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Rotodymanic pumps — Hydraulic performance acceptance test using a model pump

Pompes rotodynamiques — Modèle réduit de pompe utilisé pour les essais de performance hydraulique

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Foreword

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The procedures used to develop this document and those intended for its further maintenance are described in the ISO/IEC Directives, Part 1. In particular the different approval criteria needed for the different types of ISO documents should be noted. This document was drafted in accordance with the editorial rules of the ISO/IEC Directives, Part 2 (see www.iso.org/directives).

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights. Details of any patent rights identified during the development of the document will be in the Introduction and/or on the ISO list of patent declarations received (see www.iso.org/patents).

Any trade name used in this document is information given for the convenience of users and does not constitute an endorsement.

For an explanation on the voluntary nature of standards, the meaning of ISO specific terms and expressions related to conformity assessment, as well as information about ISO's adherence to the World Trade Organization (WTO) principles in the Technical Barriers to Trade (TBT) see the following URL: www.iso.org/iso/foreword.html.

This document was prepared by ISO/TC 115, *Pumps*, SC 2, *Methods of measurement and testing*.

Introduction

Wherever the capacity of a manufacturer's test facility is not appropriate to realise the necessary physical preconditions for testing a pump at realistic flow/head conditions the alternative of a model pump is taken. By means of the similitude theory, a model pump is used to assess and calculate the ability of the real pump to be built. The option using such model pump or prototype pump is chosen

— when the capacity of the pump, namely its flow rate and/or its power input (e.g. flowrate ≥35,000 m^3/h, and P_2 ≥ 5,000 kW), exceeds the limitations of the test facility, or

— one part or parts of the pump should be constructed by concrete walls and reproduction of the whole assembly is impractical.

In consideration of these given facts the application of a model pump for the hydraulic performance acceptance test is an efficient and effective alternative. The advantages using a model pump may also include:

— a higher precision due to the difference in measurement uncertainties;

— minimising costs in respect to material and other resources;

— and shorter delivery period(s) of the prototype pump(s).

For many years, manufacturers have developed and specified independent calculation approaches and collected experiences to handle the similitude theory for pumps and their specifics. Several calculation models are described in the pertinent literature. This document describes testing methods using model pumps for hydraulic performance acceptance tests in addition to other testing methods given in ISO 9906 as hydraulic performance acceptance tests for prototype pumps.

This document has been initially established based on prior standards such as the Japanese Industrial Standard JIS B 8327. This document combined with ISO 9906 presents new testing methods for hydraulic acceptance tests of pumps.

Rotodymanic pumps — Hydraulic performance acceptance test using a model pump

1 Scope

This document describes hydraulic performance tests (including cavitation tests) using a small size pump (centrifugal, mixed flow or axial pump, hereinafter referred to as a “model pump”).

This document is used for pump acceptance tests with a geometrically similar model pump to guarantee the performance of a large size pump manufactured for practical use (hereinafter, a “prototype pump”). This document, however does not preclude a temporary assembly inspection or other tests on the prototype pump. Moreover, it is preferable to conduct the tests with prototype pumps unless

— the capacity of the pump, namely its flow rate and/or its power input, is beyond the limitations of the test facility, though it is difficult to set a criterion for carrying out a model pump test instead of the prototype pump test in terms of the volume rate of flow or the power input,

— a part of the pump is to be constructed by concrete walls and reproduction of the whole assembly is impractical,

— model tests are specified by the purchaser, or

— it is difficult to carry out the prototype pump test due to any other reasons.

This document applies to performance tests under steady operating conditions corresponding to the prototype pump.

2 Normative references

The following documents are referred to in the text in such a way that some or all of their content constitutes requirements of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ISO 17769-1, *Liquid pumps and installation — General terms, definitions, quantities, letter symbols and units — Part 1: Liquid pumps*

ISO 17769-2, *Liquid pumps and installation — General terms, definitions, quantities, letter symbols and units — Part 2: Pumping system*

3 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 17769-1 and ISO 17769-2 and the following apply.

ISO and IEC maintain terminological databases for use in standardization at the following addresses:

— ISO Online browsing platform: available at https://www.iso.org/obp

— IEC Electropedia: available at http://www.electropedia.org/

3.1 General terms

3.1.1
performance test
test to examine the performance of a pump in a state free from the influence of cavitation

3.1.2
cavitation test
test to examine whether pump total head changes happen due to the occurrence of cavitation under operating conditions of a model pump corresponding to the working conditions of a prototype pump

Note 1 to entry: Cavitation test corresponds to NPSH Type III test in ISO 9906:2012.

3.1.3
NPSH3 test
test to reduce the *NSPH* of a model pump and determine the *NSPH* value at which the pump total head of a model pump is reduced by 3 % due to the occurrence of cavitation compared with the pump total head measured without the occurrence of cavitation

Note 1 to entry: NPSH 3 test corresponds to NPSH Type I or II test in ISO 9906:2012.

Note 2 to entry: NPSH is an abbreviation for “net positive suction head”.

3.1.4
four quadrant characteristic test
test to examine the characteristics of a model pump regarding its pump range, pump brake range, water turbine range, water turbine brake range and reverse pump range

Note 1 to entry: The purpose is to obtain the characteristics necessary for the calculation of pump transient phenomena.

3.1.5
specified speed of rotation
speed of rotation of a model pump selected to indicate the performance of the model pump corresponding to the requirements on a prototype pump determined by the agreement between the purchaser and manufacturer

3.1.6
test speed of rotation
measured speed of rotation of a model pump in a performance test or cavitation test on the pump

3.1.7
specified volume rate of flow
volume rate of flow at the specified speed of rotation of a model pump corresponding to the requirements on a prototype pump determined by the agreement between the purchaser and manufacturer

3.1.8
specified pump total head
pump total head at the specified speed of rotation and volume rate of flow of a model pump corresponding to the requirements on a prototype pump determined by the agreement between the purchaser and manufacturer

3.2 Terms and definitions relating to performance

3.2.1
acceleration of gravity
g
acceleration due to gravity
local value used, the local value of the acceleration of gravity is calculated by the following formula:

$$g = 9{,}780\,3 \times \left(1 + 0{,}005\,3 \times \sin^2 \varphi\right) - 3{,}0 \times 10^{-6} \cdot Z$$

where

Z is the altitude, expressed in metres (m);

φ is the latitude, expressed in degrees [°].

Note 1 to entry: In many cases, however, no notable error occurs when 9,80 m/s^2 is used.

3.2.2
Reynolds number
Re
ratio of inertial force to viscous force

The Reynolds numbers used for hydraulic efficiency conversion for a model pump and a prototype pump are given by the following formulae:

$$Re_{\text{hP}} = \frac{u_{1\text{P}} \cdot D_{1\text{P}}}{\nu_{\text{P}}}$$

for the prototype pump

$$Re_{\text{hM}} = \frac{u_{1\text{M}} \cdot D_{1\text{M}}}{\nu_{\text{M}}}$$

for the model pump

where

Re_{hM} is the Reynolds number of the model pump, dimensionless (—);

Re_{hP} is the Reynolds number of the prototype pump, dimensionless (—);

$u_{1\text{M}}$ is the peripheral velocity at the impeller inlet diameter of the model pump, expressed in metres per second (m/s), $u_{1\text{M}} = \pi \cdot D_{1\text{M}} \cdot n_{\text{M}}$;

$u_{1\text{P}}$ is the peripheral velocity at the impeller inlet diameter of the prototype pump, expressed in metres per second (m/s), $u_{1\text{P}} = \pi \cdot D_{1\text{P}} \cdot n_{\text{P}}$;

$D_{1\text{M}}$ is the inlet diameter of the impeller of the model pump, expressed in metres(m);

$D_{1\text{P}}$ is the inlet diameter of the impeller of the prototype pump, expressed in metres(m);

ν_{M} is the kinematic viscosity of liquid in the model pump, expressed in square metres per second (m^2/s);

ν_{P} is the kinematic viscosity of liquid in the prototype pump, expressed in square metres per second (m^2/s);

n_{M} is the speed of rotation of the model pump, expressed in reciprocal seconds (s^{-1});

n_{P} is the speed of rotation of the prototype pump, expressed in reciprocal seconds (s^{-1}).

3.2.3
peripheral velocity
u
speed of a rotor in the tangential direction

3.2.4
pipe friction loss coefficient
λ
coefficient used for calculating the loss of head due to friction in a pipe

3.2.5
equivalent diameter
D_e
the cross-sectional area divided by the wetted perimeter of a hydraulic passageway and multiplied by 4

3.2.6
hydraulic efficiency
η_h
proportion of the pump total head to the theoretical head (impeller head when there is no loss of head)

Note 1 to entry: It should be noted that the definition of hydraulic efficiency in this document is different from that in ISO 17769-1. In ISO 17769-1, where hydraulic efficiency involves all hydraulic losses such as those resulting from friction due to the relative motion of internal surfaces and internal leakage. In this Document, on the other hand, disc friction losses at impellers and internal leakage losses are classified into the factor for mechanical efficiency and volumetric efficiency, respectively, and out of scope of hydraulic efficiency.

3.2.7
hydraulic efficiency ratio
F_h
ratio between the hydraulic efficiency of a prototype pump and the hydraulic efficiency of a model pump at a mutually corresponding operating point

3.2.8
mechanical efficiency
η_m
proportion of the power that an impeller transmits to a liquid to the pump power input

Note 1 to entry: It should be noted that the definition of mechanical efficiency in this document is different from that in ISO 17769-1. Here, the loss of power at the seals and bearings is out of scope (it should be dealt with separately) and the loss of power due to disc friction is considered as the factor, while loss of power at seals and bearings is taken as factor as in ISO 17769-1.

