



BSI Standards Publication

Hydraulic machines, radial and axial – Methodology for performance transposition from model to prototype

National foreword

This British Standard is the UK implementation of EN IEC 62097:2019. It is identical to IEC 62097:2019. It supersedes BS EN 62097:2009, which is withdrawn.

The UK participation in its preparation was entrusted to Technical Committee MCE/15, Hydraulic turbines.

A list of organizations represented on this committee can be obtained on request to its secretary.

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Date	Text affected
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English Version

Hydraulic machines, radial and axial - Methodology for
performance transposition from model to prototype
(IEC 62097:2019)

Machines hydrauliques, radiales et axiales - Méthodologie
de transposition des performances du modèle au prototype
(IEC 62097:2019)

Hydraulische Maschinen, radial und axial -
Leistungsumrechnung vom Modell zum Prototyp
(IEC 62097:2019)

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European Committee for Electrotechnical Standardization
Comité Européen de Normalisation Electrotechnique
Europäisches Komitee für Elektrotechnische Normung

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European foreword

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- latest date by which the document has to be implemented at national level by publication of an identical national standard or by endorsement (dop) 2019-11-12
- latest date by which the national standards conflicting with the document have to be withdrawn (dow) 2022-02-12

This document supersedes EN 62097:2009.

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Endorsement notice

The text of the International Standard IEC 62097:2019 was approved by CENELEC as a European Standard without any modification.

In the official version, for Bibliography, the following notes have to be added for the standards indicated:

ISO 4287	NOTE	Harmonized as EN ISO 4287
ISO 4288	NOTE	Harmonized as EN ISO 4288

Annex ZA
(normative)

**Normative references to international publications
with their corresponding European publications**

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.

NOTE 1 Where an International Publication has been modified by common modifications, indicated by (mod), the relevant EN/HD applies.

NOTE 2 Up-to-date information on the latest versions of the European Standards listed in this annex is available here: www.cenelec.eu.

<u>Publication</u>	<u>Year</u>	<u>Title</u>	<u>EN/HD</u>	<u>Year</u>
IEC 60193	-	Hydraulic turbines, storage pumps and pump-turbines - Model acceptance tests	EN 60193	-

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CONTENTS

FOREWORD.....	7
INTRODUCTION.....	10
0.1 General remarks	10
0.2 Basic features	11
1 Scope	12
2 Normative references	12
3 Terms, definitions, units, subscripts and symbols	12
3.1 Terms and definitions.....	12
3.2 List of definitions by topic	13
3.3 Subscripts and symbols	14
3.4 Geometric terms	15
3.5 Physical quantities and properties.....	15
3.6 Discharge, velocity and speed terms.....	16
3.7 Pressure terms	16
3.8 Specific energy terms	16
3.9 Head terms	17
3.10 Power and torque terms	18
3.11 Efficiency terms	19
3.12 Fluid dynamics and scaling terms	20
3.13 Dimensionless terms.....	20
4 Scale-effect formula	22
4.1 General.....	22
4.1.1 Scalable losses	22
4.1.2 Basic formulae of the scale effect on hydrodynamic friction losses	23
4.2 Specific hydraulic energy efficiency	25
4.2.1 General	25
4.2.2 Radial flow machines.....	26
4.2.3 Axial flow machines	26
4.3 Volumetric efficiency.....	26
4.4 Power efficiency (disk friction)	27
4.4.1 Radial flow machines.....	27
4.4.2 Axial flow machines	27
5 Surface roughness of model and prototype	28
5.1 General.....	28
5.2 Measurement of surface roughness	28
5.2.1 Procedure.....	28
5.2.2 Roughness of model and prototype.....	29
5.2.3 Measurement of very rough surfaces	30
5.3 Surface roughness ranges	30
6 Standardized values of scalable losses and pertinent parameters	32
6.1 General.....	32
6.2 Specific speed	33
6.3 Parameters for specific hydraulic energy efficiency transposition	33
6.3.1 General	33
6.3.2 Francis turbines.....	33
6.3.3 Pump-turbines	34

6.3.4	Axial flow machines	34
6.4	Parameters for power efficiency (disk friction) transposition	35
6.4.1	Francis turbines	35
6.4.2	Pump-turbines	35
6.4.3	Axial flow machines	35
7	Transposition to prototype	36
7.1	General	36
7.2	Assumed maximum hydraulic efficiency	37
7.3	Hydraulic efficiency	38
7.4	Specific hydraulic energy	39
7.4.1	Turbine operation	39
7.4.2	Pump operation	39
7.5	Discharge	40
7.5.1	Turbine operation	40
7.5.2	Pump operation	40
7.6	Torque	40
7.6.1	Turbine operation	40
7.6.2	Pump operation	40
7.7	Power	41
7.7.1	Turbine operation	41
7.7.2	Pump operation	41
8	Calculation procedure	41
8.1	General	41
8.2	Reference model	42
8.3	Comparison of different models	42
8.4	Normalization procedure of the test data: Step 1	42
8.5	Prototype transposition procedure: Step 2	44
8.6	Alternative one step method for the optimum point	46
8.7	required input data	48
Annex A (informative)	Basic formulae and their approximation	50
A.1	Basic concept of loss structure and scale effect	50
A.1.1	General	50
A.1.2	Loss structure and efficiency components	50
A.1.3	Homologous operating condition	52
A.1.4	Shifting of performance [6]	53
A.1.5	Scalable losses	54
A.2	Derivation of the scale effect formulae and the approximation introduced for simplifications	55
A.2.1	Scalable loss ratio in specific hydraulic energy δ_E and specific hydraulic energy efficiency η_E	55
A.2.2	Transposition of specific hydraulic energy efficiency η_E	56
A.2.3	Transposition of volumetric efficiency η_Q	58
A.2.4	Transposition of power efficiency (disk friction) η_T	59
Annex B (informative)	Scale effect on specific hydraulic energy losses of radial flow machines	61
B.1	Scale effect on friction loss	61
B.1.1	Scale effect on friction loss coefficient	61

B.1.2	Relationship between sand roughness k_s and arithmetical mean roughness Ra	63
B.2	Componentwise transposition of specific hydraulic energy efficiency	63
B.2.1	Friction loss coefficient of each component [9].....	63
B.2.2	Derivation of the scale effect formula for component wise transposition.....	66
B.3	Standardized relative scalable hydraulic energy loss of radial flow machines.....	67
B.3.1	Definition	67
B.3.2	Standardized relative scalable hydraulic energy loss δ_E of Francis turbine	68
B.3.3	Standardized relative scalable hydraulic energy loss δ_E of reversible pump-turbine	70
B.4	Flow velocity factor κ_{uCO} and dimension factor κ_{dCO} of radial flow machines [9]	71
B.4.1	Definition	71
B.4.2	κ_{uCO} and κ_{dCO} for Francis turbine.....	71
B.4.3	κ_{uCO} and κ_{dCO} for pump-turbine	73
B.5	Standardized scalable loss index d_{ECOref}	76
B.5.1	Definition	76
B.5.2	Standardized d_{ECOref} and d_{Eref} for Francis turbine	76
B.5.3	Standardized d_{ECOref} and d_{Eref} for pump-turbine	77
Annex C (informative)	Scale effect on specific hydraulic energy losses of axial flow machines [9]	79
C.1	Scalable losses of axial flow machines	79
C.2	Scale effect formula for runner blades [8]	79
C.3	Scale effect formula for stationary parts	80
C.4	Scale effect for other efficiency components	81
C.4.1	Volumetric efficiency.....	81
C.4.2	Power efficiency (disk friction)	81
C.5	Scale effect of hydraulic efficiency	81
C.6	Determination of δ_{ECOref} of axial flow turbines.....	81
C.7	Determination of δ_{ECOref} of bulb turbines	82
C.8	Derivation of scalable hydraulic energy loss index, d_{Eref}	83
C.8.1	Scalable loss index for runner blades	83
C.8.2	Scalable loss index for stationary parts.....	83
C.9	Summary of the scale effect formula for axial flow machines.....	84
Annex D (informative)	Scale effect on disk friction loss	85
D.1	Loss coefficient formula for disk friction	85
D.2	Transposition formula for power efficiency	86
D.3	Standardized dimension factor κ_T and disk friction loss index d_{Tref}	87
D.3.1	Disk friction loss ratio δ_{Tref}	87
D.3.2	Dimension factor of the disk κ_T	88
D.3.3	Disk friction loss index d_{Tref}	89
Annex E (informative)	Comparison of IEC 60193 and IEC 62097 hydraulic efficiency transposition methods for reaction machines	

