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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).

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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:

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

for the prototype pump

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

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 (—);

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

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

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

$D_{1\mathrm{P}}$ 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.

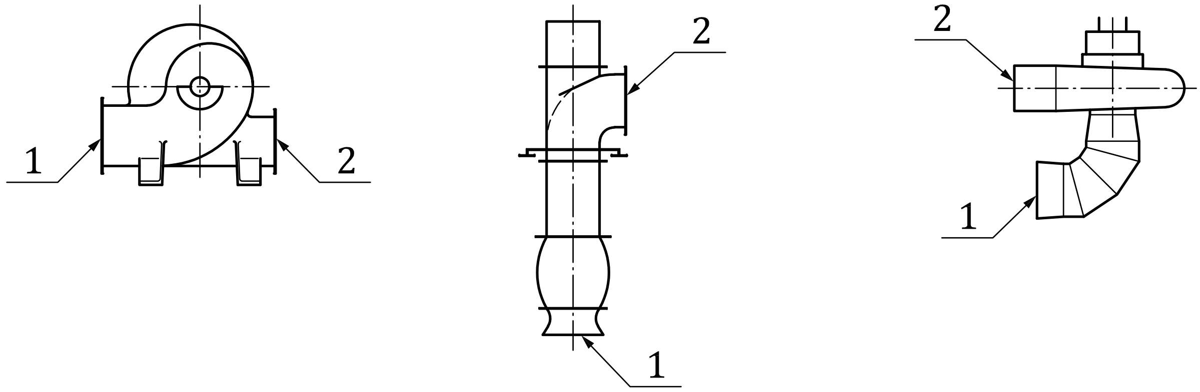
Table 3 — Contents of tests

<table>
<tr><th>Type of test</th><th>Measurement items</th></tr>
<tr><td>1. Performance test</td><td rowspan="2">Pump total head, volume rate of flow, speed of rotation, pump shaft torque or power input, pump efficiency, and NPSH</td></tr>
<tr><td>2. Cavitation test or NPSH3 test</td></tr>
<tr><td>3. Additional tests</td><td>See Annex A.</td></tr>
</table>

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.