3.2.9
mechanical efficiency ratio
F_m
ratio between the mechanical efficiency of a prototype pump and the mechanical efficiency of a model pump at a mutually corresponding operating point

3.2.10
volumetric efficiency
η_v
proportion of the volume rate of flow of a pump and the volume rate of flow passing through the impeller

Note 1 to entry: It should be noted that the definition of volumetric efficiency in this document is different from that in ISO 17769-1. The definition given in ISO 17769-1 seems applicable only for positive displacement pumps, while the definition in this Technical Report is for rotodynamic pumps.

3.2.11
volumetric efficiency ratio
F_v
ratio between the volumetric efficiency of a prototype pump and the volumetric efficiency of a model pump at a mutually corresponding operating point

3.2.12
scale effect coefficient
V
proportion of the loss due to the scale effect to the combination of scalable and non-scalable losses

Note 1 to entry: The loss due to the scale effect is equal to a loss due to friction of wall surface of flow passage.

3.2.13
cavitation coefficient
σ
NPSH divided by the velocity head for the peripheral velocity at the impeller inlet given by the following formula:

$$\sigma = \frac{g \cdot NPSH}{u_1^2/2}$$

where

$NPSH$	is the net positive suction head, expressed in metres (m);
u_1	is the peripheral velocity at the inlet diameter of the impeller, expressed in metres per second (m/s);
σ	is the cavitation coefficient, dimensionless (—).

Note 1 to entry: The cavitation coefficient is a quantity deduced from the hydraulic similarity rule of pumps for the best efficiency point and is nearly constant among similar pumps regardless of size and speed of rotation.

4 Symbols and suffixes

Table 1 — Main symbols and units used in this document

Symbol	Quantity	Unit
A	Area	m^2
D	Diameter	m
e	Surface roughness	m
e	Uncertainty	Unit of corresponding measuring quantity
F	Efficiency ratio	Dimensionless
F_a	Axial force	N
f	Frequency	s^{-1}
g	Acceleration of gravity	m/s^2
H	Head, Loss of head	m
H	Pump total head	m
K	Type number	Dimensionless
k	Coverage factor	Dimensionless
L, l	Length or distance	m
N	Number of measurement sets	Dimensionless
$NPSH$	Net positive suction head	m
$NPSHA$	Net positive suction head available	m
$NPSH3$	Net positive suction head required for a drop of 3 % of the pump total head of the first stage of the pump	m
n	Speed of rotation	s^{-1}
P (P_2)	Pump power input	W
P_h	Pump power output	W
p	Pressure	Pa
Q	Volume rate of flow	m^3/s
Re	Reynolds number	Dimensionless
s	Standard deviation	Unit of corresponding measuring quantity

Table 1 *(continued)*

Symbol	Quantity	Unit
T	Torque	Nm
t_d	Student's t-distribution	Dimensionless
t	Time	s
U	Expanded uncertainty, relative expanded uncertainty	Unit of corresponding measuring quantity or %
$\bar{v}$	Mean velocity (for flow in pipe), peripheral velocity (for flow in pump)	m/s
u	Uncertainty, relative uncertainty	Unit of corresponding measuring quantity or %
V	Scale effect coefficient	Dimensionless
v	Local velocity	m/s
X, x	Measuring quantity	Unit of corresponding measuring quantity
Z	Altitude	m
α	Influence factor of pump total head in hydraulic efficiency ratio between prototype and model pumps	Dimensionless
β	Influence factor of pump power input in hydraulic efficiency ratio between prototype and model pumps	Dimensionless
Δ	Increment of variation	Unit of corresponding measuring quantity
ε	Fluctuation width	Dimensionless
η	Efficiency	Dimensionless
λ	Friction coefficient of pipe	Dimensionless
υ	Kinematic viscosity	m^2/s
ρ	Density	kg/m^3
σ	Cavitation coefficient	Dimensionless
τ	Tolerance	Dimensionless
φ	Latitude	degree (°)

Table 2 — Characters used as suffixes and their meanings

Suffix	Meaning
1	Suction or inlet
2	Discharge or outlet (except for P_2)
a	Axial direction
B	Wetted perimeter
c	Combined uncertainty
d	Discharge pipe
e	Equivalent
e	Expanded uncertainty
ED	Dimensionless coefficient for four quadrant characteristics
f	Frictional resistance
G	Guarantee point
H	Pump total head
h	Hydraulic
i, j	Integer numbers of measurement sets (1, 2, 3, ...)
M	Model pump

Table 2 *(continued)*

Suffix	Meaning
m	Mechanical
N	Number of measurement sets
P	Prototype pump
Q	Volume rate of flow
r	Type A uncertainty
r	Radial direction
s	Suction pipe
s	Type B uncertainty
t	Total
V	Volumetric
x	Coordinate axis
y	Coordinate axis

5 Test types and measurement items

The tests shown in Table 3 should be carried out. Tests 2. and 3. should be conducted when specified in the agreement between the purchaser and manufacturer. As a rule, the same model pump should be used in both these tests.

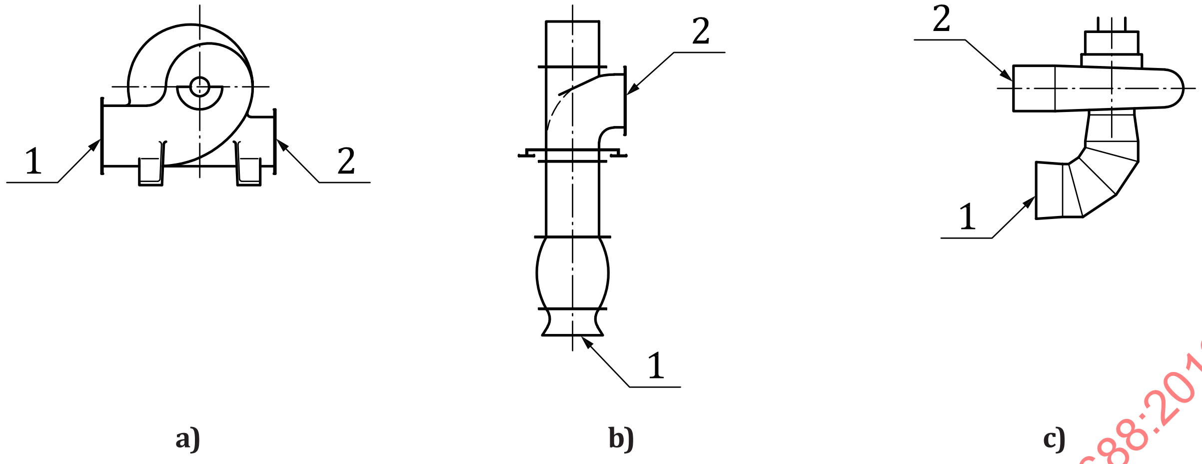
Table 3 — Contents of tests

Type of test	Measurement items
1. Performance test	Pump total head, volume rate of flow, speed of rotation, pump shaft torque or power input, pump efficiency, and NPSH
2. Cavitation test or NPSH3 test	
3. Additional tests	See Annex A.

6 Model pump

6.1 Extent of model pump

The extent of a model pump should be the segment between the inlet section and the outlet section of the pump (see Figure 1). When a part of the suction channel or discharge channel has a form that can be regarded as part of the pump and a suction opening or discharge opening cannot be clearly recognised, a cross section where the flow velocity distribution is considered uniform should be designated as an inlet or outlet of the model pump. The extent of the model pump may be otherwise defined by the agreement between the purchaser and manufacturer.



Key

1 inlet section of pump

2 outlet section of pump

Figure 1 — Extent of model pump

6.2 Dimensional ranges of model pump

6.2.1 Reynolds number

The Reynolds number of a model pump, Re_{hM}, should be no less than 2,0 × 10^6 for either a centrifugal, mixed flow or axial pump.

6.2.2 Dimension of impeller

The largest diameter of the impeller of a model pump should be no less than 300 mm. For an adjustable vane type pump, the largest diameter of the impeller should be the largest diameter at the designed vane setting angle. When manufacturing, precision can be ensured, the largest diameter of the impeller may be otherwise defined by the agreement between the purchaser and manufacturer.

6.2.3 Pump total head

The pump total head of a model pump should be determined to satisfy 6.2.1 and 6.2.2 and ensure the necessary precision of performance measurement.

6.3 Construction of model pump

All parts forming the hydraulic passageways of the model pump should be geometrically similar as the corresponding parts of the prototype pump. When this is difficult to attain, another arrangement may be agreed between the purchaser and manufacturer.

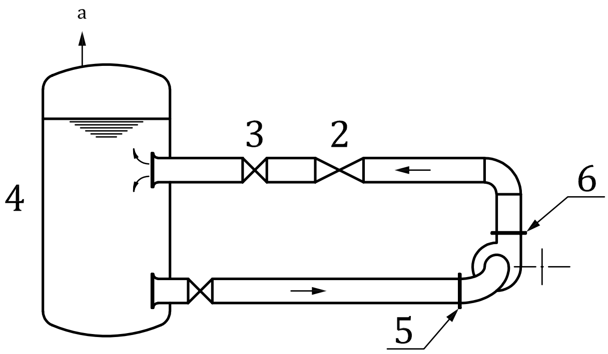
Similarity of the model pump should be proven by comparing measured dimensions of the model pump with the values of the model pump drawings. If necessary, vane profiles and degree of surface finish may also be measured and evaluated. Dimensions and items to be measured, measuring methods and permissible deviations may be agreed between the purchaser and manufacturer.