E.1	IEC 60193 transposition method	91
E.1.1	Applications	91
E.1.2	Limitations	91
E.2	IEC 62097 transposition method	91
E.2.1	Applications	91
E.2.2	Limitations	91
Annex F (informative)	Leakage loss evaluation for non homologous seals	93
F.1	Loss coefficient of runner seal	93
F.2	General formula to obtain $\Delta\eta_Q$ for non-homologous seal	97
F.3	Evaluation of scale effect in case of a homologous straight seal	97
F.4	Straight seal with non-homologous radial clearance	98
Annex G (normative)	Guide for detailed calculations by means of the attached Excel workbook	100
G.1	Normalization of test data: Step 1 of the “2 step method”	100
G.2	Prototype transposition: Step 2 of the “2 step method”	101
G.3	Alternative one step method for the optimum point.....	102
Annex H (informative)	Example of a calculation with the attached Excel workbook.....	104
H.1	Cover page	104
H.2	Example of STEP 1 for a Pump-Turbine in Turbine Mode.....	105
H.2.1	« Input Form » data sheet for step 1	105
H.2.2	Results for Step 1	107
H.3	Example of STEP 2 for a Pump-Turbine in Turbine Mode.....	115
H.3.1	« Input Form » data sheet for step 2	115
H.3.2	Results for Step 2	117
Bibliography.....		125
Figure 1 – Scale effect considering surface roughness		24
Figure 2 – Impact of surface roughness on turbine efficiency and costs		28
Figure 3 – Surface roughness regions for Francis runner blades.....		31
Figure 4 – Surface roughness region for axial flow runner blade		32
Figure 5 – Normalization of test data from Re_{M_i} to Re_{M^*} and Re_{M^*} to Re_P		36
Figure 6 – Reference Assumed maximum hydraulic efficiency		38
Figure 7 – Calculation procedure: “Two step method”: First step – From model to reference model.....		43
Figure 8 – Calculation procedure: “Two step method”: Second step – From reference model to prototype		45
Figure 9 – Calculation procedure: Alternative one step method for the optimum point from model to prototype		47
Figure A.1 – Flux diagram for a turbine		51
Figure A.2 – Flux diagram for a pump		52
Figure B.1 – Loss coefficient versus Reynolds number and surface roughness		62
Figure B.2 – Different characteristics of λ in transition zone		62
Figure B.3 – Representative dimensions of component passages		65
Figure B.4 – Standardized relative scalable hydraulic energy loss in each component of Francis turbine		69
Figure B.5 – Standardized relative scalable hydraulic energy loss in each component of pump-turbine in turbine operation		70

Figure B.6 – Standardized relative scalable hydraulic energy loss in each component of pump-turbine in pump operation.....	71
Figure B.7 – κ_{uCO} and κ_{dCO} in each component of Francis turbine.....	72
Figure B.8 – κ_{uCO} and κ_{dCO} in each component of pump-turbine in turbine operation	74
Figure B.9 – κ_{uCO} and κ_{dCO} in each component of pump-turbine in pump operation.....	75
Figure B.10 – Standardized d_{ECOref} and d_{Eref} for Francis turbine.....	76
Figure B.11 – Standardized d_{ECOref} and d_{Eref} for pump-turbine in turbine operation	77
Figure B.12 – Standardized d_{ECOref} and d_{Eref} for pump-turbine in pump-operation	78
Figure C 1 – δ_{ECOref} for Kaplan turbines	82
Figure D.1 – Disk friction loss reference ratio δ_{Tref}	88
Figure D.2 – Dimension factor κ_T	89
Figure D.3 – Disk friction loss index d_{Tref}	90
Figure F.1 – Examples of typical design of runner seals (crown side)	95
Figure F.2 – Examples of typical design of runner seals (band side)	96
Table 1 – Permissible deviation of the geometry of model seals from the prototype	27
Table 2 – Recommended roughness range for the model.....	30
Table 3 – Minimum recommended prototype roughness for new radial or diagonal machines	31
Table 4 – Maximum recommended prototype roughness for new radial or diagonal machines	31
Table 5 – Minimum recommended prototype roughness for new axial turbines.....	32
Table 6 – Maximum recommended prototype roughness for new axial turbines.....	32
Table 7 – Standardized scalable loss index d_{ECOref} and standardized velocity factor κ_{uCO} for Francis turbines.....	33
Table 8 – Standardized scalable loss index d_{ECOref} and standardized velocity factor κ_{uCO} for pump-turbines in turbine operation.....	34
Table 9 – Standardized scalable loss index d_{ECOref} and standardized velocity factor κ_{uCO} for pump-turbines in pump operation.....	34
Table 10 – Standardized scalable loss index d_{ECOref} and velocity factor κ_{uCO} for axial flow machines.....	34
Table 11 – Calculation of $\eta_{hAmaxMref}$	37
Table 12 – Reference roughness of the reference model, Ra_{COMref}	42
Table 13 – Required input data for the calculation of the prototype performance.....	48
Table A.1 – Definitions of scalable loss ratios	56
Table A.2 – Definitions of transposition of specific energy	57
Table A.3 – Definitions of transposition of volumetric efficiency	58
Table A.4 – Definitions of power efficiency transposition terms	59
Table C.1 – Ratio of $\frac{d_{EST}}{\delta_{EST}}$ for Francis turbines and pump-turbines	84

INTERNATIONAL ELECTROTECHNICAL COMMISSION

**HYDRAULIC MACHINES, RADIAL AND AXIAL – METHODOLOGY FOR
PERFORMANCE TRANSPOSITION FROM MODEL TO PROTOTYPE**

FOREWORD

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International Standard IEC 62097 has been prepared by IEC technical committee 4: Hydraulic turbines.

This second edition cancels and replaces the first edition published in 2009. This edition constitutes an editorial and technical revision.

This edition includes the following significant technical changes with respect to the previous edition:

- a) In introduction, clarifications have been brought such as addition of a sentence which declares the precedence of IEC 62097 over IEC 60193 if any mismatch is found between them
- b) In Clauses 3 and 4, corrections of the typographical errors
- c) In Clause 3: changes to be in accordance with presentation of the terms and structure of IEC 60193 (except for the water temperature)
- d) In Clause 4:
 - Deletion of the clause providing the direct step-up procedures for a whole turbine

- Introduction of a global view by using turbine A and turbine B instead of model turbine, reference model turbine and prototype turbine
 - Move of section dealing with “surface roughness of model and prototype” in a new Clause 5
- e) In Clause 5:
- Introduction of additional chapters to answer comments raised at the CDV stage and to clarify the subject of surface roughness of model and prototype
 - Introduction of new tables for minimum recommended prototype roughness for new radial or diagonal machines and for new axial turbines
 - Addition of the explanation about roughness measurement of heavily rusted surface
- f) In Clause 7 (former Clause 6):
- Introduction of a new subclause for clarifications about the assumed maximum hydraulic efficiency, η_{hAmax}
 - Deletion of the requirement of mutual agreement for the application of the step-up formula for very high efficiency machines exceeding η_{hAmax}
 - Clarifications of the equations from 22 to 33 by doubling the equations for suiting the “two step method”
- g) In Clauses 6 and 7, correction of typographical errors
- h) In Clause 8 (former Clause 7), introduction of new figures for clarifying the “2 step method” and the alternative method
- i) In Annex A, modification of the flux diagram to be in compliance with IEC 60193
- j) In Annex B:
- Correction of the equation to obtain Δ_{ECO}
 - Deletion of the clause which describes the direct step-up procedures for radial flow machines
- k) In Annex C, deletion of the clause which describes the direct step-up procedures for axial flow machines
- l) In Annex D:
- notes become main text
 - change of variable names in Subclause D.1 for clarifications
- m) Addition of Annex E, about comparison of IEC standards dealing with models: 60193 and 62097
- n) In Annex F, clarifications of equations by adding more subscripts
- o) The Excel sheets attached to the standard are revised as itemized below
- Deletion of the routine regarding the direct step-up procedures for a whole turbine
 - Deletion of the notice which requires mutual agreement when the step-up is applied to high efficiency machines exceeding η_{hAmax}
 - Addition of the routine to process the normalization of test data obtained at optimum test conditions
- p) Simplification of structure, calculation of optimum and individual point, step up calculation with η_{hAmax}

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

FDIS	Report of voting
4/359/FDIS	4/364/RVD

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

This publication has been drafted in accordance with the ISO/IEC Directives, Part 2.

The committee has decided that the contents of this publication will remain unchanged until the stability date indicated on the IEC website under "<http://webstore.iec.ch>" in the data related to the specific publication. At this date, the publication will be

- reconfirmed,
- withdrawn,
- replaced by a revised edition, or
- amended.

IMPORTANT – The 'colour inside' logo on the cover page of this publication indicates that it contains colours which are considered to be useful for the correct understanding of its contents. Users should therefore print this document using a colour printer.

INTRODUCTION

0.1 General remarks

IEC 62097 forms an element of a series of standards which deals with model testing of hydraulic machines. The series has two groups describing

- a) Hydraulic turbines, storage pumps and pump-turbines – Model acceptance tests (IEC 60193);
- b) Hydraulic machines, radial and axial – Performance conversion method from model to prototype (IEC 62097).

Advances in the technology of hydraulic machines for hydroelectric power plants provided background for updating and revising the scale effect methodology of IEC 60193. The advance in knowledge of scale effects originates from work done by research institutes, manufacturers and relevant working groups within the organizations of IEC and IAHR. See IEC 60193 and [1]¹ to [6].

The method of calculating prototype efficiencies is supported by experimental work and theoretical research on flow analysis and has been simplified for practical reasons and agreed as a convention by [7] to [9]. The methodology is representing the present state of knowledge of the transposition of performance from model to a homologous prototype.

Homology is not limited to the geometric similarity of the machine components; it also calls for homologous velocity triangles at runner inlet and outlet [1].

According to the present state of knowledge, the formula for the efficiency transposition calculation given in IEC 60193 and earlier standards often overestimates the transposition increment of the efficiency for the prototype.

Limitations and applications of performance conversion of both standards (IEC 60193 and IEC 62097) are given in Annex E.

This document is intended to be used mainly for the assessment of the results of contractual model tests of hydraulic machines. If it is used for other purposes such as evaluation of refurbishment of machines having very rough surfaces, special care is taken as described in Annex B.

Due to the lack of sufficient knowledge about the loss distribution in Deriaz turbines, multi-stage pump-turbines and storage pumps, this document does not provide the scale effect formulae for them.

An Excel workbook concerning the conversion procedures of hydraulic machine performance is attached as a complement of this document to facilitate the calculation of the scaled value for a given test point.

When using this document, if any mismatch is found with IEC 60193, the information in IEC 62097 prevails. Annex E provides additional information for performance conversion method.

¹ Numbers in square brackets refer to the bibliography.

0.2 Basic features

A fundamental difference compared to the IEC 60193 transposition is the standardization of scalable losses. In IEC 60193, a loss distribution factor V has been defined and standardized, with the disadvantage that turbine designs which are not optimized will benefit from their lower technological level.

