = p_{amb2} = p_{amb}$

Further it is assumed: $Z \cdot \rho_1/\overline{\rho} = Z$

Hence: $p_{abs1} = p_{M1} + (z_{M1} - z_1)\rho_1 g + p_{amb}$

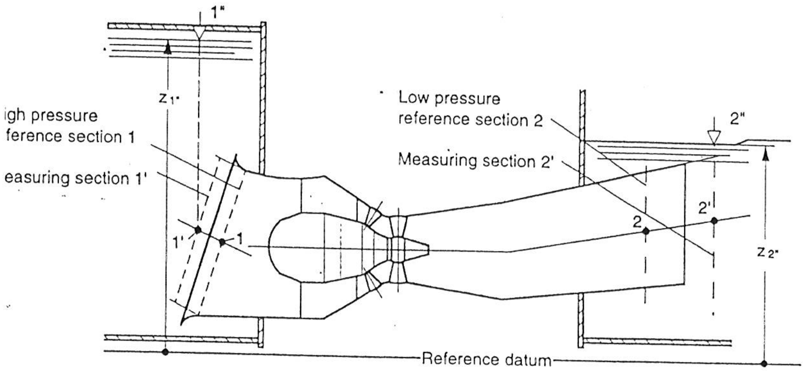
where p_{M1} is the gauge pressure measured at z_{M1}

$$p_{abs\,2} = p_{amb}$$

Assuming $v_2 = 0$, the simplified formula is:

$$E = \;_{M1}/\overline{\rho} + \; \cdot (\;_{M1} - \;_2) + \;^2_1/2 = \;_{M1}/\overline{\rho} + \; \cdot \; + v_1^2/2$$

Figure 43 – Pelton turbines with horizontal axis. Determination of specific hydraulic energy of the machine



$$E = gH = \left(p_{abs1} - p_{abs2}\right)/\bar{\rho} + \left(v_1^2 - v_2^2\right)/2 + g\left(z_1 - z_2\right)$$

ections 1' and 2' are chosen as measuring sections.

$$E = gH = \left(p_{abs1'} - p_{abs2'}\right)/\bar{\rho} + \left(v_{1'}^2 - v_{2'}^2\right)/2 + g\left(z_{1'} - z_{2'}\right) \pm E_{L1'-1} \pm E_{L2-2'}$$

osses $E_{L1'-1}$ between 1' and 1 and $E_{L2-2'}$ between 2 and 2' are subtracted for a turbine and dded for a pump with the situation described in the sketch above[1]).

e compressibility of water is neglected because the difference of pressure between 1' and 2' small.

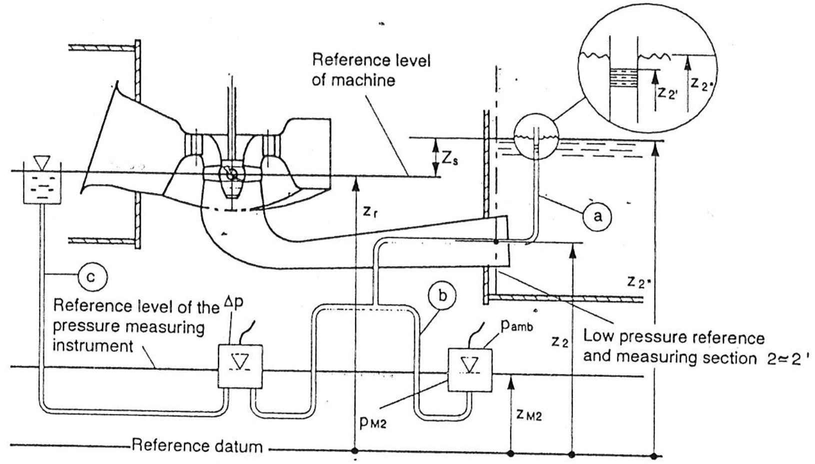
erefore $\rho_{1'} = \rho_{2'} = \bar{\rho} = \rho$

e simplified formula becomes (see figure 41):

$$E = g\cdot\left(z_{1''} - z_{2''}\right)\left(1 - \rho_a/\rho\right) + \left(v_{1'}^2 - v_{2'}^2\right)/2 \pm E_{L1'-1} \pm E_{L2-2'}$$

Figure 44 – Low-head machines. Determination of specific hydraulic energy of the machine using water levels

It is difficult to establish methods of calculation of energy losses $E_{L1'-1}$ and $E_{L2-2'}$, particularly in non-/mmetrical or swirling flow (high values of the kinetic energy coefficient α), which would enable in this standard to ve some guidance generally valid. Particularly when the intake and outlet of the unit are not fully modelled, the /aluation method of these losses is to be agreed upon before the tests.



$Z_s = z_r - z_{2'}$[1])

$$NPSE = g\cdot NPSH = \left(p_{abs2} - p_{va}\right)/\rho_2 + v_2^2/2 - g\left(z_r - z_2\right)$$

Case a) Liquid (water column) manometer at point 2:

$$p_{abs2} = \rho_2 \cdot g\left(z_{2'} - z_2\right) + p_{amb}$$

The simplified formula is:

$$NPSE = \left(p_{amb} - p_{va}\right)/\rho_2 + v_2^2/2 - g\left(z_r - z_{2'}\right)$$
$$= \left(p_{amb} - p_{va}\right)/\rho_2 + v_2^2/2 - g\cdot Z_s$$

Case b) Pressure gauge at level z_{M2}, measuring at point 2:

$$p_{abs2} = p_{M2} + g\cdot\rho_2\cdot\left(z_{M2} - z_2\right) + p_{amb}$$

The simplified formula is:

$$NPSE = \left(p_{M2} + p_{amb} - p_{va}\right)/\rho_2 + v_2^2/2 - g\left(z_r - z_{M2}\right)$$

Case c) Differential manometer connected with a pot situated at the reference level of the machine:

The simplified formula is: $NPSE = \left(\Delta p + p_{amb} - p_{va}\right)/\rho_2 + v_2^2/2$

Figure 45 – Determination of net positive suction energy NPSE and net positive suction head NPSH

1) Z_s is positive when the level $z_{2'}$ is lower than the reference level of machine z_r and vice versa.

Shaft torque measurement

.1 General

e calculation of mechanical power of the runner/impeller P_m requires the determination of torque applied to the runner/impeller T_m:

$$P_m = 2 \cdot \pi \cdot n \cdot T_m$$

h $T_m = T \pm T_{Lm}$ $\left[\begin{array}{l} + \text{ for turbine rotational direction} \\ - \text{ for pump rotational direction} \end{array}\right.$

$_n$ is the friction torque due to seal and bearing arrangement.

ncipally, two different measuring systems can be applied for torque measurement:

those of the type "swinging frame", where T_{Lm} is a so-called "inner torque", i.e. T_{Lm} is taken into account by the system itself (in the following, described as "bearing of rotating parts in balance", see for instance figures 46 and 47).

those where T and T_{Lm} are measured separately (in the following, described as "bearing of rotating parts not in balance", see for instance figure 49).

e shaft torque T of runner/impeller may be absorbed or generated by:

an electrical machine, usually a motor/generator with variable speed,

absorbed by different types of brakes:

eddy-current brake,

hydraulic brake,

mechanical brake.

6.2 Methods of torque measurement

6.2.1 Primary method

the primary method, the torque T is determined by the force F applied to a lever arm ultiplied by the radius r at which it is applied: $T = F \cdot r$

e actual force applied to balance the swinging frame may be measured by:

weighing masses on a lever system (calibrated weights and calibrated lever arm). This is in principle the basic primary method

) one of the following methods, which can be calibrated *in situ* by the basic primary method a):

- force transducer,
- manometer (via a rotating piston),
- mechanical balance.

o increase the accuracy in determining the total force, it is recommended to counterbalance a art of the force acting on the arm by means of calibrated weights.

3.6.2.2 Secondary method

A torquemeter may be used provided its accuracy is acceptable to all parties and it is calibrated by the primary method. A torquemeter comprises a length of shafting whose torsional strain, when rotating, is converted to an electrical output quantity by optical, electrical or other means. The design and the arrangement of this type of torquemeter shall be such that the measurement is not influenced by speed, temperature, axial thrust or radial thrust.

3.6.3 Methods of absorbing/generating power

3.6.3.1 Motor/generator with variable speed

This method comprises a machine for absorbing and generating power electrically, suitably mounted so that the mechanical torque can be measured. This device can be used for both turbine and pump models.

3.6.3.2 Eddy-current brake

The operation of this electromagnetic brake is restricted to absorbing power.

3.6.3.3 Hydraulic brake

This brake absorbs power hydrodynamically. It is unsuitable for use at low speeds as its power absorption varies with n^3.

3.6.3.4 Mechanical brake

This brake absorbs power by friction. It has the advantage that high torques can be applied at low speeds, even down to zero rotational speed. The torque applied shall be steady and the mechanical system shall be vibration free.

3.6.4 Layout of arrangement

3.6.4.1 General

Figures 46 to 54 show practical arrangements comprising primary and secondary methods of torque measurement. All arrangements shown can be adapted for either a horizontal axis or a vertical axis test stand.

Figures 46 and 47 illustrate the principle of the balance arrangement. The torque acting on the runner/impeller is measured at the torque arm of the swinging frame.

If the swinging frame, as shown in figure 48, is formed by two separate frames, the torques acting on each frame shall be measured and added algebraically.

Figure 49 shows an arrangement not fully in balance, therefore the losses due to bearing and seal shall be measured separately.

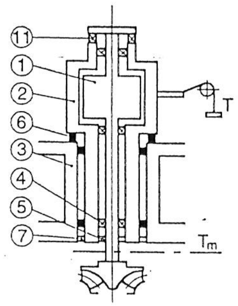
Figure 50 shows an arrangement with the model shaft through a draft tube elbow. Normally, such an arrangement is not fully in balance and losses have to be accounted for.

Special arrangements are necessary for testing multistage pumps or pump-turbines as shown in figure 51. Particular attention is to be paid to mechanical losses P_{Lm} which shall be accurately known and accounted for over the whole range of speed and pressure occurring during the test.

Figure 52 illustrates an arrangement using a torquemeter. Figures 53 and 54 show variations with an additional guide bearing which may be balanced or not.

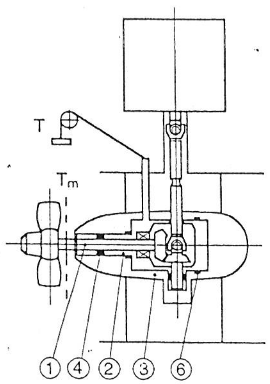
ımenclature for figures 46 to 54

rotating part
swinging frame
stationary part
bearing of rotating part in balance
mechanical seal in balance
low friction bearing of the swinging frame
labyrinth seal, membrane
bearing of rotating part not in balance
mechanical seal of rotating part not in balance
torquemeter
axial thrust bearing
– — reference section for torque measurement



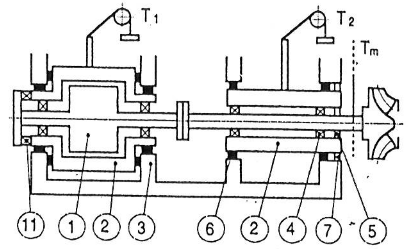
$T_m = T$

Figure 46 – Balance arrangement



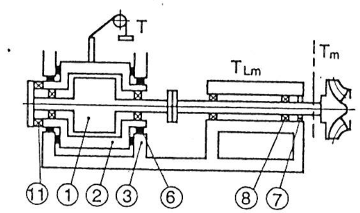
$T_m = T$

Figure 47 – Balance arrangement with gear



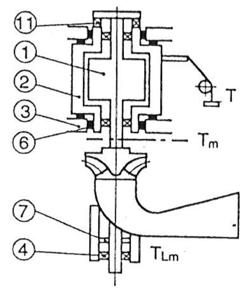
$T_m = T_1 \pm T_2$

Figure 48 – Balance arrangement with two separate frames



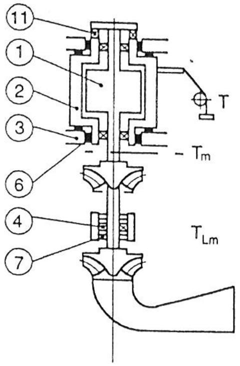
$T_m = T \pm T_{Lm}$

Figure 49 – Arrangement with machine bearings and seals not in balance



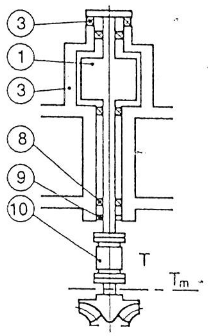
$T_m = T \pm T_{Lm}$

Figure 50 – Arrangement with lower bearing and seal not in balance



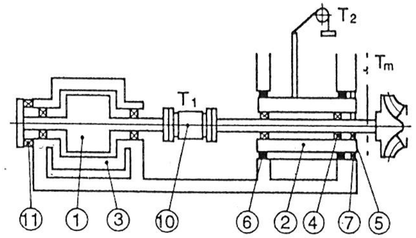
$T_m = T \pm T_{Lm}$

Figure 51 – Arrangement with intermediate bearing and seal not in balance



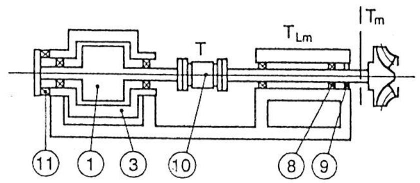
$T_m = T$

Figure 52 – Arrangement using a torquemeter



$T_m = T_1 \pm T_2$

Figure 53 – Arrangement using a torquemeter with machine bearings and seals in balance



$T_m = T \pm T_{Lm}$

Figure 54 – Arrangement using a torquemeter with machine bearings and seals not in balance

3.6.4.2 Suspension of swinging frame

To meet the requirements regarding uncertainty of measurement over the measured range, special low friction bearings are necessary for suspending the swinging frame, i.e. oil or water hydrostatic bearings. The swinging frame should be perfectly balanced, otherwise it is necessary to limit the rotational movement.

3.6.4.3 Windage losses

Whatever speed of the motor/generator, there shall be no torque reaction caused by windage or by a blower; if there is, this shall be taken into account.

3.6.4.4 Cooling fluid connections

The torque measuring device shall be designed in such a way that the cooling fluid enters and leaves without introducing errors due to tangential velocity components. Flexible pipes (if used) shall impose no measurable tangential restraint, especially when under pressure. Dash-pots (if used) shall be shown to impose equal resistance to motion in either direction. Furthermore, the shaft glands retaining the liquid shall either impose no sensible frictional torque or be provided with a torque measurement device.

;.4.5 Seals

;ealing between swinging and fixed parts is made by means of friction or membrane seals, ;se shall be calibrated.

;.4.6 Electrical leads

;ctrical connections shall impose no measurable tangential restraint. Braided flexible copper ;ds or mercury pots are suitable for this purpose.

;.5 Checking of system

;e checks described below are recommended to verify the correct operation of the whole 'que measuring arrangement; nevertheless, the measuring device shall be calibrated.

6.5.1 Sensitivity test

;e sensitivity of a testing arrangement indicates the lowest torque difference which can be ;asured through the system. The sensitivity strongly depends on the layout and capacity of ; arrangement. Reduced sensitivity indicates to incorrect functioning.

;pending on the layout and capacity of the arrangement used, the sensitivity shall be in a nge of 0,05 N·m to 0,5 N·m whereby the low value is valid for $T_{m,max}$< 500 N·m.

6.5.2 Speed test

;is test is carried out with a dismantled runner/impeller or a disconnected shaft. The system nctions correctly if the mechanical torque T_m remains zero for the whole range of speeds.

6.5.3 Counterbalancing

uring this check the applied torque/force, or part of it, is counterbalanced by means of ;rtified masses. The system functions correctly if the display is reduced according to the ;unterbalancing mass.

6.6 Calibration

.6.6.1 Primary method

uring calibration the following quantities shall be measured:

- length of brake lever;
- force on brake lever;
- tare weight of the brake lever if necessary.

he force acting on the brake lever shall be applied by means of certified masses in the irection of both increasing and decreasing loads. Metal tapes and frictionless pulleys shall be sed for applying the torque balancing weights.

.6.6.2 Secondary method

Vhen using this method, a calibration of the arrangement against a primary method shall be arried out.

3.6.6.3 Friction torque T_{Lm}

If part of bearing/sealing arrangement is not included in the swinging frame, the relevant total friction torque T_{Lm} shall be determined by an appropriate test taking into account the dependency on speed and shaft seal pressure.

3.6.7 Uncertainty in torque measurement

The expected relative systematic uncertainties (at a confidence level of 95 %) are set out below.

3.6.7.1 Uncertainty in the shaft torque measurement by primary method

3.6.7.1.1 Length of lever arm r

The length of lever arm r should be measured within:

$$f_{r,s} = \pm 0{,}05\ \%\ \text{to}\ 0{,}1\ \%.$$

3.6.7.1.2 Force F

The force F acting on the lever arm should be measured within:

$$f_{F,s} = \pm 0{,}05\ \%\ \text{to}\ 0{,}1\ \%.$$

3.6.7.2 Uncertainty in the shaft torque measurement by secondary method

The systematic uncertainty in the shaft torque strongly depends on the arrangement used. The expected uncertainty should be within:

$$f_{T,s} = \pm 0{,}15\ \%\ \text{to}\ 0{,}25\ \%.$$

3.6.7.3 Uncertainty in the measurement of the friction torque T_{Lm}

Where the bearing/seal arrangement is not included in the swinging frame system, the friction torque T_{Lm} should be measured within:

$$f_{T_{Lm},s} = \pm 0{,}02\ \%\ \text{to}\ 0{,}05\ \%,\ \text{of}\ T_{m,max}$$

3.6.7.4 Systematic uncertainty in the runner/impeller torque measurement

Using the above uncertainties (see 3.6.7.1 to 3.6.7.3) the relative systematic uncertainty in the runner/impeller torque can be calculated as shown below:

a) for the primary method with T_{Lm}, measured in the common swinging frame (see figures 46 and 47):

$$f_{Tm,s} = \sqrt{\left(f_{r,s}^2 + f_{F,s}^2\right)}$$

b) for the secondary method (see figure 52):

$$f_{Tm,s} = f_{T,s},\ \text{defined in 3.6.7.2}$$

for the primary method with T_{Lm}, not measured with swinging frame (see figures 49, 50 and 51):

The absolute systematic uncertainty is:

$$e_{Tm} = \sqrt{T^2 \cdot f_{T,s}^2 + T_{Lm}^2 \cdot f_{TLm,s}^2} = \sqrt{T^2\left(f_{r,s}^2 + f_{F,s}^2\right) + T_{Lm}^2 \cdot f_{TLm,s}^2}$$

then the relative systematic uncertainty is:

$$f_{Tm} = \frac{e_{Tm}}{T_m}$$

for the secondary method with T_{Lm} not measured with a swinging frame (see figure 54) the absolute systematic uncertainty is:

$$e_{Tm} = \sqrt{T^2 \cdot f_{T,s}^2 + T_{Lm}^2 \cdot f_{Lm,s}^2}$$

then the relative systematic uncertainty is:

$$f_{Tm} = \frac{e_{Tm}}{T_m}$$

' Rotational speed measurement

'.1 General

termination of the mechanical power of the runner/impeller requires knowledge of the ational speed of the runner/impeller shaft.

'.2 Methods of speed measurement

e rotational speed of the turbine/pump model may be measured by one of the following ethods:

counting of pulses generated by the model shaft, using an electronic counter and timebase. The pulse generator may be electrical or optical;

electrical frequency meter connected with a generator directly driven by the model shaft;

electrical high-precision tachometer comprising a stable permanent magnet directly driven by the model shaft.

7.3 Checking

sually, the speed measurement device is not truly calibrated, but checked

either by comparison with another speed measurement device;

or by checking separately the counting of pulses and the accuracy of the time base.

case of malfunction, possible errors are:

missing pulses;

change of time base.

3.7.4 Uncertainty of measurement

Using the aforementioned instrumentation, systematic uncertainty is expected to be within

$$f_{n,s} = \pm 0{,}01\ \%\ \text{to}\ 0{,}05\ \%.$$

3.8 Computation of test results

3.8.1 General

The main hydraulic performance guarantees verifiable by model test are (see 1.4.2): power, discharge and/or specific hydraulic energy, efficiency, steady-state runaway speed and/or discharge.

The model test results shall be transformed into quantities directly comparable with the data specified or guaranteed in the contract. The procedures to calculate these quantities are described below, and summarized as a flow chart in figure 62. The procedures shall be agreed between the parties prior to the beginning of the tests.

Subclause 3.8.2 deals with the computation of mechanical power of runner/impeller, discharge and/or specific hydraulic energy, and hydraulic efficiency in the guarantee range and includes the influence of cavitation (see 3.8.2.3.7 and 3.8.2.4.2).

Subclause 3.8.3 deals with the computation of steady-state runaway speed and discharge, and includes the influence of cavitation (see 3.8.3.2).

Subclauses 3.8.2.5 and 3.8.3.4 provide the formulae to be used.

Annex E provides an abstract of the test and calculation procedure.

For the hydraulic performance test of a machine, table 6 shows:

– the geometric parameters;
– the independent hydraulic variables;
– the dependent hydraulic variables.

Examples of performance diagrams for a Francis turbine, a Kaplan turbine, a radial pump, a double-regulated (axial) pump and a Pelton turbine are given respectively in figures 55 to 60. In the case of a single-regulated turbine, two examples of hill diagrams are given using either discharge and speed factors (see figure 55) or discharge and energy coefficients (see figure 56). An example of a four quadrant diagram for a single-regulated (radial) pump-turbine is shown in figure 61.

Hydraulic efficiencies measured on two geometrically similar reaction machines at any hydraulically similar operating points (see 2.3.1.2) of the guaranteed efficiency range are generally different because of the different values of test Reynolds number, which affects hydraulic efficiency (and consequently the mechanical power of runner/impeller), as explained in annex F.

Therefore, even for comparison with the guarantees given on the model, all values of hydraulic efficiency computed during the tests on a given model shall be referred to a constant Reynolds number, usually stated in the contract, by using the scale effect formula (see 1.4.1.4 and 3.8.2.2). The relevant symbols become η_{hM^*} and P_{ED^*} or P_{nD^*}.

here the model tests can be performed at the Reynolds number specified in the contract, no :ale effect formula is applied.

Table 6 – Variables defining the operating point of a machine

	MACHINE		
	Single-regulated	Double-regulated	Non-regulated
›ometrical parameter	α or β or s	α and β	–
łependent hydraulic variables	E_{nD}, Q_{nD}, σ_{nD} or n_{ED}, Q_{ED}, σ	E_{nD}, Q_{nD}, σ_{nD} or n_{ED}, Q_{ED}, σ	E_{nD} or Q_{nD}, σ_{nD} or n_{ED} or Q_{ED}, σ
›pendent hydraulic variables	η_h P_{nD} or P_{ED}	η_h P_{nD} or P_{ED}	η_h Q_{nD} or E_{nD}, P_{nD} or Q_{ED} or n_{ED}, P_{ED}

For runaway tests $\eta_h = 0$ and $P_{nD} = P_{ED} = 0$:
- for single-regulated machines, only one of the quantities E_{nD}, Q_{nD} (or n_{ED}, Q_{ED}) is an independent variable;
- for non-regulated machines, there is only one runaway point (neglecting the influence of cavitation).

ıe prototype usually has a well defined Reynolds number Re_P. The hydraulic efficiency easured on a reaction machine model shall be transformed into the prototype hydraulic ficiency taking into account the scale effect due to the Reynolds number.

ɔ scale effect due to the Reynolds number is assumed for n, Q, E and the relevant mensionless terms unless otherwise agreed (see 3.8.2.5.1): therefore $n_{EDM} = n_{EDP}$ and $_{EDM} = Q_{EDP}$ (or $E_{nDM} = E_{nDP}$ and $Q_{nDM} = Q_{nDP}$).

'ith multistage machines, if the model is tested with a reduced number of stages (see 1.3.4.2), the method of calculation of model data, taking into account the effect of reduced ages (labyrinth leakage and power loss) and the transformation of these to prototype ɔnditions shall be agreed.

ɔr impulse turbines (Pelton), no scale effect in efficiency is taken into account, unless :herwise specified in the contract (see 1.4.1.4a) and 3.8.2.2b)).

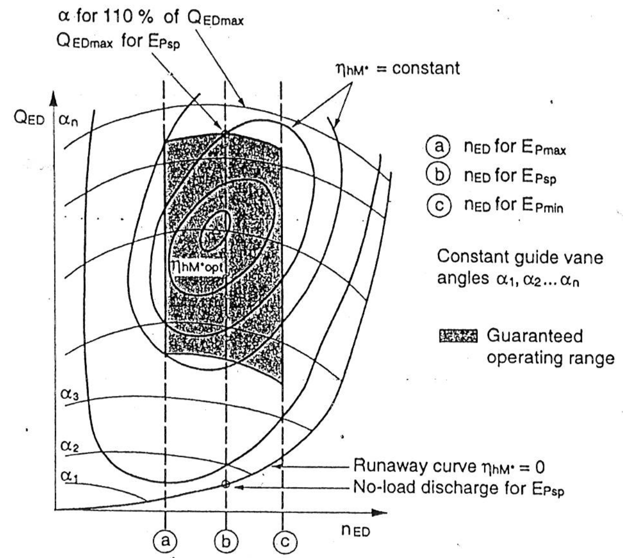


Figure 55 – Single-regulated (Francis) model turbine: performance hill diagram (discharge factor versus speed factor)

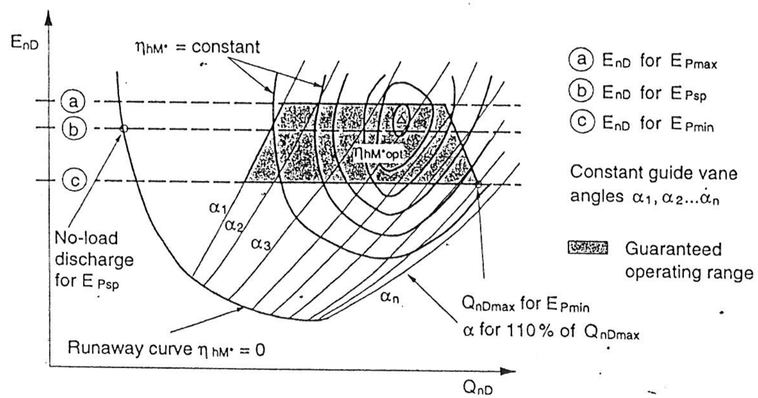


Figure 56 – Single-regulated (Francis) model turbine: performance hill diagram (energy coefficient versus discharge coefficient)

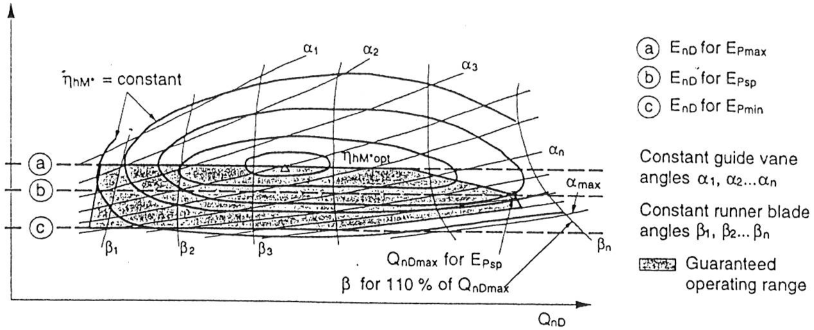


Figure 57 – Double-regulated (Kaplan) model turbine: performance hill diagram

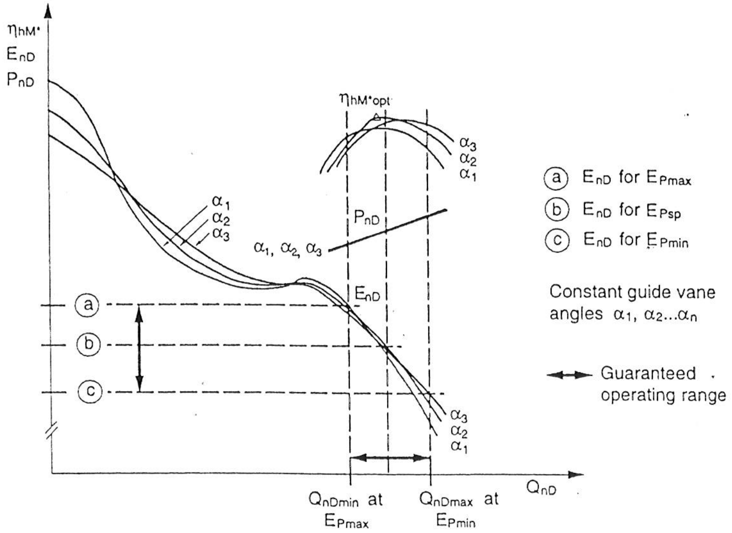


Figure 58 – Single-regulated (radial) model pump: performance diagram

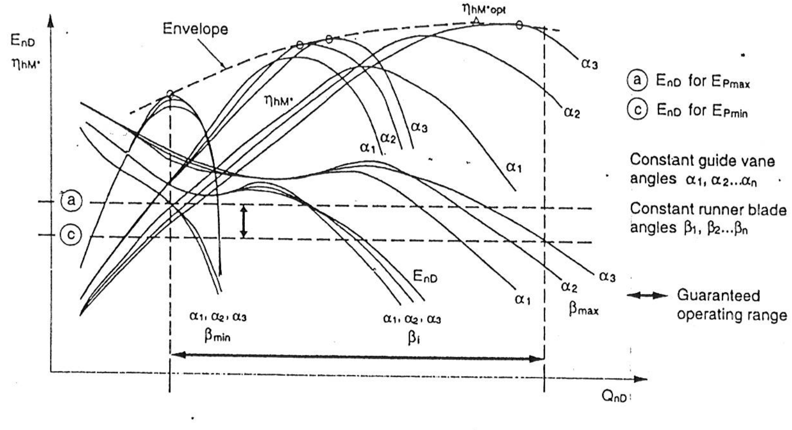


Figure 59 – Double-regulated model pump: performance diagram

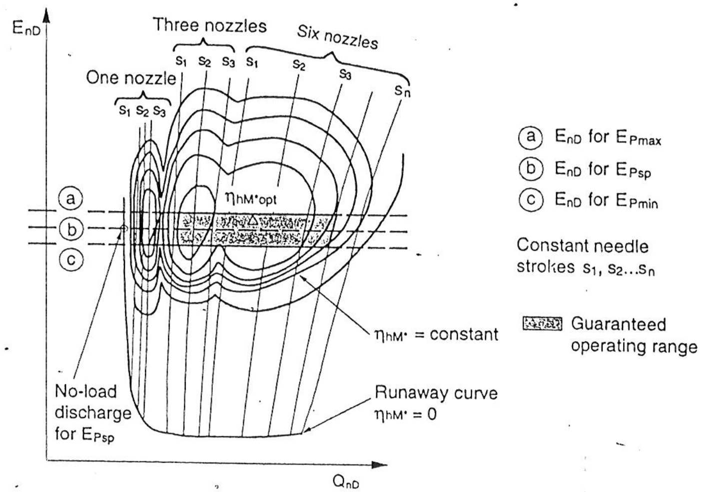


Figure 60 – Pelton model turbine: performance hill diagram (example for a six-nozzle machine)

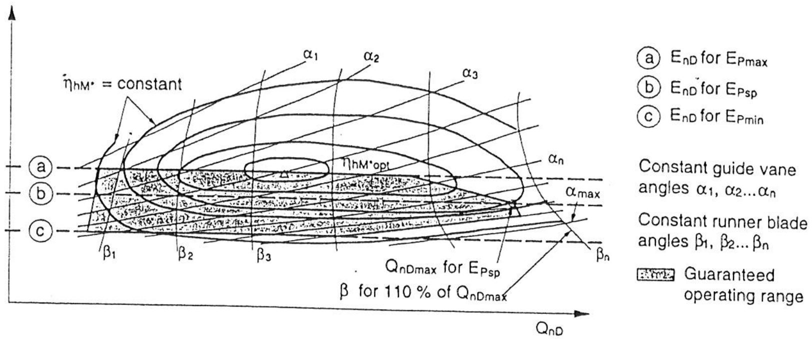


Figure 57 – Double-regulated (Kaplan) model turbine: performance hill diagram

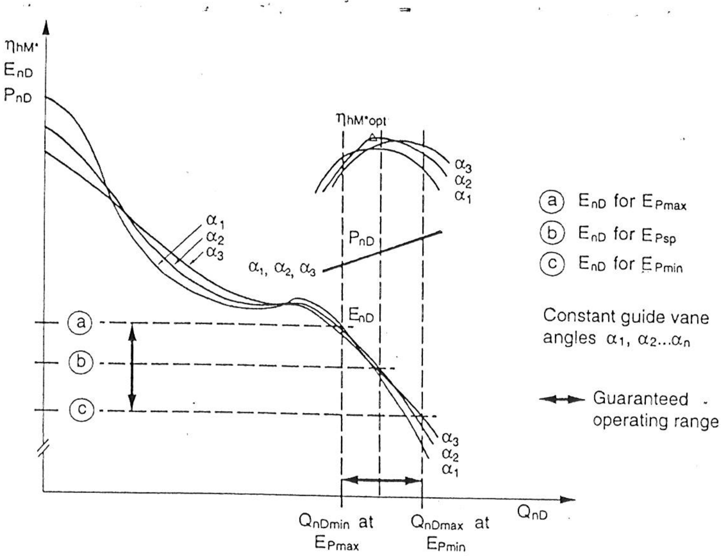


Figure 58 – Single-regulated (radial) model pump: performance diagram

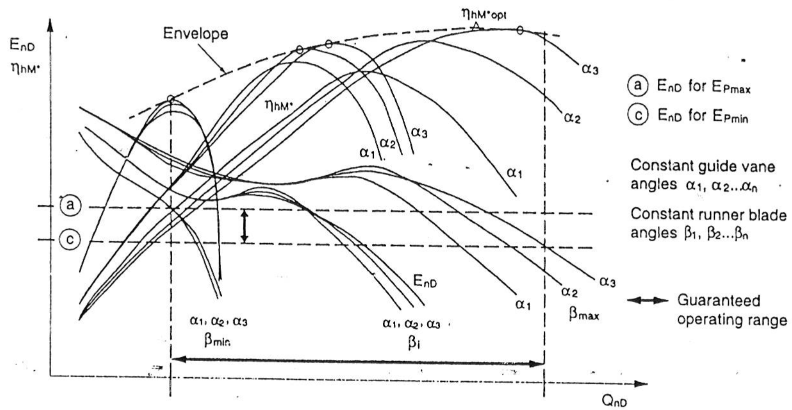


Figure 59 – Double-regulated model pump: performance diagram

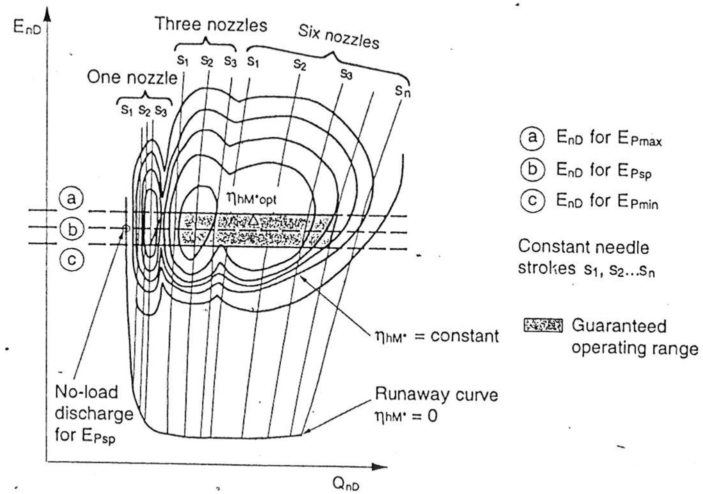


Figure 60 – Pelton model turbine: performance hill diagram (example for a six-nozzle machine)

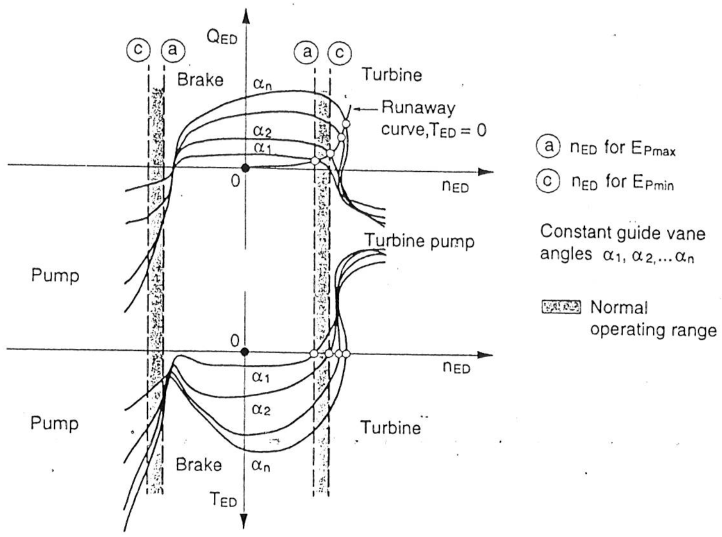


Figure 61 – Radial model pump-turbine: general four quadrant diagram

2 Computation of power, discharge and efficiency in the guarantee range

2.1 Computation of the model performance at a point

each point a set of one or more readings and/or recordings of the physical quantities used 'etermine the hydraulic performance of the model (see 2.4) is made.

› average values of E_M, Q_{1M}, n_M, P_{mM} and $NPSE_M$ are then computed (see 3.1); the nulae shown in 2.4.1.4 enable the hydraulic efficiency η_{hM} of the model to be calculated. › Reynolds number Re_M is computed by the formula in 1.3.3.11.1.

.2.2 Computation of model performance referred to a constant Re_{M^*}

Reaction machines

rmally, model tests are carried out at constant Reynolds number Re_{M^*}. If the guarantees are en on the model at a specified Reynolds number Re_{Msp}, it is good practice to choose Re_{M^*} = $_{Msp}$. If the model test cannot be carried out at constant Reynolds number, the hydraulic ciency calculated for each point with a different Reynolds number Re_M shall be scaled to $_{M^*}$ (see figures 62 and 63).

The following formula[1] is applied:

$$(\Delta\eta_h)_{M\to M^*} = \delta_{ref}\left[\left(\frac{Re_{ref}}{Re_M}\right)^{0,16} - \left(\frac{Re_{ref}}{Re_{M^*}}\right)^{0,16}\right]$$

where

$$\delta_{ref} = \frac{1-\eta_{hoptM}}{\left(\frac{Re_{ref}}{Re_{optM}}\right)^{0,16} + \frac{1-V_{ref}}{V_{ref}}} \quad \text{and } Re_{ref} = 7\cdot 10^6$$

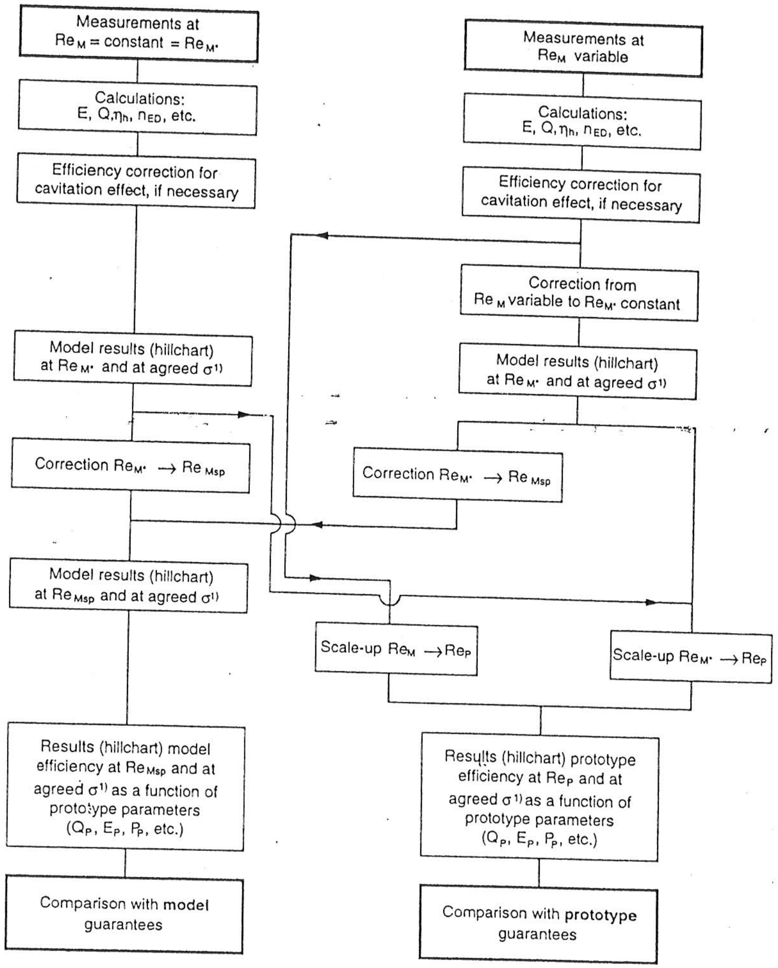
– Re_{optM} is the Reynolds number at which the optimum hydraulic efficiency η_{hopt} in each operating mode of the model is measured (see 3.8.2.2.1);

– the value of V_{ref}' is taken from 3.8.2.2.2.

b) Impulse turbines

Experience of different manufacturers has shown scale effects on impulse (Pelton) turbines to be primarily influenced by Froude, Reynolds and Weber numbers. A procedure for taking these effects into account is proposed in annex K and may be used by mutual agreement.