Regarding the clearance in the wearing part of a closed impeller, a geometrical similarity should be kept between the model pump and the prototype pump for the number of annular clearance steps, axial length, clearance average diameter, etc. The annular clearance may, however, be increased when it is possible to conduct operational testing of the model pump. The effect of increased clearance may be taken into consideration when converting performance of the model pump to that of the prototype.

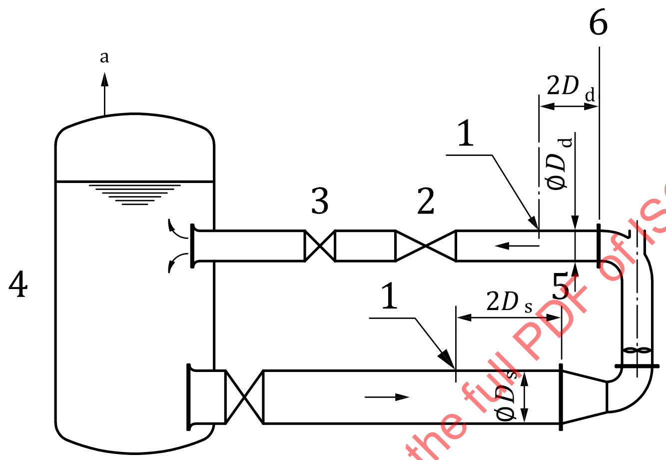
7 Performance test

7.1 Test installation and measuring instruments

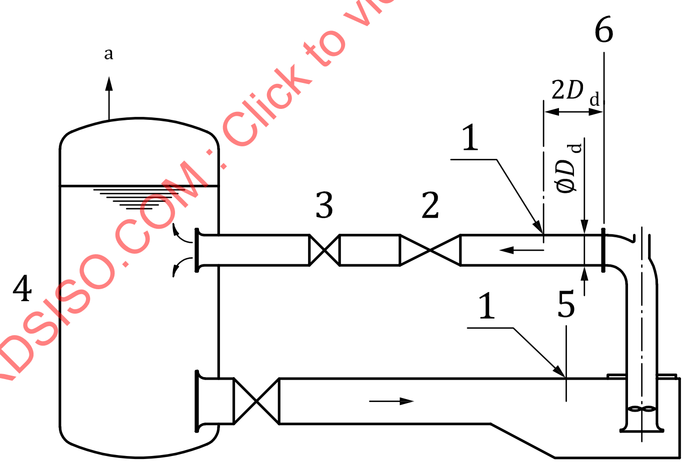
A test installation comprising a water reservoir or tank, piping, a discharge control valve, etc. providing a normal flow of water and allowing stable operation of a model pump and a performance measurement should be used. An example of a test arrangement is shown in Figure 2.



a) Centrifugal pump



b) Mixed flow and axial pump



c) Pump with suction sump

Key

1 pressure tapping
2 flow-meter
3 throttle valve
4 tank
5 inlet flange (Inlet boundary of model pump)
6 outlet flange (Outlet boundary of model pump)
a To vacuum and pressure control.

Figure 2 — Performance test installation

The best measuring conditions are obtained when the flow in a measured section has

— an axially symmetrical distribution,

— a uniform static pressure distribution, and

— no swirls induced by piping installation.

Although it is difficult to achieve the above conditions completely, for practical purposes it is adequate to make a measurement under the conditions given in a) to e) below.

In addition, any excessive curvature or steps in the approach piping and hydraulic passageways of the pump should be avoided to make the pump inlet flow uniform and minimize disturbances as much as possible.

Furthermore, it is preferable to implement a flow rectifier if there is a possibility of uneven flow or disturbance.

a) Any bend, combination of bends, expanded pipe section, discontinuous pipe section, etc. should be avoided in the vicinity of the measuring section.

b) A test arrangement where a suction line is provided by a closed channel extending from a sump with a free surface or a large stilling vessel, the length of the straight suction pipe L is determined by Formula (1)

$$L / D \geq 1{,}5\,K + 5{,}5 \qquad (1)$$

where

D is the pipe diameter;

K is the type number.

As long as this condition is fulfilled, there it is not required to install a flow rectifier between a bend and the model pump. On the other hand, a flow rectifier is required in a closed channel having no sump or stilling vessel on the upstream side of the model pump.

c) A flow control valve should be installed in the discharge piping as a rule (it is preferable to refrain from installing a flow control valve in the suction piping). If a flow control valve is installed in the suction piping and cannot be fully opened (for example, in the case of a cavitation test), a flow rectifier should be installed between the control valve and the pump inlet, or a straight pipe having a length at least 12 times the pipe diameter should be installed. When using a flow control valve in a throttled state, it should be noted that the pump cavitation performance may change due to the occurrence of cavitation in the valve.

d) In the case of a vertical shaft pump, the geometry of the suction sump of the prototype pump should be taken into consideration in the testing of the model pump, whose performance should be measured between 5 and 6 [see Figure 2 b)]. Other measuring locations may, however, be adopted according to the agreement between the purchaser and manufacturer.

e) Changes in water temperature should be minimized. In cases where liquid temperature rises more than 5 degrees K during a test due to heat input, the liquid temperature should be measured before and after each test and the average temperature should be used to evaluate physical properties.

Measuring instruments should have enough accuracy to measure the pump total head, volume rate of flow, pump power input, and speed of rotation of the model pump. The permissible relative uncertainties of instruments should be within the values listed in Table 4. Suitable periods for instrument calibration as provided in ISO 9906:2012, Annex C should be recorded. The calibration should be guaranteed by a calibration system that can be traced back to the relevant international metrology standard. In cases where such an international calibration standard is not available, the procedure of calibration should be documented. In cases where instruments other than those listed in Table 4 are used, the agreement between the purchaser and manufacturer should apply.

Table 4 — Permissible relative uncertainties of instruments

Measuring quantity	Uncertainty
Volume rate of flow[a]	0,5 %
Pump total head (differential head)	0,3 %
Outlet head	0,3 %
Inlet head	0,3 %
Driver power input[b]	0,3 %
Speed of rotation	0,1 %
Torque	0,2 %
Temperature	0,1 K

[a] Uncertainty in the measurement of volume rate of flow using an orifice, nozzle, Venturi tube or electromagnetic flowmeter should include the uncertainty of the mass method or the volumetric method used in addition to the uncertainty of the instrument itself. The calculation for estimating uncertainty is shown in Annex B for reference purposes.

[b] In the case when using watt meter with calibrated motor for pump power input, uncertainty of pump power input should include the uncertainty of motor efficiency in addition to the uncertainty of the instrument. See ISO 9906:2012, D.4.3.

7.2 Test conditions

7.2.1 Test operation

The test operating conditions are as follows.

a) Tests should be conducted with clean, cold water, whose characteristics should meet the requirement specified in ISO 5198.

b) While conducting a performance test, the fluctuation around the mean of the measured values and the extent of variation during the repeated measurement period, as defined in 7.2.2.1, should satisfy the criteria given in Table 5 and 7.2.2.3, respectively. These conditions should be confirmed close to the specified pump total head before conducting the performance test.

c) The speed of rotation in the test should be within ±5 % of the specified speed of rotation.

7.2.2 Stability of operation

7.2.2.1 Fluctuation and variation

The following definitions apply to this document.

7.2.2.1.1 Fluctuation

The range of fluctuation around the mean during one measurement ε is defined by the following Formula (2):

$$\varepsilon = \frac{X_j - X_i}{X_i} \tag{2}$$

where

X_j is the instantaneous value during one measurement;

X_i is the measured value (arithmetic mean of one measurement).

7.2.2.1.2 Variation

Changes in a measuring quantity observed between one reading and the next reading. The uncertainty of variation is discussed in detail in 7.2.2.3, and the method of calculation is described in Annex B.

7.2.2.2 Allowable fluctuation in readings and use of fluctuation damper

7.2.2.2.1 Measurement of signals from measurement system

The allowable fluctuation of each measuring quantity is shown in Table 5. The fluctuations are confirmed at the same time when the test operating conditions are confirmed. The confirmation is made by conducting a set of measurements within 10 seconds with a sampling cycle of not less than once per second close to the specified pump total head.

Table 5 — Allowable fluctuation around the mean value of measuring quantity

Measuring quantity	Allowable fluctuation
Volume rate of flow	±2 %
Pump total head (differential pressure)	±3 %
Outlet pressure	±2 %
Pump Inlet pressure	±2 %
Driver power input	±2 %
Speed of rotation	±0,5 %
Torque	±2 %
Temperature	±0,3 K

The allowable fluctuation of the pump total head is obtained as the square root of the square sum of the fluctuation ranges of inlet and outlet pressure. When using a differential pressure-type instrument for measuring the volume rate of flow, the allowable fluctuation of measured differential pressure is ±4 %. When measuring the pump total head of suction and the pump total head of discharge separately, the allowable fluctuation is determined about the pump total head.

When the pump generates vibration of larger amplitude due to its construction or operating conditions, a damper may be introduced in the instrument or connecting pipe to reduce fluctuation according to the limits shown in Table 5.

Since the damper could influence the accuracy of readings, a damper that has symmetrical and linear characteristics and is appropriate to show an integral value for at least one cycle of fluctuation (e.g. capillary tube) should be used.