This is certainly not correct, since a low efficiency design typically produces high non-scalable losses, like incidence losses, whereby the amount of scalable losses is about constant for all hydraulic machines, for a given type and a given specific speed of a hydraulic machine.

This document avoids all the inconsistencies connected with IEC 60193. A new basic feature of this document is the separate consideration of losses in specific hydraulic energy, leakage losses and disk friction losses [4], [7] to [9].

Above all, in this document, the transposition of the hydraulic performance is not only driven by the dependence of friction losses on Reynolds number Re , but also the effect of surface roughness Ra has been implemented.

Since the roughness of the actual machine component differs from part to part, scale effect is evaluated for each individual part separately and then is finally summed up to obtain the overall step-up for a complete machine [9]. For radial flow machines, the evaluation of scale effect is conducted on five separate parts; spiral case, stay vanes, guide vanes, runner and draft tube. For axial flow machines, the scalable losses in individual parts are not fully clarified yet and are dealt with in two parts; runner blades and all the other stationary parts inclusive.

The calculation procedures according to this document are summarized in Clause 8.

In case that the Excel workbook is used for evaluation of the results of a contractual model test, each concerned party executes the calculation individually for cross-check using common input data agreed on in advance for at least one test point.

HYDRAULIC MACHINES, RADIAL AND AXIAL – METHODOLOGY FOR PERFORMANCE TRANSPOSITION FROM MODEL TO PROTOTYPE

1 Scope

This International Standard establishes the prototype hydraulic machine efficiency from model test results, with consideration of scale effect including the effect of surface roughness.

This document is intended to be used for the assessment of the results of contractual model tests of hydraulic machines.

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.

IEC 60193, *Hydraulic turbines, storage pumps and pump-turbines – Model acceptance tests*

3 Terms, definitions, units, subscripts and symbols

For the purposes of this document, the following terms, definitions, units, subscripts and symbols given in IEC 60193 and the following apply.

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

- IEC Electropedia: available at <http://www.electropedia.org/>
- ISO Online browsing platform: available at <http://www.iso.org/obp>

The International System of Units (SI, see ISO 80000-4) is used throughout this document. All terms are given in SI base units or derived coherent units². The basic formulae are valid using these units. Any other system of units may be used after written agreement of the contracting parties.

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

3.1 Terms and definitions

3.1.1 point

position 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

² N = kg·m·s⁻² Pa = kg·m⁻¹·s⁻² J = kg·m²·s⁻² W = kg·m²·s⁻³

3.1.2**test**

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

3.1.3**hydraulic performance**

all performance parameters attributable to the machine due to hydrodynamic effects

3.1.4**main hydraulic performance data**

subset of the hydraulic performance parameters, i.e. power, discharge and/or specific hydraulic energy, efficiency, steady-state runaway speed and/or discharge

Note 1 to entry: The influence of cavitation shall be considered.

3.1.5**additional data**

subset of hydraulic performance data which can be determined for information on the model

Note 1 The prediction of the corresponding prototype data is less accurate than that achievable for the main hydraulic performance data, due to application of approximate similarity rules.

3.1.6**guarantees**

specified performance data contractually agreed to

3.1.7**scaling**

transposition of machine efficiency

3.2 List of definitions by topic

Subclause	Title
3.3	Subscripts and symbols
3.4	Geometric terms
3.5	Physical quantities and properties
3.6	Discharge, velocity and speed terms
3.7	Pressure terms
3.8	Specific energy terms
3.9	Head terms
3.10	Power and torque terms
3.11	Efficiency terms
3.12	Fluid dynamic and scaling terms
3.13	Dimensionless terms

3.3 Subscripts and symbols

Subclause	Term	Definition	Subscript or symbol
3.3.1	high pressure a reference section	high pressure section of the machine to which the performance guarantees refer [SOURCE: IEC 60193]	1
3.3.2	low pressure a reference section	low pressure section of the machine to which the performance guarantees refer (see IEC 60193)	2
3.3.3	specified	subscript denoting values of quantities such as rotational speed, discharge etc. for which other quantities are guaranteed	sp
3.3.4	turbine	subscript denoting either Model (M), Reference Model (M*) or Prototype (P)	A, B
3.3.5	prototype	subscript denoting values related to the full size machine	P
3.3.6	model	subscript denoting values related to the model	M
3.3.7	reference model	subscript denoting values related to a model and referred to a constant value of Reynolds number, a referred diameter, a referred water temperature $t_{WM^*} = 20\text{ °C}$ and a referred water density at ambient pressure $p_{amb} = 101\,325\text{ Pa}$ and at the reference value of each component surface roughness, Ra_{COM^*} according to Table 12	M* or *
3.3.8	specific energy	subscript referring to specific hydraulic energy	E
3.3.9	volumetric	subscript referring to discharge	Q
3.3.10	torque or disk friction	subscript referring to torque or disk friction	T
3.3.11	reference	subscript denoting values related to a specified reference condition	ref
3.3.12	optimum point	subscript denoting the best efficiency point	opt
3.3.13	hydraulic diameter	subscript referring to a hydraulic diameter value	d
3.3.14	velocity	subscript referring to a velocity value	u
3.3.15	hydraulic	subscript referring to hydraulic value	h
3.3.16	flat plate	subscript referring to a flat plate	f
3.3.17	individual	subscript referring to an individual test point	i
3.3.18	component	subscript referring to a component of the studied machine	CO
3.3.19	spiral case	subscript referring to the spiral case of the studied machine	SP
3.3.20	stay vane	subscript referring to the stay vane of the studied machine	SV
3.3.21	guide vane	subscript referring to the guide vane of the studied machine	GV
3.3.22	runner	subscript referring to the runner of the studied machine	RU
3.3.23	draft tube	Subscript referring to the draft tube of the studied machine	DT
3.3.24	stationary part	subscript referring to the stationary part of the studied axial machine	ST
3.3.25	rotating part	subscript referring to the rotating part of the runner for disk friction power loss	TR
3.3.26	stationary part	subscript referring to the stationary part facing the runner for disk friction power loss	TS

In general these terms are represented by CO

^a 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.

3.4 Geometric terms

Subclause	Term	Definition	Symbol	Unit
3.4.1	radial flow machines	Francis turbines and Francis type reversible pump-turbines	-	-
3.4.2	axial flow machines	Kaplan turbines, bulb turbines and fixed blade propeller turbines	-	-
3.4.3	reference diameter	reference diameter of the hydraulic machine [SOURCE: IEC 60193]	D	m
3.4.4	hydraulic diameter	4 times sectional area divided by the circumference of the section	d_h	m
3.4.5	diameter of runner crown or runner band	maximum diameter of runner crown or runner band, whichever larger	D_d	m
3.4.6	radius of runner crown or runner band	maximum radius of runner crown or runner band, whichever larger	r_d	m
3.4.7	opening of the concerned component	opening between two adjacent blades at high pressure and low pressure side of the runner/impeller, guide vanes or stay vanes.	a	m
3.4.8	sand roughness	equivalent sand roughness [10]	k_s	m
3.4.9	arithmetical mean roughness	deviation of the surface profile represented by the arithmetical mean value	Ra	m

3.5 Physical quantities and properties

Subclause	Term	Definition	Symbol	Unit
3.5.1	acceleration due to gravity	local value of gravitational acceleration at the place of testing as a function of altitude and latitude [SOURCE: IEC 60193]	g	m s^{-2}
3.5.2	temperature	temperature of water in Celsius [SOURCE: IEC 60193]	t_w	$^{\circ}\text{C}$
3.5.3	density of water	mass per unit volume of water [SOURCE: IEC 60193] Note 1 to entry: ρ is commonly used instead of ρ_w	ρ, ρ_w	kg m^{-3}
3.5.4	dynamic viscosity	quantity characterizing the mechanical behaviour of a fluid [SOURCE: ISO 8000-4]	μ	Pa s
3.5.5	kinematic viscosity	ratio of the dynamic viscosity to the density of the fluid. Values are given as a function of temperature [SOURCE: IEC 60193]	ν	$\text{m}^2 \text{s}^{-1}$

3.6 Discharge, velocity and speed terms

Subclause	Term	Definition	Symbol	Unit
3.6.1	discharge	volume of water per unit time flowing through any section in the system	Q	$\text{m}^3 \text{s}^{-1}$
3.6.2	mass flow rate	mass of water flowing through any section of the system per unit time	(ρQ)	kg s^{-1}
3.6.3	discharge of machine	discharge flowing through the high pressure reference section	Q_1	$\text{m}^3 \text{s}^{-1}$
3.6.4	leakage flow rate	volume of water per unit time flowing through the runner seal clearances	q	$\text{m}^3 \text{s}^{-1}$
3.6.5	net discharge	volume of water per unit time flowing through runner/impeller Note 1 to entry: This corresponds to $Q_1 - q$ in case of turbine and $Q_1 + q$ in case of pump.	Q_m	$\text{m}^3 \text{s}^{-1}$
3.6.6	mean velocity	discharge divided by the sectional area of water passage	v	m s^{-1}
3.6.7	peripheral velocity	peripheral velocity at the reference diameter $u = \pi D n$	u	m s^{-1}
3.6.8	rotational speed	number of revolutions per unit time	n	s^{-1}

3.7 Pressure terms

Subclause	Term	Definition	Symbol	Unit
3.7.1	ambient pressure	absolute pressure of the ambient air Note 1 to entry: Values for standard atmosphere are given as a function of elevation in IEC 60193.	p_{amb}	Pa

3.8 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 document 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 3.9. They differ only by the factor g , which is the local value of acceleration due to gravity.