[1] This formula is a particular form of the general scale formula given in annex F.



1) Generally, the agreed value of σ is equal to σ_{pl}

Figure 62 – Reaction machines: procedure for calculating test results in view of comparison with guarantees

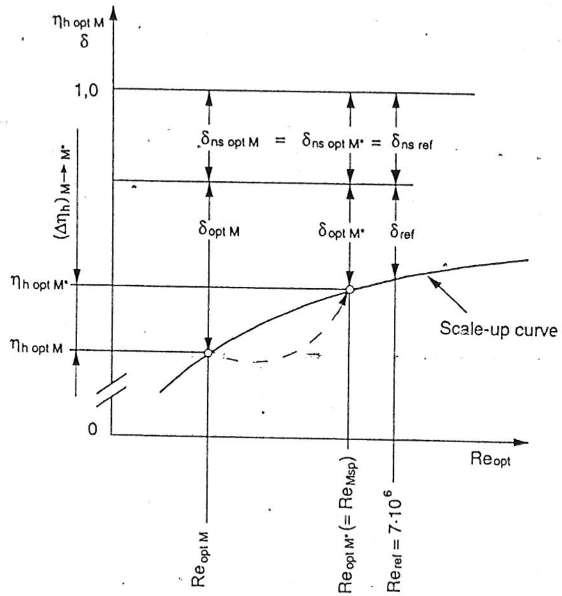


Figure 63 – Scale-up curve for best efficiency point

3.8.2.2.1 Determination of the efficiency scale-up

A series of tests performed according to 2.3.3.3.5 enables the optimum value of the hydraulic efficiency of the model η_{hoptM} and the corresponding Reynolds number Re_{optM} at non-cavitating conditions to be determined.

Using these values in the equations given in 3.8.2.2, δ_{ref} and $(\Delta\eta_h)_{M \rightarrow M^*}$ may then be calculated (see figure 63 and annex F).

In the case of a pump-turbine, this procedure shall be followed separately for both turbine and pump operation.

In the case of axial or diagonal machines having fixed runner blade angles and/or fixed guide vane angles (see table 7), η_{hoptM} is the optimum efficiency of the model tested with the same openings as the prototype.

Table 7 – V_{ref} values

Type of reaction machine [1]	V_{ref}
bines	
iial turbine (Francis)	0,7
il or diagonal turbine (Kaplan, tubular[2] and Deriaz) with adjustable ner blades and adjustable or fixed guide vanes	0,8
il or diagonal turbine with fixed runner blades (propeller turbine)	0,7
rage pumps	
iial storage pumps (single-stage or multi-stage)	0,6
il or diagonal storage pump	0,6
np-turbines	
iial pump-turbine (single-stage or multi-stage) operating as **turbine**	0,7
iial pump-turbine (single-stage or multi-stage) operating as **pump**	0,6
al or diagonal pump-turbine with adjustable runner blades operating as **turbine**	0,8
al or diagonal pump-turbine with adjustable runner blades operating as **pump**	0,6
al or diagonal pump-turbine with fixed runner blades operating as **turbine**	0,7
al or diagonal pump-turbine with fixed runner blades operating as **pump**	0,6

For hydraulic machines of special design (e.g. double flow machines, Francis turbines with splitter vanes, outer turbines) scaling-up formulae with other V_{ref} values and other values of the exponent of ratio Re_{ref}/Re_{M^*} in the iations given in 3.8.2.2 basing on individual loss considerations and experience may be applied after mutual eement.

Tubular turbines include: bulb turbines, pit turbines, rim-generator turbines, S-turbines.

3.2.2.2 Values of loss distribution coefficient V_{ref}

ie values of V_{ref} listed in table 7 are referred to $Re_{ref} = 7 \times 10^6$. They represent the ratio of iative scalable losses to relative total losses $(1 - \eta_{hopt})$ for the point of optimum hydraulic iciency at the reference Reynolds number $Re_{ref} = 7 \times 10^6$ for different types of reaction achines (see annex F).

8.2.2.3 Computation of the runner/impeller mechanical power factor P_{ED^*} (or coefficient P_{nD^*}) referred to a constant value Re_{M^*} for reaction machine

hen $Re_M \neq Re_{M^*}$ it is necessary to correct P_{ED} to P_{ED^*} or P_{nD} to P_{nD^*} as follows:

Turbines

$$P_{ED^*} = P_{ED}\frac{\eta_{hM^*}}{\eta_{hM}}$$

$$P_{nD^*} = P_{nD}\frac{\eta_{hM^*}}{\eta_{hM}}$$

Pumps

$$P_{ED^*} = P_{ED}\frac{\eta_{hM}}{\eta_{hM^*}}$$

$$P_{nD^*} = P_{nD^*}\frac{\eta_{hM}}{\eta_{hM^*}}$$

3.8.2.3 Presentation of model performance

The three basic types of hydraulic machines:

- single-regulated machine;
- double-regulated machine;
- non-regulated machine

are dealt with separately. Each type of machine is divided into turbine (or pump-turbine operating as turbine) and into pump (or pump-turbine operating as pump).

As the model efficiency guarantees for a specified value of Re_{Msp} are usually stated as a function of the prototype data E_P and Q_{1P} (or P_{mP}) for turbines or Q_P (or E_P) for pumps, the model performance data are converted to the prototype data using the relevant formulae (see 3.8.2.5). The influence of cavitation on the model performance and on the efficiency scale-up are dealt with in 3.8.2.3.7 and 3.8.2.4.2.

For all the following cases the first step is the determination of η_{hoptM}, δ_{ref} and $\Delta\eta_h$ (see 3.8.2.2).

3.8.2.3.1 Single-regulated turbine (figure 64)

The procedure described below applies to any type of impulse or reaction hydraulic machine.

The guaranteed efficiency is usually given for one specified speed and one or more specified specific hydraulic energies. As a result, it is necessary to obtain from the model test data a sufficient number of points or curves to cover the guarantees.

If the tests are carried out by choosing energy coefficients or speed factors nearly equal to the specified ones, it is possible to obtain a set of points or curves[1] $\eta_{hM^*}(Q_{nD})$ or $\eta_{hM^*}(Q_{ED})$ to be used for comparison with guarantees. Because testing at exactly constant speed factor is not possible, the following procedure is recommended:

- measurement of a number of points sufficient to draw the three-dimensional surface (hill diagram) of: $\eta_{hM^*}(E_{nD}, Q_{nD})$ or $\eta_{hM^*}(n_{ED}, Q_{ED})$;
- sectionalize the three-dimensional surface representing the hydraulic efficiency at the energy coefficients or speed factors specified. Figure 64 shows the three-dimensional representative surface η_{hM^*} drawn versus energy and discharge coefficients and its section at the specified values of $E_{nD} = E_{nDsp}$.

The determination of the three-dimensional surface (hill diagram) is necessary if the guarantees are based on annual energy production.

For each E_{nDsp} (or n_{EDsp}) the η_{hM^*} values obtained following one of the above procedures, enable the mechanical runner power coefficient P_{nD^*} (or factor P_{ED^*}) curve to be calculated, for comparison with model guarantees.

[1] For the determination of the best smooth curves, see for instance annex H.

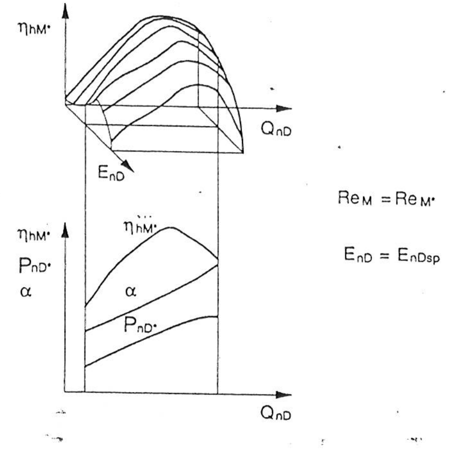


Figure 64 – Single-regulated turbine. Three-dimensional surface of hydraulic efficiency and curves of performance at E_{nD} constant

3.2.3.2 Single-regulated pump (figure 65)

e guaranteed efficiency and discharge are usually given for one specified speed and a ecified range of specific hydraulic energy. As a result, it is necessary to obtain from the odel test data a sufficient number of points or curves to cover the guarantees.

r different guide vane openings the $\eta_{hM^*}(Q_{nD})$ and $E_{nD}(Q_{nD})$ or $\eta_{hM^*}(E_{nD})$ and $Q_{nD}(E_{nD})$ rves[1] are drawn and the relevant mechanical power coefficients P_{nD^*} are calculated (see ure 65). The guide vane openings will be chosen taking into account the guaranteed discharge d efficiency and the power limits.

3.2.3.3 Double-regulated turbine (figure 66)

e guaranteed efficiency is usually given for one specified speed and one or more specified ecific hydraulic energy. As a result, it is necessary to obtain from the model test data a fficient number of points or curves to cover the guarantees.

the tests are carried out by choosing energy coefficients or speed factors nearly equal to the ecified ones, it is possible to obtain a set of points or curves [1] $\eta_{hM^*}(Q_{nD})$ or $\eta_{hM^*}(Q_{ED})$ to mpare with guarantees: the tests are usually performed considering the double-regulated rbine as a set of single-regulated turbines having different constant runner blade angles. gure 66 shows the performance curve drawn for a Kaplan model turbine, measuring a umber of points at six different runner blade angles β_1, β_2, etc. and keeping E_{nD} constant qual to E_{nDsp}): the on cam hydraulic efficiency η_{hM^*} curve is the envelope curve, determining e optimum guide vane/runner blade relationship [2].

For the determination of the best smooth curves, see for instance annex H.

The optimum relationships (α, β) on model and on prototype are only approximately the same (see 3.4).

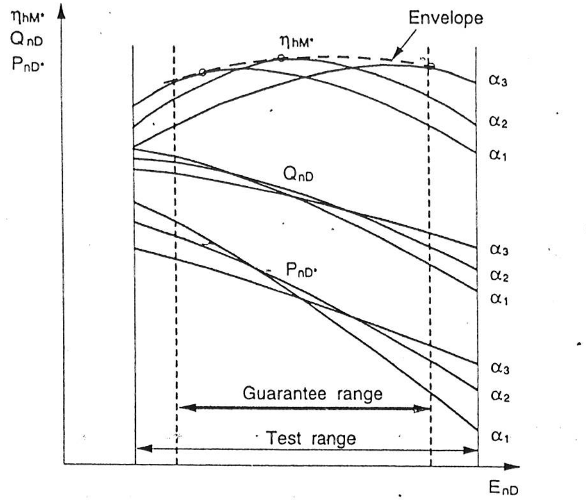


Figure 65 – Single-regulated pump. Performance curves

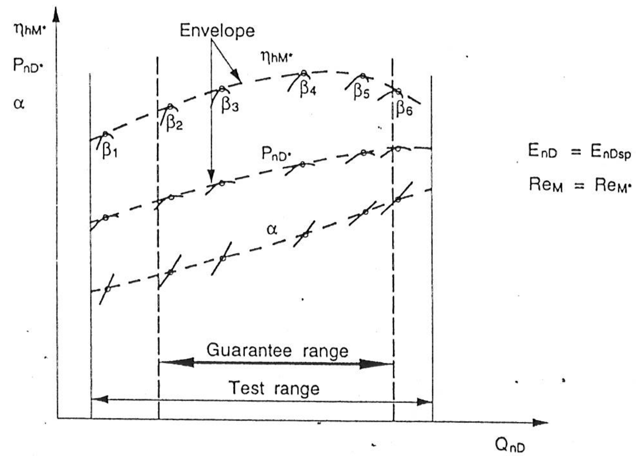


Figure 66 – Double-regulated turbine. Performance curves at E_{nD} constant

Since it is impossible to carry out testing at exactly constant energy coefficient or speed factor, the following procedure is recommended:

- measure a sufficient number of points to draw a three-dimensional surface (hill diagram) of: $\eta_{hM^*}(E_{nD}, Q_{nD})$ or $\eta_{hM^*}(n_{ED}, Q_{ED})$, for the chosen runner blade angles;
- sectionalize the hill diagram representing the hydraulic efficiency at the energy coefficients or speed factors specified.

ı this way the on-cam η_{hM^*} values are determined as in figure 66.

he determination of the hill diagram is necessary if the guarantees are based on annual nergy production.

or each E_{nDsp} (or n_{EDsp}) the η_{hM^*} values obtained, following one of the above procedures, nable the mechanical runner power coefficient P_{nD^*} (or factor P_{ED^*}) curve to be calculated, for omparison with model guarantees.

.8.2.3.4 Double-regulated pump (figure 67)

he guaranteed efficiency and discharge are usually given for one specified speed and a pecified range of specific hydraulic energy. As a result, it is necessary to obtain from the ıodel test data a sufficient number of points or curves to cover the guarantees.

he procedure is the same as that for double-regulated turbines (see 3.8.2.3.3). Figure 67 hows the performance curves drawn for a double-regulated model pump keeping E_{nD} constant $= E_{nDsp}$).

'or each E_{nDsp} (or n_{EDsp}) the η_{hM^*} values obtained following one of the above procedures, nable the mechanical runner power coefficient P_{nD^*} (or factor P_{ED^*}) curve to be calculated, for omparison with model guarantees.

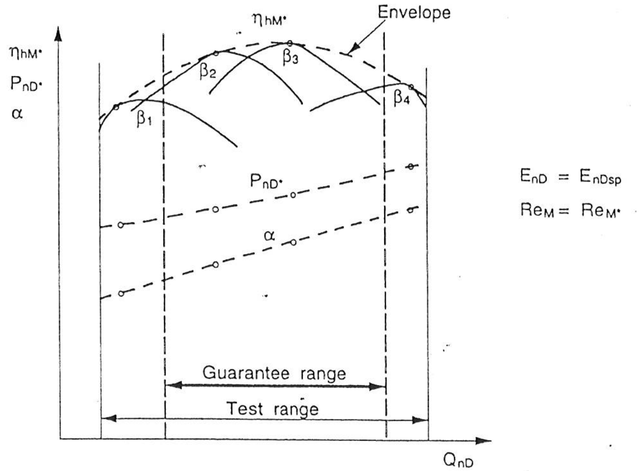


Figure 67 – Double-regulated pump. Performance curves at E_{nD} constant

3.8.2.3.5 Non-regulated turbine (figure 68)

The guaranteed efficiency is usually given for one specified speed and a specified range of specific hydraulic energy. There is only one independent variable: E_{nD} (or Q_{nD}) or n_{ED} (or Q_{ED}) as shown in table 6.

The performance curves1) including the mechanical power coefficient P_{nD^*} or factor P_{ED^*}, computed through the hydraulic efficiency η_{hM^*}, are shown in figure 68. They are directly used for the comparison with the model guarantees.

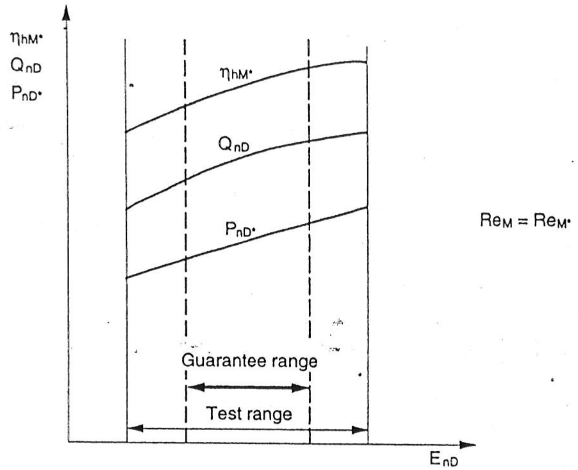


Figure 68 – Non-regulated turbine. Performance curves

3.8.2.3.6 Non-regulated pump (figure 69)

The guaranteed efficiency and discharge are usually given for one specified speed and a specified range of specific hydraulic energy. There is only one independent variable: E_{nD} (or Q_{nD}).

The performance curves1) including the mechanical runner power coefficient P_{nD^*}, computed through the hydraulic efficiency η_{hM^*}, are shown in figure 69. They are directly used for the comparison with the model guarantees.

1) For the determination of the best smooth curves, see annex H.

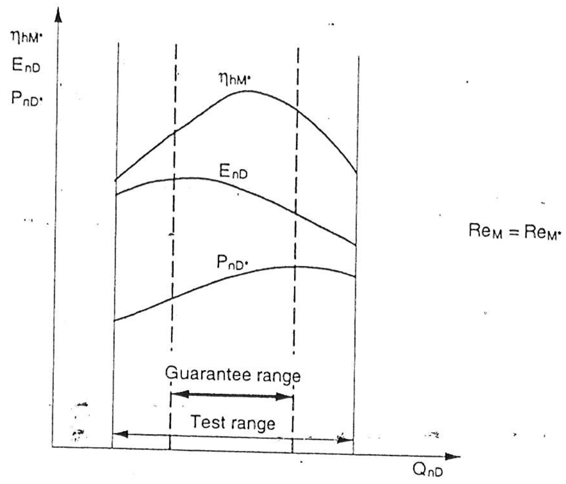


Figure 69 – Non-regulated pump. Performance curves

3.2.3.7 Influence of cavitation on model power, discharge and/or specific hydraulic energy and efficiency

is recommended to verify the influence of cavitation, characterized by the Thoma number (or ə cavitation coefficient), on model performance[1]). In 2.3.3.3.5 and 2.3.3.3.6 the test ocedure is explained.

these tests reveal that within the range of guarantees an influence exists, figure 70 explains ə procedure for correcting the efficiency curve, measured at $\sigma_M > \sigma_{pl}$, at a sufficient number agreed operating points. The cavitation influences on discharge and efficiency found at σ_{pl} e superimposed on the performance curves, measured at $\sigma_M > \sigma_{pl}$. The values of the draulic performance quantities, to be taken into account for the comparison with the ıarantees, shall be those that consider the influence of cavitation under plant conditions, if ıy.

gure 71 shows the curves of P_{ED}, Q_{ED}, and η_{hM} drawn at one measured point by varying the ıoma number σ for a Francis model turbine or pump-turbine operating as a turbine; the ırves of figure 72 refer to a model pump, or pump-turbine operating as a pump. In this last ıse, the curve of Q_{ED} has been replaced by the curve of E_{nD} because it is more relevant to ımp performance. In the case of a pump, σ_{nD} replaces σ as E is variable during cavitation sts[2]).

The NPSE and σ-values are determined in the low pressure reference section 2 of the model (see 1.3.3.6.5 and 3.6.6). Since only the free water levels in the suction channel of the plant are usually known, it is necessary, when lculating σ-factors at guaranteed performance points, to take into account the specific hydraulic energy ssipated between the tailwater level and section 2'. In the case of a pump having a suction channel where the free ıter level can be measured close to section 2, the pump inlet losses can be considered negligible and the specific draulic energy at section 2 is assumed to correspond to its submergence. In other cases, an agreement shall to reached between the parties.

For pumps, it may also be useful to keep E_{nD} nearly constant, as shown in figure 71.

The Thoma number measured on the model is transferred to the prototype $NPSE_P$ using the formulae given in 3.8.2.5.3.

Where guarantees are given on the prototype, the cavitation influence on the scale-up formula is given in 3.8.2.4.2.

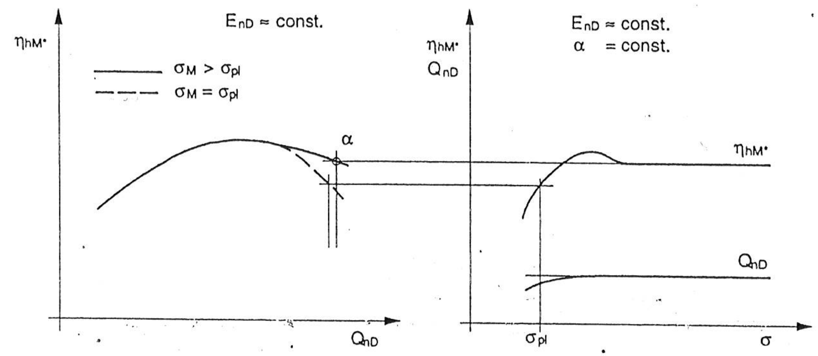


Figure 70 – Efficiency curve correction in order to take into account cavitation influence (e.g. tubular machines at overload operation)

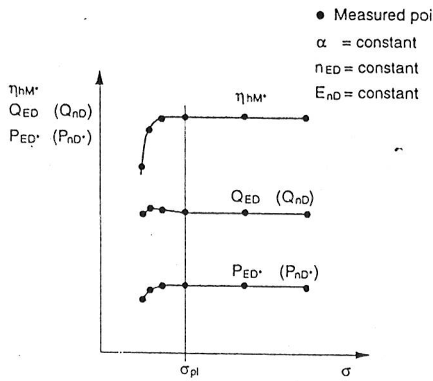


Figure 71 – Francis model turbine. Cavitation curves

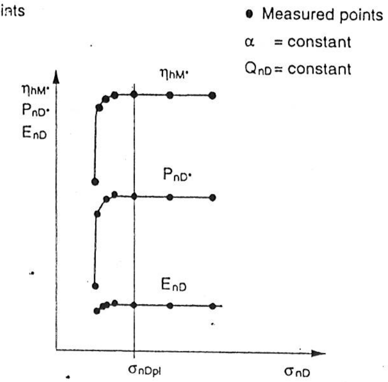


Figure 72 – Model pump. Cavitation curves

8.2.4 Computation of prototype performance

8.2.4.1 Efficiency scale-up

ɔr a reaction machine, if the guarantees are referred to prototype, the model efficiencies η_{hM} easu:ed at different Reynolds numbers Re_M are scaled up to the prototype Reynolds number e_P using the following formula:

$$(\Delta\eta_h)_{M\rightarrow P} = \delta_{ref}\left[\left(\frac{Re_{ref}}{Re_M}\right)^{0,16} - \left(\frac{Re_{ref}}{Re_P}\right)^{0,16}\right]$$

he value of Re_{ref} and the formula to calculate δ_{ref} are given in 3.8.2.2 a).

the model hydraulic efficiency has been measured at constant Reynolds number Re_{M*}, or has ɜen scaled to a constant Reynolds number Re_{Msp}, $\Delta\eta_h$ is a constant value within the range of uaranteed efficiencies if, as usually happens, n_P and consequently Re_P are constant (see ɡure F.3).

the model hydraulic efficiency has been measured at different Reynolds numbers, $\Delta\eta_h$ shall e calculated for each measured point, taking into account the relevant Re_M (see figure F.4).

or an impulse turbine, if it has been contractually agreed to take into account a scale effect on fficiency, $(\Delta\eta_h)_{M\rightarrow P}$ may be calculated according to annex K.

ther data concerning the main hydraulic performance of the prototype (discharge, specific ydrauiic energy and mechanical runner/impeller power) are obtained from the formulae of .8.2.5. The prototype mechanical runner/impeller power is computed taking into account the cale effect on the hydraulic efficiency.

he procedure of drawing[1] the curves of prototype performance, and of determining the curves ɔ be compared with the prototype guarantees is the same as the one established for uarantees given on the model (see 3.8.2.3).

.8.2.4.2 Influence of cavitation on applicability of scale up formula

Vhile no scientifically founded theory for scale-up under cavitating conditions exists, it is enerally agreed that the scale-up calculated for non-cavitating conditions can be applied at :-values where the performance is not influenced by cavitation.

ły convention, this scale-up may continue to be applied if the increase or decrease of hydraulic :fficiency, due to the decrease of Thoma number σ, does not exceed 0,5 %, unless otherwise ıgreed (see figure 73).

Vhere the efficiency is affected by more than 0,5 %, the relationship between model and ɔrototype performance is less certain and an *a priori* agreement regarding this relationship ;hall be reached between the parties.

ſhe value of P_{mP} is determined on the basis of the values of η_{hP}, Q_P, E_p at σ_{pl}.

) For the determination of the best smooth curves, see annex H.

In large tubular turbines, if Froude's similarity cannot be fulfilled (see 2.3.1.5.1), a scale-up method of the cavitation characteristics from model to prototype should take into account the vertical distribution of cavitation, as for example shown in [17].

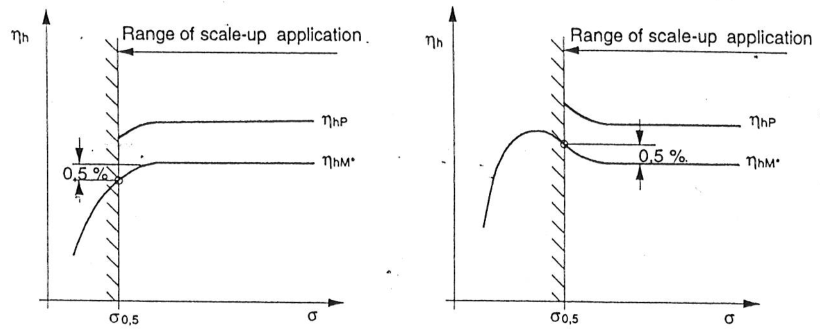


Figure 73 – Francis model turbine: cavitation curves. Examples of limits for application of scale-up formula

3.8.2.5 Formulae for computation of prototype performance within the guaranteed efficiency range

The model test data provide for each point, the discharge Q_{1P}, the specific hydraulic energy E_P and the mechanical runner/impeller power P_{mP} of the geometrically similar prototype operating in hydraulically similar conditions (see 2.3.1.2), by using the formulae listed below.

3.8.2.5.1 Reaction machines

The scale effect due to a Reynolds number influence on hydraulic efficiency is taken into account on mechanical runner/impeller power[1]. Since available data for scaling-up discharge and specific hydraulic energy do not show consistent trends, it has been assumed that only efficiency and power (due to efficiency increase) are influenced by scale effect[2].

Two procedures are possible:

a) Direct calculation from model measured data

$$\eta_{hP} = \eta_{hM} + (\Delta\eta_h)_{M\rightarrow P}$$

where $(\Delta\eta_h)_{M\rightarrow P}$ is calculated by the formula given in 3.8.2.4.1.

[1] In the case of axial turbines operating under low specific hydraulic energy far from the best efficiency point, some measurements show a power different from that calculated according to this standard.

[2] Sometimes, tests on prototypes show shifting effects on $Q_{1P} = f(E_P)$ curves and consequently on $P_{mP} = f(E_P)$ curves compared to corresponding model curves. Shifting effects on $Q_{1P} = f(E_P)$ have to be taken into account in determining the maximum mechanical power of a pump. A possible approach is given in the JMSE S008 Standard [18], which applies the formula:

$$P_{mP} = P_{mM}\frac{\rho_{1P}}{\rho_{1M}}\left(\frac{n_P}{n_M}\right)^3\left(\frac{D_P}{D_M}\right)^5$$

$$Q_{1P} = Q_{1M}\left(\frac{D_P}{D_M}\right)^2\left(\frac{E_P}{E_M}\right)^{0,5} = Q_{1M}\left(\frac{D_P}{D_M}\right)^3\frac{n_P}{n_M}$$

$$E_P = E_M\left(\frac{D_P}{D_M}\right)^2\left(\frac{n_P}{n_M}\right)^2$$

or turbine:

$$P_{mP} = \rho_{1P}Q_{1P}E_P\eta_{hP} = P_{mM}\frac{\rho_{1P}}{\rho_{1M}}\left(\frac{D_P}{D_M}\right)^2\left(\frac{E_P}{E_M}\right)^{1,5}\frac{\eta_{hP}}{\eta_{hM}} = P_{mM}\frac{\rho_{1P}}{\rho_{1M}}\left(\frac{D_P}{D_M}\right)^5\left(\frac{n_P}{n_M}\right)^3\frac{\eta_{hP}}{\eta_{hM}}$$

eing $P_{mM} = \rho_{1M}\,Q_{1M}\,E_M\,\eta_{hM}$

or pump:

$$P_{mP} = \frac{\rho_{1P}Q_{1P}E_P}{\eta_{hP}} = P_{mM}\frac{\rho_{1P}}{\rho_{1M}}\left(\frac{D_P}{D_M}\right)^2\left(\frac{E_P}{E_M}\right)^{1,5}\frac{\eta_{hM}}{\eta_{hP}} = P_{mM}\frac{\rho_{1P}}{\rho_{1M}}\left(\frac{D_P}{D_M}\right)^5\left(\frac{n_P}{n_M}\right)^3\frac{\eta_{hM}}{\eta_{hP}}$$

eing $P_{mM} = \frac{\rho_{1M}Q_{1M}E_M}{\eta_{hM}}$

) Calculation from the dimensionless factors (or coefficients) previously computed from model measured data

$$\eta_{hP} = \eta_{hM} + (\Delta\eta_h)_{M\to P}$$

$$Q_{1P} = Q_{ED}\,D_P^2\,E_P^{0,5} = Q_{nD}\,D_P^3\,n_P$$

$$E_P = \frac{1}{n_{ED}^2}D_P^2\,n_P^2 = E_{nD}\,D_P^2\,n_P^2$$

or turbine: $P_{mP} = P_{ED}\,\rho_{1P}\,D_P^2\,E_P^{1,5}\frac{\eta_{hP}}{\eta_{hM}} = P_{nD}\,\rho_{1P}\,D_P^5\,n_P^3\frac{\eta_{hP}}{\eta_{hM}}$

or pump: $P_{mP} = P_{ED}\,\rho_{1P}\,D_P^2\,E_P^{1,5}\frac{\eta_{hM}}{\eta_{hP}} = P_{nD}\,\rho_{1P}\,D_P^5\,n_P^3\frac{\eta_{hM}}{\eta_{hP}}$

he above formulae can also be applied when the measured hydraulic efficiency of the model ias been referred to a constant Reynolds number (η_{hM^*}) (see 3.8.1).

3.8.2.5.2 Impulse turbines (Pelton)

The formulae in 3.8.2.5.1 apply under the following conditions:

- if no scale effect is taken into account, it is assumed that $\eta_{hP} = \eta_{hM}$;
- if it is contractually agreed to take a scale effect into account, then $(\Delta\eta_h)_{M\to P}$ may be calculated according to annex K.

3.8.2.5.3 Formulae for computation of prototype $NPSE_P$

The net positive suction specific energy of the prototype is calculated by one of the following formulae:

$$NPSE_P = \sigma \cdot E_P = \sigma_{nD} \cdot n_P^2\,D_P^2$$

3.8.3 Computation of steady-state runaway speed and discharge

3.8.3.1 Determination of the model steady-state runaway curves

The Reynolds number scale effect is assumed zero in the range near to the runaway operation. The effect of the Thoma number on the runaway curves may be significant (see 3.8.3.2).

In the case of a single-regulated machine model, for each point a set of readings and/or recordings of physical quantities used to determine the model steady-state runaway speed and discharge is recorded, with $T_{mM} = 0$ (see 2.3.3.3.7).

The average values of E_M, Q_{1M}, n_M and $NPSE_M$ are then computed; $n_{ED,R}$ and $Q_{ED,R}$ (or $E_{nD,R}$ and $Q_{nD,R}$) are finally derived using the formulae of 1.3.3.12.

The runaway curve[1]) is drawn for different openings α or s in figure 74 to obtain the maximum steady state runaway speed $n_{ED,Rmax}$ and discharge $Q_{ED,Rmax}$.

For a double-regulated machine model, a runaway curve is usually drawn for each runner/impeller blade angle β. The envelope curve of these curves is drawn to define the maximum runaway speed and discharge (see figures 78 and 79).

For a non-regulated machine model, the runaway curve is reduced to a point when using dimensionless factors or coefficients.

The runaway tests shall be carried out by varying the guide vane opening, runner/impeller blade angle or needle stroke, over all the guaranteed range.

For Pelton turbines, the maximum runaway speed is determined taking into account the influence of the number of nozzles (see figure 75).

If, during the tests, it is not possible to reach $T_{mM} = 0$, the runaway conditions can be determined by extrapolation (see figure 76).

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[1]) For a pump, the runaway speed and discharge are usually called reverse runaway speed and reverse runaway discharge.

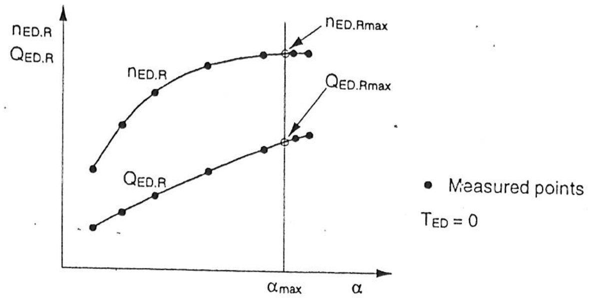


Figure 74 – Runaway curves for a single-regulated turbine (Francis)

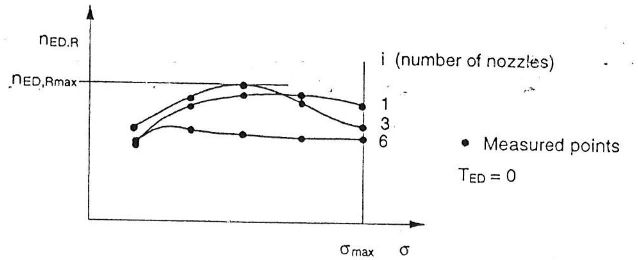


Figure 75 – Runaway curves for a single-regulated turbine (six-nozzle Pelton)

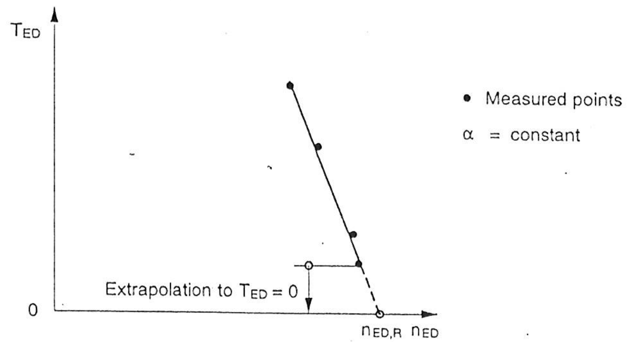


Figure 76 – Runaway speed determined by extrapolation. Example for a single-regulated turbine (Francis)

3.8.3.2 Influence of cavitation on steady-state runaway speed and discharge

It is recommended that the influence of Thoma number on model performance is also verified at runaway conditions. Subclause 2.3.3.3.7 details the test procedure.

Figure 77 shows the cavitation influence for a medium specific speed Francis turbine model at guide vane opening α_{max}.

Cavitation has a large influence on runaway curves of a Kaplan turbine model. Figure 78 shows the cavitation curves $n_{ED,R}$ and $Q_{ED,R}$ at different guide vane openings α and runner blade angles β. The same phenomenon is represented in figure 79, which shows the $n_{ED,R}(Q_{ED,R})$ curves at high σ and at $\sigma = \sigma_{pl}$.

The Thoma number or the cavitation coefficient measured on the model is usually transferred to prototype $NPSE_P$ using the formulae given in 3.8.2.5.3.

In large tubular turbines, if the Froude similitude cannot be fulfilled (see 2.3.1.5.1), a conversion method of cavitation characteristics from model to prototype should take into account the vertical distribution of cavitation as, for example, shown in reference [19].

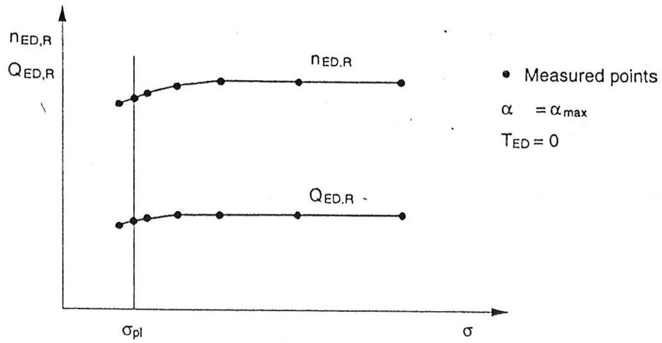


Figure 77 – Influence of Thoma number on runaway speed and discharge of a single-regulated turbine (Francis)

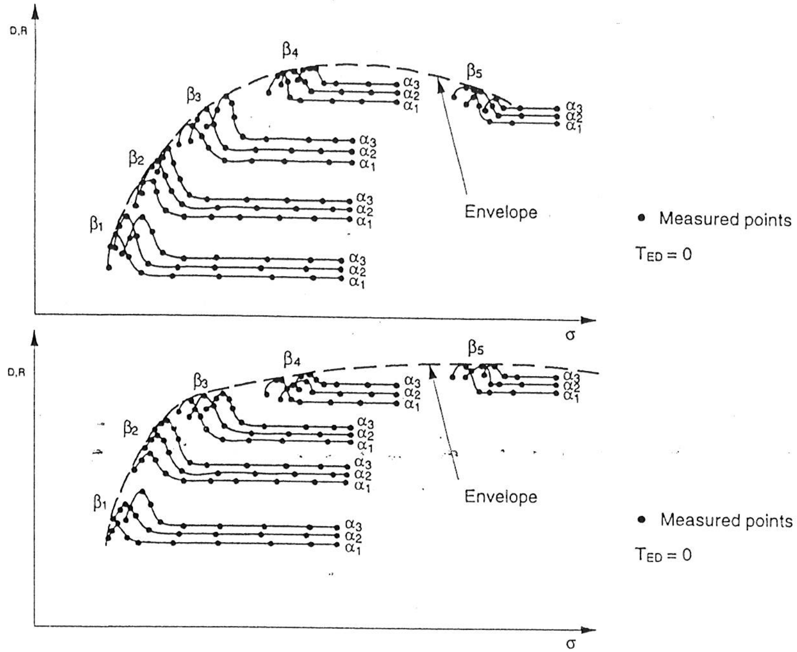


Figure 78 – Influence of the Thoma number on runaway speed and discharge of a double-regulated turbine (Kaplan)

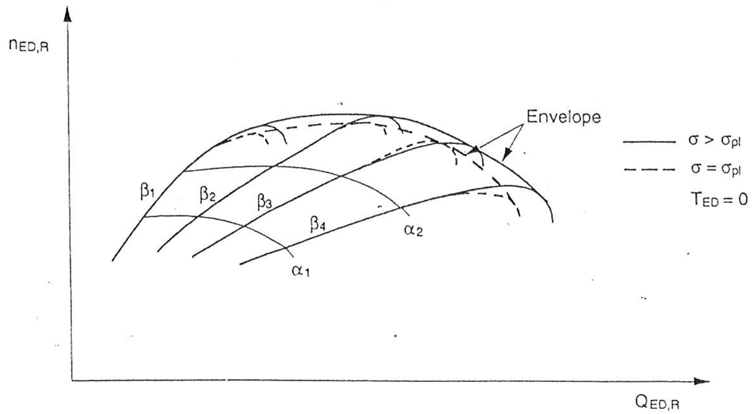


Figure 79 – Influence of the Thoma number on the off cam runaway curves of a double-regulated turbine (Kaplan)

3.8.3.3 Computation of the prototype steady-state runaway curves

If runaway guarantees are given for the prototype, no scale effect is usually considered: this means that $P_{ED,M} = P_{ED,P}$, in the range near to the runaway point. The prototype runaway data are computed at σ_{pl} from the model test results using formulae based on the affinity laws (see 3.8.3.4) and hence the maximum runaway speed and discharge points are determined. If required, the friction losses of the unit thrust bearing, guide bearings and shaft seal and the mechanical and windage losses of the electrical machine are taken into account, unless otherwise agreed, as described in annex G.