7.2.2.2.2 Automatic reading or automatic calculation of signals in measurement system

When automatically recording, or integrating signals obtained from an instrument by a measurement system, the ranges of fluctuation of such signals are permitted to exceed those given in Table 5 if the following conditions are met.

a) The measurement system is equipped with a device that can automatically calculate a mean with sufficient accuracy for a period specified longer than the response time of the system.

b) When integration is made to calculate a mean from continuously recorded or sampled analogue signals in the form of $x(t)$ (time-dependent measuring quantity) (the sampling conditions should be described in a test report).

7.2.2.3 Limit of variation

7.2.2.3.1 Number of measurement sets

At each operating point, the measurement should be conducted repeatedly at random intervals of not less than 10 seconds, and multiple sets of data should be recorded. During measurement, only the speed of rotation and temperature can be adjusted. All other settings such as control valves, water level, sealing water of the packing, and balancing water should be kept at the same conditions as the initial conditions of measurement.

Variations among repeated readings under the same operating conditions imply, at least partially, the unsteadiness of the test conditions, which are affected by the equipment and the pump being tested. At each test point, no less than three sets of measurement should be obtained and the arithmetic mean of them should be taken as a measured value.

7.2.2.3.2 Calculation of uncertainty due to the number of measurement sets

If data around the mean value are distributed randomly because of repeated observation of the same measuring quantity, the uncertainty of measurement can be estimated statistically. The method of estimation is shown in Annex B.

7.3 Number of measurement points

The number of measurement points should be as follows.

a) Normally, it is recommended to conduct measurement at minimum of seven points between the minimum and the maximum allowable operating rate of flow.

b) When the prototype pump is of variable speed type, a variable speed test on the model pump can be omitted if the range of speed variation is within 20 %.

c) In the case of an adjustable vane-type pump, the vane angle for testing should be determined by the agreement between the purchaser and manufacturer and the measurement defined in a) should be conducted at the respective angles. In axial and mixed flow pumps, measurements at flow rates of 60 % or less of the specified volume rate of flow can be omitted if specified in the mutual agreement between the purchaser and manufacturer. The number of measuring points in this case should also be specified in this mutual agreement.

7.4 Pump total head

7.4.1 General

Pump total head should be measured in accordance with ISO 9906:2012, Annex A, Grade 1, whereby the content of 7.4.2 to 7.4.7 should be considered.

7.4.2 Measuring instruments

A liquid column manometer, spring pressure gauge (including a vacuum gauge) or digital pressure gauge should be used as a measuring instrument.

7.4.3 Liquid column manometer

The liquid column manometer should be as follows.

a) The inside diameter of the glass tube of the liquid column manometer should be constant and at least 6 mm.

b) When using a U-tube or inverse-U-tube liquid column manometer, both high and low liquid manometers should be read simultaneously.

c) When determining the differential pressure by using a liquid column manometer, the density of the liquid used should be considered for precise conversion into pressure.

d) When using a liquid column pressure gauge of single pipe type, the liquid column of the single pipe should be read and corrected for changes in the liquid level of the liquid reservoir. The inside diameter of the liquid reservoir should be at least 10 times the inside diameter of the glass pipe.

e) The scale used for reading a liquid column should be graduated in millimetres and be approved by official inspection. The calibration of the scale should be conducted using an accuracy class I metal rule based on the OIML R35-1:2007.

7.4.4 Spring pressure gauge

The spring pressure gauge to be used should have been calibrated in reference to the increasing and decreasing directions of pressure using a pressure standard (of weight type, liquid column type, etc.).

7.4.5 Digital pressure gauge

The pressure gauge to be used should have been calibrated in reference to the increasing and decreasing directions of pressure using the calibration method stipulated in OIML TC10-SC1 Ver. 4.0.

7.4.6 Pressure tappings

The pressure tappings to be arranged should be as follows.

a) Connect a straight pipe for measurement on each of the pump outlet and inlet sides, each pipe having a length of at least four times its bore (equivalent diameter in the case of a non-circular pipe) except for test arrangements described in 7.1 b) which require a minimum length of 1,5 K +5,5 times its (equivalent) bore for inlet side. Pressure tappings should be provided at a distance two times the bore diameter from the pump flange surface. The diameter of each tapping hole should be 3 to 6 mm or 0,08 D, whichever is smaller, and the depth of the hole should not be less than 2,5 times the diameter. Nevertheless, a connection pipe may be provided between the pump and the straight pipe for measurement according to a contract between the purchaser and manufacturer or for convenience in using the test installation. Other requirements regarding measuring sections and arrangement of the pressure tappings should be as per ISO 9906:2012, A.4.1 to A.4.3), Grade 1.

b) If the pump is operated in the range of partial flow rate and a recirculation flow is generated so that true inlet pressure cannot be measured, correct this problem as described in ISO 9906:2012, A.4.1.

c) Join the four pressure tappings in one section by connection tubes to measure the average pressure unless otherwise agreed between the purchaser and manufacturer.

d) The loss of head, H_f, due to the frictional resistance between the pressure measuring section and the model pump should be determined by the following formula and be added to the differential head between the pressure tappings. Nevertheless, the addition of H_f is not required when H_f is negligibly small compared with the pump total head at the specified volume rate of flow.

$$H_f = \lambda \cdot \frac{l}{D} \cdot \frac{v^2}{2g} \quad (3)$$

where D is the diameter of the pipe, expressed in metres (m).

An equivalent diameter is used in the case of a non-circular pipe.

$$D_e = \frac{4 \times A}{l_B} \quad (4)$$

A	is the cross-sectional area, expressed in square metres (m^2);
l_B	is the length of the wetted perimeter, expressed in metres (m);
g	is the acceleration of gravity, expressed in metres per square second (m/s^2);
H_f	is the loss of head, expressed in metres (m);
l	is the distance between the pressure measuring section and the model pump, expressed in metres (m);
v	is the flow velocity, expressed in metres per second (m/s);
λ	is the pipe friction coefficient (the numerical value or formula for calculation should be stipulated in the agreement between the purchaser and manufacturer.), dimensionless (—).

7.4.7 Damper

When the indication of a measuring instrument overly fluctuates, a mechanical damper may be used (e.g., throttling a gauge cock or joint tube). In this case, the damper effect should not have any directionality.

7.5 Volume rate of flow

The methods and instruments for measuring the volume rate of flow should be in accordance with either of the following.

7.5.1 Orifice plate nozzle and venturi tube

These flowmeters are specified in ISO 5167-1.

7.5.2 Electromagnetic flowmeter

These flowmeters are specified in ISO 20456.

7.5.3 Mass method or volumetric method

These flowmeters are specified in ISO 4185 or ISO 8316.

7.6 Speed of rotation

7.6.1 Measurement method

The method for measuring speed of rotation should be as follows.

a) Speed of rotation should be measured in connection with the shaft of a model pump, or the dynamometer shaft or motor shaft directly connected to the model pump.

b) Speed of rotation should be determined by counting signals generated by the rotation of the main shaft for a specified period.

7.6.2 Measuring instruments

The instruments for measuring speed of rotation should be as follows.

a) The digital counter used in the counting part of the measuring instrument should be appropriate in setting the duration of counting input signals to a level of 0,01 % or less.

b) When connecting a detector to an axis of rotation, the connecting part should be so designed to avoid any impairment of rotational precision.

7.7 Pump power input

7.7.1 Method for measuring pump power input

The method for measuring pump power input should be as follows.

a) Pump power input should be determined by measuring torque and speed of rotation.

b) If there is any loss in pump power input that is not present in the prototype pump due to the design of the measuring installation, the pump power input measured should be corrected for the loss.

7.7.2 Measurement of torque

Torque should be measured as follows.

a) An electric dynamometer or torque meter should be used for the measurement of torque.

b) When using an electric dynamometer, care should be taken

— so, that the flow of air caused by the rotator does not affect the torque measurement;

— so, that the rigidity of the connecting part such as the wiring of the electric dynamometer does not affect the torque measurement.

c) The minimum detectable scale of the electric dynamometer and the torque meter should be 0,1 % or less of the torque measured at the specified volume rate of flow.

d) The effective arm length of the electric dynamometer should be measured with the expanded uncertainty being ±0,05 % of the length.

e) A dynamometer having a mechanism to convert the direction of control force from horizontal to vertical should not have friction that would cause a problem with accuracy due to the mechanism. The lever ratio of the mechanism should be measured with the expanded uncertainty being ±0,05 %.

f) The electric dynamometer and torque meter should be calibrated in the increasing and decreasing directions of driving force by using a certified weight.

7.8 Measurement uncertainty

The uncertainty of measured values may be estimated in accordance with Annex B.

7.9 Calculation of pump power input, pump power output, and pump efficiency

The calculation of pump power input, pump power output and pump efficiency should be in accordance with the following formulae:

$$P_2 = 2\pi \cdot T \cdot n \tag{5}$$

$$P_h = \rho \cdot g \cdot Q \cdot H \tag{6}$$

$$\eta = \frac{P_h}{P_2} \tag{7}$$

where

- g is the acceleration of gravity, expressed in metres per square second (m/s^2);
- H is the pump total head, expressed in metres (m);
- n is the speed of rotation, expressed in reciprocal seconds (s^{-1});
- P_2 is the pump power input, expressed in watts (W);
- P_h is the pump power output, expressed in watts (W);
- Q is the volume rate of flow, expressed in cubic metres per second (m^3/s);
- T is the torque, expressed in newton metres (Nm);
- η is the pump efficiency, dimensionless (—);
- ρ is the density of liquid, expressed in kilograms per cubic metre (kg/m^3).