Subclause	Term	Definition	Symbol	Unit
3.8.1	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 SEE IEC 60193	E	J kg^{-1}
3.8.2	specific hydraulic energy of runner/impeller	<turbine> net specific hydraulic energy working on the runner <pump> Specific hydraulic energy produced by the impeller	E_m	J kg^{-1}
3.8.3	specific hydraulic energy loss of machine	specific hydraulic energy loss which corresponds to the energy dissipated between any two sections in the turbine/pump $E_L = E - E_m = E_{Ls} + E_{Lm} = \sum E_{Lf} + \sum E_{Lk}$	E_L	J kg^{-1}
3.8.4	specific hydraulic energy loss in stationary part	specific hydraulic energy loss in stationary part which includes both friction loss and kinetic loss	E_{Ls}	J kg^{-1}
3.8.5	specific hydraulic energy loss in runner/impeller	specific hydraulic energy loss in runner/impeller which includes both friction loss and kinetic loss	E_{Lm}	J kg^{-1}
3.8.6	friction loss of specific hydraulic energy	specific hydraulic energy loss caused by the friction on the surface of water passages	E_{Lf}	J kg^{-1}
3.8.7	kinetic loss of specific hydraulic energy	specific hydraulic energy loss caused by the hydraulic phenomena other than surface friction, such as turbulence, separation of flow, abrupt change of water passage, etc.	E_{Lk}	J kg^{-1}

3.9 Head terms

Subclause	Term	Definition	Symbol	Unit
3.9.1	turbine net head or pump delivery head	specific energy per local unit weight of water at any section $H = E / g$ See 3.7.1 for the definition of E	H	m

3.10 Power and torque terms

Subclause	Term	Definition	Symbol	Unit
3.10.1	turbine output or pump input	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 Figures A.1 and A.2)	P	W
3.10.2	hydraulic power	hydraulic power available for producing power (turbine) or imparted to the water (pump) $P_h = E(\rho Q)_1$	P_h	W
3.10.3	mechanical power of runner/ impeller	mechanical power transmitted through the coupling of the runner(s)/impeller(s) and the shaft	P_m	W
3.10.4	power of runner/impeller	<turbine> Power produced by the runner corresponding to $E_m(\rho Q_m)_1$ or $P_m \cdot P_{Ld}$ <pump> Power produced by the impeller represented by $E_m(\rho Q_m)_1$ or $P_m \cdot P_{Ld}$	P_r	W
3.10.5	disk friction loss	power loss caused by the friction on the outer surface of the runner/impeller	P_{Ld}	W
3.10.6	mechanical power losses (bearing loss power)	mechanical power dissipated in guide bearings, thrust bearings and shaft seals of the hydraulic machine	P_{Lm}	W
3.10.7	runner/impeller torque	torque transmitted through the coupling of the runner/impeller and the shaft corresponding to the mechanical power of runner/impeller, P_m	T_m	N m

3.11 Efficiency terms

Subclause	Term	Definition	Symbol	Unit
3.11.1	hydraulic efficiency	<turbine> ratio of mechanical power of runner to the hydraulic power $\eta_h = P_m / P_h$ <pump> ratio of hydraulic power to the mechanical power of the impeller $\eta_h = P_h / P_m$	η_h	-
3.11.2	specific hydraulic energy efficiency	<turbine> $\eta_E = E_m/E$ <pump> $\eta_E = E/E_m$ SEE Figures A.1 and A.2	η_E	-
3.11.3	volumetric efficiency	<turbine> $\eta_Q = Q_m/Q_1$ <pump> $\eta_Q = Q_1/Q_m$ SEE Figures A.1 and A.2	η_Q	-
3.11.4	power efficiency (disk friction efficiency)	<turbine> $\eta_T = P_m/P_r$ <pump> $\eta_T = P_r/P_m$ SEE Figures A.1 and A.2	η_T	-
3.11.5	mechanical efficiency	<turbine> $\eta_m = P/P_m$ <pump> $\eta_m = P_m/P$ SEE Figures A.1 and A.2	η_m	-
3.11.6	efficiency step-up	difference between efficiencies at two hydraulically similar operating conditions	$\Delta\eta$	-
3.11.7	efficiency step-up ratio	ratio of efficiency step-up against considered efficiency $\Delta = \frac{\Delta\eta}{\eta}$	Δ	-
3.11.8	assumed maximum hydraulic efficiency	efficiency of an ideal model at a specified Reynolds number and with smooth surface roughness without any hydraulic kinetic losses other than friction and leakage losses	η_{hAmax}	-
3.11.9	efficiency correction coefficient	correction of standardized loss terms. $k_{corr} = \frac{(1 - \eta_{hMopt})}{(1 - \eta_{hAmaxM})}$	k_{corr}	-

3.12 Fluid dynamics and scaling terms

Subclause	Term	Definition	Symbol	Unit
3.12.1	Reynolds number	Reynolds number of the machine $Re = \frac{D \cdot u}{\nu}$	Re	-
3.12.2	Reynolds number of component passage	Reynolds number of component passage $Re_d = d_h \cdot v / \nu$	Re_d	-
3.12.3	friction loss coefficient for pipe flow	friction loss coefficient for a pipe $\lambda = \frac{E_{Lf}}{L \cdot \frac{v^2}{2}}$ where d pipe diameter (m) L pipe length (m)	λ	-
3.12.4	friction loss coefficient for a flat plate	friction loss coefficient for a flat plate $C_f = \frac{E_{Lf}}{\frac{B \cdot L}{Q} \cdot \frac{w^3}{2}}$ where B width of a flat plate (m) L length of a flat plate (m) Q discharge passing by the plate (m ³ /s) w relative flow velocity (m/s)	C_f	-
3.12.5	disk friction loss coefficient	friction loss coefficient for a rotating disc $C_m = \frac{P_{Ld}}{\frac{\pi^4}{8} \rho_w \cdot n^3 \cdot D_d^5}$ where D_d diameter of the rotating disk (m)	C_m	-
3.12.6	relative scalable hydraulic energy loss	scalable specific hydraulic energy loss divided by E , which is dependent on Reynolds number and roughness (in most cases, it is represented in %) $\delta_E = E_{Lf}/E$	δ_E	-
3.12.7	relative non-scalable hydraulic energy loss	non-scalable specific hydraulic energy loss divided by E , which remains constant regardless of Reynolds number and roughness $\delta_{ns} = E_{Lk}/E$	δ_{ns}	-
3.12.8	reference scalable hydraulic energy loss	δ_E value for a model with smooth surface operating at a reference Reynolds number $Re = 7 \times 10^6$	δ_{Eref}	-
3.12.9	reference scalable hydraulic energy loss in component passage	δ_{Eref} for each component passage	δ_{ECOref}	-
3.12.10	relative disk friction loss	disk friction loss P_{Ld} divided by P_m $\delta_T = \frac{P_{Ld}}{P_m}$	δ_T	-
3.12.11	reference disk friction loss	δ_T value for a model with smooth surface operating at a reference Reynolds number $Re_{ref} = 7 \times 10^6$	δ_{Tref}	-

3.13 Dimensionless terms

Machine performance may be characterized by dimensionless terms based on $E = 1$, $D = 1$ and $\rho = 1$ or on $n = 1$, $D = 1$ and $\rho = 1$. The units are: H (m); D (m); E (J·kg⁻¹); n (s⁻¹); ρ (kg·m⁻³); T (N·m); P (W); Q (m³·s⁻¹). The relations of these dimensionless terms to other existing definitions are given in annex of IEC 60193.

Subclause	Term	Definition	Symbol	Unit
3.13.1	flow velocity factor for each component passage	ratio of the maximum relative flow velocity in each component passage against the peripheral velocity u $\kappa_{uCO} = \frac{v_{CO}}{u}$	κ_{uCO}	-
3.13.2	dimension factor for each component passage	ratio of the hydraulic diameter of each component passage against the reference diameter $\kappa_{dCO} = \frac{d_{hCO}}{D}$	κ_{dCO}	-
3.13.3	dimension factor for disk friction loss	ratio of the diameter of the runner crown or runner band against the reference diameter $\kappa_T = \frac{D_d}{D}$	κ_T	-
3.13.4	scalable hydraulic energy loss index for each component passage	<all components of radial machine> $d_{ECOref} = \frac{\delta_{ECOref}}{1 + 0,351 \times (\kappa_{uCO} \cdot \kappa_{dCO})^{0,2}}$ <runner of axial machine> $d_{ECOref} = d_{ERUref} = \frac{\delta_{ECOref}}{1 + 0,25 \times (\kappa_{uCO} \cdot \kappa_{dCO})^{0,2}}$ <stationary component of axial machine> $d_{ECOref} = d_{ESTref} = 0,0123$	d_{ECOref}	-
3.13.5	scalable disk friction loss index	$d_{Tref} = \frac{\delta_{Tref}}{1 + 0,154 \cdot \kappa_T^{0,4}}$	d_{Tref}	-
3.13.6	loss distribution factor	ratio of scalable loss to total loss $V = \frac{\delta}{1 - \eta_h}$	V	-
3.13.7	specific speed	specific speed $N_{QE} = \frac{n \cdot Q_1^{0,5}}{E^{0,75}}$	N_{QE}	-
3.13.8	speed factor	speed factor $\eta_{ED} = \frac{n \cdot D}{E^{0,5}}$	η_{ED}	-
3.13.9	discharge factor	discharge factor $Q_{ED} = \frac{Q_1}{D^2 \cdot E^{0,5}}$	Q_{ED}	-
3.13.10	power factor	power factor $P_{ED} = \frac{P_m}{\rho_{1w} \cdot D^2 \cdot E^{1,5}}$	P_{ED}	-
3.13.11	energy coefficient	energy coefficient $E_{nD} = \frac{E}{n^2 \cdot D^2}$	E_{nD}	-
3.13.12	discharge coefficient	discharge coefficient $Q_{nD} = \frac{Q_1}{n \cdot D^3}$	Q_{nD}	-
3.13.13	power coefficient	power coefficient $P_{nD} = \frac{P_m}{\rho_{1w} \cdot n^3 \cdot D^5}$	P_{nD}	-

4 Scale-effect formula

4.1 General

4.1.1 Scalable losses

The energy flux through hydraulic machines and the various losses produced in the energy conversion process in a hydraulic machine can be typically illustrated by the flux diagrams shown in A.1.