3.8.3.4 Formulae for computation of prototype runaway characteristic

Two procedures are possible:

a) Direct calculation from model measured data

$$n_{R,P} = n_{R,M}\frac{D_M}{D_P}\left(\frac{E_P}{E_M}\right)^{0,5} \qquad Q_{1,RP} = Q_{1,RM}\left(\frac{D_P}{D_M}\right)^{2}\left(\frac{E_P}{E_M}\right)^{0,5}$$

The formula

$$P_{mP} = P_{mM}\frac{\rho_{1P}}{\rho_{1M}}\left(\frac{D_P}{D_M}\right)^{2}\left(\frac{E_P}{E_M}\right)^{1,5}$$

is used to draw a curve $P_{mP}(n_P)$ necessary to take into account the bearings and shaft seal friction losses and the windage losses (see annex G).

b) Calculation from the dimensionless factors (or coefficients) previously computed from model measured data

$$n_{R,P} = n_{ED,R}\frac{E_P^{0,5}}{D_P} \qquad Q_{1,RP} = Q_{ED,R}\,D_P^2\,E_P^{0,5}$$

The formula

$$P_{mP} = P_{ED}\cdot\rho_{1P}\cdot D_P^2\cdot E_P^{1,5}$$

is used to draw a curve $P_{mP}(n_P)$ necessary to take into account the bearings and shaft seal friction losses and the windage losses (see annex G).

For the computation of $NPSE_P$, see 3.8.2.5.3.

3.9 Error analysis

3.9.1 Basic principles (see ISO 5168)

Starting from the measurements made on the model the different sources of error shall be examined to determine the corresponding uncertainties.

3.9.1.1 Definition of the error

The error in the measurement of a quantity is the difference between that measurement and the true value of the quantity.

ı measurement of a physical quantity is free from uncertainties arising from systematic and ıdom errors.

stematic errors cannot be reduced by repeating measurements since they arise from the aracteristics of the measuring apparatus, the installation and the operating conditions. ıwever, random errors can be reduced by repetition of measurements since the random error the mean of n independent measurements is $\sqrt{n}$ times smaller than the random error of an lividual measurement (see annex L).

).1.2 Definition of uncertainty

e range within which the true value of a measured quantity can be expected to lie, with a itably high probability, is termed the uncertainty in the measurement. For the purpose of this ındard, the probability to be used shall be 95 % confidence level.

e uncertainty in the measurement of a quantity X may be expressed as an absolute value e_X as a relative value: $f_X = e_X / X$.

).1.3 Types of errors

ıree types of error shall be considered:

- spurious errors (see 3.9.1.3.1);
- random errors (see 3.9.1.3.2);
- systematic errors (see 3.9.1.3.3).

).1.3.1 Spurious errors

ıese are errors such as human errors, or instrument malfunction, which invalidate a ɜasurement. For example, the transposing of numbers in recording data or the presence of ckets of air in leads from a water line to a manometer.Such errors should not be corporated into any statistical analysis and the corresponding measurement shall be scarded. Where the error is not large enough to make the result obviously invalid, the point all be repeated or some rejection criteria may be applied to decide whether the data point ould be rejected (see for example [20]).

9.1.3.2 Random errors and associated uncertainty

ındom errors are caused by numerous, small, independent influences which prevent a easurement system from delivering the same reading (repeatability of the measurement stem) when supplied with the same input value of the quantity to be measured. The easurements deviate from their mean in accordance with the laws of chance, such that their stribution usually approaches a normal (gaussian) distribution as the number of measure-ents is increased.

ıe random error is influenced by the care taken during the measurements, the number of easurements and the operation conditions. The scatter of the readings observed during a test sults from the combination of the random error arising from the instrumentation and of the fluence of the operating conditions. The repetition of points at a given operating condition ıables the value of the uncertainty associated with random errors to be established by atistical methods (see 3.9.2.2.1 and annex L).

'hen the sample size (i.e. the number of measurements) is small, it is necessary to correct e statistical results that are based on the assumption of a normal distribution, by means of e Student's t value, as explained in annex L. Student's t is a factor which compensates for ıe uncertainty in the standard deviation increasing, for a given confidence level, as the sample ze is reduced.

3.9.1.3.3 Systematic errors and associated uncertainty

A systematic error is one that invariably has the same magnitude and the same sign under the same conditions of measurement. Therefore it cannot be reduced by increasing the number of measurements if the equipment and conditions of measurements remain unchanged.

Systematic errors do not affect the repeatability of measurements during a test.

The uncertainty associated with systematic errors cannot be assessed experimentally without changing the equipment or conditions of measurements. The only way to verify the main measuring system and to obtain an order of magnitude of the systematic error is to measure each basic quantity by two different systems, if they are available.

The alternative is to make a subjective judgement on the basis of experience and consideration of the equipment involved.

If the error has a unique known value then this should be added to (or subtracted from) the result of the measurement and there is no longer systematic uncertainty in the measurement due to this source.

If the systematic error of a measuring device is unknown but its error limits (class of accuracy) are specified, the interval between them may be assumed as the systematic uncertainty of that device with a confidence level better than 95 %.

Notwithstanding the difference shown above between systematic and random uncertainties, the probability distribution of the uncertainty values of each systematic component is essentially gaussian and the accepted convention calculating the total systematic uncertainty f_s from the individual systematic uncertainties is the root-sum-square method.

3.9.1.4 Total uncertainty

The total uncertainty in a measurement (f_t) is obtained by combining the systematic (f_s) and random (f_r) uncertainties (see 3.9.1.3.3 and 3.9.1.3.2). It defines a range within which the true value is assumed to lie with a probability of 95 % and any point in this range is equally valid.

Given the same type of probability distribution of the systematic and random uncertainties, they can be combined by the root-sum-squares method. Subclause 3.9.2.2.4 explains how to determine the total uncertainty in model tests.

3.9.2 Determination of uncertainties in model tests

3.9.2.1 Sources of errors

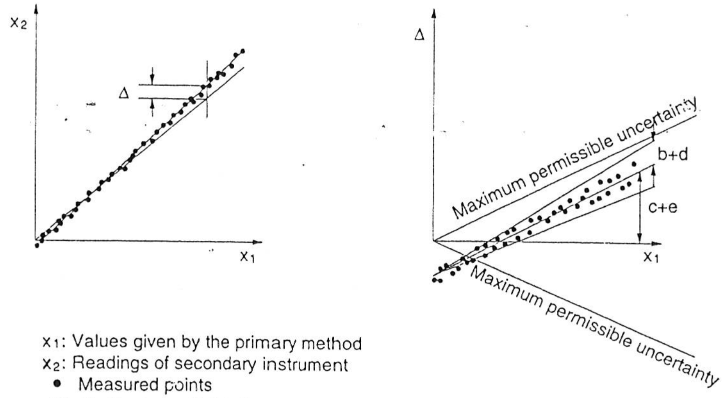
Table 8 provides a summary of the most important sources of error which can occur during model tests.

Clause J.1 gives an example of analysis of the sources of error and of uncertainty evaluation when measuring a quantity by a secondary electronic device.

.2.1.1 Errors arising during the calibration of a secondary instrument

sides possible spurious errors which should be eliminated, systematic and random errors y occur during the calibration of a secondary measuring instrument. The bias[1] of the nary method and of the secondary instrument and the errors in physical properties are tematic errors, whereas the repeatability of the primary method and of the secondary trument are random errors; the errors due to physical phenomena and influence quantities y be of partly systematic and partly random nature.

e total uncertainty in the calibration of the secondary instrument may be calculated by nbining, by the root-sum-square method, the component uncertainties (see table 8 and use J.1), as far as each of them can be evaluated. In practice, a calibration result resented by the example shown in figure 80 may be used to estimate the value of errors ns b) to e) of table 8. Item f) is generally negligible, item a) (bias of the primary method) uld be obtained from a higher level in the calibration chain (traceability of the primary thod) or certified by a neutral authority.



x_1: Values given by the primary method
x_2: Readings of secondary instrument
• Measured points
b), c), d), e): see Table 8

Figure 80 – Example of calibration curve

).2.1.2 Errors arising during the tests

natever the nature of its individual components, the total error in the calibration (item g) scribed in 3.9.2.1.1 becomes a systematic error in the measured quantity when this libration is used for subsequent model tests.

[1] The bias error is the systematic component of the error of a measuring instrument.

Table 8 - Summary of errors

During calibration

- a) Bias of primary method
- b) Repeatability of primary method
  - → Total error of primary method
- c) Bias of secondary instrument
- d) Repeatability of secondary instrument
- e) Error due to physical phenomena and influence quantities[1]
  - → Total error of secondary instrument
- f) Errors in physical properties
- → g) Total error in the calibration of the secondary instrument ($f_{t\,cal}$)

During test

- g) Total error in the calibration of the secondary instrument ($f_{t\,cal}$)
- h) Additional systematic error in secondary instrument
- i) Errors in physical properties
- k) Errors due to physical phenomena and influence quantities[1] → Systematic / Random
- l) Repeatability of secondary instrument
- Systematic → Systematic error in the measurement (f_s)
- Random → Random error in the measurement (f_r)
- → Total error in the measurement (f_t)

NOTE: Items a) to l) are cross-referenced with annex J

[1] Influence quantity: a quantity which is not the subject of the measurement but that affects the result of the measurement

The errors due to physical phenomena and influence quantities occurring during the tests (item k) may be neglected if the conditions of measurement (ambient temperature, voltage and frequency of the power supply, flow pattern, etc.) are maintained within a reasonable range during calibration and tests.

Since error in the determination of physical properties (item j) is generally small, the systematic error is largely controlled by the choice of the calibration method, the characteristics of the measuring apparatus, the installation and the operating conditions. For example, the value of the kinetic energy calculated with the mean velocity differs from its true value if the velocity distribution at the measuring section is not uniform (see 3.5.2.4, note 1).

The repeatability of the secondary instrument appears once again during the tests (item l) and results in a random error which may be characterized as stated in 3.9.2.2.1.

3.9.2.2 Estimation of the uncertainty

3.9.2.2.1 Uncertainty associated with random errors

An agreement between the parties prior to the test should specify the maximum permissible value of the uncertainty, f_r , for each quantity subject to guarantee. In the absence of such an agreement, the maximum permissible value of the random uncertainty in the hydraulic efficiency near the optimum should be $(f_{\eta h})_r = \pm 0{,}1$ %.

The actual value of the random uncertainties shall be estimated during the test at some operating points in a range in which the model runs in stable conditions (e.g. in the range near the point of maximum efficiency). For each of these points, the measurements shall be repeated a sufficient number of times (e.g. at least five) for applying the procedure described in annex L.

If, in these check points, the observed random uncertainties are lower than the previously agreed values, the maximum permissible levels of the random uncertainty are deemed satisfied in the whole guaranteed operating range, even if the direct evaluation of the random uncertainty in operating points with disturbed conditions gives higher values than the agreed value. In disturbed operating conditions (for instance a Francis turbine operating at partial load), the scatter of the measurements may be greatly increased; these higher values may nevertheless be accepted, for they arise from the model and not from the instrumentation.

If, in the check points, more than 5 % of the results are outside the agreed range, an accurate analysis of the conditions of measurement shall be made and the measurements shall be repeated or a new uncertainty bandwidth value due to the random errors agreed.

3.9.2.2.2 Uncertainty associated with systematic errors

The first step in estimation of this uncertainty is to identify each component which can influence its value. The second step is to allocate uncertainty limits to allow for each of these components. This may be done, in part at least, by statistical analysis (see for example ISO 5168).

The systematic error of a measurement is given mainly by the systematic error due to the calibration of the secondary instrument, and by the errors in physical properties.

The following considerations are useful in evaluating the systematic uncertainty (see table 8 and annex J).

a) As explained in 3.9.2.1.2, almost all the sources of systematic errors are covered by the calibration of the secondary instrument. In most cases, the systematic uncertainty in the measurement of a quantity may be taken as equal to the total uncertainty in the calibration of the secondary instrument used for the test $f_s \equiv f_{tcal}$, but it shall be borne in mind that in some circumstances other sources of error may need to be taken into account.

b) According to the measuring method and instrument used, systematic uncertainty consists of some of the following components:

   – the total uncertainty f_{t1} proper to the primary method shall be established before the calibration (see 3.9.2.1.1);

   – the uncertainty f_d is due to the random error of the secondary instrument during a calibration and to the scatter of several calibrations carried out at various periods if no systematic trend appears. For instance, if the calibration coefficient used during the test results from the average of n calibrations characterized by a standard deviation s_c, this component uncertainty may be taken as equal to:

   $$f_d = \pm \frac{t s_c}{\sqrt{n}}$$

   where t is the Student's coefficient for (n – 1) degrees of freedom (see table L.2);

   – the bias of the secondary instrument and the uncertainty due to the effect of physical phenomena and influence quantities are covered by the calibration, and the residual uncertainty in the applied correction can generally be omitted;

   – the errors in physical properties, if any, are small; for instance, the uncertainty f_ρ in the water density is lower than ±0,05 %;

   – an additional uncertainty may arise from the regression process used to determine the calibration curve. Although this uncertainty can be evaluated in accordance with ISO 7066, a conventional value of say ±0,05 % may be assumed.

c) The systematic uncertainty f_{s2} in a measurement made by the secondary method may then be obtained by combining these component uncertainties by the root-sum-square method.

   Each clause concerning the measurement of one of the quantities necessary to determine the hydraulic performance of the model gives an indication of the relevant systematic uncertainty.

   This value applies when the measurements are made in normal conditions by experienced personnel with apparatus of high quality, in accordance with the provisions of this standard, and can be used as a guide to establish the value of systematic uncertainty.

Prior to the test, an agreement between the parties shall specify the systematic uncertainty bandwidth for the different quantities, including the hydraulic efficiency. The actual value of the systematic uncertainties, like those of the random uncertainties, depends on many factors, some of which can only be evaluated after completion of the test. A review of these factors shall be made and agreement established as to whether the expected uncertainties have to be changed on technical grounds or not.

.2.2.3 Uncertainty in a derived quantity

ɘ uncertainty (systematic or random) of a derived quantity is determined by combining the ꞉ertainties of the component measurements by the root-sum-square method.

· example, the systematic uncertainty in hydraulic efficiency $(f_{\eta h})_s$ is computed from the ividual systematic uncertainties in discharge $(f_Q)_s$, specific hydraulic energy $(f_E)_s$, torque $)_s$, speed of rotation $(f_n)_s$ and density of water $(f_\rho)_s$ [1] by:

$$(f_{\eta h})_s = \pm\sqrt{(f_Q)_s^2 + (f_E)_s^2 + (f_T)_s^2 + (f_n)_s^2 + (f_\rho)_s^2} \quad \text{(see annex J)}$$

r contractual purposes it is conventionally accepted to omit the uncertainty associated with scale-up formula of the hydraulic efficiency.

ı.2.2.4 Total uncertainty

e total uncertainty (see table 8) in any quantity is given by:

$$f_t = \pm\sqrt{f_s^2 + f_r^2}$$

ıen the random uncertainty (evaluated as prescribed in 3.9.2.2.1) is lower than or equal to ɘ maximum permissible value (generally ±0,1 %) it is assumed to be conventionally equal to s value, which shall then be used to calculate the total uncertainty.

ıen, in some operating points, the conditions of measurement are disturbed and the scatter the readings (see 3.9.2.2.1) results in an increase of the observed random uncertainty, it is ɜsonable to take into account the observed value instead of the previously agreed one for lculating the total uncertainty.

ı0 Comparison with guarantees

ı0.1 General

is recommended that the test results, calculated according to 3.8, are compared with ıarantees using the methods of presentation and analysis described below, taking into count the total uncertainty bandwidth (see 3.10.2) and the contractual limits (see 3.10.3). For nplicity's sake, only prototype power, discharge and/or specific hydraulic energy, hydraulic ficiency and steady state runaway speed and discharge are considered hereafter (see 1.4.2).

ɔmparison with prototype guarantees shall consider the effect of cavitation (see 3.8.2.4.2 and 8.3.2).

ɔmparison with guarantees given on the model is made directly using the same procedures.

is recommended that hydraulic efficiency is presented versus discharge (or specific hydraulic ıergy, in case of non-regulated machines) and not versus power.

[1] The systematic uncertainty $(f_\rho)_s$ in density of water may generally be neglected.

3.10.2 Interpolation curve and total uncertainty bandwidth

Different methods and criteria can be used to draw the interpolation curve ranging from the manual to the more sophisticated ones (one of the possible methods of drawing is briefly described in annex H). The final choice of the interpolation method shall be clearly defined and agreed between the parties.

Taking into account the total uncertainties calculated per 3.9.2, each measured point can be represented on a diagram by an ellipse. The semi-axes of this ellipse represent the total uncertainty, at a confidence level of 95 %, in the two quantities chosen as co-ordinates of the diagram. Any point within this ellipse is equally valid.

An uncertainty band corresponding to the upper and lower envelopes of these ellipses is superimposed on the curve drawn through the test points (interpolation curve). All the points within this band are equally valid and hence this band constitutes an acceptable bandwidth for the comparison with the guarantees.

The ellipses need only be used when evaluating the guaranteed points or whenever the result of the comparison is not sufficiently clear (see figure 83, details X and Y). In most other cases, it is possible to simplify the procedure and to determine the total uncertainty bandwidth by reducing the ellipses to their principal axis when for example the error on the abscissa can be neglected (see figure 86), or the measured curve across the guaranteed point is almost horizontal, or has only a small gradient.

If the guarantees are given by points, it is recommended that the measured points be selected as near as possible to the guaranteed points: figures 81 and 82 show two examples concerning the hydraulic efficiency of a single and a double-regulated turbine, respectively.

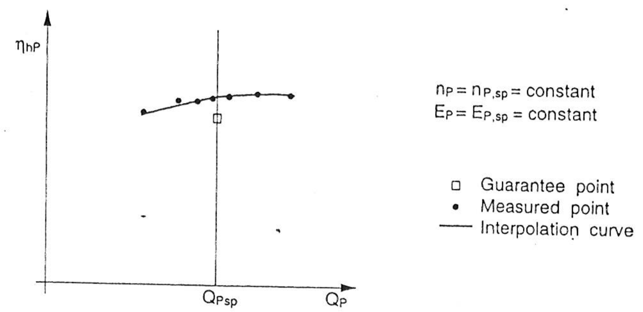


Figure 81 – Single-regulated machine

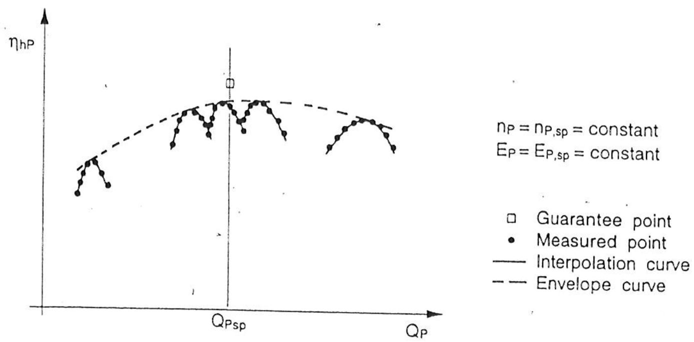


Figure 82 – Double-regulated machine

.10.3 Power, discharge and/or specific hydraulic energy and efficiency in the guarantee range

'he following classes of machines will be dealt with:

regulated turbine;

non-regulated turbine;

non-regulated or regulated pump.

.10.3.1 Regulated turbine

the hydraulic efficiency guarantee is given at one or more specified powers or discharges, it ; met if, at the specified speed and specified specific hydraulic energy, the guaranteed single alues lie below the upper limit of the total uncertainty bandwidth at the specified powers or ischarges.

the hydraulic efficiency guarantee is given as a weighted or arithmetic average efficiency, it is ıet if, at the specified speed and specified specific hydraulic energy, the guaranteed average fficiency is less than the average efficiency calculated at the same specified discharges (or ower) using the upper limit of the total uncertainty bandwidth.

ı case of guarantees given at different E_{Psp}, a diagram similar to figure 83 shall be drawn for ach specified specific hydraulic energy.

'or double-regulated turbines, the curves to be compared with the guarantees are drawn as ·nvelope curves.

'igure 83 gives an example of comparison with guarantees given at four operating points, at a ·pecified E_P, in the case of a single-regulated turbine. It shows that:

ı) the hydraulic efficiency guarantee is not met at D (see detail X);

ı) the mechanical runner power guarantee is not met because the guaranteed power is not reached even if the upper limit of the uncertainty bandwidth is taken into account (see detail Y).

The curve α (Q_P) is drawn to determine:

– the maximum opening α_{max} for the runaway test (see figure 86);

– if a sufficient safety margin exists between the guaranteed power and the mechanical runner/impeller power saturation.

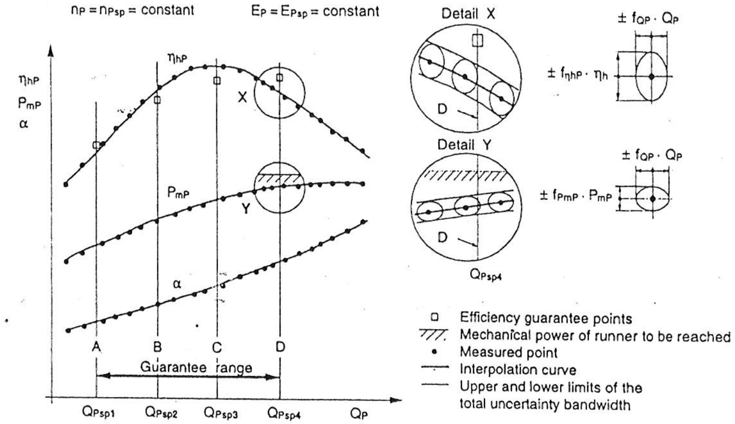


Figure 83 – Single-regulated turbine. Comparison between guarantees and measurements

3.10.3.2 Non-regulated turbine

If the hydraulic efficiency guarantee is given at one or more specific hydraulic energy, it is met if, at the specified speed, the guaranteed single values lie below the upper limit of the total uncertainty bandwidth.

If the hydraulic efficiency guarantee is given as weighted or arithmetic average efficiency, it is met if, at the specified speed, the guaranteed average efficiency is less than the average efficiency calculated at the same specific hydraulic energies using the upper limit of the total uncertainty bandwidth.

The mechanical runner power limit is usually defined, if not otherwise agreed, by a lower limit kP_{mPsp} and a higher limit (k + 0,03) P_{mPsp}, k being a mutually agreed value lying somewhere between 0,97 and 1,0. Normally, the value of k is 0,985. The choice of k shall be compatible with the guaranteed limit of P_{mP}.

Figure 84 gives an example of comparison with guarantees given at three operating points and with limit values:

a) the hydraulic efficiency guarantees are met in points A, B and C;

b) the discharge limit[1] to be exceeded in point A is satisfied;

c) the power limit not to be exceeded in point C is satisfied (see detail X), because it has been chosen k = 0,970 and the lower limit of the total uncertainty bandwidth is lower than the guaranteed higher limit P_{mP} = (0,970 + 0,030) P_{mPsp}.

[1] The prototype guaranteed discharge should be referred to the ambient pressure. Although the symbol of the prototype discharge should be Q_{1P} (see 1.3.3.4.5), the symbol Q_P is normally used.

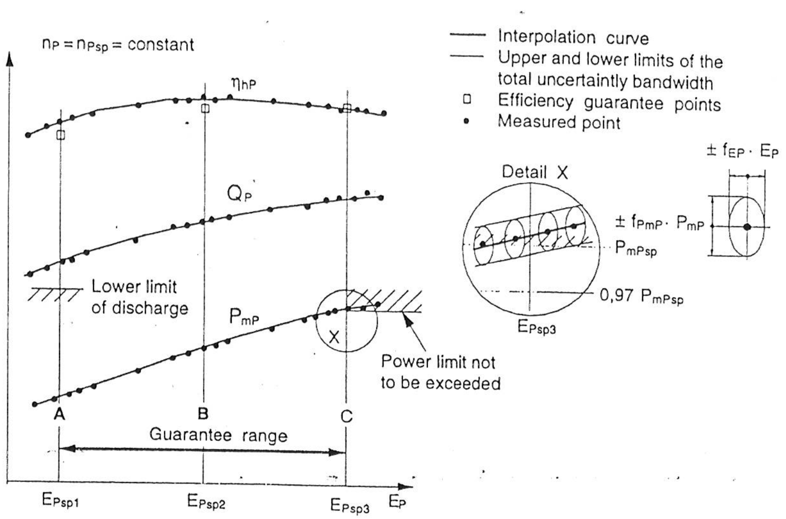


Figure 84 – Non-regulated turbine. Comparison between guarantees and measurements

ı.3.3 Non-regulated/regulated pump

discharge limit is usually defined, if not otherwise agreed, at one or more points by a lower kQ_{Psp} and a higher limit (k +0,03) Q_{Psp}, k being a mutually agreed value lying somewhere veen 0,97 and 1,0. Normally, the value of k is 0,985.

discharge guarantee is met if, at the specified specific hydraulic energy, there is inter- ion or contact between the band defined by the discharge limits and the total uncertainty- dwidth defined by the envelope of the uncertainty ellipses of the measured points defining discharge characteristic (see figure 85).

checking the hydraulic efficiency guarantee, the value to be compared with the guaranteed ie is the upper limit of the total uncertainty bandwidth on efficiency at the operating point ned by the intersection of the interpolation curve drawn through the measured points with characteristic curve $E_P = f(Q_P)$ (see for example point A' of figure 85).

regulated pump is operated at different openings, the above considerations apply to the vant envelope curves.

ɹre 85 gives an example of comparison with guarantees given for three operating conditions a non-regulated pump:

he hydraulic efficiency guarantees are met at points A' and B', but not at point C';

he minimum discharge limit is not met at point A';

he power limit not to be exceeded at point C' is satisfied.

variable speed pumps, the change of the E(Q) and E(P) characteristics in function of speed to be taken into account.

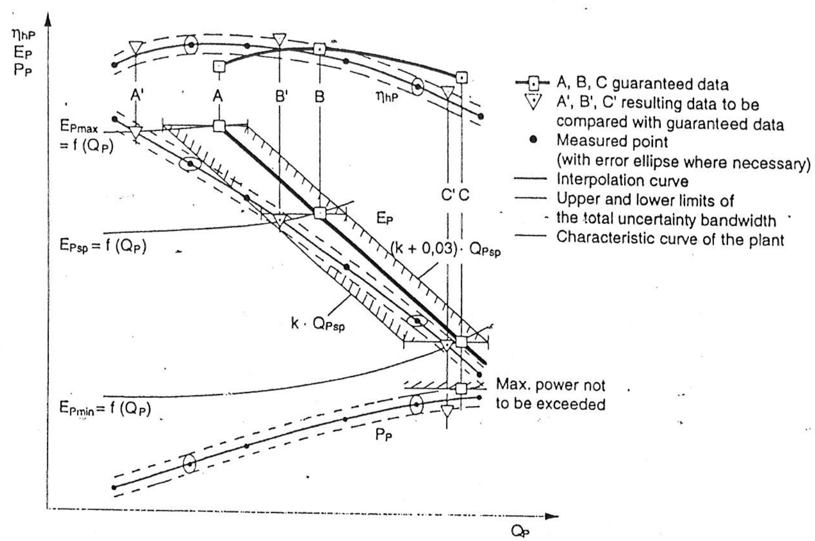


Figure 85 – Non-regulated pump. Comparison between guarantees and measurements

3.10.3.4 Prototype mechanical losses

If the prototype power P_P is guaranteed, the prototype mechanical losses shall be taken into account (see 1.4.2.1.1).

If the prototype efficiency η_P is guaranteed, the following formula applies:

$$\eta_P = \eta_{hP} \cdot \eta_{mP} \quad \text{(see 1.3.3.9.3)}$$

3.10.3.5 Penalty and premium

It is recommended that the contract should state in detail the method of calculating penalties and/or premium from the test results.

To determine the penalty on efficiency, the guarantee shall be compared with the upper limit curve of the total uncertainty band.

To determine the premium on efficiency, the guarantee shall be compared with the lower limit curve of the total uncertainty band.

).4 Runaway speed and discharge

› shape of runaway curves and the influence of cavitation for different types of machines are ered in 3.8.

› example of figure 86 refers to a Francis turbine. It shows the prototype steady-state away speed curve, calculated from the measured model speed factors, against guide vane ·ning.

his example, the guarantees of maximum steady-state runaway speed and discharge are : as shown in details X and Y: at openings less than α_{max} the lower limit of the uncertainty ıdwidth is lower than the value not to be exceeded.

a double-regulated machine, the guarantee shall be verified under the worst runaway dition which can occur in the guarantee ranges of the Thoma number and specific hydraulic ·rgy, depending on guide vane opening and runner blade angle.

a non-regulated machine, there is only one measured point to be compared with the ıranteed value.

ess otherwise agreed, the mechanical and windage losses of the motor/generator and the :hanical losses of the hydraulic machine (see annex G) are taken into account.

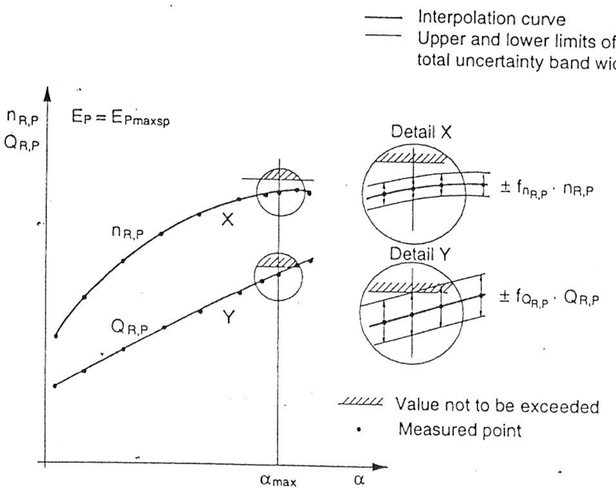


Figure 86 – Francis turbine. Runaway speed and discharge curves. Comparison between guarantees and measurements

0.5 Cavitation guarantees

e influence of the cavitation on the performance of the machine is dealt with in 2.3 and 3.8, ɗ some recommendations are given for comparison of prototype guarantees with ·formance data resulting from model tests.

When determining the Thoma number σ_0, which is the lowest value of σ for which efficiency remains unchanged, the uncertainty bandwidth of the non-cavitating efficiency shall be taken into account.

Figure 87 shows a test curve η_{hM} (σ). If the guarantee prescribes $\sigma_1 \leq k\ \sigma_{pl}$, the guarantee in this case is not fulfilled, since the Thoma number σ_1 for which a drop of 1 % in efficiency is observed is higher than the plant Thoma number σ_{pl} reduced by a mutually agreed safety coefficient k, even if the total uncertainty bandwidth is taken into account.

In the prototype, the cavitation guarantees may also be given in term of $NPSE_{P1} \leq (NPSE_{pl} - K)$, K being a mutually agreed safety margin.

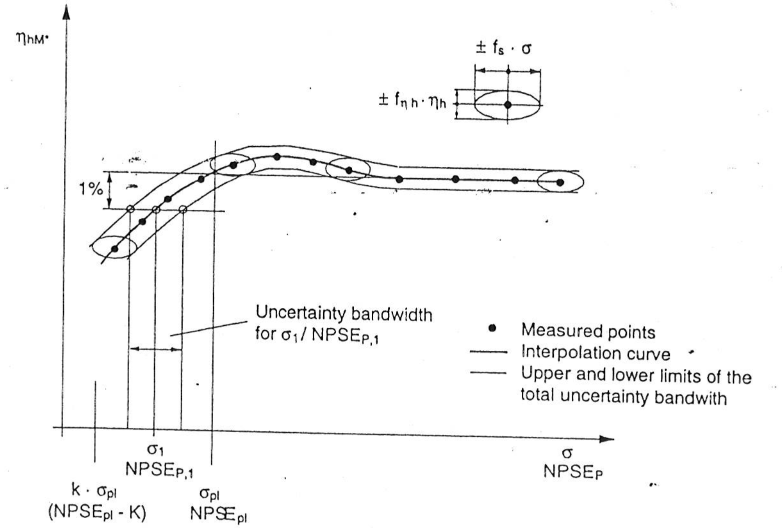


Figure 87 – Model turbine. Cavitation curve and comparison with the guarantee on the influence of the cavitation on the efficiency

Additional performance data – Methods of measurement and results

1 Introduction to additional data measurement

1.1 General

ıe so called "additional data" (torque, forces, pressure fluctuations etc.) defined in ıbclause 1.4.4 provide information for the design and operation of the hydraulic machine thin the hydroelectric power plant. Therefore, additional types of measurements are needed, ıich can be specified.

is currently neither possible nor desirable to prescribe requirements for measurements of dditional data" as rigorously as prescribed for the main hydraulic performance. The easurement methods and evaluation techniques for "additional data" are rapidly evolving.

formation stated in this clause shall thus be taken as recommendation or guidance to the ser in order to carry out measurements with the needed accuracy and under comparable onditions.

ach operating point of a hydraulic machine is characterized by a variety of mechanical and /draulic quantities (usually of oscillating nature) which occur during both steady state and ansient operations. The model is always operated in steady state conditions. In general, it is ɔt possible to simulate on the model the prototype transient operating sequences, the data for hich can only be derived from a series of steady-state operations.

ubclause 4.2 describes requirements for data acquisition and processing in addition to those escribed in subclause 3.1.

is often unnecessary to measure some of the additional data on the model if these values an be predicted with sufficient accuracy based on data from similar hydraulic machines (e.g. ade and guide vane torque, radial thrust etc.). The measurements of additional data shall be efined by the technical program, (see 2.3.3.3.2).

he hydraulic machine has to be considered as a component of the whole hydro-electric ıstallation. In this connection it is advisable to investigate unstable operation due to excitation f natural frequencies of the hydraulic circuit. Model tests can be used to identify possible xcitation frequencies and mode shapes of the machine at various operating points. ubclauses 4.3 and 4.4 are devoted to these procedures of identification.

or a "safe" mechanical design of the prototype, hydraulic loads acting on various components f the prototype machine can be derived from model test data scaled up using appropriate ansposition laws. Subclauses 4.5 and 4.6 describe the methods and the test conditions for eriving such hydraulic loads with their mean and dynamic components.

tart up, shut down and/or any change of operation modes will lead to transient operation of ıe machine far from the "normal" operating range. Therefore, in certain cases it is required to xplore this extended operating range with respect to the relevant hydraulic and mechanical uantities. Subclause 4.7 deals with measurements of the hydraulic characteristics in the xtended operating range (the so-called 4-quadrants range for a pump-turbine).

Finally, subclause 4.8 describes how to investigate during model tests the feasibility of index tests on prototype.

4.1.2 Test conditions and test procedures

The additional measurements are usually performed with the same model machine on the same test installation and require the same instrumentation as used for the main hydraulic performance tests according to clauses 1 to 3 of this standard. It has to be checked if the same test conditions can be applied as during the main hydraulic performance tests or shall be adapted. In any case, disturbing effects due to vibrations, resonance, mechanical deformations, increased leakage flow, etc., or defects of the additional measuring equipment shall be minimized. Possible exceptions to the requirements of clauses 2 and 3, if any, are dealt with in the relevant subclauses of clause 4.

IEC 60994 provides specification for instrumentation requirements and the measurement of fluctuating quantities. In addition, relevant specifications are given in the corresponding subclauses of clause 4. Supplementary recommendations for measurement of these additional data can be found in the existing IEC and ISO standards.

Depending on the measuring arrangement, the scope of various test series and the admissible and required test conditions, the test procedure and the test program can be defined. The measurement of various types of additional data can be combined with each other and/or with the measurement of the main hydraulic performance.

The same procedures with respect to calibration, preliminary and acceptance tests, check of zero readings, etc., as described in subclause 2.3 are also applicable to measurements of additional data.

The test conditions and basic analysis methods shall be agreed upon prior to the tests.

4.1.3 Uncertainty in measurements

The total uncertainty in the measurements of additional data is generally higher than the uncertainty in the measurement of the main hydraulic performance quantities, for the following reasons:

a) the off-design operating range in which the quantity is measured;

b) the unsteady nature of the measured quantities;

c) the limitation of the available instrumentation and calibration procedures.

The measuring methods shall be selected such that the uncertainty corresponds to the mutually agreed accuracy for any given purpose. In many cases, figures for uncertainties should be given in physical units (Pa, N, N·m, ...).

The individual aspects of uncertainties are described in the relevant subclauses 4.3 to 4.8.

4.1.4 Model to prototype conversion

Prototype values can be calculated by converting the model test data according to the general similarity laws. The appropriate conversion procedures are described in the following subclauses. It is common practice to transpose the results from model to prototype using the appropriate dimensionless terms.

ɔwever, it shall first be checked whether hydraulic and mechanical similarity laws are ıfficiently established for the particular quantity. If not, the conversion shall consider the 'namic-structural aspects of the prototype machine within the entire hydroelectric system, cluding such factors as flow pattern at entrance and exit, resonance, external excitation, etc.

ıctors in the model test facility, such as the following which could affect the test results, ıould be eliminated to the extent possible:

influences due to flow regimes;

influences of mechanical structures;

others.

2 Data acquisition and processing for measurement of fluctuating quantities

2.1 General

ıbclause 3.1 describes the measurement of average values of the main hydraulic perform-ce quantities. Many of the methods used allow recording of the fluctuating component of the antity as well.

ıis subclause 4.2 describes requirements for data acquisition and processing for the termination of the fluctuating quantities. Data processing includes the calculation, evaluation d presentation of statistical quantities representing the model measurements. IEC 60994 all be taken into account.

e procedure for measurement, data acquisition and data processing shall be agreed upon or to the tests.

e measured quantities will be:

either of periodic nature;

or of non-periodic, stochastic and intermittent nature.

e fluctuating quantities can be measured:

with their mean components (measurement of $\overline{X}$ in Figure 7);

or as fluctuating component only (measurement of $\tilde{X}$ in Figure 7).

ıe measurement methods shall be able to record fluctuating quantities in sufficient resolution describe their characteristic occurrence. Measurements can be made using:

pressure transducers;

accelerometers;

strain gauges;

other transducers for mechanical quantities.

ı allow further data analysis, the measurement chain from transducer to data storage system all fulfill various criteria provided by the signal analysis theory (see reference [21] in ınex P). In particular the criteria described in subclause 3.1 shall be applied. However, the ıctuating nature of the quantities considered in this subclause leads to additional quirements described below.

4.2.2 Data acquisition

4.2.2.1 Signal conditioning

The purpose of measurement and the methods of data acquisition and processing determine the appropriate signal conditioning. The main purposes of conditioning are:

– removal of high frequency noise to prevent aliasing effects (analog filter);
– elimination of irrelevant signal components (analog, digital filter, or software methods);
– offset adaptation.

4.2.2.2 Analog to digital conversion

Most data acquisition systems are based on periodic sampling followed by analog to digital conversion. Figure 88 shows a typical data acquisition system.

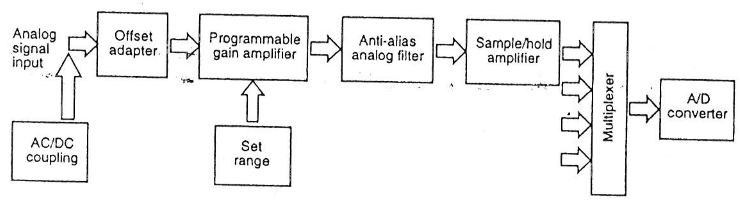


Figure 88 – Typical data acquisition system

When the steady-state value of a signal is not relevant and is greater than the peak to peak value, offset adaptation can improve the quality of the stored fluctuating component of the signal. AC coupling of the converter inputs can induce signal distortions in the low frequencies. Selectable DC offset adds a constant to the analog signal. The signal range can be centered on the converter range without low frequency distortions.

Programmable gain amplifiers adapt the analog signal range to the dynamic range of the A/D converter.

Analog anti-alias low-pass filters shall be used for periodic sampling. The sampling frequency is limited by the filter characteristics. Simultaneous sampling of all signals can be done with a sample and hold amplifier on each measurement channel. Sampled signals can then be multiplexed and converted sequentially. If the signals are sampled sequentially, the resulting delay shall be considered (see 4.2.2.4).