The pump efficiency should have three significant digits and the fourth digit should be rounded according to the provision of ISO 80000-1. Annex B shows a method for calculating the uncertainty in the calculated pump efficiency.

8 Cavitation test and NPSH3 test

8.1 Concept of test

The cavitation test and NPSH3 test are aimed at confirming the existence and extent of deviation in pump total head due to the occurrence of cavitation under the operating conditions of a model pump corresponding to the operating conditions of a prototype pump. In the case of multistage pumps, the head drop should be relative to the head of the first stage, which should be measured if accessible.

8.2 Test method

8.2.1 General

The test method should be any one of the following.

Clause 7 should apply correspondingly to the measurement of other aspects of pump performance.

When it is necessary to perform readings repeatedly because of unstable testing conditions, the fluctuation of *NPSH* should be allowed to be as much as 1,5 times the fluctuation given in 7.2.2.2 or 0,2 m, whichever is greater.

In a pump having a wide operating range, it is difficult to evaluate the influence of cavitation in the range of excessive flow rate, where the pump total head is significantly lower than the specified total head. The criteria for evaluation in such a case should be based on the agreement between the purchaser and manufacturer.

8.2.2 Cavitation test

Within the operating range of a prototype pump, three or four similar points close to the volume rate of flow are selected as test points based on the agreement between the purchaser and manufacturer. Then, a test is conducted at *NPSH* where the cavitation coefficient of the prototype pump is equal to that of the model pump, and the Q–H curve obtained by connecting these points is compared with the Q–H curve at an ample *NPSH* level within the operating range of the prototype pump.

8.2.3 NPSH3 test

Within the operating range of a prototype pump, one to four points close to the volume rate of flow are selected as test points based on the agreement between the purchaser and manufacturer. Then, the pump head is measured while keeping the same volume rate of flow and changing suction pressure, and the critical *NPSH* level at each volume rate of flow is determined to obtain a *Q*–*NPSH3* curve.

8.3 Characteristics of the test liquid

As far as possible, free gas should be removed from water before testing. In case it is necessary to avoid de-gassing in any part of the pump, the water of the circuit should not be supersaturated.

8.4 Test installation

Figure 2 shows an installation for testing a pump by adjusting the *NPSH* of the pump. Additional requirements on the installation and optional equipment such as temperature control system and de-aerator should be as per ISO 9906:2012, B.5.

9 Indication of performance and evaluation of test results

9.1 Arrangement of measured values and indication of performance test results

The measured values should be arranged and the performance test results should be indicated as follows.

9.1.1 Conversion at specified speed of rotation

$$Q_M = Q \cdot \frac{n_M}{n} \tag{8}$$

$$H_M = H\left(\frac{n_M}{n}\right)^2 \tag{9}$$

$$P_M = P\left(\frac{n_M}{n}\right)^3 \tag{10}$$

$$NPSH_M = NPSH\left(\frac{n_M}{n}\right)^X \tag{11}$$

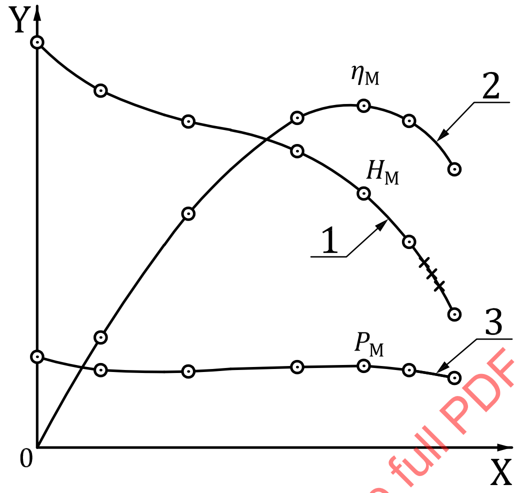
where

H_M	is the pump total head for the specified speed of rotation, expressed in metres (m);
n	is the test speed of rotation of the model pump, expressed in reciprocal seconds (s^{-1});
n_M	is the specified speed of rotation of the model pump, expressed in reciprocal seconds (s^{-1});
$NPSH_M$	is the NPSH for the specified speed of rotation, expressed in metres (m);
P_M	is the pump power input for the specified speed of rotation, expressed in watts (W);
Q_M	is the volume rate of flow for the specified speed of rotation, expressed in cubic metres per second (m^3/s).

For the conversion of NPSH, the value x = 2 may be used as a first approximation, however values of exponent x between 1,3 and 2 have been observed for different conditions and an agreement between the purchaser and the manufacturer is recommended to establish the conversion formula to be used. See ISO 9906:2012, 6.1.1 for more information.

9.1.2 Performance curves of model pump

The performance of the model pump should be shown by indicating the value of the specified speed of rotation, by plotting measurement points on a graph with Q on the horizontal axis and H, P, and efficiency η on the vertical axis using the values obtained in 9.1.1, and drawing a smooth curve that passes through the measurement points. The cavitation test points should also be shown in the graph (see Figure 3). This does not apply when conducting an NPSH3 test to determine NPSH3.



Key

X volume rate of flow Q_M

Y pump total head H_M, pump efficiency η_M, pump power input P_M

o point of measurement for the performance test

× point of the cavitation test

1 pump total head H_M

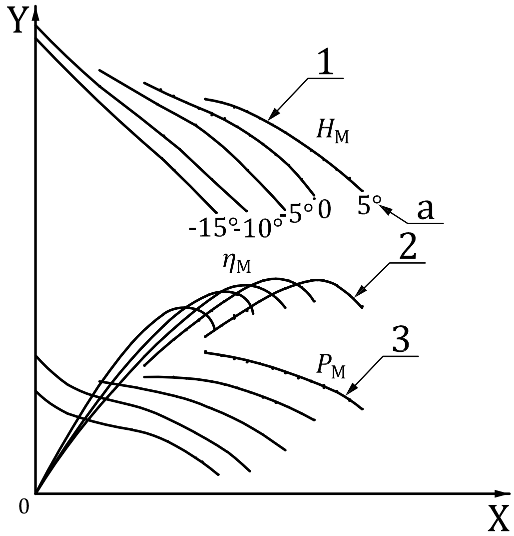
2 pump efficiency η_M

3 pump power input P_M

Figure 3 — Performance curves for fixed vane type pump

9.1.3 Performance curves of adjustable vane type model pump

In the case of a pump with adjustable vanes or adjustable diffuser vanes, performance curves for different vane angles should also be noted (see Figure 4).



Key

X volume rate of flow Q_M

Y pump total head H_M, pump efficiency η_M, pump power input P_M

1 pump total head H_M

2 pump efficiency η_M

3 pump power input P_M

a Vane angle (degrees).

Figure 4 — Performance curves for adjustable vane type pump

9.2 Conversion of various quantities from model to prototype pump

9.2.1 Conversion of volume rate of flow, pump total head and pump power input

The volume rate of flow, pump total head and pump power input of a prototype pump should be determined from various quantities obtained by the performance test of a model pump according to the following formulae:

$$Q_P = Q_M \frac{n_P}{n_M}\left(\frac{D_P}{D_M}\right)^3 F_v \tag{12}$$

$$H_P = H_M \left(\frac{n_P}{n_M}\right)^2 \left(\frac{D_P}{D_M}\right)^2 \frac{g_M}{g_P} F_h^{\alpha} \tag{13}$$

$$P_P = P_M \frac{\rho_P}{\rho_M}\left(\frac{n_P}{n_M}\right)^3 \left(\frac{D_P}{D_M}\right)^5 \Big/ \left(F_h^{\beta} \cdot F_m\right) \tag{14}$$

$$\eta_P = F_h \cdot F_m \cdot F_v \cdot \eta_M = \frac{\eta_{hP}}{\eta_{hM}} \frac{\eta_{mP}}{\eta_{mM}} \frac{\eta_{vP}}{\eta_{vM}} \eta_M \tag{15}$$

where

D_M	is the representative dimension of the model pump, expressed in metres (m);
D_P	is the representative dimension of the prototype pump, expressed in metres (m);
F_h	is the hydraulic efficiency ratio $F_h = \eta_{hP}/\eta_{hM}$, dimensionless (—);
F_m	is the mechanical efficiency ratio $F_m = \eta_{mP}/\eta_{mM}$, dimensionless (—);
F_v	is the volumetric efficiency ratio $F_v = \eta_{vP}/\eta_{vM}$, dimensionless (—);
g_M	is the acceleration of gravity at the installation location of the model pump, expressed in metres per square second (m/s^2);
g_P	is the acceleration of gravity at the installation location of the prototype pump, expressed in metres per square second (m/s^2);
H_M	is the pump total head of the model pump, expressed in metres (m);
H_P	is the pump total head of prototype pump, expressed in metres (m);
n_M	is the speed of rotation of the model pump, expressed in reciprocal seconds (s^{-1});
n_P	is the speed of rotation of the prototype pump, expressed in reciprocal seconds (s^{-1});
P_M	is the pump power input of the model pump, expressed in watts (W);
P_P	is the pump power input of the prototype pump, expressed in watts (W);
Q_M	is the volume rate of flow of the model pump, expressed in cubic metres per second (m^3/s);
Q_P	is the volume rate of flow of the prototype pump, expressed in cubic metres per second (m^3/s);
α	is the influence factor of the pump total head in the hydraulic efficiency ratio between the prototype and model pumps, dimensionless (—);
β	is the influence factor of the pump power input in the hydraulic efficiency ratio between the prototype and model pumps, dimensionless (—);
η_M	is the efficiency of the model pump, dimensionless (—);
η_P	is the efficiency of the prototype pump, dimensionless (—);
η_{hM}	is the hydraulic efficiency of the model pump, dimensionless (—);
η_{hP}	is the hydraulic efficiency of the prototype pump, dimensionless (—);
η_{mM}	is the mechanical efficiency of the model pump, dimensionless (—);
η_{mP}	is the mechanical efficiency of the prototype pump, dimensionless (—);
η_{vM}	is the volumetric efficiency of the model pump, dimensionless (—);
η_{vP}	is the volumetric efficiency of the prototype pump, dimensionless (—);
ρ_M	is the density of liquid in the model pump, expressed in kilograms per cubic metre (kg/m^3);
ρ_P	is the density of liquid in the prototype pump, expressed in kilograms per cubic metre (kg/m^3).