One of the main features of the transposition formula as stated in this document is the separate consideration of three efficiency components. They are specific hydraulic energy efficiency η_E , volumetric efficiency η_Q and power efficiency η_T . Scale effect on each of these efficiency components is considered in this document.

Among the losses corresponding to these efficiency components, the following losses are subject to scale effect by the difference of Reynolds number and the relative roughness. Then these losses are referred to as “scalable losses” in this document.

- Specific hydraulic energy loss due to friction: E_{Lf}
- Leakage loss: q
- Disk friction loss: P_{Ld}

The relative magnitude of each scalable loss to each corresponding performance parameter is given as a function of the specific speed for each type of machine, except for discharge ($\delta_E = E_{Lf}/E$ and $\delta_T = P_{Ld}/P_m$).

E_{Lf} is the sum of the friction loss in various parts of the machine and it is expressed as the sum of the friction loss in each component as $E_{Lf} = \sum E_{LfCO}$. The scale effect on this loss is caused by the difference of Reynolds number and relative roughness between turbine A and turbine B and assessed by the formula 1.

The rest of the specific hydraulic energy loss is called “kinetic loss” or “non-scalable loss” and expressed as $E_{Lk} = \sum E_{LkCO}$. It is considered that the ratio E_{Lk}/E_m remains the same through the turbine A and the turbine B.

Similarly to E_{Lf} , the scale effect on the disk friction P_{Ld} is caused by the difference in Reynolds number and the relative roughness of the outer surface of the runner/impeller between the turbine A and the turbine B. However, due to the presence of the radial flow and the distortion of the boundary layer in the limited space between the runner/impeller and the stationary parts, the scale effect on P_{Ld} appears in a slightly different manner than on E_{Lf} . It is considered in this document that the scale effect on the disk friction is assessed by a scale effect shown in formulae D.7 to D.9.

In case of axial flow machines, the friction loss of the surface of runner hub is negligibly small and its scale effect is disregarded.

The scale effect on the leakage loss, q , is caused by the change of the friction loss coefficient of the seal clearance of the runner/impeller. In most cases, the leakage loss itself is minor and the scale effect on this loss is relatively very small.

In case that the geometry of the seal is maintained homologous between a turbine A and a turbine B within the criteria given in Table 1, the scale effect on the leakage loss is disregarded and the volumetric efficiencies of turbines A and B are considered equal.

Otherwise it is recommended to use the correction formula for η_Q as set out in Annex F.2.

In summary, the following efficiencies are addressed:

- For radial machines:
 - scale effect on losses: η_E , η_T
 - correction in case of non homologous seals on η_Q
- For axial machines:
 - scale effect on losses: η_E

4.1.2 Basic formulae of the scale effect on hydrodynamic friction losses

One feature of the scale effect formula is the consideration of surface roughness.

Following mathematical models are used to describe the physical effect of surface roughness on water flow:

- friction in a pipe
- friction on a flat plate
- disk friction

For pipe friction flow, the Colebrook diagram is used and the implicit Colebrook formula can be simplified into an explicit expression as shown below (see B.1).

$$\lambda = \lambda_0 \left[0,74 \left(8 \times 10^4 \frac{k_s}{d_h} + \frac{Re_{ref}}{Re_d} \right)^{0,2} + 0,26 \right] \quad (1)$$

where

Re_{ref} is 7×10^6

λ_0 is a constant equal to 0,008 5

k_s is the sand roughness

d_h is the hydraulic diameter of the water passage

Re_d is the Reynolds number of the water passage defined as $Re_d = \frac{d_h \cdot v}{\nu} = \frac{d_h \cdot v}{D \cdot u} Re$

Regarding the relationship between the sand roughness k_s and Ra , wide spread results have been reported so far. In this document, however, it is considered that the relationship can be expressed by the following formula:

$$\frac{k_s}{d_h} \cong 5 \frac{Ra}{d_h} \quad (2)$$

Then, Formula 1 is rewritten as follows:

$$\lambda = \lambda_0 \cdot \left[0,74 \times \left(4 \times 10^5 \frac{Ra}{d_h} + \frac{D \cdot u}{d_h \cdot v} \cdot \frac{Re_{ref}}{Re} \right)^{0,2} + 0,26 \right] \quad (3)$$

Figure 1 sketches the basic concept for the loss transposition from a turbine A to turbine B including different examples of surface roughness:

- B_1 : machine with very rough surface with decrease of efficiency compared to turbine A

- B₂: machine of reasonable roughness
- B₃: smooth machine.

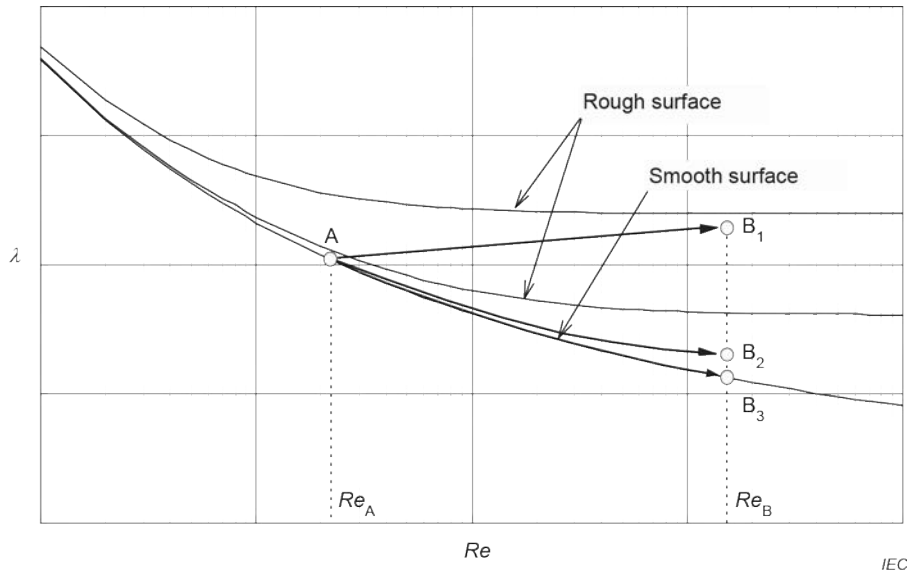


Figure 1 – Scale effect considering surface roughness

In order to calculate the difference of hydraulic efficiency between two hydraulically similar operating points A and B at different Reynolds numbers and different surface roughness conditions, the following formulae can be derived by using Formula 3 (see Annex A.2.2).

$$\Delta E_{A \rightarrow B} = \frac{\Delta \eta_E}{\eta_{EA}} = \delta_{Eref} \cdot \left(\frac{\lambda_A - \lambda_B}{\lambda_{ref}} \right) \quad (4)$$

The Colebrook diagram is valid for pipe flow, but it can be demonstrated that also friction loss coefficients of flat plate flow can be approximated with sufficient accuracy by similar formulae as shown below.

$$\begin{aligned} C_f &= C_{f0} \cdot \left[0,80 \times \left(10^5 \frac{k_s}{L} + \frac{Re_{ref}}{Re_f} \right)^{0,2} + 0,20 \right] \\ &= C_{f0} \cdot \left[0,80 \times \left(5 \times 10^5 \frac{Ra}{L} + \frac{D \cdot u}{L \cdot w} \cdot \frac{Re_{ref}}{Re} \right)^{0,2} + 0,20 \right] \end{aligned} \quad (5)$$

where

Re_{ref} is 7×10^6

C_{f0} is a constant equal to 0,003 2

Re_f is the Reynolds number of the plate $Re_f = \frac{L \cdot w}{\nu} = \frac{L \cdot w}{D \cdot u} \cdot Re$

L is length of the plate

w is the relative flow velocity on the plate

By replacing λ in Formula 4 by C_f given by Formula 5, Formula 4 is used to calculate the scale effect of the friction loss of runner blades of axial flow machines.

Similar formula of friction loss coefficient for disk friction is established as follows (See Annex D and [8]).

$$C_m = C_{m0} \cdot \left[0,85 \times \left(1,5 \times 10^4 \frac{k_{sT}}{r_d} + \frac{Re_{ref}}{Re_T} \right)^{0,2} + 0,15 \right]$$

$$= C_{m0} \cdot \left[0,85 \times \left(7,5 \times 10^4 \frac{Ra_T}{r_d} + \frac{D^2}{2r_d^2} \cdot \frac{Re_{ref}}{Re} \right)^{0,2} + 0,15 \right] \quad (6)$$

where

Re_{ref} is 7×10^6

C_{m0} is a constant equal to 0,001 9

k_{sT} is the equivalent sand roughness corresponding to Ra_T $k_{sT} = 5 \times Ra_T$

Ra_T is the weighted average of the arithmetical mean roughness of the outer surface of the runner and the surface of the stationary part facing to the runner as given by Formula 13

Re_T is the Reynolds number of the disk defined by

$$Re_T = \frac{r_d^2 \cdot \omega}{\nu} = \frac{r_d^2 \cdot \omega}{D \cdot u} \cdot Re = \frac{2 r_d^2}{D^2} \cdot Re = \frac{D_d^2}{2D^2} \cdot Re$$

r_d radius of runner crown or band, whichever larger (m)

ω angular velocity of the disk (rad/s)

By using Formula 6, the transposition formula for power efficiency (disk friction) is obtained as follows (see A.2.4):

$$\Delta_{TA \rightarrow B} = \frac{\Delta \eta_T}{\eta_{TA}} = \delta_{Tref} \cdot \left(\frac{C_{mA} - C_{mB}}{C_{mref}} \right) \quad (7)$$

4.2 Specific hydraulic energy efficiency

4.2.1 General

The scalable losses δ_{Eref} as appeared in Formula 4 refer to those of a model with smooth surface operating at a reference Reynolds number $Re_{ref} = 7 \times 10^6$. They have been established as a function of type and specific speed of a hydraulic machine. They are standardized and set out in Annex B for radial flow machines and Annex C for axial flow machines.