Most analog to digital converters used in signal analysis deliver a stream of integer values corresponding to discrete increments of the physical input. This operation induces an irreversible loss of data and shall be done with care.

he resolution and precision of a analog to digital converter determine its quality. The ›solution is normally stated in bits. The sign bit is not always counted.

he quality of the digitized signal also depends on the matching between the analog signal xtreme values and the converter range. Therefore, each measurement channel shall be onditioned so that the signal best fits the converter range.

.2.2.3 Sampling rate

eriodic sampling requires all energy in the signal to be contained between zero and half the ampling frequency, (see reference [21] in annex P). If this condition is not fulfilled, aliasing ffects will introduce an irreversible corruption of the sampled signal.

o prevent aliasing, an analog low-pass filter shall be used before sampling. If the filter has a at gain function up to f_{max}, the required sampling frequency is:

$$f_s > 2\, f_{max} + f_{trans}$$

here f_{trans} is the transition frequency (see Figure 89), which depends on:

the type and order of the analog anti-aliasing filter;

the filter cut-off frequency;

the characteristics of the analog to digital converter;

the possible high frequency content of the signal to be measured;

the acceptable noise level.

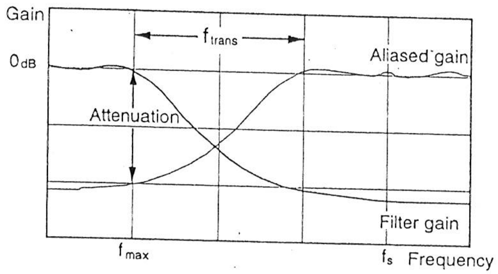


Figure 89 – Frequency response of analog anti-aliasing filter

.2.2.4 Phase information

ı some cases, the phase relationship between different fluctuating quantities is important. In ıch cases, the information on phase shift shall be carefully dealt with to prevent possible ıase distortions caused by the data processing system.

/hen the measurement channels have different signal conditioning electronics and the phase important, this influence shall be determined and corrected.

If the channels are sampled sequentially, the phase distortion $\Delta\varphi$ for channel i (i = 1, 2, ..., n) compared to channel 0 depends on:

- the time Δt between sampling of consecutive channels;
- the position of channel i;
- the frequency of interest f.

The phase distortion is:

$$\Delta\varphi = 2\pi \cdot \Delta t \cdot i \cdot f \qquad \Delta t \text{ can be constant or equal to } 1/(n \cdot f_s)$$

4.2.2.5 Data storage

Both analog recorders and digital media can be used for data storage. In the first case, raw signals are for example recorded on magnetic tapes and can be recovered for visualization and further analysis. The magnetic recorder is considered as part of the acquisition system and its frequency response should match acquisition criteria. In the second case, digitized data are stored in mass storage media such as disks, tapes, etc.

4.2.2.6 Data acquisition procedure

The following information should be available in order to identify the data acquisition procedure:

- resolution of the analog to digital conversion in terms of number of bits;
- sampling frequency;
- frequency response of anti-aliasing filters;
- frequency responses of the measurement chain;
- number of records and the number of samples in each record;
- data storage procedure.

4.2.3 Data processing

Once data acquisition criteria are fulfilled, data processing is performed in order to extract information of interest. A variety of transformations can be applied to time-history signals in order to analyze:

- their statistical behavior;
- their spectral content (in the frequency domain);
- correlation or other relationships between signals.

Time domain analysis parameters are:

- averaged values;
- characteristic amplitudes[1)], standard deviations, r.m.s. values;
- probability density functions, probability distributions;
- others.

1) A characteristic amplitude means e.g. half the difference between the maximum and minimum values of a signal. It can be determined with the aid of probability distribution applying counting methods and assuming a certain amount of probability (e.g. 97 %). Amplitudes occurring outside of this probability are ignored.

ꝫquency domain analysis parameters are:

amplitude spectrum (square root of auto power spectrum);

auto power spectrum (energy spectrum for finite energy signals);

cross power spectrum;

transfer functions;

coherence functions;

others.

bclause 4.3.6 provides more detailed information on data processing of pressure ctuations. This information, especially the subclause 4.3.6.1, is generally also valid for data ɔcessing of other measured fluctuating quantities, such as shaft torque fluctuations ıbclause 4.4), axial and radial thrust (subclause 4.5), guide vane torque (subclause 4.6.2).

ꜣ Pressure fluctuations

ꜣ.1 General

ꜣ.1.1 Pressure fluctuations in hydraulic machinery

essure fluctuations are a natural occurrence in hydraulic machinery. They can be of a riodic or stochastic nature. They are due to the action of the water passages and vanes on ꜫ flow fields within the machine. They are influenced by machine design, operating conditions d by the dynamic response of the water conduits and rotating components. Pressure ctuations are actually a part of hydro-acoustic phenomena involving unsteady pressure and w velocity distributions. They can also be associated with mechanical fluctuations of shaft que, rotational speed, hydraulic load on guide vanes etc. as well as with vibrations of the ıchine.

w frequency disturbances are of special interest because they can propagate to the whole ıter conduit and the rotating parts of the electric machine. They occur typically between 0,2 d 3 times the runner rotational frequency.

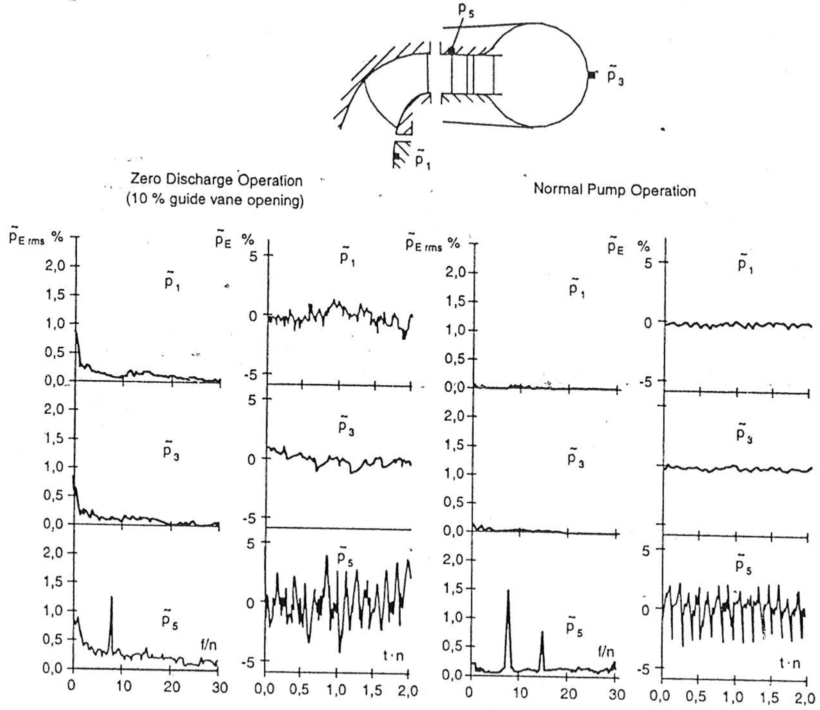


Figure 90 – Normal pump mode and zero discharge operations of an n_{oE} = 0,102 pump-turbine model

The draft tube surge of Francis and propeller turbines and pump-turbines is perhaps the most commonly identified phenomenon among low frequency pressure fluctuations. In these machines, a strong runner outlet swirl can develop, inducing pressure fluctuations. In addition, draft tube cavitation can change the natural frequencies of the hydraulic system.

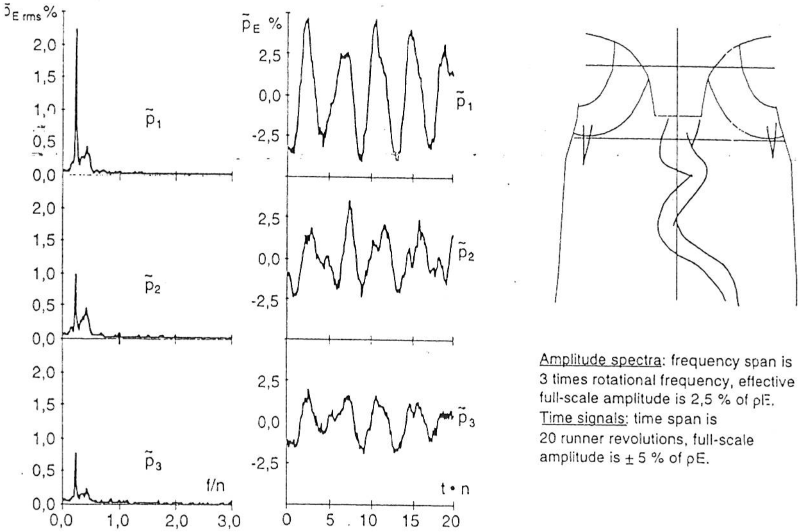
In double-regulated Kaplan or diagonal (Dériaz) turbines, on-cam control of the guide vane opening and of the runner blade setting leads to a minimal runner outlet swirl and no serious draft tube surge is generated.

In impulse turbines, the runner is separated from the nozzle by a constant-pressure condition. For this reason, there is no interaction between runner and water conduits. Impulse turbines are not considered in this subclause.

ɹrbines and pumps produce an excitation at the rotational frequency multiplied by the number runner/impeller blades usually defined as the blade passing frequency. Due to the interaction the runner/impeller blades with the guide vanes / stay vanes / spiral case, higher frequencies 'e generated up to k multiplied by the blade passing frequency, k being typically between 1 ıd 2 for turbines and 1 and 4 for pumps and pump turbines.

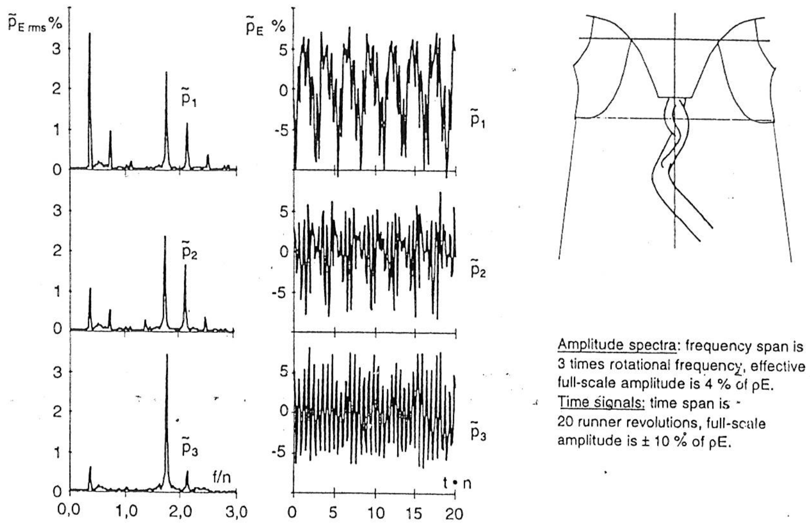
tochastic pressure fluctuations due to turbulent flow separation or intermittent pressure ɹlses due to vortex breakdown can take place in various conditions of the extended operating ınge.

ome examples of pressure fluctuations associated with reaction machines are shown in igure 90 through Figure 93 for various operating conditions. Pressure transducer locations are ; proposed in Figure 94.



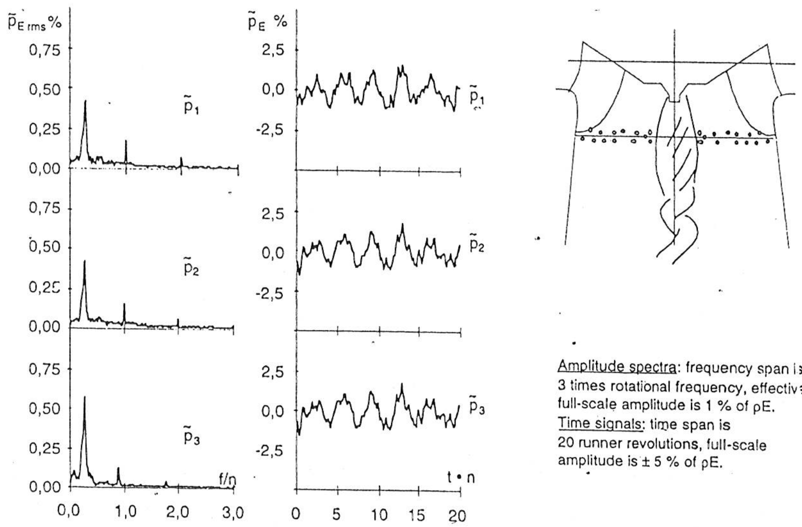
Amplitude spectra: frequency span is 3 times rotational frequency, effective full-scale amplitude is 2,5 % of ρE.
Time signals: time span is 20 runner revolutions, full-scale amplitude is ± 5 % of ρE.

Figure 91 – Part load operation of an n_{QE} = 0,321 Francis turbine model: Q_{nD}/Q_{nDopt} = 0,719



Amplitude spectra: frequency span is 3 times rotational frequency, effective full-scale amplitude is 4 % of ρE.
Time signals: time span is 20 runner revolutions, full-scale amplitude is ± 10 % of ρE.

Figure 92 – Higher part load operation of an n_{QE} = 0,226 Francis turbine model: Q_{nD}/Q_{nDopt} = 0,764



Amplitude spectra: frequency span is 3 times rotational frequency, effective full-scale amplitude is 1 % of ρE.
Time signals: time span is 20 runner revolutions, full-scale amplitude is ± 5 % of ρE.

Figure 93 – Full load operation of an n_{QE} = 0,173 Francis turbine model: Q_{nD}/Q_{nDopt} = 1,218

.1.2 Influence of the hydraulic circuit and rotating machinery

del tests of an isolated hydraulic machine do not fully reproduce all possible prototype iamic phenomena because hydro-acoustic waves can be significantly influenced by the indary conditions.

ssure fluctuations are not only characteristics of the model but they can also be affected by characteristics of the test facility. Possible conditions affecting similitude include the owing:

water conduit (pipe length, wall stiffness);

test fluid characteristics (free gas content);

dynamic behavior of rotating components;

Reynolds and Froude numbers.

.1.3 Purpose of the measurements

ssure fluctuation measurements are usually conducted to obtain the following information:

relative magnitude of the pressure fluctuations within specified operating range;

nature of the pressure fluctuations, periodic or stochastic;

dominant frequency of the pressure fluctuations, if any;

effect of palliative methods such as air admission (effectiveness and suitable location);

intensity of the pressure fluctuations of the model compared with other models of similar specific speed.

favourable conditions where no significant interactions with the external systems, see jure 99, are involved, quantitative model test results can be directly transposed to the ototype.

vertheless, due to the various interactions with the external systems or the differences of id characteristics between model and prototype, the amplitude and frequency of the ototype pressure fluctuations can deviate considerably from those directly transposed from model tests. Since the present state of the art does not permit to quantify such deviations, measurements of pressure fluctuations of the model are conducted in most cases to obtain least qualitative information or to assess the magnitude of pressure fluctuations.

.2 Special requirements for model and installation

ie model and test installation shall comply with general requirements for the testing of draulic machinery, see subclause 2.1. It shall provide full control of test specific hydraulic iergy, rotational speed and pressure at the low pressure side. Once the desired test nditions are set, they shall remain steady during the time necessary for the measurement of ictuating quantities.

ansparent parts on the low pressure side of the runner should be large enough for servation of cavitation not only on the runner blades, but also in the upper portion of the draft be.

In order to avoid resonance effects between the test circuit and the model, the natural frequency of the test circuit should be sufficiently outside of the range of interest of model frequencies (see 4.3.4.1). Hydraulic disturbances from the feed pump, throttling devices, by-passes, bends etc. shall not affect the model in this range of frequencies.

The draft tube should be connected to a conduit or tank with a cross-sectional area large enough to overcome dynamic coupling of the model with the low pressure part of the test circuit.

The model construction shall be stiff enough to prevent excessive deformations. Vibrations of the model test rig and rotational speed fluctuations due to the governing system should not induce pressure fluctuations in the considered frequency band.

The test installation should operate in closed circuit mode, so the gas content can be kept low and constant. The model inlet flow shall be free of traveling bubbles.

In the case of models with a full spiral case, a straight uniform pipe, if feasible, at least 6 diameters long, should be installed on the high pressure side of the model. This allows an estimation of both the pressure waves propagation conditions and the hydro-acoustic power of these pressure waves at the spiral case inlet using the signals of the p_3 and p_6 pressure transducers outlined in Figure 94.

4.3.3 Instrumentation and calibration

4.3.3.1 Instruments for pressure fluctuation measurements

Pressure transducers should be mounted with their membrane flush with the hydraulic profile. When this cannot be done, cavities shall be carefully bled and their natural frequencies evaluated: they shall not induce distortions in the frequency band of interest (see 4.3.1.1 and IEC 60994, 6.3).

The transducers shall be sensitive enough to measure effective pressure fluctuations of ± 0,1 % of ρE.

The maximum permissible error of the measuring chain shall be smaller than ± 5 % of the measuring range used. This error can be reduced by a preliminary calibration procedure (see 4.3.3.2).

The maximum permissible error of the signal processing equipment shall be smaller than ± 1 % for the amplitude and smaller than ± 10° for the phase.

Figure 94 shows an example of suggested locations of transducers for a Francis or propeller turbine or pump-turbine. The measurement of p_1, p_2 and p_3 as listed below is strongly recommended. Transducers p_1 and p_2 should be placed 0,3 to 1,0 diameter from the low pressure side of the runner/impeller.

p_1 pressure transducer on the downstream side of the draft tube cone;

p_2 pressure transducer on the upstream side of the draft tube cone;

p_3 pressure transducer at the spiral case inlet.

Depending on pressure fluctuation data desired, the following transducers can also be installed:

p_4 additional pressure transducers in the draft tube cone: in the same section as p_1 and p_2 preferably 90° apart, in the draft tube bend or at locations corresponding to manholes on the prototype;

p_5 pressure transducers in the distributor (e.g. between runner / impeller and guide vanes);

p_6 additional pressure transducers along the intake;

p_7 pressure transducers in the draft tube outlet.

rce and torque measurements as listed below can also be performed concurrently with ›ssure fluctuation measurements:

f_1 axial and radial thrust transducers on the shaft – runner coupling flange (see subclause 4.5);

t_1 torque transducer on the shaft (see subclause 4.4).

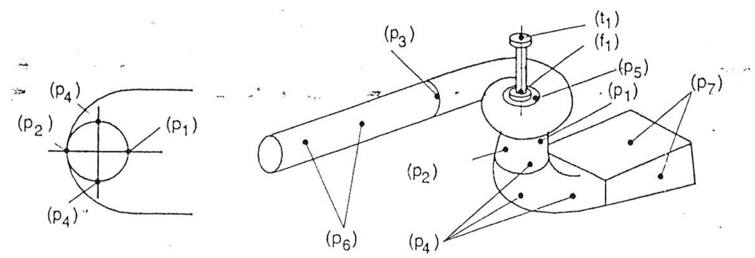


Figure 94 – Suggested locations of transducers

‹.3.2 Calibration

e dynamic calibration for pressure measurement systems can be performed. It consists of ‹ determination of the transfer function between the input pressure signal and the output ›ctric signal.

ıe absolute phase shift between the pressure and the output signal is not usually required. ıase shifts between output signals due to different signal conditioning systems, however, ould be known or compensated.

ıe gain and phase corrections can be determined by exciting all transducers with the same essure fluctuation, in the same range of frequencies and amplitudes as for the model test. ıe calibration shall ensure that differences in gain and phase between channels are within the ıcertainty margin of the signal processing equipment (see 4.3.3.1).

3.4 Detailed procedures

3.4.1 Test specific hydraulic energy

ıe test specific hydraulic energy is selected to provide good conditions for the adjustment of eady-state operation parameters. Also, the test specific hydraulic energy shall be chosen so at pressure fluctuation frequencies and amplitudes are within the limits of the strumentation.

Moreover, if a resonance between the model and the test installation is suspected, it is recommended to conduct the measurements under different test specific hydraulic energies (see 4.3.2).

Test specific hydraulic energy according to Froude similitude should be adopted whenever it is practical. In the case of large machines with small specific hydraulic energy, the Froude influence is considerably increased (see 2.3.1.5).

4.3.4.2 Cavitation reference level for pressure fluctuation measurements

If Froude similitude cannot be fulfilled, the flow field in the draft tube cone and consequently the pressure fluctuations will be influenced by vapour cavity development.

A cavitation reference level shall be agreed upon (see 2.3.1.5.1). For vertical units, this level can be at or below the low pressure side of the runner / impeller.

4.3.4.3 Model test operating conditions

Typical exploration paths, ranges of test points, for a pump or a pump turbine are represented in Figure 95 and typical exploration paths for a Francis or propeller turbine are represented on the E_{nD} - Q_{nD} diagram in Figure 96.

In the case of a turbine, the test points should cover at least the specified discharge operating range under a constant test specific hydraulic energy and at the Thoma number related to the cavitation reference level defined in 4.3.4.2. This can be completed by additional explorations.

Detailed tests at part load and at full load, at guide vane openings for which remarkable fluctuations occur, give a fair idea of fluctuation phenomena with slightly different operating conditions. These consist in one variation of energy coefficient with constant test specific hydraulic energy, Thoma number and guide vane opening, one variation of Thoma number and possibly a variation of test specific hydraulic energy.

In the extended operating range, considerable pressure fluctuations can occur. This refers particularly to turbines in runaway conditions and to pump-turbines (or pumps) in extreme operating conditions (for example in reverse rotation pump mode).

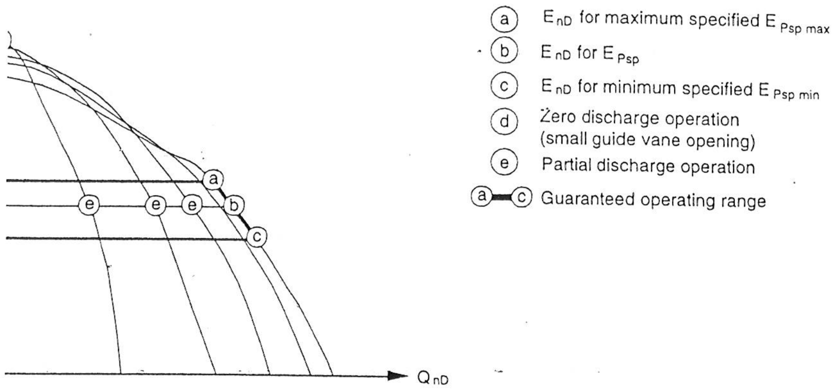


Figure 95 – Pump diagram with exploration paths

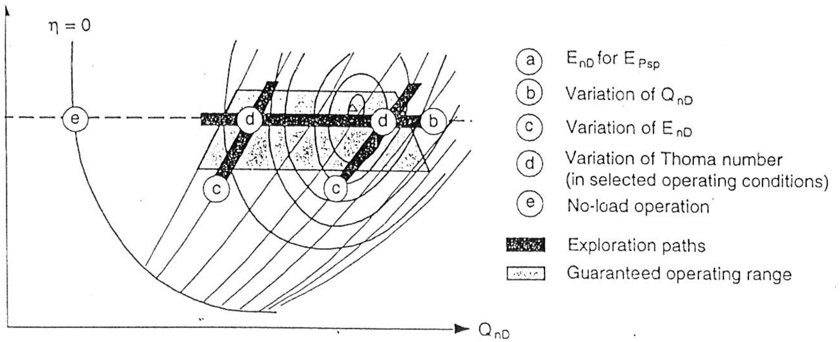


Figure 96 – Turbine hill-chart with exploration paths

ɔr plants with large variations of specific hydraulic energy, tests should also be performed for fferent energy coefficients, with corresponding values of Thoma number.

ɿis amount of data is desirable for an appropriate diagnosis of pressure fluctuations.

.3.4.4 Air admission

ir admission for the reduction of pressure fluctuations can be tested on models. Such tests ive only an approximate idea of the effects of air admission to be expected on the prototype, ecause similitude cannot be sufficiently ensured.

4.3.5 Measurement

For each test point, the fluctuating signal is sampled over a time period, sufficient for the purpose of the test, e.g. investigation of low frequency surges.

Draft tube cavitation should be observed and cavity patterns should be recorded.

Sampling and recording of signals shall comply with subclause 4.2.

4.3.6 Analysis, presentation and interpretation of results

4.3.6.1 Analysis

According to either the periodic or non-periodic nature of the pressure fluctuations, a frequency domain or a time domain analysis should be chosen.

4.3.6.1.1 Time-domain analysis

Time-domain analysis is mostly useful for processing random and intermittent fluctuations (see Figure 90 to Figure 93). For pressure fluctuation tests, time-domain analysis can be made by determination of the standard deviation of the signal. The comparison of the standard deviation with the frequency-domain estimation of the signal amplitude gives an indication of the random fluctuations superimposed on the periodic signal, while the characteristic amplitude determined by statistical methods gives an indication of the absolute magnitude of fluctuations.

The signals can be viewed in the time-domain also for checking purposes.

4.3.6.1.2 Frequency-domain analysis

Frequency domain analysis of the pressure fluctuations should be done with a Fourier analyzer. By the use of a multi-channel analyzer it is possible to track periodic phenomena simultaneously occurring at different frequencies and to evaluate phase shifts between channels. The Fourier analyzers usually express the magnitude of the spectrum components in engineering units.

Frequency spectra can be estimated by an average of discrete Fourier transforms computed on a succession of time records. In order to minimize leakage effects due to the finite time records and to preserve a good definition of frequencies appropriate weighting windows, such as Hanning or Kaiser-Bessel, should be used.

Frequency spectra obtained by the discrete Fourier transforms provide the best fit, over a given time record, between the sampled time domain data and superposition of sine waves. The time record length is the same for all frequencies within a transform. Higher frequencies are thus averaged over a greater number of periods than low frequencies. For this reason, the discrete Fourier transform is not suitable for the characterization of signals with a time-varying frequency content.

In wavelet / joint time frequency methods, the time weighting window is a function of frequency. The same number of periods is used for the computation of all frequency-coefficients. The result accurately represents the frequency content of the signal of interest as a function of time.

3.6.1.3 Non-dimensional frequencies and pressures

essure fluctuation frequencies are made non-dimensional by the runner/impeller rotational equency n.

Frequency coefficient $f_n = \frac{f}{n}$

essure fluctuation amplitudes are made non-dimensional by the pressure ρE representative the test specific hydraulic energy.

Pressure fluctuation factor $\tilde{p}_E = \frac{\tilde{p}}{\rho E}$

3.6.2 Presentation and interpretation of pressure fluctuations

3.6.2.1 General

ıe oscillatory data represented in non-dimensional terms ($f_n, \tilde{p}_E$) should be presented versus test parameter so as to provide a global information on the pressure fluctuations to be pected on the prototype in the investigated operating conditions. The test parameter can be:

- discharge coefficient or factor;
- energy coefficient or speed factor;
- Thoma number;
- test specific hydraulic energy;
- air flow-rate or other.

ıe analysis and presentation of measured pressure signals p_1, p_2 and p_3 (see Figure 94) are rongly recommended.

ıe diagrams below give examples of presentation of results.

3.6.2.2 Waterfall diagram

ıe waterfall diagram in Figure 97 presents amplitude-frequency spectra as a function of the ılected test parameter in a 3-D style display. It provides a fast overview of all pressure ıctuations in the considered frequency band and operating range.

3.6.2.3 Summarized diagram

ıe summarized diagram in Figure 98 presents spectral data associated with the dominant equency for each measurement channel as functions of the selected test parameter in 2-D splays.

he dominant frequency is that associated with the maximum spectral amplitude of the signal ıannel in the frequency band of interest.

he summarized diagram indicates:

- the dominant frequency;
- the phase shift to the reference channel at the dominant frequency;
- the narrow-band amplitude at the dominant frequency;
- the wide-band amplitude.

The narrow-band effective amplitude is estimated as the effective magnitude, $\tilde{p}_{rms}$ (f) of the discrete Fourier transform coefficient at the dominant frequency. The wide-band effective amplitude is estimated as the standard deviation, $\tilde{p}_{eff}$ of the time signal.

Iso-amplitude curves of the pressure fluctuations can be drawn in the E_{nD}-Q_{nD} or n_{ED}-Q_{ED} diagrams, provided that a sufficient number of paths have been explored, see Figure 96.

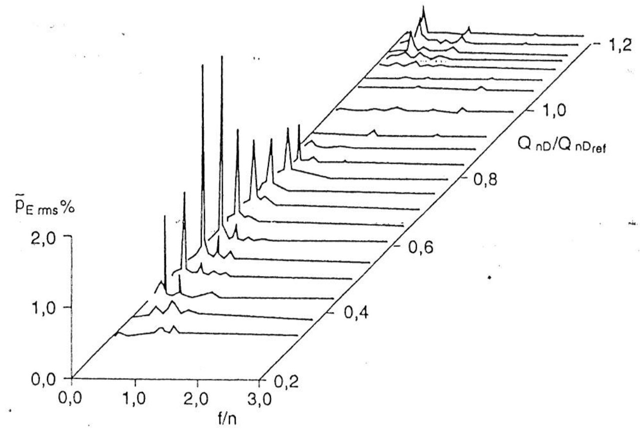


Figure 97 – Example of waterfall diagram of pressure fluctuations in the draft tube of a Francis turbine for path (b) of Figure 96, transducer p_1

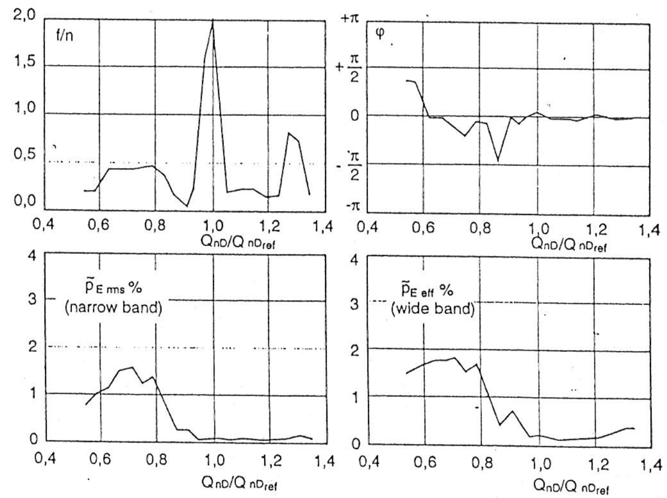
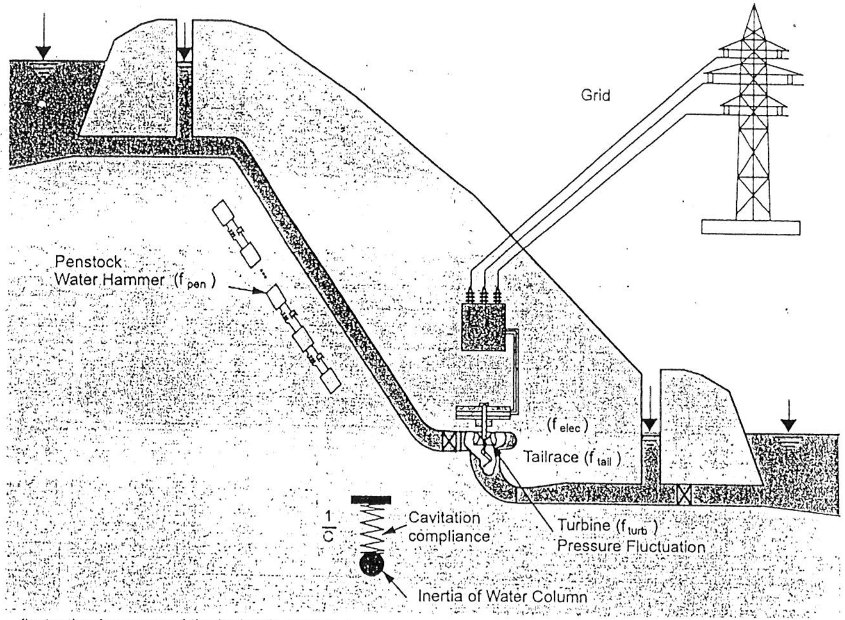


Figure 98 – Example of summarized diagram of pressure fluctuations in the draft tube of a Francis turbine for path (b) of Figure 96, transducer p_2

.7 Transposition to prototype

.7.1 Pressure fluctuation amplitudes

favourable conditions for the model tests (see 4.3.1.2) and in case that no significant eraction with the external system is present, pressure fluctuation amplitudes are fairly well nsposable from the model to the prototype.



rb fluctuation frequency of the hydraulic machine;

n natural frequency of the penstock;

il natural frequency of the tailrace;

ec natural frequency of the electric machine

Figure 99 – Interaction of the external systems on the machine pressure fluctuations

'hen any interaction with external systems exists, the prediction of prototype pressure ɪctuation amplitudes requires a dynamic response analysis of the full-size machine layout, cluding water conduits, manifold, gate chambers, tailrace tunnels etc. Due to the complexity the prototype plant configuration, it is practically very difficult to make such analysis on a omprehensive numerical model including all relevant system components as outlined in igure 99.

s a part of such analysis, it is only possible to check the occurrence of resonance by omparing the dominant frequency of the pressure fluctuations with the natural frequency of ach of the following external system components:

penstock;

tailrace;

electrical layout.

In case that any resonance with the external system is anticipated, the prediction of the prototype pressure fluctuation amplitudes is not included in the processing of model test pressure fluctuation measurements.

4.3.7.2 Fluctuation frequencies

The frequency coefficient defined in 4.3.6.1.3 is transposable from model to prototype for:

a) pressure fluctuations due to inertial forces within the runner outlet flow, such as the precession of the swirl;

b) free oscillations of the water plug in the draft tube against the gaseous volume due to either cavitation or air injection, if any;

c) pressure fluctuations due to the interaction of runner/impeller blades and guide vanes.

4.3.7.3 Excitation magnitude

The magnitude of pressure fluctuations has the same scale ratio as the pressure associated with the turbine specific hydraulic energy. This is true only at the emission of disturbances. Pressure waves propagate in the piping system, are reflected, and the observed amplitudes are strongly influenced by standing waves. The similitude of pressure amplitudes is lost in case of dynamic interaction with the hydraulic circuit.

The excitation magnitude can be more adequately determined by the estimation of the active hydro-acoustic power associated with the hydraulic oscillation. This power is defined for a given cross-section as the product of the instantaneous pressure fluctuations with the instantaneous discharge fluctuations. It is practically determined from the measurements of p_3 and p_6, see ref. [28] in Annex P. As the active hydro-acoustic power is not influenced by standing waves in the water conduit, it characterizes the power of the disturbance emission either from the machine (positive value) or from the test rig conduit (negative value).

4.3.7.4 Natural frequency of the draft tube water column

The stability of operation of a fixed runner blade reaction turbine is strongly dependent on the value of the draft tube natural frequency with respect to the frequency range of the pressure fluctuations. The physics related to the natural frequency of a draft tube is complex and under research.

The natural frequency of a draft tube f_0 can be conceptually considered as the frequency of the free oscillation of the water column against the elastic gaseous volume due to either cavitation or air injection, if any.

Then, the draft tube natural frequency f_0 is estimated by the following formula:

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{1}{\dfrac{-\partial V_{vap}}{\partial NPSE}\displaystyle\int\frac{dL}{A}}}\ ^{1)}$$

where

V_{vap} is the gas cavity volume in the draft tube, and

$\int dL/A$ is the ratio of length to cross-sectional area, integrated along the draft tube center line over the domain where the water column behaves as an inertia mass against the elastic gaseous volume.

---

1) The term $\frac{-\partial V_{vap}}{\partial NPSE}$ is called the "cavitation compliance" of the draft tube.

ınsequently, the draft tube natural frequency depends on the following operating parameters:

discharge ratio $\frac{Q_{nD}}{Q_{nDref}}$;

Thoma number;

Froude number;

injected air flowrate.

it is possible to identify the draft tube natural frequency of the model by a method as scribed below, the possibility of the occurrence of resonance can be evaluated for each reed operating point along the exploration path by comparing it to the frequency range of the ɜssure excitations.

ɔan be identify by an indirect method according to the following procedure:

a given cross-section, the free oscillations of the draft tube are characterized by pressure ctuations of a same phase. It is then possible to identify the corresponding frequencies from least 2 pressure transducers, e.g. p_1 and p_2, and to check if one of those frequencies is ected by any variation of the cavity volume with the discharge. From the analytical pression of f_0, it can be seen that the draft tube natural frequency value is decreased with ɜ onset and the growth of the cavity volume. Moreover, resonance can occur, which leads to ɜ amplification of the pressure fluctuations either at the swirl precession frequency or at its ɜt harmonic and, then, this resonance can help in identifying f_0.

example of the influence of the discharge coefficient on the natural frequency of a model ancis turbine draft tube is shown in Figure 100 at part load. Sometimes the f_0 curve ersects the frequency curve of the swirl precession which corresponds to a resonance risk.

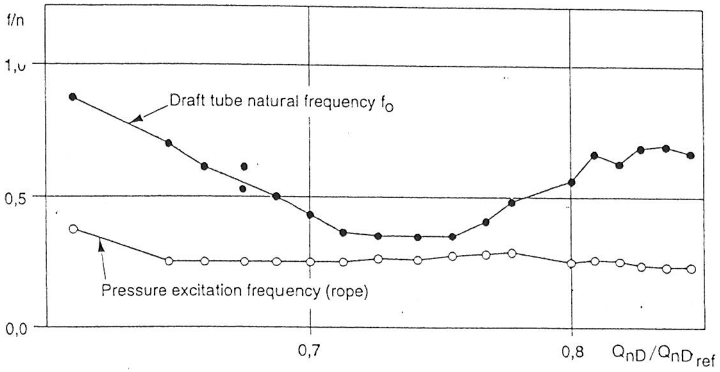


Figure 100 – Example of discharge coefficient influence on the natural frequency of a model Francis turbine draft tube

In case of doubt, a confirmation of this identification can be obtained by varying the following test parameters:

- the Thoma number (decreasing the Thoma number increases the volume of the cavity and leads to a decrease of f_0);
- the Froude number by varying the test specific hydraulic energy (decreasing the Froude number increases the volume of the cavity and leads to a decrease of f_0).

Two other ways can be also used for the determination of f_0 but they both involve more measuring time and the use of specific systems. They consist in:

- exciting the draft tube with a specific system which forces an external pressure fluctuation, and analyzing the frequency response of the pressure transducers;
- measuring the various volumes of cavity in the draft tube when varying operating discharge coefficient Q_{nD} and Thoma number value in order to estimate the frequency value with the analytical expression of f_0.

If the Thoma similitude and Froude similitude are fulfilled, the natural frequency of the draft tube can be transposed from the model to the prototype as follows:

$$\frac{f_{0M}}{f_{0P}} = \frac{n_M}{n_P} = \frac{D_P}{D_M} \cdot \sqrt{\frac{E_M}{E_P}}$$

4.3.8 Uncertainties

The uncertainty of the pressure fluctuation measured on the model is influenced by the following factors:

a) uncertainty of the instrumentation;

b) interaction with the test circuit;

c) interaction with the electric machine.

With good measurement conditions, it is possible to determine the pressure fluctuation of the model with uncertainties as shown below.

a) amplitude: ± 10 %;

b) dominant frequency: ± 2 % of the model rotation frequency;

c) phase difference: ± 10°.

Conversion from model to prototype value can involve error caused by the following factors:

a) deviation in Froude similitude;

b) interaction with water conduit;

c) interaction with the electric machine.

In case that significant influence caused by the above factors is involved, it is difficult to estimate the prototype values accurately. On the contrary, with favourable conditions, the uncertainties of the predicted values for the prototype are expected to be as follows:

a) amplitude: ± 30 %;

b) dominant frequency: ± 5 % of the prototype rotation frequency;

c) phase difference: ± 30°.

.4 Shaft torque fluctuations

.4.1 General

luctuations of shaft torque in hydraulic machines can be induced by:

) variations of the pressure forces acting on the runner/impeller blades;

) variations of the electro-magnetic forces acting on the generator/motor.

ıbservations of torque fluctuations are considered only as an extension to the processing of ressure fluctuations (see subclause 4.3). Torque fluctuations induced by the electric machine overning system can occur and shall be identified as such.