NOTE The efficiency ratios used here are calculated as described in 9.2.2.

9.2.2 Calculation of volumetric, mechanical and hydraulic efficiency ratios

If specified by purchaser or agreed between the purchaser and manufacturer, a performance conversion method may be applied to obtain efficiency ratios given in 9.2.1. Some examples of conversion methods are listed in Annex C. If no performance conversion method is applied, all efficiency ratios should be equal to unity.

The Reynolds numbers of the model and prototype pumps are calculated using the temperature of clean, cold water during the performance test in the model pump and the temperature of clean, cold water during actual operation (specified operating point) in the prototype pump, respectively.

9.3 Evaluation of test results

When performing a model test as a delivery test of a prototype pump, the following criteria should be used to evaluate the test results unless otherwise specified in a prior agreement between the purchaser and manufacturer (see Figure 5).

9.3.1 Performance curve

Convert the results of the model test to the performance of the prototype pump in accordance with the provisions of 9.2 under the same conditions of cavitation coefficient σ. Plot the measurement points on a graph with volume rate of flow Q_P on the horizontal axis and pump total head H_P, pump power input P_P and pump efficiency η_P on the vertical axis, and draw a smooth curve that passes through the measurement points [see Figure 5 a)].

9.3.2 Pump total head

Draw a horizontal line with length $\tau_Q \times Q_G$ and a vertical line with length $\tau_H \times H_G$ from the guarantee point (Q_G, H_G), both in the positive direction, to form an L-shaped figure having a size of permissible tolerance τ_Q and τ_H. The values of pump total head and volume rate of flow should be regarded as satisfactory if the $H(Q)$ curve crosses or touches the L-shaped figure [see Figure 5 b)]. Here, τ_Q should be in the range between 0 and +0,05, and τ_H should be in the range between 0 and +0,03.

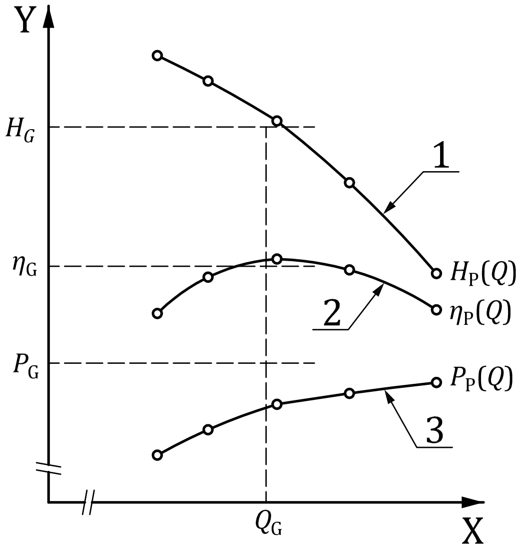
9.3.3 Pump efficiency

Efficiency is given by the intersection of $\eta(Q)$ and the vertical line that passes the point where the straight line connecting the specified point (Q_G, H_G) and the origin of the Q_G axis intersects the $H(Q)$ curve. The guarantee conditions are satisfied if the efficiency value at this intersection is equal to or greater than η_G [see Figure 5 b)].

NOTE The zone of measured values for the $P(Q)$ curve should not exceed the driving motor capacity P_G within the range of volume rate of flow used [see Figure 5 c)].

9.3.4 Cavitation performance

It should be evaluated that the guarantee condition of the cavitation test is satisfied if the decrease rate of pump total head regarding the performance test value is 3 % or less. In the NPSH3 test, on the other hand, if the Q–$NPSH3$ curve drawn as stipulated in 8.2 lies below the guarantee point (Q_G, $NPSH_G$) under the same coefficient of cavitation σ, it should be evaluated that the guarantee condition of the cavitation performance is satisfied [see Figure 5 d)].



a) **Performance curve of prototype pump**

Key

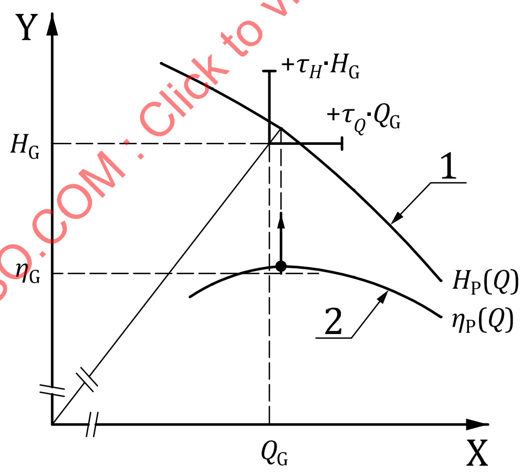
X volume rate of flow Q_P

Y pump total head H_P, pump efficiency η_P, pump power input P_P

1 pump total head H_P

2 pump efficiency η_P

3 pump power input P_P



b) **Evaluation of *H* and *η***

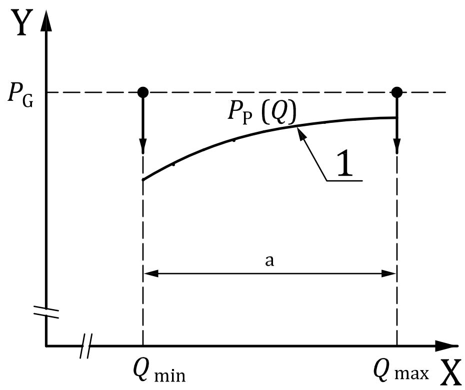
Key

X volume rate of flow Q_P

Y pump total head H_P, pump efficiency η_P,

1 pump total head H_P

2 pump efficiency η_P



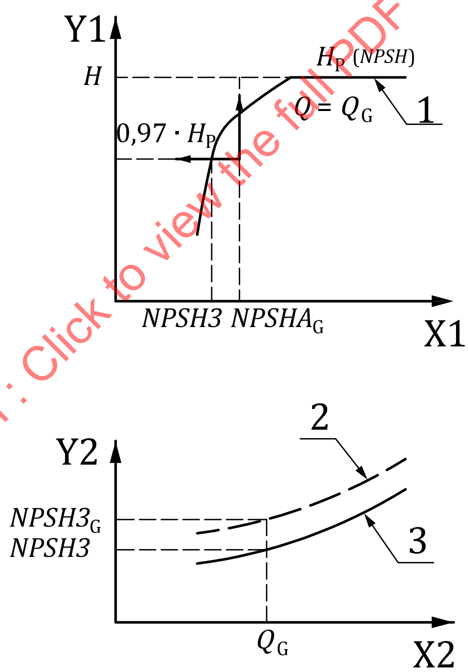
c) Evaluation of *P*

Key

X volume rate of flow Q_P

Y pump power input P_P

1 pump power input P_P



d) Evaluation of *NPSH*

Key

X_1 NPSH $NPSH_P$

X_2 volume rate of flow Q_P

Y_1 pump total head H_P

Y_2 NPSH3 $NPSH3_P$

1 pump total head H_P

2 NPSH3 $NPSH3_G$

3 NPSH3 *NPSH3*

Figure 5 — Evaluation of test results

9.4 Preparation of test results sheet

Manufacturer may prepare test results sheet in which the following items may be included:

a) date and place of the test and the name of the person in charge of the test;

b) requirements on the prototype pump and model pump;

c) matters of agreement;

d) range of similarity of the model pump;

e) measuring installation, instruments and calibration data;

f) procedures of performance conversion from the model pump to prototype pump;

g) performance curves and cavitation performance of the model pump at the specified speed of rotation;

h) converted performance curves of the prototype pump;

i) measuring quantities and results of measurement (including formulae);

j) evaluation of test results;

k) others (evaluation of measurement uncertainties, etc. as necessary).

10 Prototype pump

The parts forming the main hydraulic passageways of the model pump should be geometrically similar to the corresponding parts of the prototype pump.

Similarity of the prototype pump should be proven by measuring dimensions of the prototype pump regarding the prototype pump drawings.

If necessary, vane profiles and degree of surface finish should also be measured and evaluated. Dimensions and items to be measured, measuring methods and permissible deviations should be based on agreement between the purchaser and manufacturer.

Annex A
(informative)

Additional tests

A.1 General

Additional tests described in this annex are not mandatory and will normally be performed when it is specified by the purchaser.

A.2 Four quadrant test

A.2.1 General

Since four quadrant test is very complex and needs additional cost, it should only be performed when its necessity is recognised by the purchaser and manufacturer.

A.2.2 Operating conditions and directions

The operation modes are listed in Table A.1.