By putting the scale effect formula of Formula 3 into Formula 4, the following formula for the individual step-up for a machine component is derived (see B.2).

$$\Delta_{ECO A \rightarrow B} = \frac{\Delta \eta_{ECO}}{\eta_{EA}} = \delta_{ECOref} \cdot \left(\frac{\lambda_{COA} - \lambda_{COB}}{\lambda_{COref}} \right)$$

$$\Delta_{ECO A \rightarrow B} = d_{ECOref} \cdot \left[\left(4 \times 10^5 \kappa_{uCO} \frac{Ra_{COA}}{D_A} + \frac{7 \times 10^6}{Re_A} \right)^{0,2} - \left(4 \times 10^5 \kappa_{uCO} \frac{Ra_{COB}}{D_B} + \frac{7 \times 10^6}{Re_B} \right)^{0,2} \right] \quad (8)$$

where

- δ_{ECOref} is the reference scalable loss for each component passage (see A.2.2 and B.2.2)
- κ_{uCO} is the flow velocity factor for each component passage (see B.2.1)
- d_{ECOref} is the reference scalable loss index for each component passage (see B.2.2)

Then the transposition of the specific hydraulic energy for the whole machine can be calculated by the formula below:

$$\Delta E_{A \rightarrow B} = \frac{\Delta \eta_E}{\eta_{EA}} = \sum \Delta E_{\text{COA} \rightarrow B} \quad (9)$$

The structure of formula is valid for all types of hydraulic reaction machines.

4.2.2 Radial flow machines

For radial flow machines, Formula 8 allows to calculate the individual scaling in the various components, using d_{ECOref} and κ_{uCO} which have been established in 6.3.2 and 6.3.3 for each individual component from spiral case to draft tube.

4.2.3 Axial flow machines

For axial flow machines, the scalable loss is divided into two parts, stationary parts and runner blades. The efficiency scaling ratio for the scalable loss of stationary parts, ΔE_{ST} , can be obtained by Formula 8 in the same way as for radial flow machines. In this case, it is considered that the representative flow velocity factor κ_{uST} for all stationary parts can be represented by 0,8 times the flow velocity factor for guide vanes; namely, $\kappa_{\text{uST}} = 0,8 \times \kappa_{\text{uGV}}$. The value of κ_{uST} is shown in 6.3.4.

As stated in 4.1.2, scale effect formula for flat plate represented by Formula 5 is supposed to be applied to runner blades.

$$\Delta E_{\text{ERU}} = d_{\text{ERU ref}} \cdot \left[\left(5 \times 10^5 \kappa_{\text{uRU}} \cdot \frac{Ra_{\text{RUA}}}{D_A} + \frac{7 \times 10^6}{Re_A} \right)^{0,2} - \left(5 \times 10^5 \kappa_{\text{uRU}} \cdot \frac{Ra_{\text{RUB}}}{D_B} + \frac{7 \times 10^6}{Re_B} \right)^{0,2} \right] \quad (10)$$

4.3 Volumetric efficiency

The influence of Reynolds number on the volumetric efficiency is considered negligible in case that the geometrical configuration of clearances, labyrinths, balancing holes/pipes is similar at both model and prototype.

The influence of the seal geometry is considered negligible when the deviation is within the limits of Table 1 hereafter.

Table 1 – Permissible deviation of the geometry of model seals from the prototype

Dimensions and design	Permissible deviation from the prototype
Radial clearance of runner seals ^a	0 ~ + 20 %
Diameter of seal	± 5 %
Axial length of seal clearances ^a	0 ~ - 20 %
Number of steps or grooves	should be the same
Shape of steps or grooves (see Annex F)	shall be homologous
NOTE In case of axial flow machines, the words marked by ^a should be read as "blade tip clearances" and "thickness of blade tip", respectively. Only these two criteria for radial clearance and thickness of blade tip should be applied.	

If the homology of the runner seals of model and prototype can not be fulfilled, the volumetric efficiency has to be adjusted according to the following formula:

$$\Delta_{Q_{A \rightarrow B}} = \frac{\Delta \eta_Q}{\eta_{QA}} = \frac{\eta_{QB}}{\eta_{QA}} - 1 \quad (11)$$

Unless there is a different agreement between parties, it is recommended to use the formula given in F.2 for the calculation of Δ_Q .

4.4 Power efficiency (disk friction)

4.4.1 Radial flow machines

Disk friction has a significant impact on the efficiency of low specific speed radial machines. The following power transposition formula, Formula 12, is obtained by putting Formula 6 into Formula 7. It describes the variation of power efficiency of radial flow machines due to the difference in Reynolds number and surface roughness (see Annex D).

$$\Delta_{T_{A \rightarrow B}} = \frac{\Delta \eta_T}{\eta_{TA}} = \delta_{Tref} \cdot \left(\frac{C_{mA} - C_{mB}}{C_{mref}} \right)$$

$$\text{Therefore } \Delta_{T_{A \rightarrow B}} = d_{Tref} \cdot \left[\left(7,5 \times 10^4 \kappa_T \frac{Ra_{TA}}{D_A} + \frac{7 \times 10^6}{Re_A} \right)^{0,2} - \left(7,5 \times 10^4 \kappa_T \frac{Ra_{TB}}{D_B} + \frac{7 \times 10^6}{Re_B} \right)^{0,2} \right] \quad (12)$$

where

$\delta_{Tref} = 1 - \eta_{Tref}$ is the reference disk friction loss (see A.2.4)

d_{Tref} is the scalable disk friction loss index (see D.2)

κ_T is the dimension factor for the disk relating to disk friction loss (see D.1)

Ra_T is the representative roughness given by Formula 13.

4.4.2 Axial flow machines

For axial flow machines, the runner hub friction induces neglectable losses compared to hydraulic losses. Therefore, Δ_T is set at zero for axial flow machines.

5 Surface roughness of model and prototype

5.1 General

Surface roughness has major impact on friction losses and therefore on machine efficiency.

Refining surface roughness increases efficiency but also costs, as shown in Figure 2. Moreover, the roughness of the prototype should not be finer than the roughness expected after some period of operation.

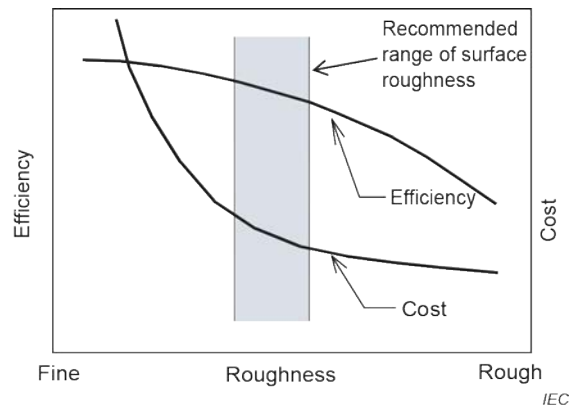


Figure 2 – Impact of surface roughness on turbine efficiency and costs

The improvement of the efficiency achievable by the replacement of some components can be assessed by comparing the efficiencies calculated with the roughness measured on the existing components and with the design values for the new components.

In case of rehabilitation projects,

- for the assessment of the efficiency improvement, the roughness of the prototype components shall be measured on the existing unit whenever possible or agreed upon between parties;
- roughness data of those components not to be replaced shall be provided by the owner with the specification or agreed upon between parties.

5.2 Measurement of surface roughness

5.2.1 Procedure

The measurements shall correctly represent the average surface roughness of each component.

It is recommended to follow the normative references presented in Clause 2.

The use of a stylus type digital microprofiler for $Ra < 50 \mu\text{m}$ and a digital depth indicator for $Ra > 50 \mu\text{m}$ is recommended. The expected Ra uncertainty is $\pm 20 \%$ of the reading for a digital microprofiler and $\pm 40 \%$ of the reading for a digital depth indicator. These uncertainty values have been estimated from measurements done by different users also writers of this document, but method of measuring (including instrument) can amplify this value of uncertainty.

The roughness should be measured at specific locations. The arithmetic average for each component should be used, keeping the distribution of points as stated below.

- Spiral case: 9 points or more; at 3 radial sections: entrance, middle, end of casing

- 2 stay vane channels: 6 points or more per stay vane channel; 2 points per side of the vane, 1 point on the top of the channel, 1 point on the bottom of the channel
- 2 guide vane channels: 10 points or more per guide vane channel between 2 guide vanes; 6 points on the inner side of the guide vane, 2 points on the outer side of the guide vane, 1 point on the top of the channel, 1 point on the bottom of the channel
- 3 runner channels: 10 points or more per channel considering:
 - for radial machines: 3 points in region A on suction side, 3 points in region A on pressure side, 1 point region B on suction side, 1 point region B on pressure side, 1 point on the crown and 1 point on the band (regions A and B are defined Figure 3)
 - for axial turbines: 5 points on suction side and 5 points on pressure side (see Figure 4)
- Draft tube: 10 points or more; with 70 % of them upstream of the bend
- Stationary part for disk friction: 4 points or more at the areas facing to the sample points of runner.
- Rotating part for disk friction:
 - Runner outer crown: 2 points or more near outer periphery
 - Runner outer band: 2 points or more near outer periphery

Since the roughness of the rotating part has dominant influence of the disk friction torque, the weighted mean roughness as given by the following formula shall be used for Ra_T in Formula 13.

$$Ra_T = \frac{2Ra_{TR} + Ra_{TS}}{3} \quad (13)$$

where

Ra_{TR} is the average roughness of those measured on the rotating part

Ra_{TS} is the average roughness of those measured on the stationary part

The surface roughness shall be measured as it appears in actual operation. Painted surface shall be measured over the paint coat.

