INTERNATIONAL STANDARD

ISO 5389

First edition
1992-05-15

Turbocompressors — Performance test code

Turbocompresseurs — Code d'essai des performances

Reference number
ISO 5389 : 1992 (E)

Contents

International Organization for Standardization
Case postale 56 • CH-1211 Genève 20 • Switzerland
Printed in Switzerland

Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

Draft International Standards adopted by the technical committees are circulated to the member bodies for voting. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

International Standard ISO 5389 was prepared by Technical Committee ISO/TC 118, *Compressors, pneumatic tools and pneumatic machines*, Sub-Committee SC 1, *Turbo compressors.*

Annexes A, B, C, D and E form an integral part of this International Standard. Annexes F and G are for information only.

Introduction

The terms "guarantee" or "guaranteed" and "performance" used in this International Standard are to be understood in the engineering rather than the contractual sense. A guarantee relates to a specific aspect of the plant or its operation which is defined in the contract.

With the aid of the test described in this International Standard, the actual performance data can be compared with the guaranteed values.

The contractual consequences of any deviations are not covered by this International Standard. A satisfactory test result does not signify acceptance in the contractual sense, as such acceptance may depend on other conditions stipulated in the contract.

This International Standard provides standard directions for conducting and reporting tests on compressors to establish their performance concerning one or more of the following aspects under specified conditions and for comparing the results with the guaranteed performance:

a) the quantity of gas or vapour delivered;

b) the pressure rise or pressure ratio produced;

c) the power required for compression or the efficiency of the compressor (according to specified definitions);

d) the stable working range — surge and maximum flow limits.

To meet this purpose, this International Standard establishes rules concerning

a) the test procedure (including the measurements to be taken, and the preparation and execution of the test);

b) the instrumentation to be used to provide adequate accuracy;

c) the methods of converting the test results in order to provide values that may be compared with the guaranteed figures;

d) the confidence limits of the converted test results according to the accuracy of the particular measurements.

Turbocompressors — Performance test code

1 Scope

This International Standard covers blowers or compressors and exhausters of the centrifugal, mixed flow, or axial flow types (inclusively covered by the term turbocompressors), with and without intercooling, handling any vapour or gas the physical properties of which are reliably known.

It may be applied to any compression process, with or without bleed-off or sidestreams, which takes place in one or more casings.

This International Standard gives no rules for the measurement of any other aspect of the compressor which may be the subject of a guarantee, such as

a) mechanical performance;

b) vibrations;

c) pulsations;

d) noise level;

e) service and reliability;

f) commercial questions.

The theory used in this International Standard is based on the laws of similarity of fluid flow (similar velocity triangles). The observation of these laws determines specific requirements for the execution of acceptance tests. In all cases, where a close approximation to these requirements is not possible, this International Standard can only be applied by mutual agreement. Compressors supplied to handle gases the physical properties of which are not reliably known can only be tested within certain limits.

For identical compressors, produced in series, the testing of an arbitrarily chosen sample may be agreed upon.

2 Normative references

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

ISO 31 (parts 0 to 13)[1], *Quantities, units and symbols.*

ISO 1000 : —[2], *SI units and recommendations for the use of their multiples and of certain other units.*

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

3 Definitions, formulae and reference processes

For the purposes of this International Standard, the following definitions apply.

3.1 Definitions relating to compressor performance

3.1.1 standard inlet point: The inlet point considered to be representative of the compressor. It is generally at the compressor inlet flange.

1) Currently under revision.

2) To be published. (Revision of ISO 1000 : 1981.)

3.1.2 standard discharge point: The discharge point considered to be representative of the compressor. It is generally at the compressor discharge flange.

3.1.3 Quantity of gas or vapour delivered

3.1.3.1 usable mass rate of flow for a compressor: The mass rate of flow delivered at the standard discharge point.

3.1.3.2 usable mass rate of flow for an exhauster: The mass rate of flow aspirated at the standard inlet point.

3.1.3.3 actual inlet volume flow for a compressor: The actual volume rate of flow compressed and delivered at the standard discharge point, referred to the conditions of temperature, pressure and composition (for example humidity) prevailing at the standard inlet point.

3.1.3.4 actual inlet volume flow for an exhauster: The actual volume rate of flow aspirated at the standard inlet point.

NOTES

1 Unless otherwise specified, the actual inlet volume flow will be referred to total temperature and total pressure.

2 For gas-vapour mixtures, see A.4.2.7.

3.2 Basic formulae for gases

Gas basic formulae are given in table 1.

3.3 Reference process

The determination of the internal power (3.6.5) is based on the assumption of a reversible reference process, hence the necessity of a definition of the corresponding efficiency, taking account of energy losses due to the irreversibility of the actual compression process.