.4.2 Recommendations for measurement

he torque transducer shall be mounted on the shaft between the generator/motor and the ınner/impeller. See also subclause 4.5 and Figure 49 in subclause 3.6.4. Its frequency band hall cover the range of relevant frequencies (see 4.3.1.1).

either the speed governing system and power transmission of the test rig, nor the torsional atural frequency of model shaft shall produce significant contributions to shaft torque variation ithin the frequency band of interest.

.4.3 Analysis of model test results

orque fluctuations are processed and displayed in the same way as pressure fluctuations see 4.3.6). Amplitudes are made non-dimensional as for torque factors or coefficients (see .3.3.13.1 and 1.3.3.13.3). They are expressed in relative values. The reference can be the est efficiency or full load torque factor (or coefficient).

there is no dynamic influence from the generator/motor at the frequency of interest, torque uctuation amplitudes represent the global action of fluctuating pressure forces acting on the ınner.

.4.4 Transposition to prototype

orque fluctuation frequencies are transposable to the prototype if:

) they are the same as the pressure fluctuation frequencies;

) there is no dynamic influence from the generator/motor at the frequency of interest.

Iodel tests of a hydraulic machine will not reproduce the amplitudes of torque fluctuations on ıe prototype if they are significantly influenced by non-homologous boundary conditions: lectric circuit, mechanical assembly and hydraulic piping system.

ı particular, measured torque fluctuations in runaway or condenser operation are not relevant the action of the model governing system is clearly dominant.

4.5 Axial and radial thrust

4.5.1 General

This subclause deals with steady-state measurement of forces (thrust) and moments acting on the runner/impeller of hydraulic machines.

In most cases it is sufficient to measure only the axial thrust in the course of the regular testing program. For the measurement of radial forces and moments, special arrangements have to be installed. Test arrangements are described below for the measurement of one component (axial thrust) up to the measurement of six components (all forces and moments acting on runner/impeller).

The measurement of the fluctuations of these axial and radial forces is part of the measurement methods and arrangement described later on. The data processing of these fluctuations is generally made in the same way as for pressure fluctuations (see 4.3.6).

4.5.1.1 Purpose of measurements

The forces and moments due to hydrodynamic action are part of the overall load on the rotating part in a hydraulic machine. Important design parameters, such as forces acting on axial and radial bearings and embedded parts, stresses and deflections of the shaft, etc., can be computed and / or derived from model measurements.

The purpose of the measurements is to determine the magnitude and direction of forces and moments as function of the various operating conditions.

4.5.1.2 Definitions

The forces and moments acting on the runner/impeller are defined in a stationary Cartesian system of coordinates (see Figure 101).

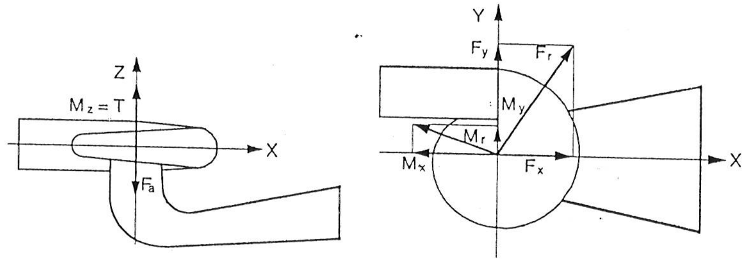


Figure 101 – Definition of coordinate system

The origin of the coordinates is determined by the reference level as specified in 1.3.3.7.6.

F_x radial force, x-component;

F_y radial force, y-component;

F_z axial force ($-\vec{F}_z = \vec{F}_a$ axial thrust);

$\vec{F}_r$ radial force $\left(\vec{F}_r = \vec{F}_x + \vec{F}_y\right)$;

M_x moment, x-axis;

M_y moment, y-axis;

M_z moment, z-axis (= shaft torque T);

M_r radial moment $\left(\vec{M}_r = \vec{M}_x + \vec{M}_y\right)$;

ıe angles of the radial force and radial moment can be calculated in the defined coordinate stem:

$$\varphi_{Fr} = \arctan\frac{F_y}{F_x} \qquad \varphi_{Mr} = \arctan\frac{M_y}{M_x}$$

5.1.3 Influencing effects

this subclause only the forces and moments caused by the hydrodynamic interaction ıtween the runner/impeller and the test fluid are considered. Therefore, forces due to the llowing effects shall also be taken into account:

- weight of runner/impeller;
- centrifugal forces;
- hydrostatic effects (buoyancy);
- hydrodynamic effects in labyrinth seals;
- mechanical forces (friction);
- resonance effects.

ıe following subclauses describe how the above effects should be considered in each ırticular case.

5.2 Axial thrust

5.2.1 Test program

ıe test should cover the whole operating range with special attention to the region where the aximum axial thrust occurs or is expected. The test shall be carried out at specified prototype ɔerating conditions. The number of test points between minimum and maximum specific ydraulic energy and from minimum to maximum discharge shall be sufficient to characterize ıe axial thrust. It is recommended that the test range extend beyond the contractually ɔecified operating range and that the possible influence of cavitation on axial thrust be ɔnsidered.

ı addition to the normal operating range, axial thrust should be measured for off-design onditions such as those listed below:

- runaway conditions;
- off-cam conditions in double regulated machines;
- operating points during transient conditions with expected high axial thrust (e.g. transition from pump to turbine mode at maximum guide vane opening in pumps and pump-turbines);
- no-load conditions which can cause up-thrust in axial machines.

4.5.2.2 Measuring arrangements

4.5.2.2.1 Direct measurement

Numerous arrangements have been developed to measure the axial thrust acting onto a runner/impeller. In a typical testing arrangement the oil pressure within a hydrostatic bearing is measured as a reference quantity related to the hydraulic force acting onto the runner/impeller along its axis of rotation (see Figure 102).

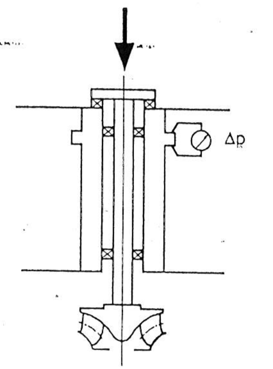


Figure 102 – Typical testing arrangement for axial thrust measurement

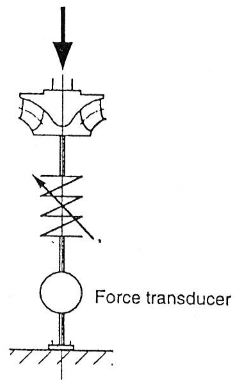


Figure 103 – Typical calibration arrangement for axial thrust measurement

Another typical arrangement uses the deflection of connecting parts between axial bearing and the housing which can be measured by means of strain gauges or inductive distance meters.

The axial thrust can also be determined with a measuring device for all six components of forces and moments (see Figure 105 d).

pending on the arrangement used, the following influencing effects shall be taken into count:

hydrostatic forces on the shaft;

oil viscosity;

gravity forces;

others.

5.2.2.2 Indirect determination

e axial thrust can be determined by a sufficient number of pressure measurements along the ter contour of the runner/impeller with the consideration of the calculated thrust component e to the change of momentum of flow. Corresponding pressure taps are shown in Figure 4. All those pressures should be referred to a reference pressure taken for instance in a ference section of the machine (see figure 1).

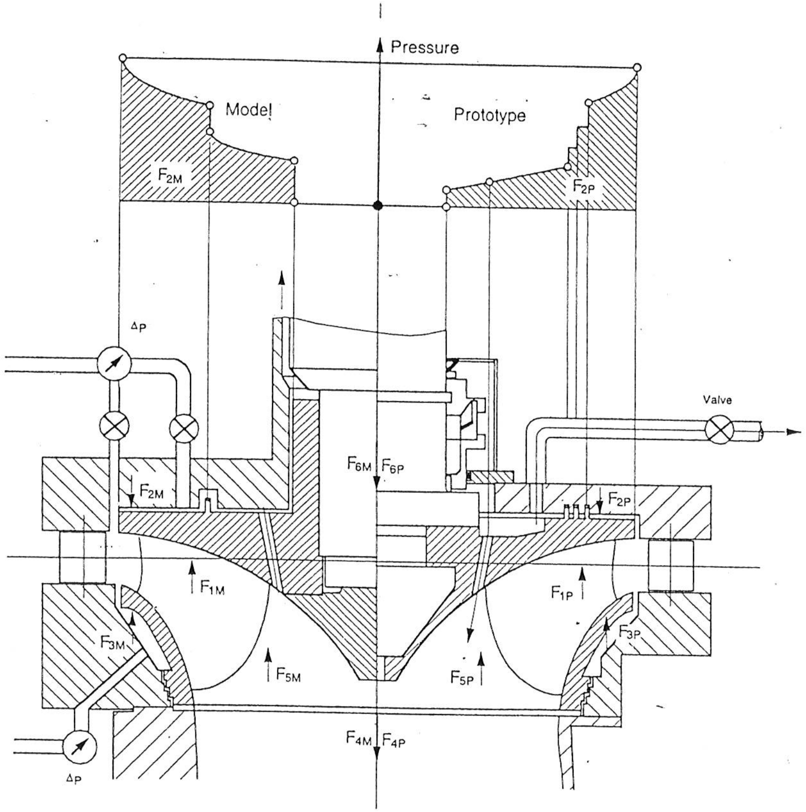


Figure 104 – Individual elements of axial force acting on a radial machine

4.5.2.3 Calibration

For calibration, a force is applied in axial direction to the model runner/impeller shaft. The magnitude of force is defined by:

- certified masses;
- masses together with a load cell;
- hydraulic jack together with a load cell (see Figure 103);
- others.

The reference forces applied to the shaft are related to the signal of the axial thrust measurement instrument in order to establish a calibration curve.

4.5.2.4 Checks during and after measurements

Before and after each test series, the measuring signal shall be recorded and checked at reference condition (e.g. at stand-still). It is recommended to check the axial thrust measurements varying the rotational speed at one operating point. This check indicates whether the calibration, compensation and evaluation are valid.

4.5.2.5 Transposition to prototype

The axial force $F_{a,M}$ measured on the model cannot always be directly transformed to prototype conditions. For a correct transposition, it is useful to consider separately the individual force elements of the total axial force for model and prototype.

Figure 104 and Table 9 show the axial force elements of total axial thrust in the example of a radial machine with a vertical shaft. The procedure as outlined can be applied to axial machines by omitting axial force elements on the runner band and crown.

Test data shall be corrected to account for all differences between the model and prototype regarding the runner/impeller (seal geometry, balance holes, balance pipes). Note that even complete geometrical similarity between model and prototype does not automatically provide similarity of flow in these regions.

The model axial thrust is composed of the following elements

$$F_{aM} = F_{1M} + F_{2M} + F_{3M} + F_{4M} + F_{5M} + F_{6M}$$

Hence

$$F_{1M} = F_{aM} - \left(F_{2M} + F_{3M} + F_{4M} + F_{5M} + F_{6M}\right)$$

The following dimensionless axial force factors/coefficients (see 1.3.3.13.2 and 1.3.3.13.4):

$$F_{1ED} = \frac{F_1}{D^2 \cdot \rho \cdot E} \qquad \text{axial force factor;}$$

$$F_{1nD} = \frac{F_1}{D^4 \cdot n^2 \cdot \rho} \qquad \text{axial force coefficient;}$$

permit the calculation of prototype axial force using the following equations:

$$F_{1P} = F_{1M} \cdot \left(\frac{D_P}{D_M}\right)^2 \cdot \frac{E_P}{E_M} \cdot \frac{\rho_P}{\rho_M} = F_{1ED} \cdot D_P^{\,2} \cdot \rho_P \cdot E_P$$

$$F_{1P} = F_{1M} \cdot \left(\frac{D_P}{D_M}\right)^4 \cdot \left(\frac{n_P}{n_M}\right)^2 \cdot \frac{\rho_P}{\rho_M} = F_{1nD} \cdot D_P{}^2 \cdot n_P{}^2 \cdot \rho_P \cdot E_P \, .$$

ıce for the axial thrust at prototype:

$$F_{aP} = F_{1P} + F_{2P} + F_{3P} + F_{4P} + F_{5P} + F_{6P}$$

prototype axial force can be determined also by a combined approach of measurement calculation. In this case the computer program used for calculation of the prototype axial e is calibrated by calculating the model axial force and subsequent comparison with model results. The same program can also be used to calculate the prototype axial force in case ıcreased crown pressure due to wear of runner/impeller seals.

Table 9 – Individual force elements of axial thrust and their treatment

Force element	Model	Prototype
Hydrodynamic [1)]	Determined from measurement by subtracting F_2 to F_6 from F_{aM}.	Transformed by using F_{1ED} or F_{1nD}.
Crown	Calculated from parabolic pressure distribution. Measurement of static pressure is advisable. Not existing for axial type machines.	Calculated from parabolic pressure distribution. Not existing for axial type machines.
Band	Calculated from parabolic pressure distribution. Not existing for axial type machines.	Calculated from parabolic pressure distribution. Not existing for axial type machines.
Runner/impeller weight	Determined by weighing or considered in calibration procedure; in case of inclined axis, only the axial component shall be considered.	Determined by calculation.
Runner/impeller buoyancy	Determination from runner/impeller volume $F_5 = V \cdot \rho_w g$ or considered in calibration procedure. In case of inclined axis, only the axial component shall be considered.	Determined by calculation. $F_5 = V \cdot \rho_w g$
Hydrostatic shaft force	Pressure acting on areas exposed to atmosphere.	Pressure acting on areas exposed to atmosphere.

.2.6 Uncertainty

ə uncertainty in determination of prototype axial force results from the uncertainty in model ıasurement and from approximations in transposition from model to prototype.

th good measurement conditions it is possible to determine the mean values of the model al hydraulic force with an uncertainty of less than ± (5 to 10) % of the extreme values ɔurring during normal operation conditions. The uncertainty in corresponding derived ɔtotype values is approximately ± (10 to 20) % of the maximum mean values.

F_1 results from the axial hydrodynamic forces acting on the runner/impeller flow passage.

4.5.3 Radial thrust

4.5.3.1 Test program

The magnitude and direction of radial thrust (forces and moments) are necessary to determine stresses and deflections of the shaft, bearing and adjacent structure. The measurements should cover all main operating ranges with particular attention to those operating points where extreme values of radial thrust occur (mean and/or fluctuating).

Typical examples of operating conditions where extreme radial forces can occur, are:

- runaway condition;
- zero discharge for pump;
- operating points associated with transient conditions;
- cavitation conditions.

4.5.3.2 Measuring arrangements

The radial thrust can be determined by measurement of the following quantities (see Figure 105):

- reaction forces in one or two shaft bearings;
- supporting forces of the bearing housing;
- deflection of the shaft;
- multidirectional strain in a special measuring section of the shaft close to the runner/ impeller.

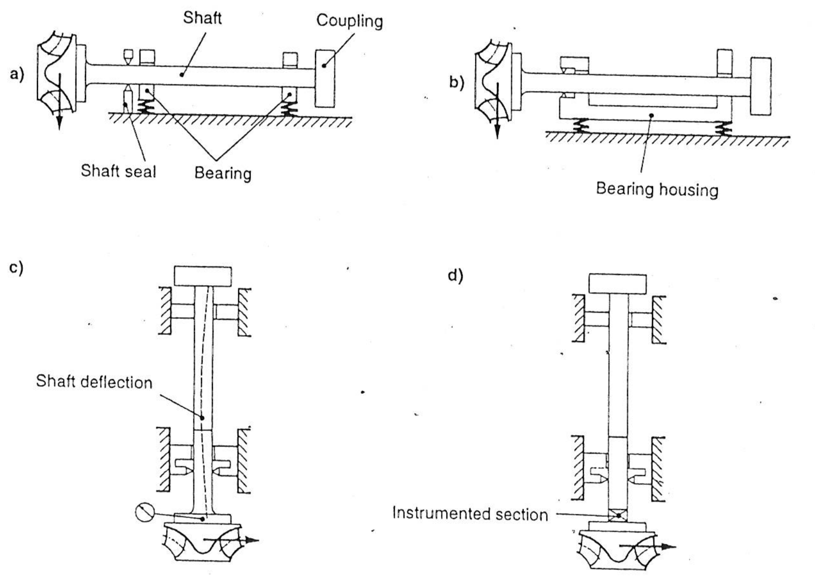


Figure 105 – Typical arrangements for radial thrust measurement (horizontal or vertical shaft).

rrangement d) of Figure 105 permits measurement of all six components of forces and ıoments $(F_x, F_y, F_z, M_x, M_y, M_z)$ in both the rotating and the stationary system. The signals of ıe strain gauge bridges are transferred from the shaft to the stationary part by means of slip ngs or a telemetry device. The angle of rotation is continuously measured and the signals are ansformed to stationary coordinates.

epending on the arrangement used, measurements of radial thrust can be affected by non-ydraulic forces as shown in Table 10.

leasurement of radial thrust can be significantly influenced by a hydrodynamic bearing effect f the radial seals. This influence can be reduced by using increased seal clearances in the ıodel for radial thrust measurement.

Table 10 – Non-hydraulic forces influencing radial thrust measurement

Origin of non-hydraulic forces		Arrangements of measurement			
		a	b	c	d
Shaft	(natural frequency)	x	x	x	x
Coupling	(inertia)	x	x	x	
Bearing housing	(inertia)		x		
Shaft seals	(reaction)	x		x	

.5.3.3 Calibration

he basic calibration is usually done by applying forces and moments on the shaft using a nown lever arm and certified masses. The actual forces and moments are related to the ɜadings of the respective instruments. If oil pressure is used to determine bearing forces, the il temperature shall be recorded and the influence of viscosity (if any) taken into account for valuation. If the measurement is done in the rotating system (arrangement d), the calibration hall be carried out in rotating condition.

ince only the hydrodynamic radial thrust shall be determined, the following effects shall be ıken into account by calibration:

weight of the runner/impeller;

buoyancy;

unbalance of rotating parts.

'he weight and buoyancy of the runner/impeller are only relevant if model and/or prototype do ot have a vertical shaft. The weight is taken into account by measuring the radial thrust at low rotation in air. Weight and buoyancy are taken into account by measuring at slow rotation ı water. The centrifugal force due to unbalance of the rotating parts is taken into account by ıeasuring the radial force at fast rotation in air.

.5.3.4 Checks before and during measurements

Before and after each test series, the measuring signal shall be recorded and checked at eference condition (e.g. at stand-still). It is recommended to check the radial thrust ıeasurements varying the rotational speed at one operating point. This check indicates vhether the calibration, compensation and evaluation are valid.

'he speed variation test can also show significant deviations for a certain speed of rotation vhich indicates a model-specific resonance effect.

4.5.3.5 Transposition to prototype

The radial force measured in the model can be transposed to prototype provided that all influencing effects are eliminated and only hydrodynamic forces are considered.

The following dimensionless radial forces and moment factors/coefficients (see 1.3.3.13):

$$F_{rED} = \frac{F_r}{D^2 \cdot \rho \cdot E} \quad \text{radial force factor}$$

$$M_{rED} = \frac{M_r}{D^3 \cdot \rho \cdot E} \quad \text{radial moment factor}$$

$$F_{rnD} = \frac{F_r}{D^4 \cdot n^2 \cdot \rho} \quad \text{radial force coefficient}$$

$$M_{rnD} = \frac{M_r}{D^5 \cdot n^2 \cdot \rho} \quad \text{radial moment coefficient}$$

permit the calculation of prototype radial force and moment by using the following equation:

$$F_{rP} = F_{rM} \cdot \left(\frac{D_P}{D_M}\right)^2 \cdot \frac{\rho_P}{\rho_M} \cdot \frac{E_P}{E_M} = F_{rED} \cdot D_P^2 \cdot \rho_P \cdot E_P$$

$$F_{rP} = F_{rM} \cdot \left(\frac{D_P}{D_M}\right)^4 \cdot \left(\frac{n_P}{n_M}\right)^2 \cdot \frac{\rho_P}{\rho_M} = F_{rnD} \cdot D_P^4 \cdot n_P^2 \cdot \rho_P$$

$$M_{rP} = M_{rM} \cdot \left(\frac{D_P}{D_M}\right)^3 \cdot \frac{\rho_P}{\rho_M} \cdot \frac{E_P}{E_M} = M_{rED} \cdot D_P^3 \cdot \rho_P \cdot E_P$$

$$M_{rP} = M_{rM} \cdot \left(\frac{D_P}{D_M}\right)^5 \cdot \left(\frac{n_P}{n_M}\right)^2 \cdot \frac{\rho_P}{\rho_M} = M_{rnD} \cdot D_P^5 \cdot n_P^2 \cdot \rho_P$$

Transposition of radial forces from model to prototype requires definition of the axial position of the plane in which the radial thrust is referred. Preferably this could be the plane corresponding to the defined reference level (see Figure 5, subclause 1.3.3.7).

The transposition from model to prototype shall also take into account non-homologous mechanical components and conditions (see 4.1.4).

4.5.3.6 Uncertainty

The uncertainty in prototype radial force depends on the model measurement uncertainty and the approximations in transposition from model to prototype.

Even with good measurement conditions, it is not possible to determine the mean values of model radial forces with uncertainties smaller than ± (5 to 10) % of the maximum mean values.

The corresponding prototype mean values of the radial forces cannot be determined with an uncertainty smaller than ± (10 to 20) % of the maximum mean value. The uncertainty of the fluctuations can be even higher.

Hydraulic loads on control components

.1 General

.1.1 Type of control components

ırder to control power or discharge, most types of hydraulic reaction machines are equipped ı:

ɡuide vanes, and/or

adjustable runner/impeller blades.

ı control elements of impulse turbines (e.g. Pelton turbines) are the nozzle(s) with ustable needle(s) and deflector(s).

netimes gates or valves can be used as control components, but they are not considered in following.

.1.2 Purpose of load measurements

ı purpose of such tests is to check or determine by means of model measurements the lraulic loads i.e. forces and torques acting on the control components of a machine. The del is always operated at steady state operating conditions (see 4.1.1).

ch model results (absolute or dimensionless values) can be used to:

check the extreme values of hydraulic loads which are relevant for the prototype design (mean values and fluctuating components);

establish at which operating conditions the fluctuating amount of a hydraulic load becomes important and which are the relevant excitation frequencies;

check at which operating conditions the hydraulic torque has opening or closing tendency;

produce input data for calculation of prototype loads during transient operating conditions;

determine torques acting on aligned and misaligned guide vanes as required for the design and adjustment of guide vane and guide vane protective devices.

.1.3 Design of control components for hydraulic loads measurements

e model components prepared for hydraulic load measurements are often of special design. strain gauges are used, the deformation of the instrumented component under hydraulic load ıst be such that sufficient strain is produced. This can be achieved by an appropriate design, wever the impact on the mechanical safety and the natural frequency of the measuring mponent must be checked. It is also important to check how much the model measurements ll be affected by friction forces or torques. If friction effects are not negligible, their effects all be eliminated from the final model test results because their relative value is different an on prototype.

ı a consequence of the design, it can be necessary to perform such tests at a reduced test ecific hydraulic energy, in order to avoid inadmissible hydraulic loads or resonance nditions. Therefore, it is important to determine the natural frequency in water of such easuring components.

Sometimes it is preferable to prepare additional measuring components which are only installed for hydraulic load tests, so that the main hydraulic performance tests can be done without limitations and without risk of damaging measuring components.

4.6.1.4 Signal processing

The measuring signal from calibration and/or measurement can be recorded manually or automatically. However, for extensive measurements with many test points or many measuring components, automatic data acquisition is recommended. This also facilitates the data-processing and display of tests results in the same way as for pressure fluctuation (see 4.3.6).

4.6.2 Guide vane torque

4.6.2.1 Number and position of measuring guide vanes

Experiences from many tests demonstrate that due to hydraulic and/or structural design, the inflow and outflow conditions for distributors of reaction machines can change along the circumference of a spiral case or of an intake of a tubular turbine. Therefore, the torque shall be measured on several guide vanes located at representative circumferential positions:

a) for a spiral case: two in the zone influenced by the nose vane and one opposite to it;

b) for a semi-spiral case: more than three measuring guide vanes can be required;

c) for a tubular turbine: two in the zone influenced by the bulb support or by the pit and one in the perpendicular direction.

When similar test data already show the circumferential influence, it can be agreed that only one or two measuring guide vanes are used.

When the number of stay vanes and guide vanes are different, the torque shall be measured on two adjacent guide vanes, unless data of similar arrangements are available which show no significant differences between two adjacent guide vanes.

If the impact of a misaligned guide vane on the neighboring guide vane torques shall be established, it is necessary to measure the torque on three adjacent guide vanes, whereby the one in the middle is misaligned.

4.6.2.2 Number of test points

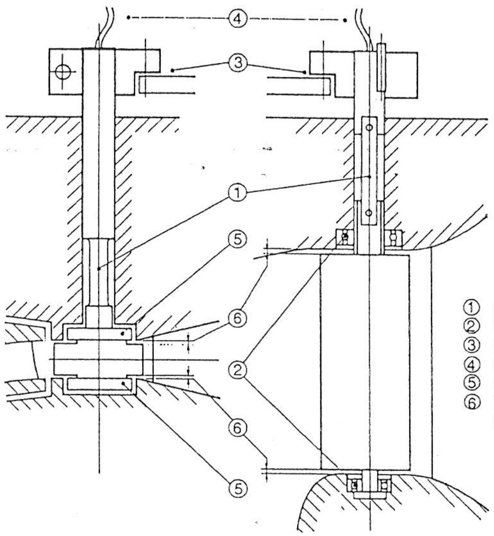
The operating conditions and the number of test points depend strongly on the type of the machine and the purpose of the test. In turbine operation the main governing parameter is the discharge factor or guide vane opening. The speed factor influence is small and practically negligible for high specific speed turbines. It is often sufficient to measure the guide vane torque only at the limits of the operating range defined by the specified values E_{nDmax} and E_{nDmin} (see Figure 107). For higher specific speed Francis turbines and for axial turbines, it is sufficient to measure the guide vane torque only at one constant specified value E_{nD} within the operating range. In this case, the evaluation shall be made by using a guide vane torque factor $T_{G,QD}$ defined in 4.6.2.6.

The number of test points shall be increased if the extended operating range is to be investigated. Transient analysis requires a sufficient number of test points.

If guide vane torques are measured with a misaligned guide vane, it is important to agree on the possible geometric configurations and hydraulic operating conditions so that the measuring program does not become too extensive.

4.6.2.3 Measuring arrangement

In most cases the hydraulic torque acting on a guide vane is determined by measuring the torsional deformation of the guide vane stem using bonded strain gauges. The upper end of the measuring stem is fixed into the adjusting mechanism of the guide vane. It is possible to use the normal guide vane stem, often with a reduced diameter, or to replace the normal stem by a special measuring stem, sometimes of a different material with a suitable elastic modulus and load hysteresis. Figure 106 illustrates two typical design examples.



① Stem with strain gauge bridge
② Ball bearing
③ Guide vane adjusting mechanism
④ Connecting cable for signal transmission
⑤ Guide vane disk
⑥ End clearance of instrumented guide vane

Figure 106 – Design examples for torque measuring guide vanes

In order to avoid disturbing effects by friction due to lateral and/or axial hydraulic forces, special attention shall be paid to the design of the support of the measuring stem (for example ball bearings can be applied) and to the guide vane end clearances, which can be increased for the torque measurement. The design of the guide vane disks (if any) shall be such that during measurement no disturbing axial thrust can result.

If the measuring section is not located in dry air, it is important that a good surface protection against humidity is applied. The electric insulation resistance should be checked periodically.

4.6.2.4 Calibration

For calibration, static loads are usually applied on the measuring guide vane in opening and closing directions by means of a known lever arm and certified masses. The calibration can be made with the guide vane installed in the model machine or by installing the guide vane in a special calibration device.

The stability of the no-load operating point and the hysteresis due to loading and unloading shall be checked, as well as the drift of the measuring signal under constant load. The result of calibration is usually an averaged calibration curve with the output signal versus calibration torque.

4.6.2.5 Checks before and during measurement

It is recommended that after installation of the guide vane in the model, a check loading is made (especially when the guide vane was calibrated in a separate calibration device) in order to prove the correct mechanical installation (no friction) and to check the processing of the measuring signal and the sign for opening and closing tendency.

Preliminary tests including a rotational speed variation shall demonstrate that filtering the noise of the measuring signal does not affect the signal itself and that resonance conditions are avoided. Before and after each test series, the measuring signal at zero-load shall be recorded and checked.

In pump mode it should be possible to identify the impeller blade passing frequency as dominating excitation frequency. This can also be the case for pump-turbines in turbine mode.

4.6.2.6 Calculation of dimensionless torque factors and prototype guide vane torques

For each operating point a mean value T_G is determined which can be used to calculate a dimensionless guide vane torque factor $T_{G,ED}$ or $T_{G,QD}$.

Definition of guide vane torque factors (see 1.3.3.13.1):

$$T_{G,ED} = \frac{T_G}{\rho \cdot D^3 \cdot E}$$

or

$$T_{G,QD} = \frac{T_G \cdot D}{\rho \cdot Q^2}$$

Based on hydraulic similarity conditions, the prototype guide vane torque can be calculated either using one of the torque factors or the absolute model values.

$$T_{G,P} = T_{G,M} \cdot \left(\frac{D_P}{D_m}\right)^3 \cdot \frac{E_P}{E_M} \cdot \frac{\rho_P}{\rho_M} = T_{G,ED} \cdot D_P{}^3 \cdot E_P \cdot \rho_P$$

or

$$T_{G,P} = T_{G,M} \cdot \left(\frac{Q_P}{Q_M}\right)^2 \cdot \frac{D_M}{D_P} \cdot \frac{\rho_P}{\rho_M} = T_{G,QD} \cdot Q_P{}^2 \cdot \frac{1}{D_P} \cdot \rho_P$$

It is to be noted that the resulting hydraulic prototype guide vane torque does not include friction in the guide vane bearing, seals and adjustment mechanism. Usually the bearing friction torque is rather small due to vibrations of the guide vanes, whereas the friction in the adjusting mechanism linking guide vanes and servo motor(s) is more important. The total amount of prototype friction torque is to be determined by calculation or by experience from site measurements.

6.2.7 Graphical presentation of results

r normal operation in both turbine mode or pump mode the guide vane torque or the rresponding factor is usually represented versus the guide vane angle at defined hydraulic nditions (e.g. at a selected $E_{P,sp}$). Moreover, in pump mode the relevant value of the guide ne torque or of the corresponding factor usually corresponds to the envelope curve E_{nD} rsus Q_{nD}.

uide vane torques occurring in one or more quadrants (see 4.7.2) are usually measured at veral constant guide vane angles and are presented as a function of the speed factor n_{ED} or scharge factor Q_{ED}.

camples of test results are presented in Figure 107, Figure 108 and Figure 109.

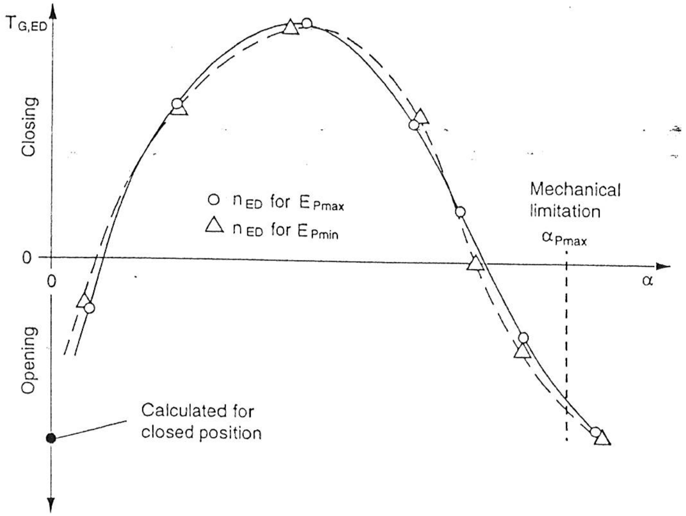


Figure 107 – Guide vane torque factor versus guide vane angle measured at different constant specific hydraulic energies in turbine mode

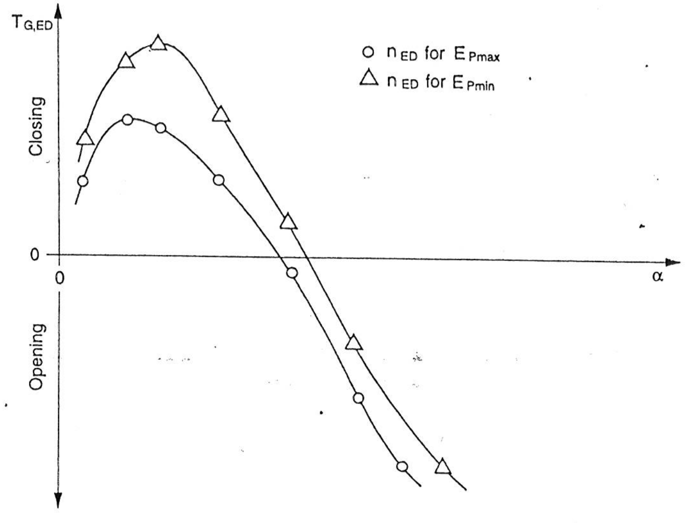


Figure 108 – Guide vane torque factor versus guide vane angle measured at different constant specific hydraulic energies in pump mode

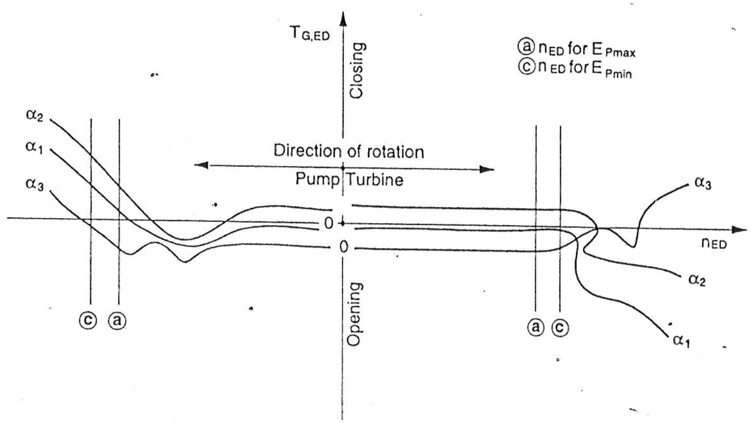


Figure 109 – Guide vane torque factor versus speed factor measured at different constant guide vane angles in the four quadrants of a pump-turbine

;.2.8 Comparison of guide vane torque fluctuations on model and prototype

most cases the guide vane torque fluctuations measured on the model should not be directly aled up to prototype conditions without applying corrections because of the following pects:

the hydroelastic similarity laws are not fulfilled;

damping effects due to water added mass and bearing design are different;

the ratios of excitation frequencies and natural frequencies are different.

;.2.9 Uncertainty

e uncertainty of model guide vane torque results is influenced by the following factors:

hysteresis of the calibration curve of the instrumented guide vanes;

repeatability and drift of the zero torque point;

influence of any friction effects.

th good measurement conditions, it is possible to determine the mean values of model draulic torque with an uncertainty of about ± 5 % of the maximum mean values. The certainty in corresponding derived prototype values is about ± (5 to 10) % of the maximum ototype mean value.

nplitudes of guide vane torque fluctuations of the model in normal operation can be termined with an uncertainty smaller than about ± 10 %, whereas predicted prototype values n have an uncertainty up to ± (50 to 100) %.

e zero torque guide vane angles, if any, can be determined with an uncertainty smaller than l° on the model and can be predicted for the prototype with an uncertainty of about ± 2°. This ro torque angle can vary along the circumference depending on the variation of inflow and/or tflow conditions.

;.3 Runner blade torque

;.3.1 Number of instrumented blades

case of diagonal and axial type machines with adjustable runner/impeller blades, the runner ade torque is usually measured on one blade.

;.3.2 Number of test points

e number of test points depends on the purpose of the test. For a double regulated machine e runner blade torque depends on the runner blade angle, guide vane angle and on the erating point. During normal operation, runner blade and guide vane angles are at on-cam sitions. In transient conditions, numerous off-cam positions occur. Therefore, it is necessary measure enough test points for checking the runner blade torque values occurring in off-cam eration.

4.6.3.3 Test procedure

In most cases such tests are performed at high Thoma number values, i.e. $\sigma > \sigma_{pl}$. However, at large discharge values, cavitation on the runner blades can affect the resulting blade torque. Therefore, at some selected operating points, the impact of variable σ on runner blade torque can also be checked.

On most models the runner blade angle is adjusted manually at stand still. As a consequence, runner blade torque tests are performed at selected constant runner blade angles whereas a hydraulic parameter, e.g. the speed factor n_{ED}, and the guide vane angle are varied systematically, covering the whole range of interest.

4.6.3.4 Measuring arrangement

Usually the hydraulic torque acting on a runner or impeller blade is determined by measuring the torsional deformation of the blade trunnion using bonded strain gauges. The end of the trunnion is fixed inside the hub. It is possible to use the normal trunnion, often with a reduced diameter, or to replace the normal trunnion by a special measuring trunnion, sometimes of a different material with a suitable elastic modulus and load hysteresis. Figure 110 illustrates a design example using telemetry.

In order to avoid disturbing effects by friction, the measuring trunnion can be supported by ball bearings, and the blade-hub and tip clearances can be somewhat increased.

The arrangement shall be such that only the torsional strain is measured.

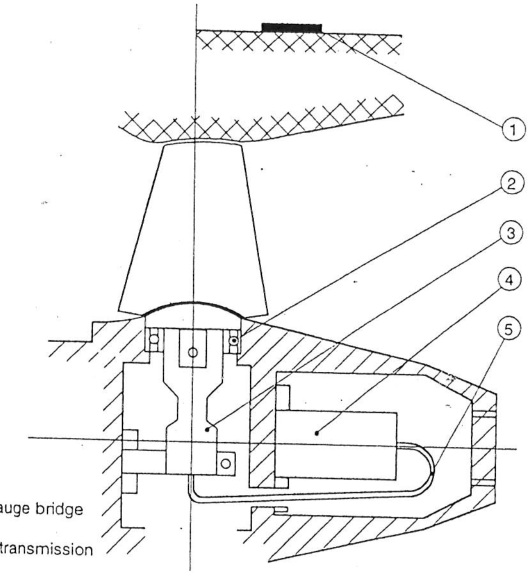
If the measuring section is not located in dry air, it is important that a good surface protection against humidity is applied. The electric insulation resistance should be checked periodically.

The measuring signal from the hub is transmitted either through the shaft to slip rings or by telemetry to the recording device outside the model machine.

4.6.3.5 Calibration

For calibration, static loads are usually applied on the measuring runner blade in opening and closing directions by means of a known lever arm and certified masses. The calibration can be made with the instrumented blade installed in the hub or by installing the instrumented blade in a special calibration device.

The stability of the zero torque point and the hysteresis due to loading and unloading shall be checked, as well as the drift of the measuring signal under constant load. The result of calibration is usually an averaged calibration curve with the output signal versus the calibration torque.



① Receiver antenna
② Ball bearing
③ Clamped stem with strain gauge bridge
④ Transmitter device
⑤ Connecting cable for signal transmission

Figure 110 – Example for runner blade torque measuring arrangement using telemetry

6.3.6 Checks before and during measurement

is recommended that after installation of the instrumented runner blade in the model, a check ading is made at standstill (especially when the runner blade was calibrated in a separate ılibration device) in order to prove the correct mechanical installation (no friction) and to ıeck the processing of the measuring signal and the sign for opening and closing tendency.

·eliminary tests, including a rotational speed variation, shall demonstrate that filtering the ɔise of the measuring signal does not affect the signal itself and that resonance conditions ·e avoided. Before and after each test series the measuring signal at zero-load shall be ·corded and checked.

6.3.7 Consideration of centrifugal effects

he measured torque is a result of the hydraulic pressure distribution around the blade and of ıe momentum due to the resulting centrifugal force which normally does not act in direction of ıe blade-trunnion centre-line. When different blade materials are used for model and ·ototype, it is necessary to separate both torque components. It is recommended to determine ıe blade torque due to centrifugal effects separately, by means of a test with the runner ɔinning in air with different rotational speeds and blade angles or by appropriate calculation.