Table A.1 — Operating conditions and directions

Zone	Operation mode	Description	*Q*	*n*	*y*	*T*
1. Pump	Pump	Zone of normal rotation and normal flow during pump operation	+	+	+	+
2. Break	Pump break	Zone of normal rotation and reverse flow	−	+	+	+
3. Turbine	Turbine	Zone of reverse flow above the runaway speed within the zone of reverse rotation and reverse flow	−	−	+	+
	Turbine break	Zone of reverse flow below the runaway speed within the zone of reverse rotation and reverse flow	−	−	+	−
	Reverse pump	Zone of reverse rotation and normal flow	+	−	+	−

NOTE The signs are defined as follows.

Q is positive in the discharging direction of pump operation;

n is positive in the rotational direction of pump operation;

y is positive when $H_2 \geq H_1$ (specific energy remains positive in all operation modes);

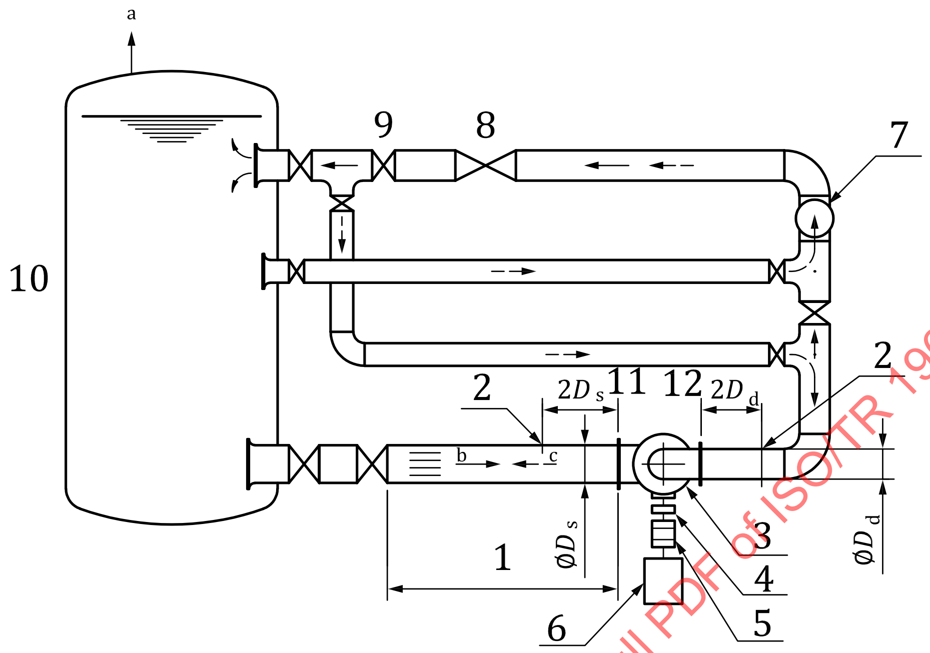
T is positive in the shaft torque direction of pump operation.

A.2.3 Model

The model should be in accordance with Clause 6.

A.2.4 Test installation and test head

The test installation should be in accordance with Clause 7 and Figure A.1. The test head should be specified in the agreement between the purchaser and manufacturer.



Key

1	suction straight pipe length
2	pressure tapping
3	model pump
4	speed meter
5	torque meter
6	motor
7	booster pump
8	flowmeter
9	throttle valve
10	tank
11	inlet flange (pump operation)
12	outlet flange (pump operation)
a	To vacuum and pressure control.
b	Flow direction (pump operation).
c	Flow direction (turbine test).

Figure A.1 — Four quadrant test installation

A.2.5 Test method and test head

The test method should be as per the following A.2.5.1 to A.2.5.4.

A.2.5.1 Measurement items

The measurement items are pump total head, volume rate of flow, speed of rotation and shaft torque in each operating mode.

A.2.5.2 Measuring instruments

The measuring instruments should be as per Clause 7.

A.2.5.3 Condition of adjustable diffuser vanes or adjustable vanes

The opening of adjustable diffuser vanes or adjustable vanes should be at the design opening.

A.2.5.4 Conversion formulae to non-dimensional values

The complete characteristics of the model pump should be converted to non-dimensional values by the following formulae, and a smooth curve that passes through the measurement points should be plotted (see Figure A.2):

$$Q_{ED} = \frac{Q_M}{D_{1M}^2 \left(g_M \cdot H_M\right)^{0,5}} \tag{A.1}$$

$$n_{ED} = \frac{n_M \cdot D_{1M}}{\left(g_M \cdot H_M\right)^{0,5}} \tag{A.2}$$

$$T_{ED} = \frac{T_M}{\rho_M \cdot D_{1M}{}^3 \left(g_M \cdot H_M\right)} \tag{A.3}$$

where

D_{1M} is the impeller inlet diameter of the model pump, expressed in metres (m);

g_M is the acceleration of gravity at the installation location of the model pump, expressed in metres per square second (m/s^2);

H_M is the pump total head, expressed in metres (m);

n_{ED} is the speed of rotation factor, dimensionless (—);

n_M is the speed of rotation of the model pump, expressed in reciprocal seconds (s^{-1});

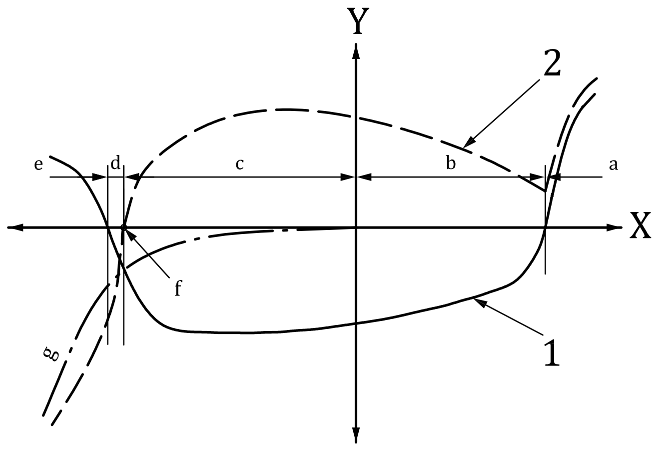
Q_{ED} is the volume rate of the flow factor, dimensionless (—);

Q_M is the volume rate of flow of the model pump, expressed in cubic metres per second (m^3/s);

T_{ED} is the torque factor, dimensionless (—);

T_M is the shaft torque of the model pump, expressed in newton metres (Nm);

ρ_M is the density of liquid in the model pump, expressed in kilograms per cubic metre (kg/m^3).



Key

X speed of rotation factor n_{ED}, dimensionless (-)
Y volume rate of flow factor Q_{ED} or shaft torque factor T_{ED}, dimensionless (-)
1 flow characteristics
2 shaft torque characteristics
a Pump region.
b Pump brake region.
c Turbine region.
d Turbine brake region.
e Reversible pump region.
f Point of runaway speed.
g Runaway speed.

Figure A.2 — Four quadrant characteristic curves

The conversion of test results from a model pump to a prototype pump is performed by determining the $n_P - Q_P$ and $n_P - T_P$ relations using the following formulae. Here, the pump total head or effective head are assumed to be constant (e.g. 100 % H_P or 80 % H_P, etc.):

$$Q_P = Q_{ED} \cdot D_{1P}^2 \left(g_P \cdot H_P \right)^{0,5} \tag{A.4}$$

$$n_P = n_{ED} \frac{\left(g_P \cdot H_P \right)^{0,5}}{D_{1P}} \tag{A.5}$$

$$T_P = T_{ED} \cdot \rho_P \cdot D_{1P}^3 \cdot g_P \cdot H_P \tag{A.6}$$

where

D_{1P} is the impeller inlet diameter of the prototype pump, expressed in metres (m);

g_P is the acceleration of gravity at the installation location of the prototype pump, expressed in metres per square second (m/s^2);

H_P is the pump total head or effective head of the prototype pump, expressed in metres (m);

n_P is the speed of rotation of the prototype pump, expressed in reciprocal seconds (s^{-1});

Q_P is the volume rate of flow of the prototype pump, expressed in cubic metres per second (m^3/s);

T_P is the shaft torque of the prototype pump, expressed in newton metres (Nm);

ρ_P is the density of the liquid in the prototype pump, expressed in kilograms per cubic metre (kg/m^3).

A.3 Pressure fluctuation test

A.3.1 Model

The model should be in accordance with Clause 6.

A.3.2 Test installation and pump total head

The installation should be in accordance with Clause 7. The test head should be the same as that used in the performance test or cavitation test. Alternative arrangements may be made by the agreement between the purchaser and manufacturer.

A.3.3 Test method

A.3.3.1 Location of measurement

The measurement locations for the pressure fluctuation test should be in the suction casing inlet and the discharge casing outlet. Alternative locations may be arranged, if necessary, by the agreement between the purchaser and manufacturer.

A.3.3.2 Method for measurement

The method for the measurement of pressure fluctuation is as follows.

a) The measured pressure is converted to an electrical signal and then to an equivalent head.

b) Pressure should be measured by arranging a pressure tapping in a pipe wall and setting the pressure detection part of a pressure measuring instrument directly in contact with the flow of water. When using a copper pipe or the like to connect between the pressure tapping and the measuring instrument, the pipe should be minimized in length and be filled with water to expel air.

c) Attention should be paid to avoid any disturbance of flow due to the shape of the pressure tapping or the installation state of the pressure measuring instrument.

A.3.3.3 Measuring instrument

The measuring instrument should be as per the following a) to d).

a) Use a pressure converter that converts pressure changes into electrical signals.