The reference process is characterized by the law

$$p = f(v)$$

which is used to determine the specific compression work:

$$W_m = \int_1^2 v \, \mathrm{d}p$$

The total specific compression work is thus determined using the equation

$$W_{m,\mathrm{t}} = W_m + \frac{c_2^2 - c_1^2}{2}$$

By approximation with the low flow speeds ($Ma < 0{,}2$) normally prevailing in the inlet and discharge nozzles, the total pressures and total temperatures can be used in the calculation directly:

$$W_{m,\mathrm{t}} = \int_1^2 (v \, \mathrm{d}p)_\mathrm{t}$$

Table 1 — Gas basic formulae

No.	Term	Formula: For a real gas	Formula: For a perfect gas
3.2.1	Equation of state	$pV = ZRT$	$pV = RT$
3.2.2	Compressibility factor	Z	$Z = 1$
3.2.3	Isothermal deviation factor	$Y = \frac{p}{V}\left(\frac{\partial V}{\partial p}\right)_T = 1 - \frac{p}{Z}\left(\frac{\partial Z}{\partial p}\right)_T$	$Y = 1$
3.2.4	Isobaric deviation factor	$X = \frac{T}{V}\left(\frac{\partial V}{\partial T}\right)_p - 1 = \frac{T}{Z}\left(\frac{\partial Z}{\partial T}\right)_p$	$X = 0$
3.2.5	Isentropic exponent	$\kappa = -\frac{V}{p}\left(\frac{\partial p}{\partial V}\right)_S = \frac{\gamma}{Y}$	$\kappa = \gamma = \frac{c_p}{c_V}$

NOTES

1 The data serving as a reference for the determination of gas properties shall be agreed between purchaser and vendor.

2 Clause A.1 deals with general recommendations relating to the thermodynamic data for gases and gas mixtures.

3 Clause A.2 deals with specific recommendations for some of the more common gases.

3.4 Reference processes for use with perfect or near-perfect gases

The following methods of computation of specific compression work are recommended to be applied

— when agreed between purchaser and vendor, or

— when the deviation of gas properties from perfect gas laws at any state point of the compression process of an uncooled compressor, or at any state point of a compression section included between two successive intercoolers of a cooled compressor, do not exceed the limits given in table 2 for the appropriate pressure ratio.

Within the limits given, the errors in specific isentropic compression work and discharge specific volume will be less than 1 % and 2 % respectively if calculations are made according to perfect gas laws instead of real gas equations.

It is recommended that in most cases polytropic compression be used as the reference process.

Polytropic compression should always be adopted for any case in which the gas used for the acceptance test has a ratio of specific heats which differs from that of the guarantee gas by more than 1 %.

3.4.1 Polytropic compression

3.4.1.1 A polytropic compression process follows the law

$$pv^n = p_1 v_1{}^n = \text{constant}$$

where for perfect gases

$$n = \frac{\lg\left(\dfrac{p_2}{p_1}\right)}{\lg\left(\dfrac{p_2}{p_1}\cdot\dfrac{T_1}{T_2}\right)}$$

3.4.1.2 The specific compression work based on static conditions is calculated using

$$W_{m,\text{pol}} = \int_1^2 (v\,\mathrm{d}p)_{\text{pol}} = \left(\frac{n}{n-1}\right) p_1 v_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

3.4.1.3 In its general form the polytropic compression offers, owing to the free choice of the exponent n, great liberty in adapting it to any change of state. With $n = \gamma$ the compression becomes isentropic. When n approaches unity, the compression approaches an isothermal process. If, with multistage compressors, a single-stage reference compression does not represent the actual process with sufficient accuracy, a multistage polytropic compression may be chosen. From the above it follows that the polytropic compression is suited for cooled and uncooled, and for single-stage and multistage, compressors.

3.4.1.4 In the case of compressors with interstage cooling the polytropic compression approaches isothermal compression at one extreme, and isentropic compression at the other, depending on whether the process takes place at a constant temperature or the aerodynamic flow losses only are removed by the cooler. An approximation has to be made by suitable choice of the exponent n and the number of stage groups according to the arrangement and effectiveness of the cooling.

3.4.1.5 For compressors without cooling (adiabatic compression) the isentropic process is often used as a reference, but here too the polytropic process offers a better basis on which to assess the aerodynamic losses of a compressor.

It takes into account the increased compression work caused by the reheat losses. This increase is particularly noticeable at either high pressure ratios or low efficiencies.

3.4.2 Isentropic compression

3.4.2.1 In this reference process, compression takes place over the whole part of the pressure range (depending on whether it is a single-stage or multistage machine) at constant entropy, i.e. $n = \kappa$.