Consequently, for each operating point the hydrodynamic blade torque T_{Bh} is the difference between the measured torque T_{Btot} and the torque due to centrifugal effects T_{Bc}:

$$T_{Bh} = T_{Btot} - T_{Bc}$$

4.6.3.8 Calculation of dimensionless torque factors and prototype hydrodynamic runner blade torque

For each operating point a mean value T_{Bh} is determined as the difference between the measured torque T_{Btot} and the torque due to centrifugal effects T_{Bc}. T_{Bh} can be used to calculate a dimensionless torque factor $T_{Bh,ED}$.

Definition of blade torque factors/coëfficients (see 1.3.3.13):

Hydraulic blade torque factor:

$$T_{Bh,ED} = \frac{T_{Bh}}{\rho \cdot D^3 \cdot E}$$

Runner blade torque coefficient due to centrifugal effects:

$$T_{Bc,nD} = \frac{T_{Bc}}{\rho_B \cdot n^2 \cdot D^5} \quad \text{where } \rho_B \text{ is the density of blade material}$$

Based on hydraulic similarity conditions, the prototype runner blade torque can be calculated either using the torque factors/coefficients or the absolute model values.

Hydraulic prototype runner blade torque:

$$T_{Bh,P} = T_{Bh,M} \cdot \left(\frac{D_P}{D_M}\right)^3 \cdot \frac{E_P}{E_M} \cdot \frac{\rho_P}{\rho_M} = T_{Bh,ED} \cdot D_P{}^3 \cdot E_P \cdot \rho_P$$

Prototype runner blade torque due to centrifugal effects:

$$T_{Bc,P} = T_{Bc,M} \cdot \left(\frac{D_P}{D_M}\right)^5 \cdot \frac{n_P^2}{n_M^2} \cdot \frac{\rho_{BP}}{\rho_{BM}} = T_{Bc,nD} \cdot D_P{}^5 \cdot n_P^2 \cdot \rho_{BP}$$

Total prototype runner blade torque:

$$T_{Btot,P} = T_{Bh,P} + T_{Bc,P}$$

It is to be noted that the hydraulic prototype runner blade torque does not include friction in the runner blade bearings and seals nor in the adjusting mechanism. Usually the bearing friction torque is rather small due to vibrations of the runner blades, whereas the friction in the adjusting mechanism linking runner blades and servo motor(s) is more important. The total amount of prototype friction torque is to be determined by calculation or by experience from site measurements.

4.6.3.9 Graphical presentation of results

For normal turbine mode operation the runner blade torque or the corresponding factor is usually represented for each runner blade angle as a function of the speed factor n_{ED} or discharge factor Q_{ED} for different constant guide vane openings. An example of test results is presented in Figure 111.

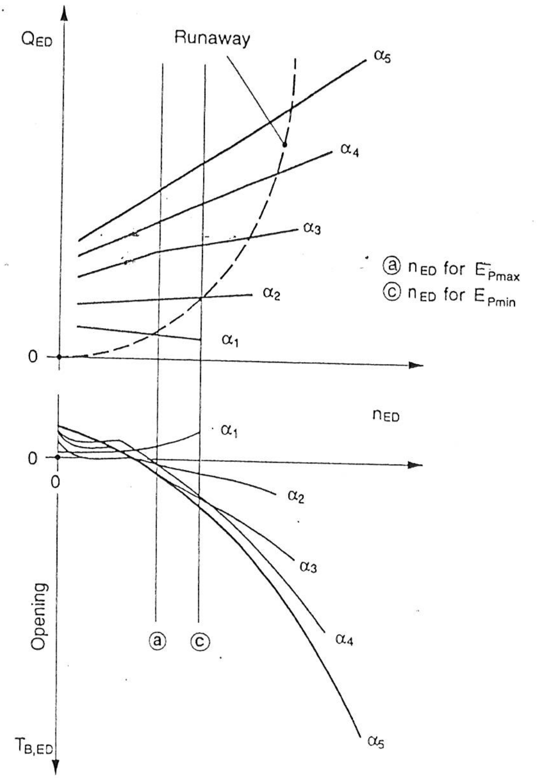
.3.10 Comparison of runner blade torque fluctuations on model and prototype

most cases the runner blade torque fluctuations measured on the model should not directly scaled up to prototype conditions without applying corrections because of the following)ects:

the hydroelastic similarity laws are not fulfilled;

damping effects due to water added mass and bearing design are different;

the ratios of exciting frequencies and natural frequency are different.



'igure 111 – Performance and hydraulic runner blade torque characteristics of an axial turbine measured at one constant runner blade angle β and various constant guide vane angles α

4.6.3.11 Uncertainties

The uncertainty of model runner blade torque results is influenced by the following factors:

a) hysteresis of the calibration curve of the instrumented runner blade;

b) repeatability and drift of the zero torque point;

c) influence of any friction effects;

d) influence of cavitation.

With good measurement conditions, it is possible to determine the mean values of model hydraulic torque with an uncertainty smaller than about ± 5 % of the maximum mean values. The uncertainty in corresponding derived prototype values is about ± (5 to 10) % of the maximum prototype mean values.

Amplitudes of runner blade torque fluctuations on the model in normal operation can be determined with an uncertainty smaller than about ± 10 %, whereas predicted prototype values can have an uncertainty of up to about ± (50 to 100) %.

4.6.4 Pelton needle force and deflector torque

In most cases it is not necessary to measure needle forces and deflector torques because either sufficient data are available from model or site tests or the data can be calculated.

4.6.4.1 Number of measuring needles and deflectors

In multi-jet Pelton turbines, it is sufficient to measure the hydraulic load on only one needle and/or one deflector, because the flow differences between the various nozzles are not significant for these measurements.

4.6.4.2 Test procedure

The needle force and the deflector torque mainly depend on their position and on the discharge. There is also an impact of the splashing water in the housing on the deflector torque. The speed factor of the runner has no influence on the needle force and only a small influence on the deflector torque. Needle force and deflector torque can therefore be determined by one test series, where the stroke is varied systematically from closed to fully open position, and at some selected stroke positions the deflector angle is also varied.

4.6.4.3 Measuring arrangement

a) Needle force: the hydraulic force acting on the needle can be directly measured by means of load cell fixed inside the inner servomotor and mechanically connected to the rod end of the needle. It is also possible to extend the needle rod through the distributor such that the load cell can be fixed outside. In addition, the static pressure should be recorded in the nozzle body to consider a possible force resulting from different diameter ratios between nozzle opening and needle rod on model and prototype.

b) Deflector torque: the hydraulic torque acting on the deflector can be determined by measuring deformations with strain gauges applied at its supporting structure or adjusting mechanism. In case of horizontal shaft Pelton turbines, deflector torque includes both hydraulic torque and gravitational torque acting on the deflector. Therefore, the gravitational torque should be measured beforehand in air to obtain the proper hydraulic torque from the measured torque.

ıe design shall be such that the force or torque measurement is not significantly affected by ction effects. If the measuring section is not located in dry air, it is important that a good ırface protection against humidity is applied. The electric insulation resistance should be ıecked periodically.

6.4.4 Calibration

·eferably the calibration is made with the same measuring arrangement as used for the tests order to check and/or to consider the friction forces of the bearing and seal system.

Needle force: the load cell installed on the needle rod is loaded statically with certified masses.

Deflector torque: provisions shall be made to apply on the deflector known forces or torques using a calibrated load cell or certified masses. In case of a force, it is important to determine its direction and the distance to the pivot (length of the lever arm).

6.4.5 Checks before and during measurement

əfore and after each test series, the measuring signal at standstill (nozzle not completely ɔsed, deflector not touching its stop) shall be recorded and checked. Preliminary tests shall ɛmonstrate that the force factor and/or torque factor are not dependent on the test rotational ıeed nor on the test specific hydraulic energy.

6.4.6 Calculation of dimensionless force and torque factors and prototype data

ır each operating point, a mean value F_N and/or T_D is determined which can be used to ılculate a dimensionless needle force factor $F_{N,ED}$ and/or a dimensionless torque factor $T_{D,ED}$.

əfinition of needle force factor (see 1.3.3.13.2):

$$F_{N,ED} = \frac{F_N}{\rho \cdot D^2 \cdot E}$$

əfinition of deflector torque factor (see 1.3.3.13.1):

$$T_{D,ED} = \frac{T_D}{\rho \cdot D^3 \cdot E}$$

ısed on hydraulic similarity conditions, the prototype needle force and/or deflector torque can ፥ calculated either using the force and/or torque factors or the absolute model values.

ılculation of hydraulic prototype needle force:

$$F_{N,P} = F_{N,M} \cdot \left(\frac{D_P}{D_M}\right)^2 \cdot \frac{E_P}{E_M} \cdot \frac{\rho_P}{\rho_M} = F_{NED} \cdot D_P{}^2 \cdot E_P \cdot \rho_P$$

needle rod diameters on model and prototype are non-homologous, a correction shall be ɔplied to get the total force F_{NtotP}.

Calculation of hydraulic prototype deflector torque:

$$T_{D,P} = T_{D,M} \cdot \left(\frac{D_P}{D_M}\right)^3 \cdot \frac{E_P}{E_M} \cdot \frac{\rho_P}{\rho_M} = T_{D,ED} \cdot D_P{}^3 \cdot E_P \cdot \rho_P$$

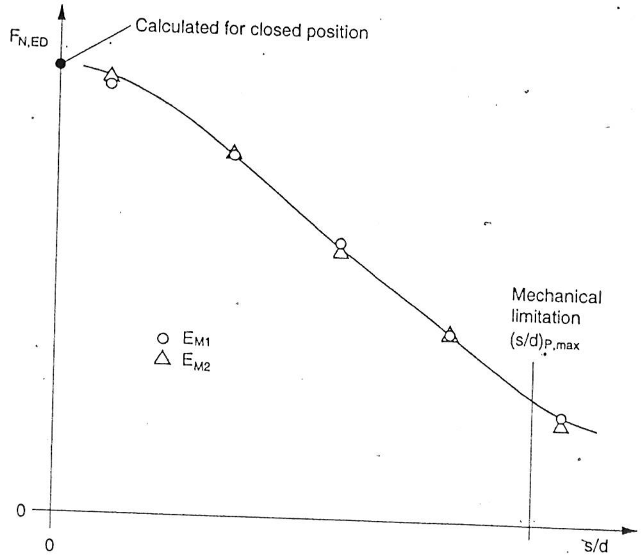
4.6.4.7 Graphical presentation of results

Usually the needle force factor is represented as a function of the needle stroke s referred for example to the nozzle outlet diameter: $F_{N,ED} = f(s/d)$. At closed position of the nozzle, s = 0. An example is presented in Figure 112.

Usually the deflector torque factor is represented as a function of the deflector position.

4.6.4.8 Comparison of fluctuations of needle force and deflector torque on model and prototype

Because there is no impact of the flow in the runner on the needle, the fluctuations of the needle force are negligible. At fully opened position, the deflector is exposed to splashing water from the runner inducing torque fluctuations. At deflecting positions, torque fluctuations are mainly due to splashing water of the deflected jet. Since in most cases the hydraulic similarity with respect to the two-phase flow in the housing is not fulfilled, deflector torque fluctuations measured on the model should not be directly scaled-up to prototype conditions.



s : needle stroke
d : nozzle outlet diameter

Figure 112 – Pelton needle force factor as a function of relative needle stroke

6.4.9 Uncertainty

ıe uncertainty in the measurement of model needle force and deflector torque is influenced ᐟ the following factors:

- hysteresis of the calibration curve of the instrumented components;
- repeatability and drift of the zero force or torque point;
- influence of any friction effects.

ith good measurement conditions, the mean values of the model deflector torque or of the ›edle force can be determined with an uncertainty of about ± 5 % of the maximum mean ılues. The uncertainty in corresponding derived prototype values could be about ± (5 to 10) % the maximum prototype mean value.

7 Testing in an extended operating range

7.1 General

addition to the determination of hydraulic characteristics of a machine in a limited range of ›ecific hydraulic energies and discharges, it is also important to know its complete ıaracteristics covering possible operating conditions outside the normal operating range. The ost extended field of operation exists for pumps and pump-turbines due to two directions of scharge and two directions of rotation (four-quadrants operation).

ꜱ stated in 4.1.1, model tests do not reproduce transient operation of the prototype. However, e obtained data are meaningful and are a necessary input for the calculations of transient ıenomena and loads for prototype design.

7.2 Terminology

comprehensive representation of hydraulic characteristics of a hydraulic machine is given by e four-quadrant-diagram.

7.2.1 Definition of the quadrants

ıe four quadrants are defined by the combination of the positive or negative direction of scharge and rotational speed as follows (see Table 11, Figure 113 and Figure 114).

Table 11 – Definition of quadrants and operating modes

Quadrant		Direction (sign) of				Mode
Number	Name	Q	n	E	T	
1	Pump quadrant	-	-	-	-	Reverse turbine
				-	+	Brake
				+	+	Pump
1/2		0	-	+	+	Zero discharge
2	Brake quadrant	+	-	+	+	Pump-brake
2/3		+	0	+	+	Zero speed
3	Turbine quadrant	+	+	+	+	Turbine
				+	0	Runaway
				+	-	Turbine-brake
				-	-	Reverse rotation pump (axial machines only)
3/4		0	+	+/-	-	Zero discharge
4	Reverse pump quadrant	-	+	+	-	Reverse rotation pump (radial machines only)
				-	-	Brake
4/1		-	0	-	-	Zero speed

In each of these four quadrants several modes of operation are possible due to:

a) the sign of power (output/input), corresponding to the direction of torque;

b) the sign of specific hydraulic energy in special cases of application (e.g. tidal power plant).

In the following subclause 4.7.2.2 the normal cases of hydroelectric application with positive value of E are described.

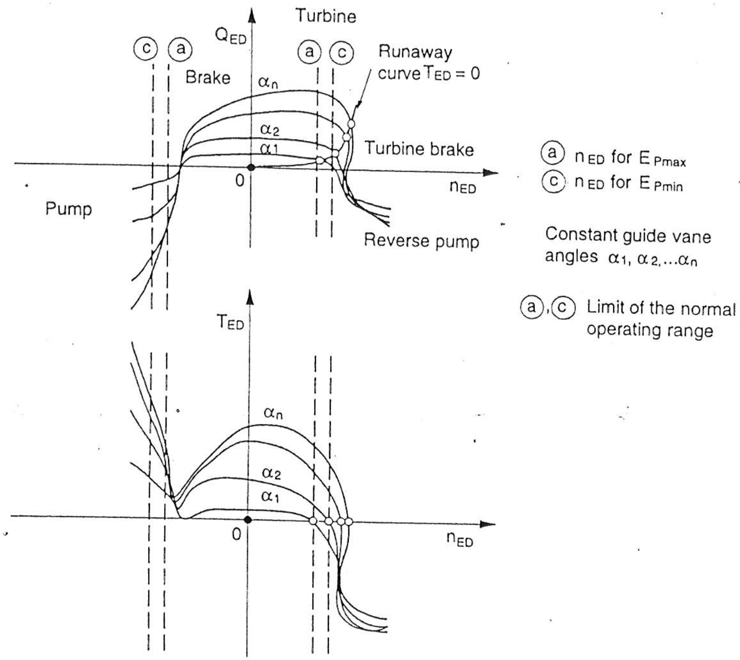


Figure 113 – Example of four quadrants operation of a radial-type pump-turbine

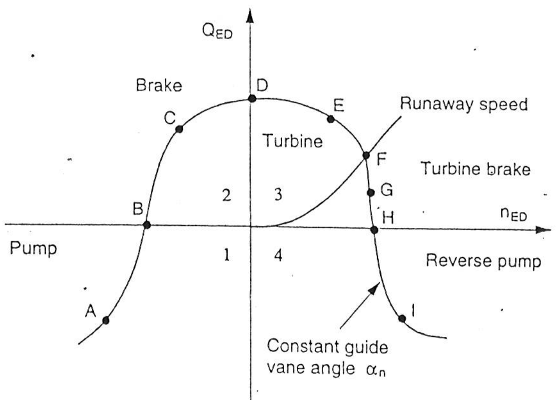


Figure 114 – Chart illustrating the various operating modes

4.7.2.2 Operating modes

4.7.2.2.1 Pump mode

The pump mode is characterized by negative discharge and negative rotational speed (see operating point A on Figure 114).

4.7.2.2.2 Pump brake mode

This mode is characterized by negative direction of rotation but positive direction of discharge (see point C). This mode is of importance in case of power failure in pump operation.

4.7.2.2.3 Turbine mode

This mode has positive direction of discharge and rotational speed and a positive torque is delivered to the machine shaft (see point E). The particular case of zero torque corresponds to turbine runaway (see point F).

4.7.2.2.4 Turbine brake mode

This mode has positive direction of discharge and rotational speed but negative torque (see point G).

4.7.2.2.5 Reverse rotation pump mode

This mode is characterized by a positive direction of rotational speed. However, the direction of discharge is negative (see point I). This mode can only be reached in transient condition.

4.7.2.3 Operation at zero speed and zero discharge

Besides the operating modes within the quadrants, the operating points at the axes of coordinate are of interest (see Figure 114):

– the zero-discharge specific hydraulic energy E_0 in pump mode (see point B);
– the zero-speed discharge and break-away torque (see point D);
– the zero-discharge specific hydraulic energy in turbine mode (see point H).

4.7.3 Scope of tests

4.7.3.1 Relevant modes of operation

Depending on the type of hydraulic machine one up to four quadrants are of interest:

a) impulse turbine: only turbine mode and turbine brake mode exist because flow and speed have both obviously positive direction;
b) reaction turbine: depending on specific speed and on guide vane opening, reverse rotation pump mode can reach importance in addition to normal turbine mode, particularly during load rejection and start up;
c) axial-type pump-turbine: operation in three quadrants is feasible. Besides quadrants for normal pump and turbine modes, the brake quadrant can be passed during transient conditions. Reverse pump quadrant cannot be reached;
d) radial-type pump or pump-turbine: operation in all four quadrants is feasible. Besides quadrants for normal pump and turbine modes, the turbine brake mode and even the reverse pump mode due to the radial extension of runner can be reached during transient conditions.

'.3.2 Performance data

performance data achieved during testing within normal operating range are taken also ring four quadrant testing. The main hydraulic quantities to be measured are:

specific hydraulic energy;
discharge;
shaft torque;
rotational speed;
net positive suction specific energy.

e intervals between the measured points can be considerably increased compared to these the guaranteed range of operation. On the other hand, it is recommended to extend the sting range to the maximum possible guide vane and/or runner/impeller blade openings which uld be useful for the future operation of the prototype.

draulic similarity laws still apply, however, secondary effects can change considerably the aracteristics found in model testing. Particular attention has to be paid to the influence of vitation on the hydraulic characteristics in extreme conditions.

'.3.3 Additional hydraulic data

addition to the above listed main hydraulic data, specific measurements can be relevant. The tent of test should be clearly defined prior to the test:

axial thrust;
radial thrust;
pressure fluctuations;
hydraulic guide vane torque;
runner blade torque;
needle force;
deflector torque;
shaft torque fluctuations.

ecific requirements for above tests are discussed in subclauses 4.2 through 4.6 of this andard. Some of them are measured simultaneously with the main hydraulic data. Some of em need to have special arrangements and data acquisition systems.

7.4 Provisions for particular tests

ere are no particular arrangements necessary for model tests outside the normal operating nge. However, as there is no demand to get these hydraulic data with the same high curacy as for the normal range, the specific hydraulic energy for these tests can be reduced order to protect the model and the measurement system.

.7.4.1 S-shape characteristics in turbine brake mode

epending on the specific speed of a hydraulic reaction machine, the Q_{ED} - n_{ED} characteristics t constant guide vane opening can be S-shaped (see Figure 115). Testing under steady state onditions can then be difficult.

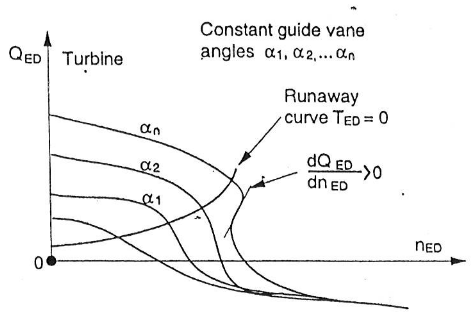


Figure 115 – S-shape characteristics in turbine brake mode

4.7.4.2 Pump characteristic

Depending on the design of a pump or pump-turbine, the slope of the $E_{nD} - Q_{nD}$ characteristic can be positive in a limited range of discharge (see Figure 116). This positive slope can cause instability in operation. Therefore, this zone should be explored carefully, particularly considering maximum specific hydraulic energy under transient conditions.

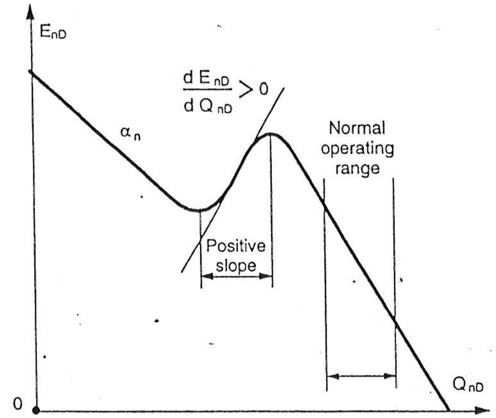


Figure 116 – Pump characteristic with positive slope in a limited discharge range

4.7.4.3 Runaway

For this type of testing a reduced specific hydraulic energy is recommended in order to protect the measurement equipment. Also, deviations from a full homology with increased clearances between rotating and stationary parts of the model machine are acceptable for the same reason. Special attention should be paid to the influence of Thoma number σ on the runaway characteristics particularly with high specific speed machines.

7.4.4 Zero-discharge specific hydraulic energy

ı keep the discharge exactly zero over the full range of guide vane opening, the test circuit ıs to be interrupted by e.g. a valve or a blind flange.

ıportant information at zero-discharge specific hydraulic energy are the power input and also e amplitude and frequency of pressure pulsation, while the runner/impeller is spinning in ıter.

can be also of interest to test the power for an impeller spinning in air. The corresponding rques are low, so a high sensitivity torque measuring system is required. Measurements of ıaft torque of an impeller spinning in air show increased uncertainties of up to ± 10 %. ırthermore, the conversion of test results to the prototype conditions is questionable due to e low Reynolds number in the model and the difference of Froude number.

ıother very specific scope of tests consists in measuring the shaft torque of the nner/impeller spinning in air with guide vanes closed but keeping the cooling water fed to the ɔyrinth seals.

7.4.5 Zero-speed discharge and break-away torque

ır this specific test it is necessary to block the model runner shaft. In case of a hydrostatic ıaring system the rotating part has to be linked to the swinging system connected with the rque measuring instrument in order to determine the break-away torque. When a torque-eter is used, the rotating shaft has to be blocked against the stationary part of the system.

8 Differential pressure measurement in view of prototype index test

8.1 General

index testing at the prototype machine (in pump or turbine mode) is anticipated, some lditional measurements can be performed during model testing. This subclause only deals th index testing using differential pressure as an index value. Index tests in the model can ɛver be a substitute for an absolute discharge measurement at the prototype.

fferential pressure measurement for index testing is made on a well chosen pair of taps ıowing a significant pressure difference related to local kinetic energy.

l index testing using differential pressure measurement is therefore well represented by the llowing equation:

$$Q = f(\Delta p) \cdot \Delta p^{0,5}$$

here

ɔ is the differential pressure measured with a differential manometer or a differential pressure transducer connected between the taps, and

$\Delta p)$ is a function of the flow condition, the Reynolds number and the wall roughness.

According to IEC 60041, clause 15, the equation can be written:

$$Q = k \cdot \Delta p^{n}$$

where k and n are constants.

The value of n is usually taken in the range of 0,48 to 0,52.

4.8.2 Purpose of test

The purposes of additional measurements in the model are:

a) to select a proper position of pressure taps giving a significant differential pressure;
b) to determine the maximum differential pressure to be expected at the prototype. This information enables a proper selection of instrumentation;
c) to check the stability of the differential pressure at the selected pressure taps;
d) to verify that the relationship between pressure difference and the discharge is not influenced by other operating parameters (e.g. guide vane opening, n_{ED}, etc.).

The purpose of the tests described is not to establish a calibration curve for an absolute determination of discharge at the prototype.

4.8.3 Execution of test

4.8.3.1 Pressure taps

Depending on the type of machine, different locations of pressure taps can be chosen. Typical examples are given in clause 15 of IEC 60041.

The design of pressure taps shall be in accordance with the requirements stated in subclause 3.3.3.

4.8.3.2 Instrumentation

Differential manometers or differential pressure transducers shall be selected to cover the complete range of expected differential pressure and used within their optimum measuring range.

4.8.3.3 Test procedure

To demonstrate that the chosen pair of pressure taps has been suitably located, the following procedures are recommended:

a) vary model discharge by varying guide vane opening while maintaining constant specific hydraulic energy;
b) in a second step vary discharge by varying specific hydraulic energy at a constant guide vane opening;
c) in case of double regulated machines the same procedure should be repeated for another blade angle in order to enlarge the range of discharge;
d) in case of pump operation consistency of results should be checked also at different speed factors rather than by variation of guide vane opening.

It is possible to simultaneously measure these differential pressures and hydraulic performance characteristics.

Suitability of a chosen pair of taps can be assumed if all the readings taken under various conditions of flow form a power function of discharge within a reasonable band.

exact determination of k and n, the least-square method shall be applied.

.4 Transposition to prototype conditions

›rder to calculate the magnitude of differential pressure to be expected at the prototype :hine the following equations can be used:

$$Q_P = k_P \cdot (\Delta p)_P^{0,5} \qquad Q_M = k_M \cdot (\Delta p)_M^{0,5}$$

ı

$$k_P = k_M \cdot \frac{D_P^2}{D_M^2}$$

s

$$(\Delta p)_P = \frac{1}{k_M^2} \cdot \frac{D_M^4}{D_P^4} \cdot Q_P^2$$

.5 Uncertainty

ən under favorable circumstances, applicable tolerances for prototype fabrication and for ation of pressure taps together with deviations in inflow conditions result in an uncertainty in ı determination of prototype discharge based on model test results of about ± 5 %. erefore, k values determined during model tests shall not be used for a determination of solute discharge for contractual purposes.

Annex A (informative)
Dimensionless terms

Definition in 1.3.3.12[3)]					Relation to other existing definitions								
					I			II			III		
Clause	Term	Symbol	Definition	Relations	Symbol	Definition	Relation to term	Symbol	Definition	Relation to term	Symbol	Definition	Relation to term
1.3.3.12.1	Speed factor	n_{ED}	$\frac{nD}{E^{0,5}}$	$=\frac{1}{E_{nD}^{0,5}}$	n_{11}	$\frac{nD}{H^{0,5}}$ (n in min^{-1})	$\frac{1}{60g^{0,5}}n_{11}$	K_u	$\psi^{-0,5}=\frac{\omega D/2}{\sqrt{2}E^{0,5}}$	$\frac{\sqrt{2}}{\pi}K_u$	ω_{ED}	$\frac{\omega D}{E^{0,5}}$	$\frac{\omega_{ED}}{2\pi}$
1.3.3.12.2	Discharge factor	Q_{ED}	$\frac{Q_1}{D^2E^{0,5}}$	$=\frac{Q_{nD}}{E_{nD}^{0,5}}$	Q_{11}	$\frac{Q_1}{D^2H^{0,5}}$	$\frac{1}{g^{0,5}}Q_{11}$	K_{cm}	$\varphi\psi^{-0,5}=\frac{Q_1}{\pi(D/2)^2(2E)^{0,5}}$	$\frac{\pi}{\sqrt{8}}K_{cm}$	Q_{ED}	$\frac{Q_1}{D^2E^{0,5}}$	1
1.3.3.12.3	Torque factor	T_{ED}	$\frac{T_m}{\rho_1D^3E}$	$=\frac{T_{nD}}{E_{nD}}=\frac{P_{ED}}{2\pi n_{ED}}$	T_{11}	$\frac{T_m}{D^3H}$ (T_m in kpm)	$\frac{1}{\rho g}T_{11}$	K_T	$\frac{T_m}{\rho_1\pi(D/2)^3E}$	$\frac{\pi}{8}K_T$	T_{ED}	$\frac{T_m}{\rho_1D^3E}$	1
1.3.3.12.4	Power factor	P_{ED}	$\frac{P_m}{\rho_1D^2E^{1,5}}$ [1)]	$=Q_{ED}'\eta_{hT}$ (turbine) $=\frac{Q_{ED}}{\eta_{hP}}$ (pump) $=P_{nD}/E_{nD}^{1,5}=P_{nD}\cdot n_{ED}^3$ $=2\pi n_{ED}T_{ED}$	P_{11}	$\frac{P_m}{D^2H^{1,5}}$ (P in kW)	$\frac{1000}{\rho g^{1,5}}P_{11}$	K_P	$\frac{2P_m}{\rho_1\pi(D/2)^2(2E)^{1,5}}$	$\frac{\pi}{2^{1,5}}K_P$	P_{ED}	$\frac{P_m}{\rho_1D^2E^{1,5}}$	1
1.3.3.12.5	Energy coefficient	E_{nD}	$\frac{E}{n^2D^2}$	$=\frac{1}{n_{ED}^2}$				ψ	$\frac{2E}{(\omega D/2)^2}$	$\frac{\pi^2}{2}\psi$	$E_{\omega D}$	$\frac{E}{\omega^2D^2}$	$4\pi^2E_{\omega D}$
1.3.3.12.6	Discharge coefficient	Q_{nD}	$\frac{Q_1}{nD^3}$	$=\frac{Q_{ED}}{n_{ED}}=Q_{ED}E_{nD}^{0,5}$				φ	$\frac{K_{cm}}{K_u}=\frac{Q_1}{\pi\omega(D/2)^3}$	$\frac{\pi^2}{4}\varphi$	$Q_{\omega D}$	$\frac{Q_1}{\omega D^3}$	$2\pi Q_{\omega D}$
1.3.3.12.7	Torque coefficient	T_{nD}	$\frac{T_m}{\rho_1n^2D^5}$	$=\frac{T_{ED}}{n_{ED}^2}=T_{ED}E_{nD}=\frac{P_{nD}}{2\pi}$				τ	$\frac{2T_m}{\rho_1\pi\omega^2(D/2)^5}$	$\frac{\pi^3}{16}\tau$	$T_{\omega D}$	$\frac{T_m}{\rho_1\omega^2D^5}$	$4\pi^2T_{\omega D}$
1.3.3.12.8	Power coefficient	P_{nD}	$\frac{P_m}{\rho_1n^3D^5}$ [1)]	$=E_{nD}Q_{nD}\eta_{hT}$ (turbine) $=\frac{E_{nD}Q_{nD}}{\eta_{hP}}$ (pump) $=P_{ED}/n_{ED}^3=P_{ED}E_{nD}^{1,5}=2\pi T_{nD}$				λ	$\frac{2P_m}{\rho_1\pi\omega^3(D/2)^5}$	$\frac{\pi^4}{8}\lambda$	$P_{\omega D}$	$\frac{P_m}{\rho_1\omega^3D^5}$	$8\pi^3P_{\omega D}$
1.3.3.12.9	Thoma number	σ	$\frac{NPSE}{E}$	$=\sigma_{nD}/E_{nD}=\sigma_{nD}n_{ED}^2$									
1.3.3.12.10	Cavitation coefficient	σ_{nD}	$\frac{NPSE}{n^2D^2}$	$=\sigma E_{nD}=\sigma/n_{ED}^2$				ψ_c	$\frac{2NPSE}{(\omega D/2)^2}$	$\frac{\pi^2}{2}\psi_c$			
1.3.3.12.11	Specific speed	n_{QE}	$\frac{nQ_1^{0,5}}{E^{0,75}}$	$=n_{ED}Q_{ED}^{0,5}=Q_{nD}^{0,5}/E_{nD}^{0,75}$	n_q [2)]	$\frac{nQ_1^{0,5}}{H^{0,75}}$ (n in min^{-1})	$\frac{n_q}{60g^{0,75}}$	ν	$\frac{\omega(Q_1/\pi)^{0,5}}{(2E)^{0,75}}=\frac{\varphi^{0,5}}{\psi^{0,75}}$	$\frac{\nu}{2^{0,25}\pi^{0,5}}$	ω_s	$\frac{\omega Q_1^{0,5}}{E^{0,75}}$	$\frac{\omega_s}{2\pi}$

1) Reference is made to the mechanical power of the runner/impeller, usually measured on the model.

... Also $n = \frac{nP_m^{0,5}}{[illegible]}$ is applied for turbines.

Annex B
(normative)

Physical properties, data

Table B.1 – Acceleration due to gravity g ($m \cdot s^{-2}$)

Latitude φ °	Altitude above mean sea level z m				
	0	1 000	2 000	3 000	4 000
0	9,780	9,777	9,774	9,771	9,768
5	9,781	9,778	9,775	9,772	9,769
10	9,782	9,779	9,776	9,773	9,770
15	9,784	9,781	9,778	9,775	9,772
20	9,786	9,783	9,780	9,777	9,774
25	9,790	9,787	9,784	9,781	9,778
30	9,793	9,790	9,787	9,784	9,781
35	9,797	9,794	9,791	9,788	9,785
40	9,802	9,799	9,796	9,793	9,790
45	9,806	9,803	9,800	9,797	9,794
50	9,811	9,808	9,805	9,802	9,799
55	9,815	9,812	9,809	9,806	9,803
60	9,819	9,816	9,813	9,810	9,807
65	9,822	9,820	9,817	9,814	9,811
70	9,826	9,823	9,820	9,817	9,814

NOTES

1 Values for g are given as a function of latitude and altitude.

2 Definition and formula: see 1.3.3.3.1 and 2.5.2.

Table B.2 – Density of distilled water ρ_{wd} ($kg \cdot m^{-3}$)

Temperature θ °C	Absolute pressure 10^5 Pa							
	1	10	20	30	40	50	60	70
0	999,8	1 000,3	1 000,8	1 001,3	1 001,8	1 002,3	1 002,8	1 003,3
1	999,9	1 000,4	1 000,9	1 001,4	1 001,9	1 002,4	1 002,9	1 003,4
2	1 000,0	1 000,4	1 000,9	1 001,4	1 001,9	1 002,4	1 002,9	1 003,4
3	1 000,0	1 000,4	1 000,9	1 001,4	1 001,9	1 002,4	1 002,9	1 003,4
4	1 000,0	1 000,4	1 000,9	1 001,4	1 001,9	1 002,4	1 002,9	1 003,4
5	999,9	1 000,4	1 000,9	1 001,4	1 001,9	1 002,4	1 002,8	1 003,3
6	999,9	1 000,4	1 000,9	1 001,4	1 001,8	1 002,3	1 002,8	1 003,3
7	999,9	1 000,3	1 000,8	1 001,3	1 001,8	1 002,3	1 002,7	1 003,2
8	999,9	1 000,3	1 000,8	1 001,2	1 001,7	1 002,2	1 002,7	1 003,2
9	999,8	1 000,2	1 000,7	1 001,2	1 001,6	1 002,1	1 002,6	1 003,1
10	999,7	1 000,1	1 000,6	1 001,1	1 001,6	1 002,0	1 002,5	1 003,0
11	999,6	1 000,0	1 000,5	1 001,0	1 001,4	1 001,9	1 002,4	1 002,9
12	999,5	999,9	1 000,4	1 000,9	1 001,3	1 001,8	1 002,3	1 002,7
13	999,4	999,8	1 000,3	1 000,7	1 001,2	1 001,7	1 002,1	1 002,6
14	999,2	999,7	1 000,1	1 000,6	1 001,1	1 001,5	1 002,0	1 002,4
15	999,1	999,5	1 000,0	1 000,4	1 000,9	1 001,4	1 001,8	1 002,3
16	998,9	999,4	999,8	1 000,3	1 000,7	1 001,2	1 001,7	1 002,1
17	998,8	999,2	999,6	1 000,1	1 000,6	1 001,0	1 001,5	1 001,9
18	998,6	999,0	999,5	999,9	1 000,4	1 000,8	1 001,3	1 001,7
19	998,4	998,8	999,3	999,7	1 000,2	1 000,6	1 001,1	1 001,5
20	998,2	998,6	999,1	999,5	1 000,0	1 000,4	1 000,9	1 001,3
21	998,0	998,4	998,9	999,3	999,8	1 000,2	1 000,7	1 001,1
22	997,8	998,2	998,6	999,1	999,5	1 000,0	1 000,4	1 000,9
23	997,5	997,9	998,4	998,8	999,3	999,7	1 000,2	1 000,6
24	997,3	997,7	998,1	998,6	999,0	999,5	999,9	1 000,4
25	997,0	997,4	997,9	998,3	998,8	999,2	999,7	1 000,1
26	996,8	997,2	997,6	998,1	998,5	999,0	999,4	999,9
27	996,5	996,9	997,4	997,8	998,3	998,7	999,1	999,6
28	996,2	996,6	997,1	997,5	998,0	998,4	998,9	999,3
29	995,9	996,3	996,8	997,2	997,7	998,1	998,6	999,0
30	995,7	996,1	996,5	996,9	997,4	997,8	998,3	998,7
31	995,3	995,7	996,2	996,6	997,1	997,5	997,9	998,4
32	995,0	995,4	995,9	996,3	996,8	997,2	997,6	998,1
33	994,7	995,1	995,5	996,0	996,4	996,9	997,3	997,7
34	994,4	994,8	995,2	995,7	996,1	996,5	997,0	997,4
35	994,0	994,4	994,9	995,3	995,8	996,2	996,6	997,1
36	993,7	994,1	994,5	995,0	995,4	995,8	996,3	996,7
37	993,3	993,7	994,2	994,6	995,0	995,5	995,9	996,3
38	993,0	993,4	993,8	994,2	994,7	995,1	995,5	996,0
39	992,6	993,0	993,4	993,9	994,3	994,7	995,2	995,6
40	992,2	992,6	993,1	993,5	993,9	994,4	994,3	995,2

NOTES

1 Values for ρ_{wd} are given as a function of temperature θ (°C) and absolute pressure p_{abs} (10^5 Pa).

2 Definition and formula: see 1.3.3.3.3 and 2.5.3.1.3.

Table B.2 *(continued)*

Temperature θ °C	Absolute pressure 10^5 Pa							
	80	90	100	110	120	130	140	150
0	1 003,8	1 004,3	1 004,8	1 005,3	1 005,8	1 006,3	1 006,8	1 007,3
1	1 003,9	1 004,3	1 004,8	1 005,3	1 005,8	1 006,3	1 006,8	1 007,3
2	1 003,9	1 004,4	1 004,8	1 005,3	1 005,8	1 006,3	1 006,8	1 007,3
3	1 003,9	1 004,4	1 004,8	1 005,3	1 005,8	1 006,3	1 006,8	1 007,3
4	1 003,8	1 003,4	1 004,8	1 005,3	1 005,8	1 006,3	1 006,7	1 007,3
5	1 003,8	1 004,3	1 004,8	1 005,3	1 005,7	1 006,2	1 006,7	1 007,2
6	1 003,8	1 004,2	1 004,7	1 005,2	1 005,7	1 006,2	1 006,2	1 007,1
7	1 003,7	1 004,2	1 004,7	1 005,1	1 005,6	1 006,1	1 006,5	1 007,0
8	1 003,6	1 004,1	1 004,6	1 005,0	1 005,5	1 006,0	1 006,5	1 006,9
9	1 003,5	1 004,0	1 004,5	1 005,0	1 005,4	1 005,9	1 006,4	1 OO6,8
10	1 003,4	1 003,9	1 004,4	1 004,8	1 005,3	1 005,8	1 006,2	1 006,7
11	1 003,3	1 003,8	1 004,3	1 004,7	1 005,2	1 005,6	1 006,1	1 006,6
12	1 003,2	1 003,7	1 004,1	1 004,6	1 005,0	1 005,5	1 006,0	1 006,4
13	1 003,1	1 003,5	1 004,0	1 004,4	1 004,9	1 005,4	1 005,8	1 006,3
14	1 002,9	1 003,4	1 003,8	1 004,3	1 004,7	1 005,2	1 005,7	1 006,1
15	1 002,7	1 003,2	1 003,7	1 004,1	1 004,6	1 005,0	1 005,5	1 005,9
16	1 002,6	1 003,0	1 003,5	1 003,9	1 004,4	1 004,8	1 005,3	1 005,8
17	1 002,4	1 002,8	1 003,3	1 003,8	1 004,2	1 004,7	1 005,1	1 005,6
18	1 002,2	1 002,7	1 003,1	1 003,6	1 004,0	1 004,5	1 004,9	1 005,4
19	1 002,0	1 002,4	1 002,9	1 003,3	1 003,8	1 004,2	1 004,7	1 005,1
20	1 001,8	1 002,2	1 002,7	1 003,1	1 003,6	1 004,0	1 004,5	1 004,9
21	1 001,6	1 002,0	1 002,5	1 002,9	1 003,3	1 003,8	1 004,2	1004,7
22	1 001,3	1 001,8	1 002,2	1 002,7	1 003,1	1 003,5	1 004,0	1 004,4
23	1 001,1	1 001,5	1 002,0	1 002,4	1 002,9	1 003,3	1 003,7	1 004,2
24	1 000,8	1 001,3	1 001,7	1 002,2	1 002,6	1 003,0	1 003,5	1 003,9
25	1 000,6	1 001,0	1 001,5	1 001,9	1 002,3	1 002,8	1 003,2	1 003,7
26	1 000,3	1 000,7	1 001,2	1 001,6	1 002,1	1 002,5	1 002,9	1 003,4
27	1 000,0	1 000,5	1 000,9	1 001,3	1 001,8	1 002,2	1 002,7	1 003,1
28	999,7	1 000,2	1 000,6	1 001,1	1 001,5	1 001,9	1 002,4	1 002,8
29	999,4	999,9	1 000,3	1 000,8	1 001,2	1 001,6	1 002,1	1 002,5
30	999,1	999,6	1 000,0	1 000,4	1 000,9	1 001,3	1 001,7	1 002,2
31	998,8	999,3	999,7	1 000,1	1 000,6	1 001,0	1 001,4	1 001,9
32	998,5	998,9	999,4	999,8	1 000,2	1 000,7	1 001,1	1 001,5
33	998,2	998,6	999,0	999,5	999,9	1 000,3	1 000,8	1 001,2
34	997,8	998,3	998,7	999,1	999,6	1 000,0	1 000,4	1 000,9
35	997,5	997,9	998,4	998,8	999,2	999,7	1 000,1	1 000,5
36	997,1	997,6	998,0	998,4	998,9	999,3	999,7	1 000,2
37	996,8	997,2	997,6	998,1	998,5	998,9	999,4	999,8
38	996,4	996,8	997,3	997,7	998,1	998,6	999,0	999,4
39	996,0	996,5	996,9	997,3	997,8	998,2	998,6	999,0
40	995,7	996,1	996,5	996,9	997,4	997,8	998,2	998,7