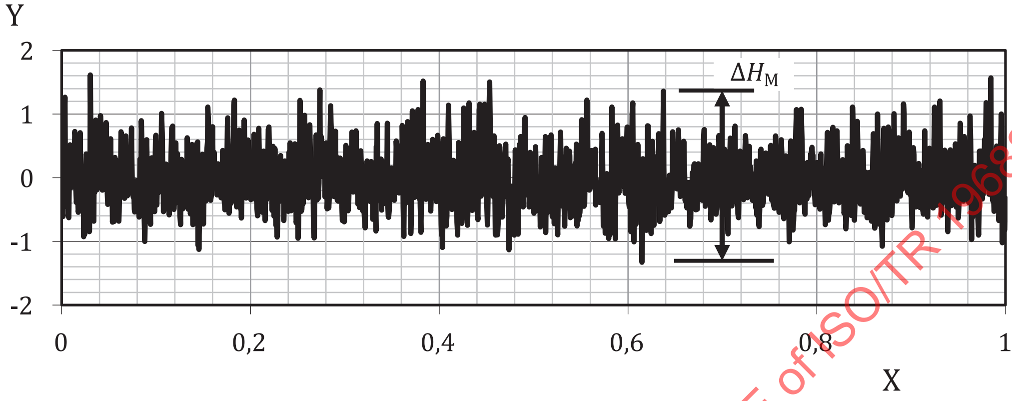
b) Pressure converters include those of electrical resistance type, piezoelectric type and condenser type. Other types may also be used for a test if they have been calibrated by an appropriate method for the purposes of the test and approved to have the necessary accuracy and characteristics.

c) The natural frequency of the pressure converter should be sufficiently higher than the pressure fluctuation frequency to be measured.

d) To estimate pressure fluctuation more accurately, a suitable electrical filter circuit or the like should be used to cut off frequency components of pressure fluctuation that are sufficiently separated from the frequency components to be measured.

A.3.3.4 Indication of measurement results

Pressure fluctuation may be indicated by peak-to-peak values of maximum amplitude or effective values (RMS: Root Mean Square) for a series of measurement results obtained in a time period considered to be sufficient for the type of fluctuation. A frequency analysis may also be conducted if necessary. An example is shown in Figure A.3. The upper chart shows a time series plot of pressure fluctuation, from which the maximum amplitude can be read. The lower chart shows the result of frequency analysis, from which frequency components of pressure fluctuation can be obtained.

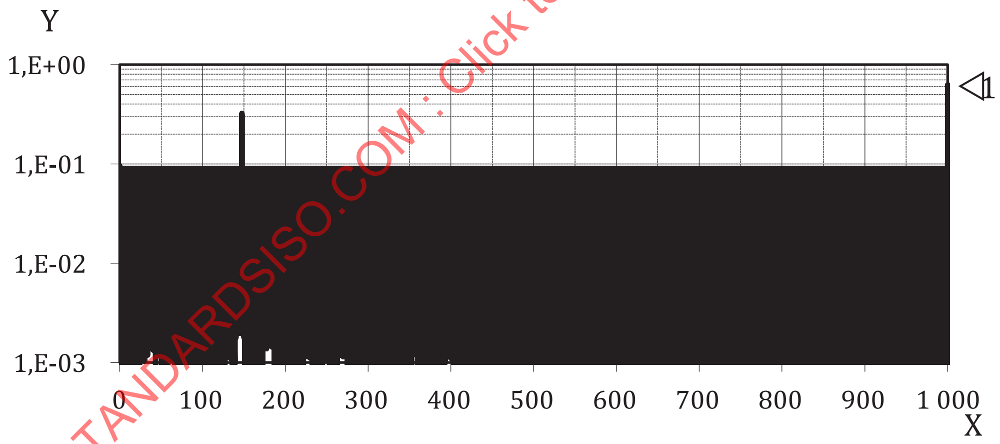


a) Pressure fluctuation pattern ΔH_M

Key

X time, t (s)

Y pressure fluctuation, ΔH_M (m)



b) Spectrum of pressure fluctuation

Key

X frequency, f_M (Hz)

Y RMS value of pressure fluctuation, $\Delta H_M/\Delta f_M$ (m/Hz)

1 ΔH_M (Overall, RMS) (m)

Figure A.3 — Example of pressure fluctuation measurements

A.3.3.5 Conversion of various quantities from model to prototype pump

The pressure fluctuation changes with the operating condition and the intensity of cavitation. Under the same operating condition and the same cavitation coefficient of the model and prototype pump, the following formulae hold true:

$$\frac{\Delta H_M}{H_M} = \frac{\Delta H_P}{H_P} \tag{A.7}$$

$$\frac{f_M}{n_M} = \frac{f_P}{n_P} \tag{A.8}$$

where

f_M is the pressure fluctuation frequency of the model pump, expressed in hertz (Hz);

f_P is the pressure fluctuation frequency of the prototype pump, expressed in hertz (Hz);

H_M is the pump total head of the model pump, expressed in metres (m);

H_P is the pump total head of the prototype pump, expressed in metres (m);

n_M is the speed of rotation of the model pump, expressed in reciprocal seconds (s^{-1});

n_P is the speed of rotation of the prototype pump, expressed in reciprocal seconds (s^{-1});

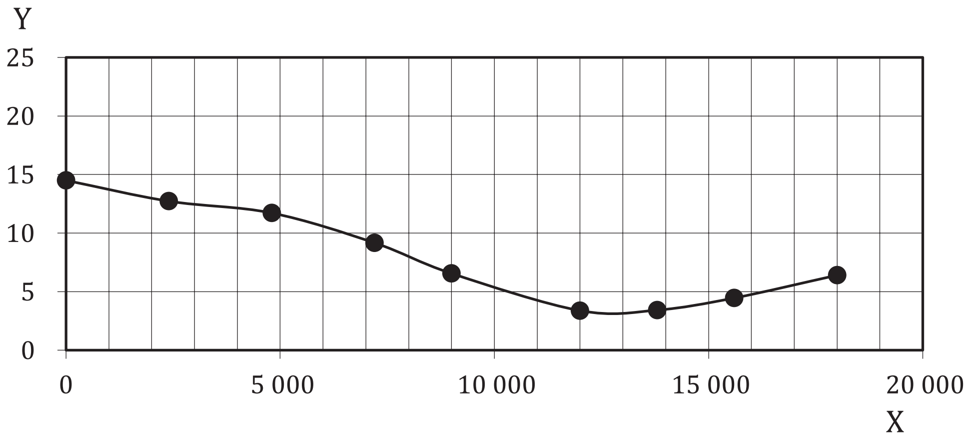
ΔH_M is the pressure fluctuation of the model pump, expressed in metres (m);

ΔH_P is the pressure fluctuation of the prototype pump, expressed in metres (m).

Note that the natural frequency of each device of the model and prototype pumps should be sufficiently separated from the above-mentioned f_M and f_P values.

A.3.3.6 Reporting of measurement results

The values of pressure fluctuation may be divided by pump total head to be expressed in non-dimensional form. The results of a pressure fluctuation measurement — as in the case of efficiency, pump total head and pump power input — are expressed on a graph with volume rate of flow on the horizontal axis. An example showing pressure fluctuation values in non-dimensional form is shown in Figure A.4.



Key

X volume rate of flow Q_P (m^3/h)

Y $\Delta H_P/H_P$ (%)

Figure A.4 — Pressure fluctuation at discharge casing outlet (example showing results of pressure fluctuation measurement)

A.4 Thrust test

A.4.1 Model

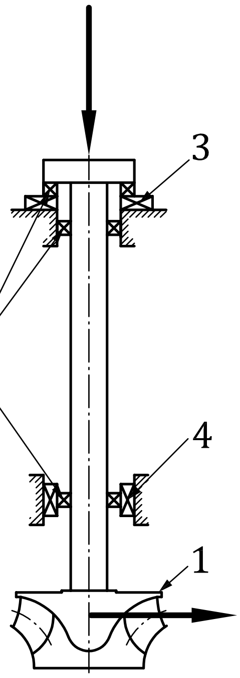
The model should be in accordance with Clause 6.

A.4.2 Test installation and test head

The installation should be in accordance with Clause 7 and Figure A.5. The test should be conducted at the same head as in the performance test or the cavitation test. Other arrangements may be determined by the agreement between the purchaser and manufacturer.

A.4.3 Method of thrust measurement

The bearing housing of the model pump should be supported by axial or radial load cells (or strain gauge type devices) and thrust is measured by electrical signals from the load cells. Before conducting measurement, the load cells should be calibrated by applying axial or radial loads on the shaft.



Key

1 impeller
2 bearing
3 load cell (axial thrust)
4 load cell (radial thrust)

Figure A.5 — Example of thrust measurement installation

A.4.4 Indication of measurement results and formulae for conversion to prototype pump

A.4.4.1 Axial thrust, applicable to pumps having identical radial sections in the casing, such as diffuser type pumps, that do not cause a significant radial thrust reaction.

The calculation of the axial thrust and its conversion from a model pump to a prototype pump should be based on the following formulae:

$$F_{aP} = F_{aM} \cdot \left(\frac{\rho_P}{\rho_M}\right) \cdot \left(\frac{n_P}{n_M}\right)^2 \cdot \left(\frac{D_{1P}}{D_{1M}}\right)^4 \tag{A.9}$$

where

D_{1M} is the impeller inlet diameter of the model pump, expressed in metres(m);

D_{1P} is the impeller inlet diameter of the prototype pump, expressed in metres(m);

F_{aM} is the axial thrust of the model pump, expressed in newtons (N);

F_{aP} is the axial thrust of the prototype pump, expressed in newtons (N);

n_M is the speed of rotation of the model pump, expressed in reciprocal seconds (s^{-1});