For axial flow machines, the roughness value given by Formula 14 shall be used as a representative roughness for all stationary parts.

$$Ra_{ST} = \frac{Ra_{SV} + Ra_{GV}}{2} \quad (14)$$

5.2.2 Roughness of model and prototype

The model is known to have a very good uniformity of roughness per component. When this is the case, 2 to 4 measuring points per component are sufficient. For repetitive built components, like stay vanes, guide vanes and runner blades, measurement on at least 2 repetitive components is recommended.

Design values for the prototype roughness, which are provided by the supplier or agreed upon, shall be used as the roughness of the prototype. When the machine components are completed in the factory, the surface roughness shall be measured. It is recommended to verify that the average value of the measured roughness of each component is equal or finer than the design roughness for the component.

5.2.3 Measurement of very rough surfaces

For the measurement of rough surfaces of old machines, the recommendations described below for R_a values larger than 50 μm such as found for heavily rusted surfaces shall be taken into consideration.

On old machines, the roughness values are often beyond any existing portable roughness tester range which makes the measurement difficult.

In these situations, one method is to take molds at most representative locations using appropriate plastic material and measure the roughness of these moulds, using mutually agreed methods to find an equivalent R_a value.

Another method is to measure the average depth of these surfaces, using a depth indicator, preferably a dial indicator to speed up the measurement. The plunger tip of the indicator shall be sharp, with an angle of approximately 30 degrees total, relative to a vertical line. The measurement locations should be as indicated in 5.2.1. The average roughness depth thus measured for each component shall be divided by a factor of 10 in order to obtain approximately the equivalent R_a value ³. With this method, there is no need to have a special consideration for surfaces having deep voids, provided that the area having these voids is considered relatively small, (less than 10 % of the component total area). If the area with deep voids is significant, mutual agreement on how to take these deep voids into account should take place.

For these heavily rusted and very rough surfaces, comparison coupons could also be used, if interpretation of those coupons and conversion to equivalent R_a is mutually agreed.

Special attention shall be given to choose the meaningful roughness values from the measurements. Based on the current state of knowledge, it is believed that areas having scattered deep voids do not create as much losses as their measured value would indicate. Indeed, the stream lines over such areas pass over the voids without reaching the bottom and do not create significantly larger losses. Therefore, it is recommended to ignore areas with recess deeper than approximately 1,5 mm.

Once the above considerations have been taken into account, the relationship between k_s and R_a as expressed by Formula 2 can be tentatively applied also to heavily rusted surfaces.

5.3 Surface roughness ranges

Table 2 shows the recommended range of model roughness in order to optimize the transposition procedure detailed in chapter 8.

Table 2 – Recommended roughness range for the model

Component	SP	SV	GV	RU	DT	TR	TS
$R_{a\text{COM}}$ (μm)	0,4 to 1,6	0,4 to 1,6	0,2 to 0,8	0,2 to 0,8	0,4 to 1,6	0,4 to 1,6	0,4 to 1,6

Table 3 and Table 5 show the minimum recommended surface roughness for prototype components below which no more increase in efficiency is expected. Table 4 and Table 6 show the maximum recommended roughness for prototype components of new turbines.

³ Note that this factor of 10 is based on a single case study of an old Francis turbine, where an efficiency test was performed before and after refurbishing the stay vane and guide vane passages. More case studies and research are needed to confirm this factor.

Table 3 – Minimum recommended prototype roughness for new radial or diagonal machines

Roughness (μm) applied on		Specific hydraulic energy, E (J/kg)				
		300	850	1 400	3 000	8 000
Spiral casing	Ra_{SP}	$\geq 5,0$	$\geq 3,2$	$\geq 3,2$	$\geq 3,2$	$\geq 3,2$
Stay vane channel	Ra_{SV}	$\geq 5,0$	$\geq 3,2$	$\geq 1,6$	$\geq 1,6$	$\geq 1,6$
Guide vane channel	Ra_{GV}	$\geq 1,8$	$\geq 1,4$	$\geq 1,0^a$	$\geq 0,8^a$	$\geq 0,8^a$
Runner channel	Ra_{RU}	$\geq 1,8$	$\geq 1,4$	$\geq 1,0^a$	$\geq 0,8^a$	$\geq 0,8^a$
Draft tube	Ra_{DT}	$\geq 5,0$	$\geq 3,2$	$\geq 3,2$	$\geq 3,2$	$\geq 3,2$
Rotating part	Ra_{TR}	$\geq 5,0$	$\geq 3,2$	$\geq 1,6$	$\geq 1,6$	$\geq 1,6$
Stationary part	Ra_{TS}	$\geq 5,0$	$\geq 3,2$	$\geq 1,6$	$\geq 1,6$	$\geq 1,6$

^a There are indications that small Ra values may deteriorate within a few months. Further studies are required in order to better understand the phenomena taking place with the surface finish once a machine is in operation for a few months.

NOTE Roughness values for intermediate specific hydraulic energy shall be linearly interpolated.

Table 4 – Maximum recommended prototype roughness for new radial or diagonal machines

Roughness (μm) applied on		Specific hydraulic energy, E (J/kg)	
		$\leq 2\,000$	$> 2\,000$
Spiral casing	Ra_{SP}	$\leq 25,0$	$\leq 12,5$
Stay vane channel	Ra_{SV}	$\leq 25,0$	$\leq 12,5$
Guide vane channel	Ra_{GV}	$\leq 12,5$	$\leq 6,3$
Runner channel	Ra_{RU}	$\leq 3,2$ for region A on suction side $\leq 6,3$ for other regions	$\leq 3,2$ for all regions
Draft tube	Ra_{DT}	$\leq 25,0$	$\leq 12,5$
Rotating part	Ra_{TR}	$\leq 25,0$	$\leq 12,5$
Stationary part	Ra_{TS}	$\leq 25,0$	$\leq 12,5$

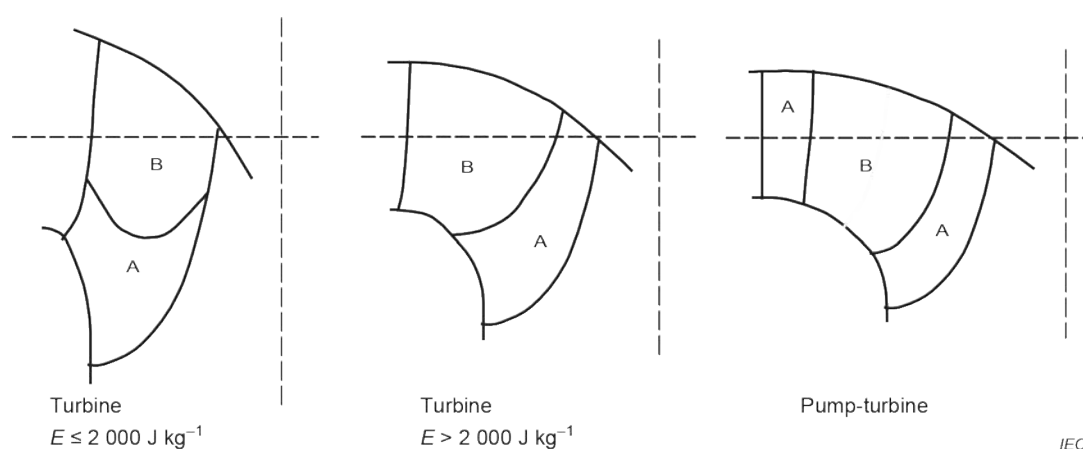


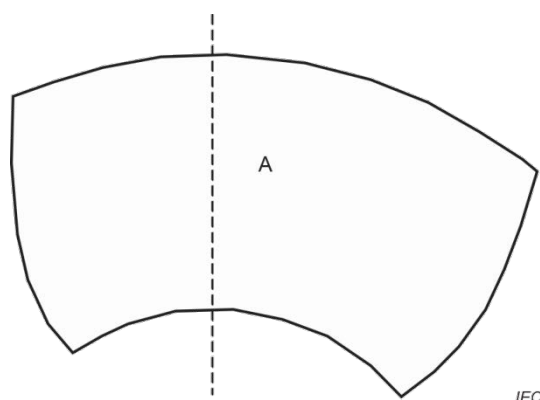
Figure 3 – Surface roughness regions for Francis runner blades

Table 5 – Minimum recommended prototype roughness for new axial turbines

Roughness (μm) applied on		Specific hydraulic energy, E (J/kg)			
		30	130	300	600
Spiral casing	Ra_{SP}	$\geq 6,3$	$\geq 4,0$	$\geq 3,2$	$\geq 2,0$
Stay vane	Ra_{SV}	$\geq 6,3$	$\geq 4,0$	$\geq 3,2$	$\geq 2,0$
Guide vane	Ra_{GV}	$\geq 6,3$	$\geq 4,0$	$\geq 3,2$	$\geq 2,0$
Runner	Ra_{RU}	$\geq 2,4$	$\geq 1,3$	$\geq 0,8$ a	$\geq 0,6$ a
Draft tube	Ra_{DT}	$\geq 6,3$	$\geq 4,0$	$\geq 3,2$	$\geq 2,0$
^a There are indications that small Ra values may deteriorate within a few months. Further studies are required in order to better understand the phenomena taking place with the surface finish once a machine is in operation for a few months.					
NOTE Roughness values for intermediate specific hydraulic energy shall be linearly interpolated.					