Table B.3 – Kinematic viscosity of distilled water ν ($m^2 \cdot s^{-1}$)

Water temperature θ °C	Kinematic viscosity ν $m^2 \cdot s^{-1}$	Water temperature θ °C	Kinematic viscosity ν $m^2 \cdot s^{-1}$
0	$1{,}791 \times 10^{-6}$		
1	$1{,}731 \times 10^{-6}$	21	$0{,}980 \times 10^{-6}$
2	$1{,}674 \times 10^{-6}$	22	$0{,}957 \times 10^{-6}$
3	$1{,}620 \times 10^{-6}$	23	$0{,}934 \times 10^{-6}$
4	$1{,}568 \times 10^{-6}$	24	$0{,}913 \times 10^{-6}$
5	$1{,}520 \times 10^{-6}$	25	$0{,}892 \times 10^{-6}$
6	$1{,}473 \times 10^{-6}$	26	$0{,}873 \times 10^{-6}$
7	$1{,}429 \times 10^{-6}$	27	$0{,}854 \times 10^{-6}$
8	$1{,}387 \times 10^{-6}$	28	$0{,}835 \times 10^{-6}$
9	$1{,}346 \times 10^{-6}$	29	$0{,}817 \times 10^{-6}$
10	$1{,}308 \times 10^{-6}$	30	$0{,}800 \times 10^{-6}$
11	$1{,}271 \times 10^{-6}$	31	$0{,}784 \times 10^{-6}$
12	$1{,}236 \times 10^{-6}$	32	$0{,}768 \times 10^{-6}$
13	$1{,}202 \times 10^{-6}$	33	$0{,}753 \times 10^{-6}$
14	$1{,}170 \times 10^{-6}$	34	$0{,}738 \times 10^{-6}$
15	$1{,}140 \times 10^{-6}$	35	$0{,}723 \times 10^{-6}$
16	$1{,}110 \times 10^{-6}$	36	$0{,}709 \times 10^{-6}$
17	$1{,}082 \times 10^{-6}$	37	$0{,}696 \times 10^{-6}$
18	$1{,}055 \times 10^{-6}$	38	$0{,}683 \times 10^{-6}$
19	$1{,}029 \times 10^{-6}$	39	$0{,}670 \times 10^{-6}$
20	$1{,}004 \times 10^{-6}$	40	$0{,}658 \times 10^{-6}$

NOTES

1 Values for ν are given as a function of water temperature θ (°C) at absolute pressure $p_{abs} = 10^5$ Pa.

2 Definition and formula: see 1.3.3.3.6 and 2.5.3.3.

Table B.4 – Vapour pressure of distilled water p_{va} (Pa)

Temperature θ °C	Vapour pressure p_{va} Pa	Temperature θ °C	Vapour pressure p_{va} Pa
0	611		
1	657	21	2 488
2	706	22	2 645
3	758	23	2 810
4	814	24	2 985
5	873	25	3 169
6	935	26	3 363
7	1 002	27	3 567
8	1 073	28	3 782
9	1 148	29	4 008
10	1 228	30	4 246
11	1 313	31	4 495
12	1 403	32	4 758
13	1 498	33	5 034
14	1 599	34	5 323
15	1 706	35	5 627
16	1 819	36	5 945
17	1 938	37	6 280
18	2 064	38	6 630
19	2 198	39	6 997
20	2 339	40	7 381

NOTES

1 Values for p_{va} are given as a function of water temperature θ (°C).

2 Definition and formula: see 1.3.3.3.4 and 2.5.3.4.

Table B.5 – Density of dry air ρ_a (kg·m^{-3})

Air temperature θ_a (°C)	Density of dry air ρ_a (kg · m^{-3})
0	1,293
2	1,284
4	1,274
6	1,265
8	1,256
10	1,247
12	1,238
14	1,230
16	1,221
18	1,213
20	1,205
22	1,196
24	1,188
26	1,180
28	1,173
30	1,165

NOTES

1 Values are given as a function of air temperature θ_a (°C) at absolute ambient pressure $p_{amb\text{-}o}$ = 101 325 Pa.

2 Definition and formula: see 1.3.3.3.3 and 2.5.4.1.

Table B.6 – Ambient pressure p_{amb} (Pa)

Elevation z m	Ambient pressure p_{amb} Pa	Elevation z m	Ambient pressure p_{amb} Pa
0	101 325		
100	100 129	2 100	78 520
200	98 945	2 200	77 548
300	97 773	2 300	76 586
400	96 611	2 400	75 634
500	95 461	2 500	74 692
600	94 322	2 600	73 759
700	93 194	2 700	72 835
800	92 076	2 800	71 921
900	90 970	2 900	71 017
1 000	89 876	3 000	70 121
1 100	88 792	3 100	69 235
1 200	87 718	3 200	68 358
1 300	86 655	3 300	67 490
1 400	85 602	3 400	66 631
1 500	84 560	3 500	65 780
1 600	83 528	3 600	64 939
1 700	82 506	3 700	64 106
1 800	81 494	3 800	63 283
1 900	80 493	3 900	62 467
2 000	79 501	4 000	61 660

NOTES

1 Values are given as a function of elevation z (m).

2 Definition and formula: see 1.3.3.5.2 and 2.5.4.2.

Table B.7 – Density of mercury ρ_{Hg} ($kg \cdot m^{-3}$)

Temperature θ °C	Density ρ_{Hg} $kg \cdot m^{-3}$	Temperature θ °C	Density ρ_{Hg} $kg \cdot m^{-3}$
0	13 595	21	13 543
1	13 593	22	13 541
2	13 590	23	13 538
3	13 588	24	13 536
4	13 585	25	13 534
5	13 583	26	13 531
6	13 580	27	13 529
7	13 578	28	13 526
8	13 575	29	13 524
9	13 573	30	13 521
10	13 570	31	13 519
11	13 568	32	13 516
12	13 565	33	13 514
13	13 563	34	13 511
14	13 561	35	13 509
15	13 558	36	13 507
16	13 556	37	13 504
17	13 553	38	13 502
18	13 551	39	13 499
19	13 548	40	13 497
20	13 546		

NOTES

1 Values are given as a function of temperature θ (°C) at absolute ambient pressure $p_{amb\text{-}0}$ = 101 325 Pa (standard ambient pressure at sea level).

2 Definition and formula: see 1.3.3.3.3 and 2.5.5.

Annex C
(informative)

erivation of the equation for the specific hydraulic energy of a machine

I Theoretical equation

e energy balance within the inner boundaries of a hydraulic machine is given by the Bernoulli ıation in its differential form, supplemented by the energy loss term:

$$\frac{dp_{abs}}{\rho} + d\left(\frac{v^2}{2}\right) + gdz + de_L + de = 0$$

ere

$\frac{abs}{)}$ is the change of specific pressure energy;

$\left(\frac{v^2}{2}\right)$ is the change of specific kinetic energy;

z is the change of specific potential energy;

L is the specific dissipated energy;

is the specific energy exchanged between the water and the runner/impeller (de < 0 for a turbine, de > 0 for a pump).

an ideal machine without losses ($de_L = 0$), the specific hydraulic energy E of the water which available between the high and low pressure reference sections 1 and 2 of the machine is tained by integration between these two sections:

$$\int_2^1 de = \int_2^1 \frac{dp_{abs}}{r} + \int_2^1 d\left(\frac{v^2}{2}\right) + \int_2^1 gdz \qquad \text{(C.1)}$$

.2 Specific pressure energy term

may be written:

$$\int_2^1 \frac{dp_{abs}}{\rho} = \frac{p_{abs1} - p_{abs2}}{\rho^*}$$

aking into account the scope of this standard, ρ^* may be defined by the approximation:

$$\rho^* = \overline{\rho} = \frac{1}{2}(\rho_1 + \rho_2)$$

'he relative error introduced by this approximation is less than 2×10^{-4}.

C.3 Specific kinetic energy term

The value of the specific kinetic energy term in a streamline of the flow and its mean value in a cross-section $\left(e_c = v^2/2\right)$ are defined in note 1 of 3.5.2.4.

By convention, it is assumed that the change in specific kinetic energy is given by:

$$\int_2^1 d\left(\frac{v^2}{2}\right) = \frac{1}{2}\left(v_1^2 - v_2^2\right)$$

where v_1 and v_2 are the mean axial velocities (see 1.3.3.4.9) in the reference sections 1 and 2. (Strictly speaking, the tangential and radial components of the velocities should be taken into account.)

C.4 Specific potential energy term

Since the change in gravitational acceleration with elevation between the reference sections 1 and 2 is small, it may be written:

$$\int_2^1 gdz = \overline{g}(z_1 - z_2)$$

where $\overline{g} = \frac{1}{2}(g_1 + g_2)$

C.5 Practical equation

With the above simplifications, equation (C.1) defining the specific hydraulic energy of the machine (see 1.3.3.6.2) becomes:

$$E = \frac{1}{\overline{\rho}}\left(p_{abs1} - p_{abs2}\right) + \frac{1}{2}\left(v_1^2 - v_2^2\right) + \overline{g}(z_1 - z_2)$$

In practice, the value of g at the reference level of the machine may be taken as $\overline{g}$. Furthermore, for low head machines, $(p_1 - p_2) < 4 \times 10^5$ Pa for instance, the value of ρ at the low pressure reference section may be taken as $\overline{\rho}$.

Annex D
(informative)

Influence of the density of actual water ρ_{wa} on measurement and calibration

ıe to chemically dissolved components, the density of actual test water ρ_{wa} (see 3.5.2.3 and 5.3.1.2) will always be higher than the density of distilled water ρ_{wd}, as given in 2.5.3.1.3 and ɔle B.2. The deviation in general is less than 0,05 %.

the specific hydraulic energy E is determined primarily by pressure measurements and if the ɜasuring instruments are mounted at approximately the same elevation, E can be expressed :

$$E = \frac{p_1 - p_2}{\bar{\rho}} + \frac{v_1^2 - v_2^2}{2}$$

ıe hydraulic power becomes:

$$P_h = \left[\frac{p_1 - p_2}{\bar{\rho}} + \frac{v_1^2 - v_2^2}{2}\right] \cdot \rho_1 \cdot Q_1$$

$$= \left[p_1 - p_2 + \bar{\rho} \cdot \frac{v_1^2 - v_2^2}{2}\right] \cdot \frac{\rho_1}{\bar{\rho}} \cdot Q_1$$

here $\rho_1 = \rho_{wa,1}$ and $\bar{\rho} = \bar{\rho}_{wa}$.

ı practice, it may be assumed that in the model testing installation ρ_{wa} differs little from ρ_{wd} ıd that both vary with temperature and pressure in a similar manner:

$$\frac{\rho_{wa,1}}{\bar{\rho}_{wa}} = \frac{\rho_{wd,1}}{\bar{\rho}_{wd}} = \frac{\rho_1}{\bar{\rho}}$$

is completely justified to apply $\bar{\rho}_{wd}$ in term $\bar{\rho}_{wa}\frac{v_1^2 - v_2^2}{2}$ because $\frac{v_1^2 - v_2^2}{2}$ will always be less ıan 10 % of the pressure energy term, even for high specific speed machines. Thus the error ɔr E or η when applying the density of distilled water will always be less than 0,005 %. 'herefore a good approximation for hydraulic power is:

$$P_h = \left[p_1 - p_2 + \bar{\rho}_{wd}\frac{v_1^2 - v_2^2}{2}\right] \cdot \frac{\rho_{wd,1}}{\bar{\rho}_{wd}} \cdot Q_1$$

Annex E
(informative)

Summarized test and calculation procedure

E.1 Premise

Annex E contains a list of agreements, checks and operations to be carried out before, during and after the tests.

Also the main aspects of a test point calculation, together with all the information necessary to make a comparison with the guarantees, including the evaluation of the measurement uncertainties are summarized below.

For each point, reference is made to the applicable subclauses of this standard.

E.2 Agreements to be reached prior to testing

Agreement between the parties shall be reached, in due time before the test, on:

- the model size (2.3.2.2) and the scale ratio λ (1.3.3.2.9);
- the model structural characteristics (2.1.3.2);
- similitude requirements (2.3.1);
- the test conditions (2.3.2.1);
- the values of Re_{M^*} or Re_{sp} (1.4.1.4) when the guarantees refer to the model (3.8.2.2), (for Pelton turbine, the value of Froude number, 2.3.1.5.2);
- the extent of the model (2.1.3.3);
- the reference sections (1.4.1.1);
- the pressure measuring sections (3.3.2 and 3.5.2.1) and the areas for the calculation of specific hydraulic energy (3.5.2) and net positive suction specific energy (3.5.4);
- the guaranteed values of power, discharge, efficiency, steady-state runaway speed and influence of cavitation on hydraulic performance (1.4.2);
- the σ_{Pl} values (1.4.2.1.5);
- the cavitation test procedure (2.3.3.3.6 and 3.8.2.3.7) and the cavitation reference level (2.3.1.5.1);
- the injection of cavitation nuclei (2.1.2.3, 2.3.1.6.2 and 2.5.3.2) in the circuit water, if necessary;
- all other specifications for the model tests (2.3.3.1.1);
- the time schedule of the tests (2.3.3.1.2);
- the personnel and responsibilities (2.3.3.1.3);
- the type and extent of the tests to be performed (2.3.3.3);
- the calibration procedure for the instruments to be utilized (2.3.3.1.5);
- g_P prototype value of gravitational acceleration (2.5.2);
- θ_P prototype water temperature and corresponding kinetic viscosity ν_P (2.5.3.3);
- ρ_P prototype water density (2.5.3.1);

n_P steady-state prototype speed;

Re_P steady-state Reynolds number of prototype;

NPSE$_P$ range of prototype operation (1.4.2.1.5);

E_{Pmax} maximum specific hydraulic energy applied to runaway condition (1.4.2.1.4);

mechanical power losses of the prototype (1.4.2.1.1 and 2.4.1.4);

electrical machine windage losses (annex G).

Model, test facility and instrumentation

.1 Model manufacture and dimensional checks

model shall be manufactured according to 2.1.3.2 and subjected to the dimensional cks outlined in 2.2 and 2.3.3.1.6. Particular attention shall be paid to:

the value of the reference diameter (1.3.3.2.6);

checking the runner/impeller seal clearances (2.2.2.1.6 and 2.3.3.2.1);

the roughness or surface quality of wetted surfaces (2.2.3).

.2 Test facility instrumentation and data acquisition system

test stand and the complete circuit shall be carefully checked (2.3.3.2.2) before and during tests, paying particular attention to the condition of the gauge piping for any leakage.

instrumentation and the data acquisition system shall also be carefully checked and ibrated before and after the tests (2.3.3.2.3).

following is usually verified by means of specific operation tests:

the regularity of the pressure at the taps of the model inlet and outlet sections (3.3.3.1);

the proper response of the instrumentation at the same operating point tested under different test conditions (2.3.3.3.1).

4 Tests and calculation of the model values

ually, a complete preparatory test and preliminary test are carried out by, or in the presence the manufacturer to verify all the guarantees prescribed in the contract and to define the ole operating behaviour of the machine. The acceptance tests, witnessed by all the parties olved, are subsequently carried out (2.3.3.3.1).

4.1 Measurement of the main quantities during the test

r each test point, the average values of the readings of the instruments measuring the antities listed below shall be obtained. Measurements are made in accordance with 3.1 and 8.2.1:

discharge (3.2);

pressure at the high pressure and low pressure measuring sections (or alternatively differential pressure between sections 1 and 2 and suction pressure at section 2) (3.3, 3.4, 3.5);

torque (3.6);

rotational speed (3.7);

test circuit water temperature;

– ambient temperature;

– instrument temperature;

– ambient pressure.

The calibrations of all the measurement instruments shall be made available and checked (2.3.3.1.5).

The value of the constants necessary for the calculation [for example: g (2.5.2), ρ (2.5.3.1), z_c (2.3.1.5.1), p_{va} (2.5.3.4), ν (2.5.3.3), length of levers (3.6.2.1), T_{Lm} (3.6.5.3)] shall be known or calculated for the conditions of the tests.

E.4.2 Total uncertainty

E.4.2.1 Systematic uncertainty of the measured quantities

The values of the systematic uncertainty necessary to determine the uncertainty band shall be fixed and agreed upon for each quantity (3.9.2.2.2 and annex J).

E.4.2.2 Random uncertainty of the measured quantities

The value of the random uncertainty shall be calculated from a test either near the maximum efficiency or, where necessary, at partial load (2.3.2.3.1, 3.9.2.2.1 and annex L).

E.4.2.3 Total uncertainty

The total uncertainty is the combination of the above systematic and random uncertainties and defines the uncertainty band of the curves used for the comparison to guarantees (3.9.2.2.4 and 3.10.2).

E.4.3 Calculation of the quantities related to the main hydraulic performance

Using the constants and the measurements listed, and using the calibration data and the relations given in 2.4.1.1 for the hydraulic efficiency calculation, the values of ρ_M, Q_M, E_M, P_{mM}, η_{hM}, NPSE$_M$ and Re_M are calculated.

E.4.4 Calculation of the dimensionless factors or coefficients and of the Thoma number

Using the relations of 1.3.3.12, the values of Q_{nD}, E_{nD}, P_{nD} and σ_{nD} and/or of Q_{ED}, n_{ED}, P_{ED} and σ may be calculated.

E.4.5 Determination of δ_{ref} for the scale effect calculation

By specific tests, the value of $\eta_{hM\,opt}$ and the value of $Re_{M\,opt}$ at which it is measured shall be determined. From these, the value δ_{ref} to be used in the formulae of 3.8.2.2 for the efficiency scale-up calculation may be determined by the formula given in the same subclause (see also annex F).

E.4.6 Calculation of efficiency and power coefficients referred to Re_{M^*}

This calculation procedure may be applied as an alternative to the others described in the flow chart of figure 62. Values of η_{hM^*}, P_{nD^*} and P_{ED^*} (3.8.2.2.3) may be calculated and then all the operation curves of the model referred to R_{eM^*} (3.8.2.2 and 3.8.2.2.3) can be obtained.

When the guarantee is given for model efficiency referred to a specified value Re_{Msp}, Re_{M^*} may be chosen equal to Re_{Msp} (1.4.1.4 and 3.8.3.3.5).

.7 Correction of the model-measured values taking into account the influence of cavitation

r the efficiency tests, see 2.3.3.3.5 and 3.8.2.3.7.

r the runaway tests, see 2.3.3.3.7 and 3.8.3.2.

5 Calculation of prototype quantities

fer to the flow chart of figure 62.

sed on 3.8.2.4 and 3.8.2.5 for steady-state operation at a specified n_p, the values of Q_P, E_P, P, $NPSE_P$ and η_{hP} are obtained taking into account the efficiency scale-up $(\Delta\eta_h)_{M \to P}$ and the luence of cavitation (3.8.2.4.2).

e efficiency η_P and the mechanical power P_P of the prototype machine shall be calculated :ing into account the mechanical power losses of the prototype (1.3.3.8.4, 1.4.2.1.1 and 0.3.4) and the electrical machine windage losses (see annex G).

r runaway operation, see 3.8.3.3 and 3.8.3.4 for calculating $n_{R,P}$, $Q_{R,P}$ and $NPSE_P$. The luence of cavitation on steady-state runaway speed is considered in 3.8.3.2.

6 Plotting of model or prototype results

e points and/or curves of model or prototype results are plotted on the basis of a proper mber of test points (3.8.2.3 and 3.10.2) after having removed any spurious point (see).1.3.1). If an interpolation curve is necessary, an example of its determination is given in nex H.

e above points or curves are plotted with the uncertainty band where necessary (3.10.2) for mparison with guarantees.

7 Comparison with the guarantees

ılfilment of the guarantees will be established as outlined in 3.10.3.1, 3.10.3.2 and 3.10.3.3.

ır the application of penalties and premiums, if any, see 3.10.3.5.

8 Final protocol

ıe final protocol, completed by the daily log and the test results documents, duly signed, shall : drawn at the end of the acceptance tests (2.3.3.3.9) and, once signed by all the involved ırties, it closes the acceptance model tests. All guarantees which were checked for ımpliance shall be mentioned and the protocol shall clearly state if the guarantee, for each as met or not.

.9 Final test report

ıe final test report (2.3.3.5) shall contain all the documents pertaining to the official tests and ıall be completed after the tests within a time mutually agreed (normally two months).

Annex F
(normative)

Scale-up of the hydraulic efficiency of reaction machines

F.1 Basic statements and assumptions

Statements and formulae given in this annex are valid only for reaction machines[1]) (for Pelton turbines, see annex K). The scale-up of hydraulic efficiency of reaction machines is based on the dependence of friction losses on Reynolds number Re.

Scale-up in this standard applies only to efficiency and mechanical power of runner/impeller, not to discharge or specific hydraulic energy. The applied method of evaluation is based on the assumption that wetted surfaces are hydraulically smooth and no influence of roughness or other effects is taken into account [2]).

In the scale-up formula (see F.3) the exponent of the scalable losses curve (0,16) and the ratio of scalable losses to total losses (V_{ref}) are average experimentally based values derived from:

a) model tests at different Reynolds numbers on models with hydraulically smooth surfaces;
b) comparison of efficiency tests on models and homologous prototypes, having a surface roughness according to 2.2. The surface roughness as required in 2.2.3.3 does not necessarily yield hydraulically smooth flow conditions.

As long as the deviations in geometric similarity of clearances are within the limits specified in 2.2.2.1.6 (see also 2.2.2.1.7 and 2.2.2.2.5), it is assumed that the formula for hydraulic efficiency scale-up given below remains valid.

It is conventionally agreed to calculate the hydraulic efficiency increase $\Delta\eta_h$ at the point of maximum model hydraulic efficiency and to apply this value over the whole range of guaranteed efficiencies provided the hydraulic efficiency is not greatly affected by cavitation (see 3.8.2.4.2). The optimum hydraulic efficiency should not be affected by cavitation.

---

1) For reaction machines of special design, see table 7, note 1 of 3.8.2.2.1.

2) Although various procedures on how to consider the scale effects on specific hydraulic energy and discharge or the roughness effects have been presented, there is yet no commonly accepted basis to quantify these effects. For more information, see references in F.5.

2 Amount of relative scalable losses in the range of guaranteed efficiencies

ır a given hydraulic turbomachine, the amount of relative scalable losses δ as a function of ɛynolds number Re is the same over the range R of guaranteed efficiencies (range R is own schematically in figure F.1).

ıis means that for a given constant Reynolds number, for example Re_M, the amount of lative scalable losses δ is constant for each operating point within the whole range R, ıereas the amount of non-scalable losses δ_{ns} depends on the amount of the relative total loss – η_h) of the operating point.

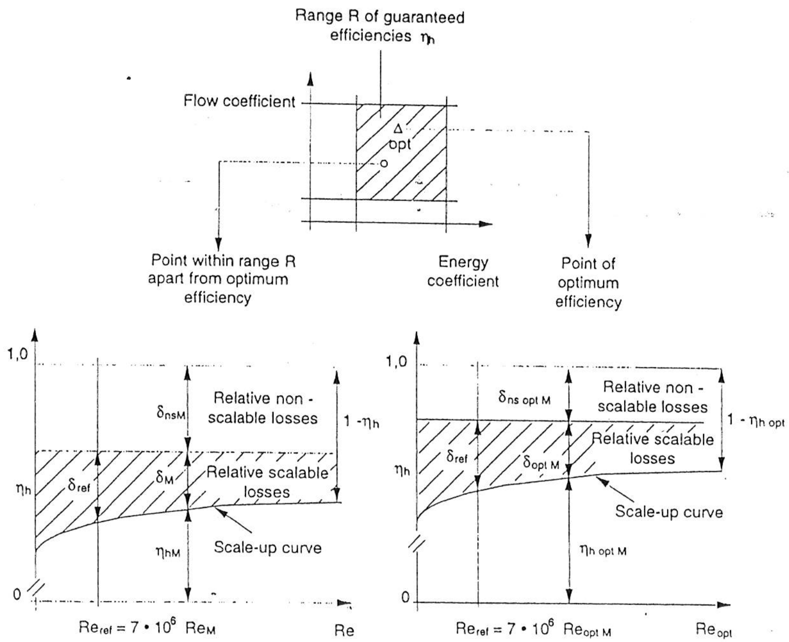


Figure F.1 – Variation of relative scalable losses

F.3 Derivation of the general scale formula

Two points A and B representing hydraulically similar operating conditions according to the assumptions in 2.3.1.2 are shown in figure F.2.

The relative scalable losses for the Reynolds numbers Re_{ref}, Re_A and Re_B (see figure F.2) are related as follows:

$$\frac{\delta_A}{\delta_{ref}} = \left(\frac{Re_{ref}}{Re_A}\right)^{0,16}$$

$$\frac{\delta_B}{\delta_{ref}} = \left(\frac{Re_{ref}}{Re_B}\right)^{0,16}$$

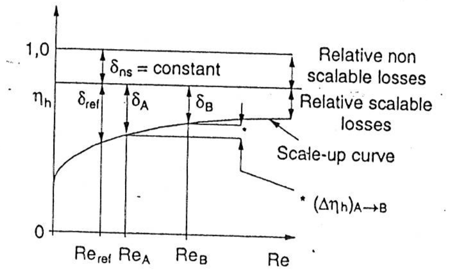


Figure F.2 – Efficiency change in hydraulically similar operating conditions A and B having different Re values

Using

$$(\Delta\eta_h)_{A\to B} = \delta_A - \delta_B$$

the following general scale formula is obtained:

$$(\Delta\eta_h)_{A\to B} = \delta_{ref}\left[\left(\frac{Re_{ref}}{Re_A}\right)^{0,16} - \left(\frac{Re_{ref}}{Re_B}\right)^{0,16}\right]$$

of which the formulae quoted in 3.8.2.2 and 3.8.2.4.1 are particular cases.

Using the assumptions of F.1 and 3.8.2.2 for the point of optimum efficiency, the following three equations result:

$$\frac{\delta_{opt\,M}}{\delta_{ref}} = \left(\frac{Re_{ref}}{Re_{opt\,M}}\right)^{0,16}$$

$$V_{ref} = \frac{\delta_{ref}}{\delta_{ref} + \delta_{ns\,opt\,M}}$$

$$\delta_{opt\,M} + \delta_{ns\,opt\,M} = 1 - \eta_{h\,opt\,M}$$

three unknown values are:

M : relative scalable losses at the point of optimum efficiency

opt M : relative non-scalable losses at the point of optimum efficiency

: relative scalable losses at the point Re_{ref} with $\delta_{ref} = (1 - \eta_{href}) \cdot V_{ref}$

e result is the equation given in 3.8.2.2:

$$\delta_{ref} = \frac{1-\eta_{h\,opt\,M}}{\left[\left(\frac{Re_{ref}}{Re_{opt\,M}}\right)^{0,16} + \frac{1-V_{ref}}{V_{ref}}\right]}$$

4 Determination of efficiency increase from model to prototype

r the following it is assumed that the Reynolds number of the prototype machine Re_P has a nstant value within the range of guarantee and that the value δ_{ref} has been determined cording to 3.8.2.2.1.

hen the model efficiencies have been referred to a constant Reynolds number Re_{M^*}, the llowing formula is applied to calculate the resulting efficiency increase $(\Delta\eta_h)_{M^* \to P}$. Within the hole range of guarantee only one constant numerical value is considered.

$$(\Delta\eta_h)_{M^* \to P} = \delta_{ref}\left[\left(\frac{Re_{ref}}{Re_{M^*}}\right)^{0,16} - \left(\frac{Re_{ref}}{Re_P}\right)^{0,16}\right]$$

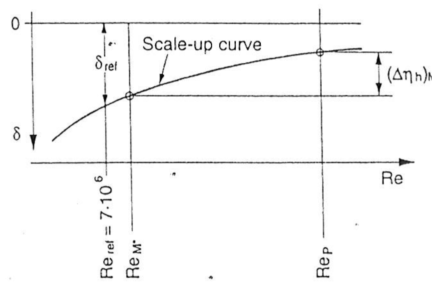


Figure F.3 – Efficiency scale-up from constant Re_{M^*} to constant Re_P

When the model efficiencies have been measured at different Reynolds numbers Re_M, the following formula is used to calculate the resulting values of efficiency increase $(\Delta\eta_h)_{M_i \to P}$ for each value Re_{M_i}. Within the whole range of guarantee more than one numerical value is considered.

$$(\Delta\eta_h)_{M_i \to P} = \delta_{ref}\left[\left(\frac{Re_{ref}}{Re_{M_i}}\right)^{0,16} - \left(\frac{Re_{ref}}{Re_P}\right)^{0,16}\right]$$

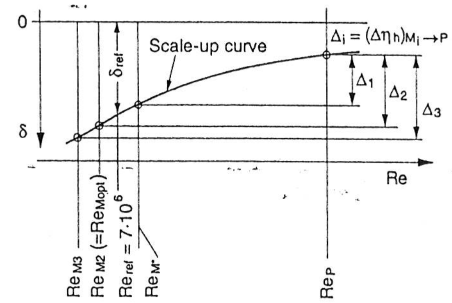


Figure F.4 – Efficiency scale-up from variable Re_M to constant Re_P

F.5 Bibliography

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Annex G
(normative)

Computation of the prototype runaway characteristics taking into account friction and windage losses of the unit

e prototype maximum runaway speed and discharge computed from the model tests :ording to 3.8.3.3 shall be corrected for the friction losses of the unit bearings and shaft seal d the windage losses of the electrical machine, which determine a decrease of the maximum iaway speed from $n_{Rmax,P}$ to $n'_{Rmax,P}$.

e procedure is the following:

ing the measured points X_1, X_2, X_3, etc. at constant guide vane opening or needle stroke in range near to runaway, plot the curve P_{mP} (n_P) for a single-regulated turbine (see figure 1). If the chosen opening corresponds to the maximum runaway speed ($\alpha = \alpha_{max}$ usually for rancis turbine), the intersection point Y of the curve P_{mP} (n_P) with the curve ($P_{Lm} + P_W$) (n_P), resenting the sum of the bearing and shaft seal friction losses of the unit and of the ctrical machine windage losses[1], is at the maximum steady-state runaway speed $n'_{Rmax,P}$ to expected for the unit.

r a double-regulated turbine, the procedure described above should be applied for each ner blade angle; the highest maximum value of $n'_{R,P}$ may then be determined.

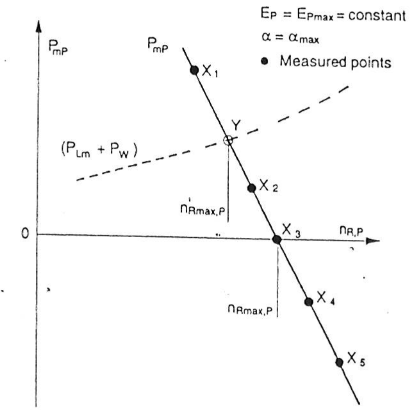

Figure G.1 – Single-regulated turbine. Determination of the maximum runaway speed of the prototype taking into account the friction and windage losses of the unit

For the determination of these losses, see IEC 60041.

Annex H
(informative)

Example of determination of the best smooth curve: method of separate segments

H.1 Premise

In model testing, unlike prototype testing, a large number of operating points are generally measured and sometimes, a measurement point is repeated very close to a previous one. Furthermore, the test range generally extends above and below the guaranteed range, but the measurement points are concentrated within the range of guarantees.

In such a case, the classical least squares method, based on a single function, may not be appropriate to determine the best fitted curve passing through the points: if the polynomial degree increases, the curve passes closer to the points but can deviate, sometimes notably, from the regular expected shape. As the mathematical form of the law expressing the physical phenomenon is in most cases unknown, a more convenient way to determine by computation the best fitted curve is to arbitrarily split the test range into segments. It is then possible to apply the least squares method to each segment using low degree polynomials which lead to reasonably smooth curves.

Several computerized methods of drawing the interpolation curves are available. There are methods based on:

– normal spline functions;
– B-splines;
– representation of surfaces.

As an example, a modified least squares method is described below.

NOTE – The measurement methods applied today result in relatively small random uncertainty. Therefore, the smoothening process can normally be restricted to averaging measurements at several points around a given operating point.

H.2 Principle of the method

A series of intervals along the abscissa are determined as described below. Three adjacent intervals (1, 2, 3 in figure H.1) form the range in which the least squares method is applied, but only the central segment 2 of the curve is directly defined by this way. The procedure is then repeated by shifting by two steps, left and right, in order to obtain a series of separate segments (a in figure H.2). The remaining intervals are then filled with segments of curve (b in figure H.2) passing through the extreme points of segments a with the same slope at these points. The two extreme segments at the ends of the test range are drawn directly by the least squares method (c in figure H.2), using the same numerical coefficients of the polynomials as in the corresponding adjacent segment.

A degree of 3 is assumed, as a rule, for the polynomials in segments of both types, a and b. Depending on the arrangement of the points and on the width of the intervals, it may happen that there are only sufficient points to define a group of two intervals, not three, at the end of the test range where the least squares methods should be applied. In this case, the polynomial is reduced to degree 2.

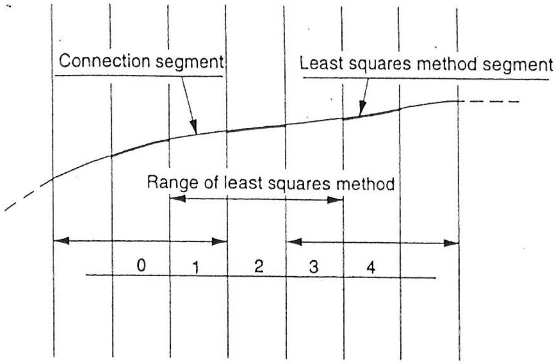

Figure H.1 – Principle of the method of separate segments

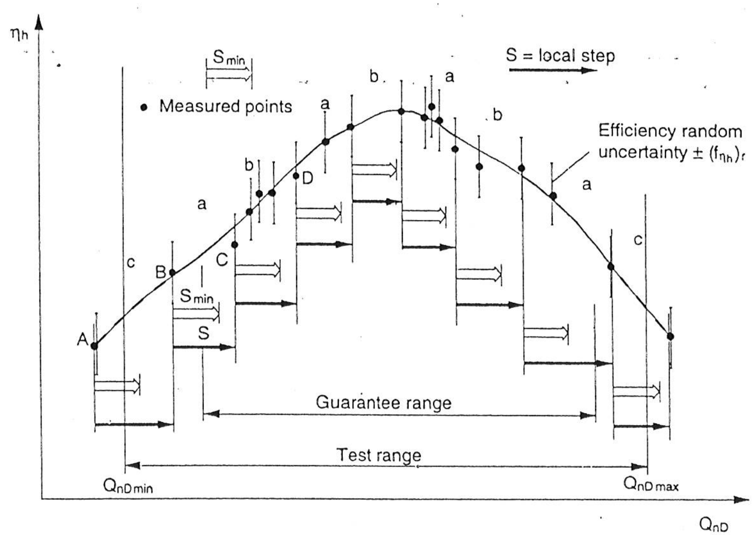

Figure H.2 – Example of determination of intervals

H.3 Choice of the minimum width of the intervals

A value s_{min} of the minimum width of the intervals shall be assumed. The most convenie value is small enough to give a good definition of the interpolation curve. Attention sl drawn to the fact that the choice of s_{min}, and thus of the number of intervals, influenc shape of the curve:

- choosing too large a s_{min} value may result in an oversmoothed curve, which m represent the actual physical phenomenon;
- choosing too small a s_{min} value may result in a curve reflecting the random scatter measurement points rather than the physical reality.

Therefore, the choice of an appropriate s_{min} value shall be left to the judgement of the (tests, taking into account that the interpolation curve shall, as far as possible, inters segments denoting the random uncertainty in each measurement point (see figure H.2).

For example, when drawing the efficiency curve $\eta_h(Q_{nD})$ of a single-regulated turbir 3.8.2.3.1), s_{min} may normally be chosen equal to approximately $(Q_{nDmax} - Q_{nDmin})/10$ Q_{nDmin} and Q_{nDmax} are the limiting values of the test range.

H.4 Determination of the intervals

To simplify the statement below, it is assumed that the process starts from the left en test range, even if in the case of an efficiency curve it may be convenient to start f centre of the range.

From the first measurement point considered, two cases can occur:

a) the next point lies at a distance along the abscissa greater than or equal to s_{min}: I interval s is then defined by these two points (A and B in figure H.2);

b) the next point lies at a distance along the abscissa less than s_{min}: proceeding tow right side, the first measurement point lying at a distance from the starting point than s_{min} shall then be sought; this point, together with the starting point, defines t s of the local interval (C and D in figure H.2).

This procedure proceeds to the end of the test range. For the least squares method wi' degree polynomial being applicable, each group of three adjacent intervals shall c least four measurement points.

Annex J
(informative)

Examples of analysis of sources of error and uncertainty evaluation

s annex may be used as a guide for analyzing sources of errors which may occur during del tests (see table 8 in 3.9.2.1). It contains the following three examples:

example of analysis of sources of error and of uncertainty evaluation in the measurement of a physical quantity (see J.1);

example of calculation of systematic uncertainty in the determination of the specific hydraulic energy, mechanical runner/impeller power and hydraulic efficiency (see J.2);

example of calculation of systematic uncertainty in the determination of the net positive suction specific energy (see J.3).

e to the various ways of determining the required quantities, it is not possible to specify in s standard a general statement for the relevant systematic errors. In the case of specific lraulic energy, for example, the value of the systematic uncertainty depends on the truments and the installation with the choice of instrumentation depending on the value of specific hydraulic energy. At lower values of specific hydraulic energy, greater values of certainty $(f_E)_s$ are generally to be expected. For the same reason, the relative systematic certainty in the determination of the net positive suction specific energy is generally higher n the systematic uncertainty in the determination of the specific hydraulic energy.

I Example of analysis of sources of error and of uncertainty evaluation in the measurement of a physical quantity[1)]

e following example illustrates how the various sources of error arising in the measurement a quantity using an electronic device can be identified and how the corresponding certainties can be evaluated and combined. Table 8 is used as a guide for this analysis. All lues given below are uncertainties at 95 % confidence level.

I.1 Errors arising during calibration

I.1.1 Component errors

Bias of the primary method: this is the systematic component of the intrinsic error of the primary method used for the calibration: $\pm f_a$.

If, for example, the quantity to be measured is the discharge and if the calibration is made using the weighing method, and if the secondary instrument is an electronic device, the main sources of systematic error of the primary method are due to the mechanical operation of the weighing machine, to its proper calibration, to the operation of the diverter, to the chronometer, to the buoyancy correction and to the determination of the density. If all the requirements of ISO 4185 are complied with, the bias of the weighing method can be estimated by combining the component systematic uncertainties of the different error sources.

All the specific terms used in this annex are defined in "International vocabulary of basic and general terms in etrology" (VIM).

b) Repeatability of the primary method: this is the random component of the intrinsic error of the primary method used for the calibration: $\pm f_b$.

In the example of item a) the main sources of random error of the primary method when using the weighing method are the scatter of the readings of the weighing machine (which may be evaluated from the scatter of the readings during the calibration of the weighing machine) and the repeatability of the motion of the diverter.

c) Bias of the secondary instrument: this is the systematic component of the intrinsic error of the secondary instrument: $\pm f_c$.

In the example of item a) it is due mainly to a systematic error in the measurement of the output signal from the electronic device.

d) Repeatability of the secondary instrument: this is the random component of the intrinsic error of the secondary instrument: $\pm f_d$.

In the example of item a) it is due mainly to the random error in the output signal measurement. It may be evaluated from the scatter of the points around the calibration curve.

e) Errors due to physical phenomena and influence quantities: $\pm f_e$.

In this category there may be various sources of error. With reference to the example of point a) are:

- the influence of the flow pattern on the response of the electronic device;
- the unsteadiness of the flow;
- the possible effect of physical properties of the water (conductivity, temperature, etc) on the response of the electronic device;
- the effect of external influence quantities (fluctuations of power supply, ambient temperature, electromagnetic field, etc).

The combination of all these errors can lead to an uncertainty in the indicated discharge which may be partly systematic and partly random in nature.

f) Errors in physical properties: these are the errors arising in the determination of physical quantities either by direct measurement or from international standardized data: $\pm f_f$.

In the example of item a), the main source of error of this type is the determination of the water density when the volume flowrate is deduced from the mass flowrate measured by weighing.

J.1.1.2 Resulting uncertainty in the calibration

All the errors listed above being small, numerous and independant of each other, the systematic and random corresponding uncertainties may be combined by the root-sum-square method (as explained in 3.9.1.4) to obtain the resulting relative uncertainty in the calibration curve:

$$f_{cal} = \pm\left[(f_a)^2 + (f_b)^2 + (f_c)^2 + (f_d)^2 + (f_e)^2 + (f_f)^2\right]^{1/2}$$

J.1.2 Errors arising during the tests

J.1.2.1 Component errors

g) Systematic error in the calibration: $\pm f_{cal}$.

Although the uncertainty determined in J.1.1.2 is partly systematic and partly random in nature, it becomes entirely systematic when the calibration previously determined is used for subsequent tests.

\dditional systematic error: this is the systematic component of the intrinsic error of the secondary instrument, not covered by the calibration: $\pm f_h$.

Errors in physical properties: $\pm f_j$.

In certain cases, for example in the case of item J.1.1.1 a), this category of errors may be omitted.

Errors due to physical phenomena and influence quantities: $\pm f_k$.

The sources of error in this category are the same as those listed in J.1.1.1 e), but their values can be different and vary with the operating point. They may be neglected if the conditions occurring during the calibration are kept the same during the tests.

If not, it is assumed that they result in a systematic component $\pm f_{ks}$ and a random component $\pm f_{kr}$.

Random error: this includes the repeatability of the secondary instrument $\pm f_l$.

This error can be measured during the tests. In the example of J.1.1.1 a), it occurs once again in a similar manner as it did during calibration.

.2.2 Total uncertainty

mbining the component uncertainties by the root-sum-square method, the relative total certainty is obtained:

systematic uncertainty:

$$= \pm[(f_{cal})^2 + (f_h)^2 + (f_j)^2 + (f_{ks})^2]^{1/2}$$

random uncertainty:

$$= \pm[(f_{kr})^2 + (f_l)^2]^{1/2}$$

total uncertainty:

$$= \pm[(f_s)^2 + (f_r)^2]^{1/2}$$

2 Example of calculation of systematic uncertainty in the determination of the specific hydraulic energy, mechanical runner/impeller power and hydraulic efficiency

ssumed that the methods of measurement and their respective uncertainties[1] are as follows:

2.1 Discharge: measurement by electromagnetic flowmeter. The systematic uncertainty is stimated equal to ±0,20 %.

.2.2 Pressure: at the high and low pressure measuring sections, measurement by dead-eight manometer.

.2.3 Specific hydraulic energy: in this case it is given by (see figure 40):

$$E = \frac{p_{abs1} - p_{abs2}}{\overline{\rho}} + g \cdot (z_1 - z_2) + \frac{v_1^2 - v_2^2}{2}$$

[1] The systematic uncertainties depend on many factors, therefore the values assumed here are to be considered only as examples.

If, in a general way, e_x is the absolute systematic uncertainty in the quantity x (thus, the relative systematic uncertainty is $f_x = \frac{e_x}{x}$) then the relative systematic uncertainty in the specific hydraulic energy is given by[1] :

$$(f_E)_s = \pm \frac{(e_E)_s}{E} = \pm \left\{ \frac{\left[(e_{pabs1}/\overline{\rho})^2 + (e_{pabs2}/\overline{\rho})^2 + (ge_{z1})^2 + (ge_{z2})^2 + \left(\frac{e_{v_1^2}}{2} \right)^2 + \left(\frac{e_{v_2^2}}{2} \right)^2 \right]^{1/2}}{\frac{p_{abs1} - p_{abs2}}{\overline{\rho}} + g \cdot (z_1 - z_2) + \frac{v_1^2 - v_2^2}{2}} + f_{\Delta E} \right\}$$

Assuming that:

p_{abs1}	= 10,5 x 10⁵	Pa	f_{pabs1}	= ±0,1	%
p_{abs2}	= 0,5 x 10⁵	Pa	f_{pabs2}	= ±0,2	%
z_1	= 4	m	e_{z1}	= ±0,01	m
z_2	= 2	m	e_{z2}	= ±0,01	m
v_1	= 6	$m \cdot s^{-1}$	f_{v1}	= ±0,2	%
v_2	= 1,5	$m \cdot s^{-1}$	f_{v2}	= ±0,4	%
$\overline{\rho}$	= 1000	$kg \cdot m^{-3}$			
g	= 9,81	$m \cdot s^{-2}$			

and supposing that the uncertainties in $\overline{\rho}$ and g may be neglected and $f_{\Delta E}$ is zero (see 3.5.3), then:

$$e_{pabs1}/\overline{\rho} = (p_{abs1}/\overline{\rho}) f_{pabs1} = \pm 10{,}5 \times (10^5/10^3) \times (0{,}1/100) = \pm 1{,}05 \quad J \cdot kg^{-1}$$

$$e_{pabs2}/\overline{\rho} = (p_{abs2}/\overline{\rho}) f_{pabs2} = \pm 0{,}5 \times (10^5/10^3) \times (0{,}2/100) = \pm 0{,}1 \quad J \cdot kg^{-1}$$

$$ge_{z1} = \pm 9{,}81 \times 0{,}01 = \pm 0{,}1 \quad J \cdot kg^{-1}$$

$$ge_{z2} = \pm 9{,}81 \times 0{,}01 = \pm 0{,}1 \quad J \cdot kg^{-1}$$

$$\frac{e_{v1}^2}{2} = v_1^2 f_{v1} = \pm 36 \times 0{,}2/100 = \pm 0{,}072 \quad J \cdot kg^{-1}$$

$$\frac{e_{v2}^2}{2} = v_2^2 f_{v2} = \pm 2{,}25 \times 0{,}4/100 = \pm 0{,}009 \quad J \cdot kg^{-1}$$

and

$$(f_E)_s = \pm \frac{\left[(1{,}05)^2 + (0{,}1)^2 + (0{,}1)^2 + (0{,}1)^2 + (0{,}072)^2 + (0{,}009)^2\right]^{1/2}}{(1050 - 50) + 9{,}81 \times (4 - 2) + \frac{(36 - 2{,}25)}{2}} = \pm \frac{1{,}07}{1037} = \pm 0{,}1\%$$

In such a case, the relative systematic uncertainty in the specific hydraulic energy is practically equal to that of pressure measurements.

[1] In fact, this formula is only an approximation, v_1^2 and v_2^2 are not independent quantities.

2.4 **Power:** torque measured by the primary method with a systematic uncertainty of ,14 %, the rotational speed by an electronic counter with a systematic uncertainty of ,075 %.

ıe power systematic uncertainty is estimated as:

$$(f_P)_s = \pm\left[(0{,}14)^2 + (0{,}075)^2\right]^{1/2} \% = \pm 0{,}16\ \%$$

2.5 **Hydraulic efficiency:** by combining the systematic uncertainties of the measured ıantities, the systematic uncertainty in the hydraulic efficiency is:

$$\left(f_{\eta_h}\right)_s = \pm \frac{\left(e_{\eta_h}\right)_s}{\eta_h} = \pm\left[(f_Q)_s^2 + (f_E)_s^2 + (f_P)_s^2\right]^{1/2} = \pm\left[(0{,}2)^2 + (0{,}1)^2 + (0{,}16)^2\right]^{1/2} \% = \pm 0{,}27\ \%$$

.3 Example of calculation of systematic uncertainty in the determination of the net positive suction specific energy

ssume that the methods of measurement and their respective uncertainties[1] are as follows:

.3.1. **Discharge:** measurement by electromagnetic flowmeter. The systematic uncertainty is stimated equal to ±0,20 %.

.3.2 **Pressure:** at the low pressure measuring section measured by a dead-weight ıanometer.

.3.3 **Net positive suction specific energy:** in this case it is given by (see figure 45):

$$NPSE = \frac{(p_{abs2} - p_{va})}{\rho_2} + \frac{v_2^2}{2} - g\cdot(z_r - z_2)$$

, in a general way, e_x is the absolute systematic uncertainty in the quantity x (thus, the relative ystematic uncertainty is $f_x = e_x/x$) then the relative systematic uncertainty in the net positive uction specific energy is given by:

$$(f_{NPSE})_s = \pm\frac{(e_{NPSE})_s}{NPSE} = \pm\frac{\left[\left(\frac{e_{pabs2}}{\rho_2}\right)^2 + \left(\frac{e_{pva}}{\rho_2}\right)^2 + \left(\frac{e_{v2}^2}{2}\right)^2 + (ge_{zr})^2 + (ge_{z2})^2\right]^{1/2}}{\frac{(p_{abs2} - p_{va})}{\rho_2} + \frac{v_2^2}{2} - g\cdot(z_r - z_2)}$$

[1] The systematic uncertainties depend on many factors, therefore the values assumed here are to be considered only as examples.

Assuming that:

p_{abs2}	$= 0{,}2 \times 10^5$	Pa	f_{pabs2}	$= \pm 0{,}3\ \%$
z_r	$= 2$	m	e_{zr}	$= \pm 0{,}01$ m
z_2	$= 1$	m	e_{z2}	$= \pm 0{,}01$ m
v_2	$= 1{,}5$	$m\cdot s^{-1}$	f_{v2}	$= \pm 0{,}4\ \%$
ρ_2	$= 1\ 000$	$kg\cdot m^{-3}$		
p_{va}	$= 0{,}03 \times 10^5$	Pa		
g	$= 9{,}81$	$m\cdot s^{-2}$		

and the uncertainties in ρ_2, p_{va} and g may be neglected, then:

$$\frac{e_{pabs2}}{\rho_2} = \frac{p_{abs2}}{\rho_2} \times f_{pabs2} = \pm 0{,}2 \times (10^5/10^3)(0{,}3/100) = \pm 0{,}06 \quad J\cdot kg^{-1}$$

$$ge_{zr} = \pm 9{,}81 \times 0{,}01 = \pm 0{,}1 \quad J\cdot kg^{-1}$$

$$ge_{z2} = \pm 9{,}81 \times 0{,}01 = \pm 0{,}1 \quad J\cdot kg^{-1}$$

$$\frac{e_{v2}^2}{2} = \pm v_2^2 \cdot f_{v2} = \pm 2{,}25 \times 0{,}4/100 = \pm 0{,}009 \quad J\cdot kg^{-1}$$

$$\left(f_{NPSE}\right)_s = \pm\frac{\left[(0{,}06)^2 + (0{,}1)^2 + (0{,}1)^2 + (0{,}009)^2\right]^{1/2}}{(20-3) + \frac{2{,}25}{2} - 9{,}81 \times (2-1)} = \pm\frac{0{,}1539}{8{,}315} = \pm 1{,}85\ \%$$

Annex K
(normative)

Efficiency scale-up for Pelton turbines

s annex, based on theoretical considerations and experimental data, summarizes the best ›roximation available at the present time. Research work currently in progress may lead to a re precise knowledge of the efficiency scale-up for Pelton turbines. Until these new results available, it is good practice to apply the scale-up formula given in K.2 for transposition n a model to the prototype.

1 Similarity considerations

can be shown by dimensional analysis (see appendix A to reference K.4.1), the losses in lton machines are controlled by five dimensionless parameters:

- the Reynolds number Re (see 1.3.3.11.1);
- the Froude number Fr (see 1.3.3.11.2);
- the Weber number We (see 1.3.3.11.3);
- the speed factor n_{ED} (see 1.3.3.12.1) or energy coefficient E_{nD} (see 1.3.3.12.5);
- the specific flow rate Φ_B defined as follows:

$$\Phi_B = \frac{4Q}{z_o \cdot \pi \cdot (2E)^{1/2} \cdot B^2}$$

ıere

	$[m^3 \cdot s^{-1}]$	is the discharge (see 1.3.3.4.1);
	[-]	is the number of nozzles;
	$[J \cdot kg^{-1}]$	is the specific hydraulic energy of machine (see 1.3.3.6.1);
	[m]	is the bucket width (see 1.3.3.2.8).

eferring to 2.3.1.1, the following characteristic values are used to define the above mensionless numbers:

- $(2E)^{1/2}$ as characteristic velocity v_C;
- bucket width B as characteristic length L_C.

or two geometrically similar Pelton turbines (e.g. a prototype machine and a corresponding ıodel machine), the following ratios of similitude numbers can be made:

$$C_{Fr} = \frac{Fr_P}{Fr_M} = \left(\frac{E_P}{E_M}\right)^{1/2} \cdot \left(\frac{B_M}{B_P}\right)^{1/2} \cdot \left(\frac{g_M}{g_P}\right)^{1/2}$$

$$C_{We} = \frac{We_P}{We_M} = \left(\frac{E_P}{E_M}\right)^{1/2} \cdot \left(\frac{B_P}{B_M}\right)^{1/2} \cdot \left(\frac{\rho_P}{\rho_M}\right)^{1/2} \cdot \left(\frac{\sigma_M^*}{\sigma_P^*}\right)^{1/2}$$

$$C_{Re} = \frac{Re_P}{Re_M} = \left(\frac{E_P}{E_M}\right)^{1/2} \cdot \frac{B_P}{B_M} \cdot \frac{\nu_M}{\nu_P}$$

These ratios are used to define functions which describe the efficiency scaling effects between two geometrically similar Pelton turbines when operating at the same speed factor n_{ED} or energy coefficient E_{nD}, with the specific flow rate Φ_B being the most important parameter for C_{Re}, C_{Fr} and C_{We}. The functions result from the analysis of numerous results of efficiency tests performed with the same model at different test conditions and from comparison of measured efficiencies of prototype and model turbines being fully or partly homologous.

The resulting differences in hydraulic efficiency $\Delta\eta_h$ are presented in figure K.1 for the influence of Froude number, in figure K.2 for the influence of Weber number and in figure K.3 for the influence of Reynolds number.

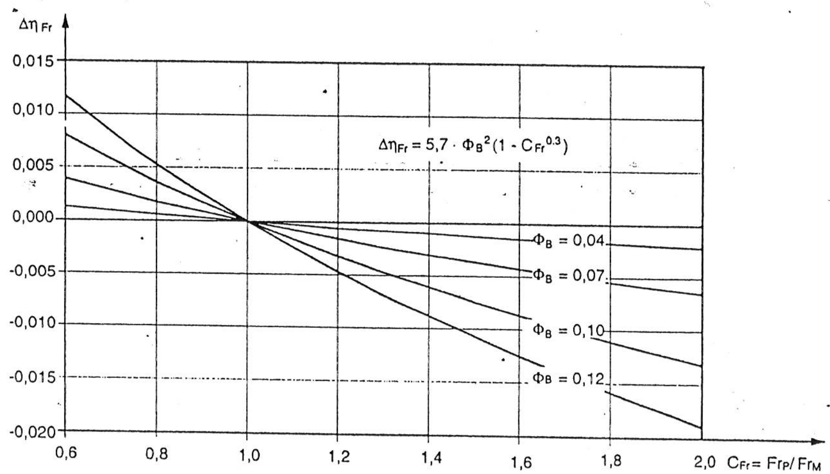

Figure K.1 – Influence of Froude number

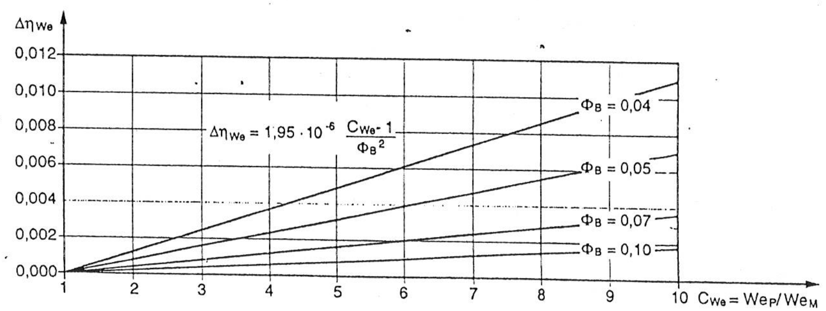

Figure K.2 – Influence of Weber number

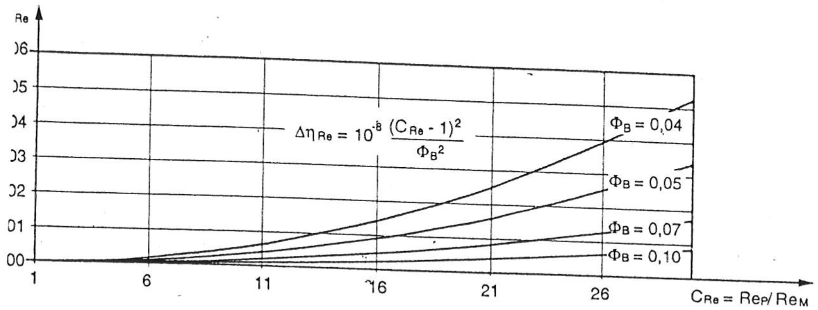

Figure K.3 – Influence of Reynolds number

2 Step-up procedure

he guarantees are referred to prototype, the model hydraulic efficiencies η_{hM} are scaled up prototype conditions using the following formula:

$$\Delta\eta_h = \eta_{hP} - \eta_{hM} = \Delta\eta_{Fr} + \Delta\eta_{We} + \Delta\eta_{Re}$$

$$\Delta\eta_h = \eta_{hP} - \eta_{hM} = 5{,}7 \cdot \Phi_B^{\,2}\left(1 - C_{Fr}^{\,0,3}\right) + 1{,}95 \cdot 10^{-6}\,\frac{C_{We} - 1}{\Phi_B^{\,2}} + 10^{-8}\,\frac{\left(C_{Re} - 1\right)^2}{\Phi_B^{\,2}}$$

3 Numerical data for surface tension σ*

Temperature θ °C	Surface tension σ* J·m⁻²
5	0,0749
10	0,0742
15	0,0735
20	0,0728
25	0,0720
30	0,0712
35	0,0696

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Annex L
(normative)

Analysis of random uncertainties for a test at constant operating conditions[1)]

Repeated measurements at one operating point may be expected to show differences in measurements but their mean value is a better estimate of the true value than any individual measurement. The accuracy of the mean value depends on the number of measurements and their individual deviations from the mean (scatter).

It is possible to calculate statistically the uncertainty in a measurement of a variable when the associated error is purely random in nature. To do this it is necessary to calculate the standard deviation and to decide on the confidence level which is to be attached to the uncertainty. For this standard, a 95 % confidence level shall be used.

L.1 Standard deviation

The exact value of standard deviation σ of any measured parameter is rarely known exactly: usually only an estimate s of σ is available, based on a limited number of observations.

If the error in the measurement of a quantity Y is purely random, then, when n independent measurements of the quantity are made, the standard deviation[2)] of the distribution of results s_Y is given by the equation:

$$s_Y = \left[\frac{\sum_{r=1}^{n}\left(Y_r - \overline{Y}\right)^2}{n-1}\right]^{1/2}$$

where

$\overline{Y}$ is the arithmetic mean of the n measurements of the variable Y;

Y_r is the value obtained by the r^{th} measurement of the variable Y;

n is the total number of measurements of the variable Y.

For brevity, s_Y is normally referred to as "the standard deviation of Y". The square of the standard deviation s_Y^2, is called the variance.

The random error in the result can be reduced by making as many measurements as possible of the variable and using the arithmetic mean value, since the standard deviation of the mean of n independent measurements is $\sqrt{n}$ times smaller than the standard deviation of the measurements themselves.

Thus, the standard deviation of the mean, $s_{\overline{Y}}$ is given by the equation:

$$s_{\overline{Y}} = \frac{s_Y}{\sqrt{n}}$$

1) The text of this annex is based on ISO 5168.

2) Standard deviation, as defined here, is what is more accurately referred to as the "estimated standard deviation" by statisticians.

2 Confidence levels

the true standard deviation σ_Y, is known (as n approaches infinity, s_Y approaches σ_Y), the nfidence level can be related to the uncertainty of measurements as indicated in table L.1.

Table L.1 – Confidence levels

Uncertainty	Confidence level
$\pm 0,674 \cdot \sigma_Y$	0,50
$\pm 0,954 \cdot \sigma_Y$	0,66
$\pm 1,960 \cdot \sigma_Y$	0,95
$\pm 2,576 \cdot \sigma_Y$	0,99

›r example, the interval $Y_r \pm 1,96 \cdot \sigma_Y$ would be expected to contain 95 % of the population. ıat is to say, where a single measurement of the variable Y is made and where the value of · is independently known, there would be a probability of 0,05 of the interval $Y_r \pm 1,96 \cdot \sigma_Y$ ›t including the true value.

practice, of course, it is only possible to obtain an estimate of the standard deviation since ı infinite number of measurements would be required in order to determine it precisely, and e confidence limits must be based on this estimate. The Student's "t distribution" for small ımples should be used to relate the required confidence level to the interval.

.3 Student's t distribution

ıe uncertainty at 95 % confidence level may be found as follows:

if n is the number of measurements, (n-1) is taken as the number of degrees of freedom, ν;

the value of t for the appropriate number of degrees of freedom is read in the table L.2;

the standard deviation s_Y of the distribution of the measurements of the quantity Y is calculated as stated in clause L.1;

) the range of values within which any reading would be expected to be with 95 % confidence is

$$\overline{Y} \pm t \cdot s_Y;$$

) the difference between a new reading and the average of the sample should be less than

$$t \cdot s_Y \cdot \sqrt{1+1/n}\,;$$

the range of values within which the true value of the quantity would be expected to lie with 95 % confidence, i.e. the band of uncertainty, is

$$\overline{Y} \pm \frac{t \cdot s_Y}{\sqrt{n}} = \overline{Y} \pm t \cdot s_{\overline{Y}}$$

Table L.2 – Values of Student's t

Degrees of freedom	Student's t	$\frac{t}{\sqrt{n}}$
ν = n -1	For 95 % confidence level	
1	12,706	8,984
2	4,303	2,484
3	3,182	1,591
4	2,776	1,241
5	2,571	1,050
6	2,447	0,925
7	2,365	0,836
8	2,306	0,769
9	2,262	0,715
10	2,228	0,672
11	2,201	0,635
12	2,179	0,604
13	2,160	0,577
14	2,145	0,554
15	2,131	0,533
20	2,086	0,455
30	2,042	0,367
60	2,000	0,256
∞	1,960	0

For other values of ν, t can be computed from the following empirical equation:

$$t = 1,96 + 2,36/\nu + 3,2/\nu^2 + 5,2/\nu^{3,84}$$

L.4 Maximum permissible value of random uncertainty

If the required range of accepted random uncertainty associated with $\overline{Y}$ is $\pm e_{r\,max}$, then

$$e_r = \frac{t \cdot s_Y}{\sqrt{n}}$$

should not exceed $e_{r\,max}$.

Alternatively, for a value of $e_{r\,max}$ associated with the 95 % confidence level, the estinated standard deviation s_Y should not exceed the value of

$$s_{Y\,max} = \frac{e_{r\,max} \cdot \sqrt{n}}{t}$$

For convenience, values of $\frac{t}{\sqrt{n}}$ are given in table L.2.

e mean value of a series of points which meet the above criteria will be acceptable. Table ! applies only to repeated points at constant operating conditions.

5 Example of calculation

e following example illustrates the computation of the estimated standard deviation and the certainty for n = 8 observations of a quantity Y.

Table L.3

Measured values Y_i	$\overline{Y}-Y_i$	$\left(\overline{Y}-Y_i\right)^2$
92,80	−0,15625	0,024414
92,70	−0,05625	0,003164
92,60	+0,04375	0,0019141
92,50	+0,14375	0,0206641
92,70	−0,05625	0,0031641
92,75	−0,10625	0,0112891
92,50	+0,014375	0,0206641
92,60	+0,04375	0,0019141

$$\overline{Y}=\frac{1}{n}\sum_{i=1}^{n}Y_i=92{,}64375 \qquad \sum_{i=1}^{n}\left(\overline{Y}-Y_i\right)^2=0{,}0871876$$

stimated standard deviation of the observations:

$$s_Y=\sqrt{\frac{\sum\left(\overline{Y}-Y_i\right)^2}{n-1}}=\sqrt{\frac{0{,}0871876}{8-1}}=0{,}111604$$

:andom uncertainty associated with the mean value at the 95 % confidence level:

$$\left(e_Y\right)_r=\pm\frac{t\cdot s_Y}{\sqrt{n}}=\pm0{,}111604\times0{,}836=\pm0{,}0933$$

$$\left(f_Y\right)_{r95}=\frac{\left(e_Y\right)_r}{\overline{Y}}=\pm\frac{0{,}0933}{92{,}6437}=\pm0{,}1\%$$

n the case of the examined measure being the efficiency, it should then be verified that this 'alue of the observed random uncertainty $(f_{\eta h})_r$ does not exceed the maximum permissible andom uncertainty agreed prior to the test (see 3.9.2.2.1).

Annex M
(normative)
Calculation of plant Thoma number σ_{pl}

M.1 Definition of σ_{pl}, NPSE and NPSH

These terms are referred to the low pressure side of the machine and are in direct relation with the cavitation phenomenon. The symbol σ_{pl} denotes the range of the Thoma number (see 1.3.3.12.9) occurring at plant conditions and is defined as follows:

$$\sigma_{pl}=\frac{NPSE}{E}=\frac{NPSH}{H}$$

Subclause 11.3 of IEC 60041 explains various possibilities for calculating NPSE of a prototype machine, as illustrated and explained below. If the pressure p_{abs2} (see figure M.1) at point 2 inside the draft tube is known, NPSE can be calculated in case of a turbine or a pump according to the following formula:

$$NPSE=g_2\cdot NPSH=\frac{p_{abs2}-p_{va}}{\rho_2}+\frac{v_2^2}{2}-g_2\cdot\left(z_r-z_2\right)$$

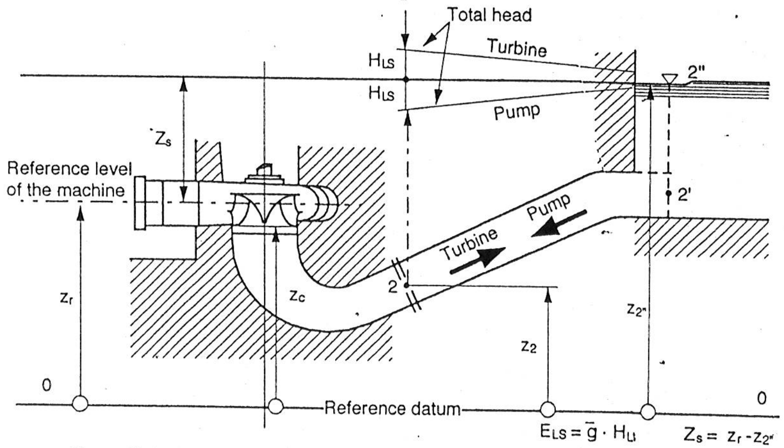

Figure M.1 – Definition for determination of net positive suction energy, NPSE, and net positive suction head, NPSH, of a prototype machine ($E_{LI}\neq 0$)

sually only the tailwater level for specified hydraulic conditions is known. In that case, NPSE calculated according to the following formula using the tailwater level $z_{2''}$ close to the draft be outlet (turbine) or inlet (pump) and considering the specific hydraulic energy losses E_{LS} etween sections 2 and 2':

$$NPSE = g_2 \cdot NPSH = \frac{p_{amb2''} - p_{va}}{\rho_2} + \frac{v_{2'}^2}{2} - g_2 \cdot \left(z_r - z_{2''}\right) \pm E_{LS}$$

$$NPSE = g_2 \cdot NPSH = \frac{p_{amb2''} - p_{va}}{\rho_2} + \frac{v_{2'}^2}{2} - g_2 \cdot Z_S \pm E_{LS}$$

(+ for turbines, – for pumps)

σ_{pl} is usually referred to the reference level of the machine z_r (see 1.3.3.7.6). As recommended 2.3.1.5.1 the cavitation reference level z_c shall be chosen corresponding to the location here the relevant cavitation occurs. Consequently, σ_{pl} shall also be referred to z_c, and the orresponding definition is identified as σ_{plc}:

$$\sigma_{plc} = \frac{NPSE - g_2\left(z_c - z_r\right)}{E}$$

$$\sigma_{plc} = \frac{\frac{p_{amb2''} - p_{va}}{\rho_2} + \frac{v_{2'}^2}{2} - g_2\left(z_c - z_{2''}\right) \pm E_{LS}}{E}$$

/hen section 2' and the corresponding tailwater level 2" are far from the draft tube opening, nd it can be assumed $v_{2'} \approx 0$ the formula changes to:

$$\sigma_{plc} = \frac{\frac{p_{amb2''} - p_{va}}{\rho_2} - g_2\left(z_c - z_{2''}\right) \pm E_{LS}}{E}$$

1.2 Data needed to calculate σ_{plc}

ıs stipulated in 1.4.1.1, most of the following site data shall be provided or specified by the ›urchaser:

ı) Constant plant data

The following data are usually assumed as constant data, i.e. they do not depend on operating conditions.

– Ambient pressure p_{amb}

If not specified, p_{amb} is taken from table B.6 as a function of the tailwater level $z_{2''}$ (average value).

– Water temperature θ_w, θ_{wmax}

The average value θ_w and the maximum value θ_{wmax} shall be specified. The value θ_w is to be used for calculation of E, P_h and η_{hM}, whereas θ_{wmax} shall be used to get the lowest possible value of σ_{plc}.

– Vapour pressure p_{va}

The value p_{va} is taken from table B.4 as a function of water temperature θ_w. To determine the minimum value of σ_{plc}, it is necessary to use the value θ_{wmax}.

– Draft tube cross section A_2 or measuring section A_2'

To calculate the mean flow velocity v_2 or $v_{2'}$, it is to be agreed on which cross-section area shall be used (except if it is assumed that $v_{2'} = 0$).

– Density of water ρ_2

This value is to be taken from table B.2 as a function of θ_w or θ_{wmax} (negligible impact on resulting σ_{plc}).

– Reference level z_r

The reference level z_r is defined by figure 5 which corresponds to figure 8 in IEC 60041. This value is usually specified on the main section drawing and/or in the technical specification.

– Cavitation reference level z_c

The level z_c shall be defined by mutual agreement. For example, for large tubular turbines with horizontal shaft, more than one level z_c may be agreed on.

b) Variable plant data

The following data usually depend on the operating conditions of the machine.

– Tailwater level $z_{2''}$

The variation of $z_{2''}$ is usually specified as a function of the specific hydraulic energy E.

– Specific hydraulic energy E

The range and the relevant values of E shall be given in the technical specification. Sometimes only the headwater and tailwater levels and the resulting values of geodesic height (gross head) are given. In this case E shall be calculated by considering the relevant energy losses E_{LS} on the high and low pressure sides of the machine.

– Mean velocity v_2 or $v_{2'}$

v_2 or $v_{2'}$ are calculated using the agreed relevant draft tube cross-section area A_2 or $A_{2'}$ and the discharge Q for each specified operating point where cavitation tests are to be performed.

– Specific hydraulic energy losses E_{LS}

In case such losses are to be considered, they are usually specified and depend on discharge, i.e. on Q^2.

In view of cavitation tests, it is recommended to summarize in a separate document the relevant data for calculation of the various σ_{plc} values. It is also helpful to add a schematic drawing similar to figure M.1. The following table M.1 illustrates as an example how the resulting σ_{plc} values and other relevant data can be summarized.

Table M.1 – Summary of calculated σ_{pl} values and other relevant data

E (J/kg)	$z_{2'}$ (m)		Q (m^3/s)		$v_{2'}$(m/s)		E_{Ls} (J/kg)		σ_{plc} (-)	
	max.	min.	max.	min.	max.	min.	max.	min.	max.	min.
Maximum										
Specified										

Minimum										

Annex N
(informative)

Detailed flux diagram of specific hydraulic energy, flow and power

As a supplement to figure 6, a more detailed analysis of internal losses in the runner/impeller of a reaction machine is presented in figures N.1 and N.2. According to recent publications, this analysis is needed to determine the scale effect on efficiency and power and also on specific hydraulic energy.

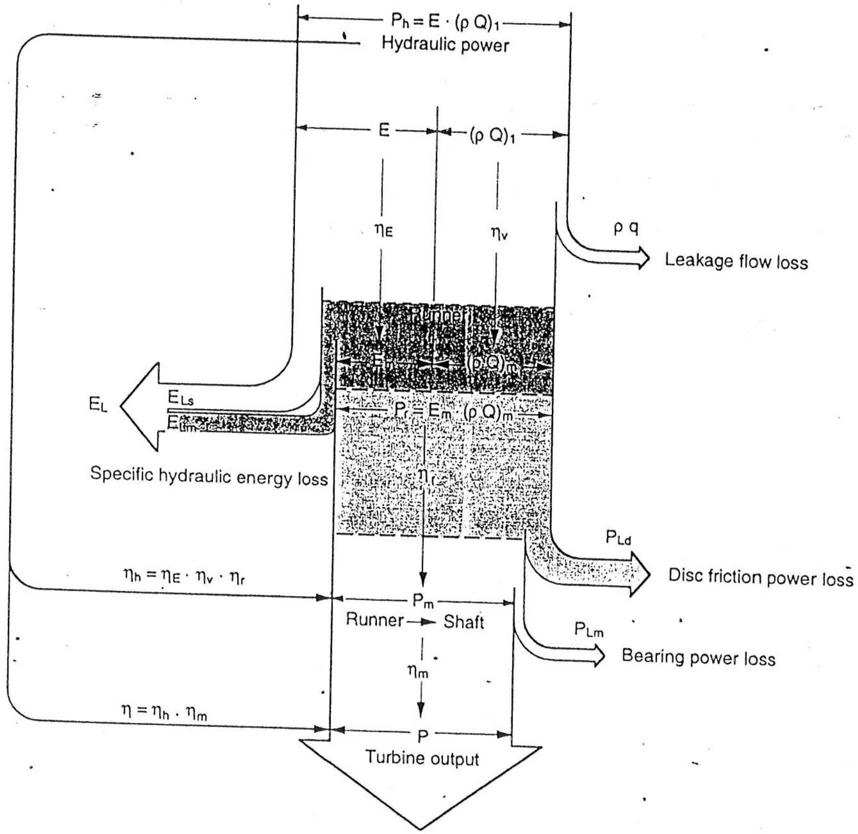

Figure N.1 – Turbine

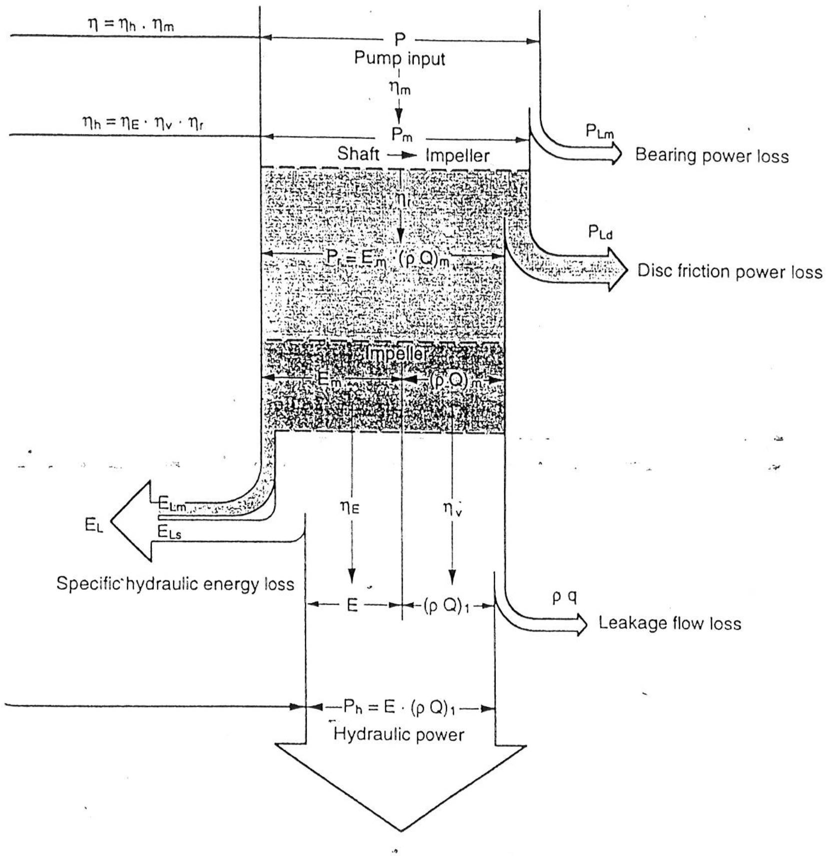

Figure N.2 – Pump

The following quantities: P, P_h, P_m, P_{Lm}, Q, q, E, E_L, ρ, η, η_h, η_m and η_v are defined in 1.3. Other quantities are defined below:

P_r is the hydraulic power transmitted from water to the runner (turbine) or from impeller to water (pump);

P_{Ld} is the hydraulic power dissipated in the chambers between the outer surface of runner/impeller and the corresponding stationary walls;

E_m is the specific hydraulic energy available for runner to produce power (turbine) or imparted to the water by impeller (pump);

Q_m is the volume of water per unit time passing through runner/impeller blades;

E_{Lm} is the specific hydraulic energy loss in runner/impeller blades;

E_{Ls} is the specific hydraulic energy loss in stationary parts;

E_L is the specific hydraulic energy loss between high (low) pressure section and low (high) pressure section of turbine (pump);

η_E is the specific energy efficiency of runner/impeller given by the ratio of E_m/E (turbine) or by the ratio of E/E_m (pump);

η_r is the power efficiency of runner/impeller given by the ratio of P_m/P_r (turbine) or P_r/P_m (pump).

Annex P
(informative)

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fairly well ❑
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Q6 If you ticked NOT AT ALL in Question 5 the reason is: *(tick all that apply)*

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standard is incomplete ❑
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title is misleading ❑
I made the wrong choice ❑
other ....

Q7 Please assess the standard in the following categories, using the numbers:
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(3) average,
(4) above average,
(5) exceptional,
(6) not applicable

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quality of writing...
technical contents...
logic of arrangement of contents ....
tables, charts, graphs, figures..............
other ....

Q8 I read/use the: *(tick one)*

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