Table 6 – Maximum recommended prototype roughness for new axial turbines

Roughness (μm) applied on		Specific hydraulic energy, E (J/kg)	
		≤ 300	> 300
Spiral casing	Ra_{SP}	$\leq 25,0$	$\leq 12,5$
Stay vane channel	Ra_{SV}	$\leq 25,0$	$\leq 12,5$
Guide vane channel	Ra_{GV}	$\leq 12,5$	$\leq 6,3$
Runner channel	Ra_{RU}	$\leq 6,3$	$\leq 3,2$
Draft tube	Ra_{DT}	$\leq 25,0$	$\leq 12,5$

**Figure 4 – Surface roughness region for axial flow runner blade**

6 Standardized values of scalable losses and pertinent parameters

6.1 General

The values of d_{ECOref} and κ_{uCO} to calculate the transposition of specific hydraulic energy efficiency and those of d_{Tref} and κ_T to calculate the transposition of power efficiency (disk friction) are defined in this clause. These values refer to machine with hydraulically smooth surface at reference Reynolds number $Re_{ref} = 7 \times 10^6$.

6.2 Specific speed

A hydraulic machine of any type can be described by its specific speed at the point of optimum efficiency which is calculated by the following formula:

$$N_{QE} = \frac{n \cdot Q_1^{0,5}}{E^{0,75}} \quad \text{or} \quad N_{QE} = n_{ED} \cdot Q_{ED}^{0,5} = \frac{Q_{nD}^{0,5}}{E_{nD}^{0,75}} \quad (15)$$

where

E is the specific hydraulic energy of machine (J kg^{-1})

Q_1 is the discharge of machine (m^3/s)

n is the rotational speed (s^{-1})

For reversible pump-turbines, the specific speed at each maximum efficiency point when operating as a turbine or as a pump, should be calculated and taken as a reference to obtain the scalable losses in turbine or pump operation, respectively.

During transposition, the shift in specific hydraulic energy, volumetric discharge and power induces a shift in N_{QE} . However this difference is very small, for example, 1 % of change in N_{QE} results in an efficiency difference less than 0,01 %.

As the specific speeds of different machines from different manufacturers for the same specified prototype conditions are quite close, it is possible to fix d_{ECOREf} , κ_{uCO} , d_{Tref} and κ_T in advance in a specification. Also, for a comparative model test, common values of d_{ECOREf} , κ_{uCO} , d_{Tref} and κ_T should be defined and agreed upon prior to set in efficiency guarantees.

6.3 Parameters for specific hydraulic energy efficiency transposition

6.3.1 General

Once an investigated hydraulic machine is described by its specific speed, the factors d_{ECOREf} and κ_{uCO} can be determined by the formulae shown in Tables 4, 5, 6 and 7.

6.3.2 Francis turbines

Table 7 – Standardized scalable loss index d_{ECOREf} and standardized velocity factor κ_{uCO} for Francis turbines

Component passage	d_{ECOREf}	κ_{uCO}
Spiral case	$d_{ESPre} = 0,40 / 100$	$\kappa_{uSP} = -0,5 N_{QE} + 0,33$
Stay vanes	$d_{ESVref} = (-N_{QE} + 0,40)/100$	$\kappa_{uSV} = -1,4 N_{QE} + 0,60$
Guide vanes	$d_{EGVref} = (-2,9 N_{QE} + 1,65)/100$	$\kappa_{uGV} = -3,3 N_{QE} + 1,29$
Runner	$d_{ERUref} = (3,4 N_{QE} + 0,55)/100$	$\kappa_{uRU} = -1,3 N_{QE} + 0,90$
Draft tube	$d_{EDTref} = (0,5 N_{QE} + 0,05)/100$	$\kappa_{uDT} = 0,28$
NOTE The above formulae are valid for $0,06 \leq N_{QE} \leq 0,30$. Outside of this range the values are not validated. In this case, if the transposition formulae are applied to the evaluation of the contractual model test results, prior agreement shall be made among the concerned parties.		

6.3.3 Pump-turbines

6.3.3.1 Turbine operation

Table 8 – Standardized scalable loss index d_{ECOref} and standardized velocity factor κ_{uCO} for pump-turbines in turbine operation

Component passage	d_{ECOref}	κ_{uCO}
Spiral case	$d_{\text{ESPref}} = 0,45 / 100$	$\kappa_{\text{uSP}} = -0,5 N_{\text{QE}} + 0,34$
Stay vanes	$d_{\text{ESVref}} = (-N_{\text{QE}} + 0,45) / 100$	$\kappa_{\text{uSV}} = -1,4 N_{\text{QE}} + 0,57$
Guide vanes	$d_{\text{EGVref}} = (-2,9 N_{\text{QE}} + 1,65) / 100$	$\kappa_{\text{uGV}} = -3,3 N_{\text{QE}} + 1,23$
Runner	$d_{\text{ERUref}} = (3,4 N_{\text{QE}} + 1,35) / 100$	$\kappa_{\text{uRU}} = -1,3 N_{\text{QE}} + 0,87$
Draft tube	$d_{\text{EDTref}} = (0,5 N_{\text{QE}} + 0,05) / 100$	$\kappa_{\text{uDT}} = 0,31$
NOTE The above formulae are valid for $0,06 \leq N_{\text{QE}} \leq 0,20$. Outside of this range the values are not validated. In this case, if the transposition formulae are applied to the evaluation of the contractual model test results, prior agreement shall be made among the concerned parties.		

6.3.3.2 Pump operation

Table 9 – Standardized scalable loss index d_{ECOref} and standardized velocity factor κ_{uCO} for pump-turbines in pump operation

Component passage	d_{ECOref}	κ_{uCO}
Spiral case	$d_{\text{ESPref}} = 0,45 / 100$	$\kappa_{\text{uSP}} = -0,5 N_{\text{QE}} + 0,31$
Stay vanes	$d_{\text{ESVref}} = (-N_{\text{QE}} + 0,50) / 100$	$\kappa_{\text{uSV}} = -1,4 N_{\text{QE}} + 0,53$
Guide vanes	$d_{\text{EGVref}} = (-2,9 N_{\text{QE}} + 1,65) / 100$	$\kappa_{\text{uGV}} = -3,3 N_{\text{QE}} + 0,96$
Runner	$d_{\text{ERUref}} = (3,4 N_{\text{QE}} + 1,55) / 100$	$\kappa_{\text{uRU}} = -1,3 N_{\text{QE}} + 0,79$
Draft tube	$d_{\text{EDTref}} = (0,5 N_{\text{QE}} + 0,05) / 100$	$\kappa_{\text{uDT}} = 0,27$
NOTE The above formulae are valid for $0,06 \leq N_{\text{QE}} \leq 0,20$. Outside of this range the values are not validated. In this case, if the transposition formulae are applied to the evaluation of the contractual model test results, prior agreement shall be made among the concerned parties.		

6.3.4 Axial flow machines

Table 10 – Standardized scalable loss index d_{ECOref} and velocity factor κ_{uCO} for axial flow machines

Component passage	d_{ECOref}	κ_{uCO}
Runner	$d_{\text{ERUref}} = 2,45 / 100$	$\kappa_{\text{uRU}} = 1,03$
All stationary parts	$d_{\text{ESTref}} = 1,23 / 100$	$\kappa_{\text{uST}} = 0,19$
NOTE The above formulae are valid for $0,25 \leq N_{\text{QE}} \leq 0,70$. Outside of this range the values are not validated. In this case, if the transposition formulae are applied to the evaluation of the contractual model test results, prior agreement shall be made among the concerned parties.		

6.4 Parameters for power efficiency (disk friction) transposition

6.4.1 Francis turbines

The following formulae shall be used to obtain d_{Tref} and κ_T (see D.3).

$$d_{Tref} = \left(0,44 + \frac{0,004}{N_{QE}^2} \right) \times \frac{1}{100} \text{ for } 0,06 \leq N_{QE} \leq 0,30 \quad (16)$$

$$\kappa_T = -5,7 N_{QE} + 2,0 \text{ or } 1,0 \text{ whichever larger} \quad (17)$$

The d_{Tref} formula is valid in the corresponding specific speed range. Outside of this range the values are not validated. In this case, if the transposition formulae are applied to the evaluation of the contractual model test results, prior agreement shall be made among the concerned parties.

6.4.2 Pump-turbines

6.4.2.1 Turbine operation

The following formulae shall be used to obtain d_{Tref} and κ_T (see D.3).

$$d_{Tref} = \left(0,97 + \frac{0,012}{N_{QE}^2} \right) \times \frac{1}{100} \text{ for } 0,06 \leq N_{QE} \leq 0,20 \quad (18)$$

$$\kappa_T = -8,3 N_{QE} + 2,7 \text{ or } 1,0 \text{ whichever larger} \quad (19)$$

The d_{Tref} formula is valid in the corresponding specific speed range. Outside of this range the values are not validated. In this case, if the transposition formulae are applied to the evaluation of the contractual model test results, prior agreement shall be made among the concerned parties.

6.4.2.2 Pump operation

The following formulae shall be used to obtain d_{Tref} and κ_T (see D.3).

$$d_{Tref} = \left(1,23 + \frac{0,015}{N_{QE}^2} \right) \times \frac{1}{100} \text{ for } 0,06 \leq N_{QE} \leq 0,20 \quad (20)$$

$$\kappa_T = -7,5 N_{QE} + 2,7 \text{ or } 1,0 \text{ whichever larger} \quad (21)$$

The d_{Tref} formula is valid in the corresponding specific speed range. Outside of this range the values are not validated. In this case, if the transposition formulae are applied to the evaluation of the contractual model test results, prior agreement shall be made among the concerned parties.

6.4.3 Axial flow machines

For axial flow machines, the runner hub friction induces neglectable losses compared to hydraulic losses. Therefore, d_{Tref} is set at zero for axial flow machines.