Table 2[1)] — Limits for the pressure ratio

Pressure ratio[2)] $\frac{p_2}{p_1}$	Maximum ratio between maximum and minimum values of κ ($= \gamma$)	X_{max}	X_{min}	Y_{max}	Y_{min}
1,4	1,12	0,279	− 0,344	1,071	0,925
2	1,10	0,167	− 0,175	1,034	0,964
4	1,09	0,071	− 0,073	1,017	0,982
8	1,08	0,050	− 0,041	1,011	0,988
16	1,07	0,033	− 0,031	1,008	0,991
32	1,06	0,028	− 0,025	1,006	0,993

1) Table taken from [1].

2) For pressure ratios between those shown, the limiting values shall be obtained by interpolation.

3.4.2.2 Isentropic compression follows the law

$$p v^{\kappa} = p_1 v_1^{\kappa} = \text{constant}$$

3.4.2.3 The specific compression work based on static conditions is expressed by the equation

$$W_{m,s} = \int_1^2 (v\,dp)_s = \frac{\kappa}{\kappa - 1} p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{\kappa - 1}{\kappa}} - 1 \right]$$

3.4.3 Isothermal compression

In this reference process, compression takes part over the whole (single stage) or part (multistage) of the pressure range at a constant temperature. As a rule the inlet temperature T_1 of the compressor or stage group under consideration is used. The exponent $n = 1$, and the specific compression work based on static conditions is defined by the equation

$$W_{m,T} = \int_1^2 (v\,dp)_T = p_1 v_1 \ln \left(\frac{p_2}{p_1} \right)$$

3.5 Reference processes for use with real gases

3.5.1 General

When tables, equations of state or charts giving the appropriate thermodynamic data are available, it is recommended that they be used to determine the specific compression work (see 3.5.2).

When such tables, equations of state or charts are not available and the limits of table 2 are exceeded, it is recommended that polytropic compression be adopted as the reference process and the specific compression work should be computed by the Schultz method of polytropic analysis (see 3.5.3 and [2]).

3.5.2 Method using tables or charts

If possible, the gas properties, especially for gas mixtures, are best determined from tables and equations of state since the diagrams are in general less accurate (see clauses A.1 to A.3).

The properties represented in diagrams and tables are today frequently compiled in the form of computer programs which can readily be included as subprograms in the calculation programs for the design of the compressor and test evaluation.

Specific information about the determination of these gas properties and changes of state cannot be listed here owing to the multiplicity of the processes and values used. The user of this International Standard is referred to the relevant literature.

3.5.3 Methods of polytropic analysis (Schultz method)

3.5.3.1 The formulae of the following method are derived from the method developed by Schultz (see [2]).

3.5.3.2 The determination of exponent n for the general case of real gases is carried out assuming that the efficiency remains constant throughout the process:

$$\eta_{pol} = v \frac{dp}{dh} = \frac{\int_1^2 (v\,dp)_{pol}}{h_2 - h_1}$$

3.5.3.3 This makes it possible to determine an average polytropic exponent with sufficient exactitude:

$$n = - \frac{1}{Y_M - m(1 + X_M)}$$

where

$$m = \frac{Z_M R}{c_{pM}} \left(\frac{1}{\eta_{pol}} + X_M \right) = \frac{\lg \left(\frac{T_2}{T_1} \right)}{\lg \left(\frac{p_2}{p_1} \right)}$$

$$\eta_{pol} = \frac{R Z_M}{m c_{pM} - R Z_M X_M}$$

with average values for the gas stream:

$$Z_M = \frac{Z_1 + Z_2}{2}$$

$$X_M = \frac{X_1 + X_2}{2}$$

$$c_{pM} = \frac{c_{p1} + c_{p2}}{2}$$

The above averages are a simplification valid for pressure ratios $\frac{p_2}{p_1} \leqslant 4$.

For higher pressure ratios, it is recommended that Schultz's method be followed, i.e. averages are formed with double weight given to the mid-point. The mid-point can be chosen at the square root of the pressure ratio and half the temperature rise.

For pressure ratios up to 4, the difference between the results of both methods is less than 0,2 % for specific polytropic compression work and 0,5 °C for discharge temperature.

3.5.3.4 Theoretically, the exact assessment of the specific polytropic compression work assumes the previous computation of factor ξ by use of the formula

$$\xi = \frac{h'_2 - h_1}{\frac{\kappa}{\kappa - 1} R \left(Z_2 T_2 - Z_1 T_1\right)}$$

where

h_1 is the enthalpy at the inlet;

h'_2 is the resulting enthalpy at the outlet if the process had been isentropic.

NOTE — Where specific values of the factors X, Y and Z are not available values may be obtained from the generalized curves given in annex D.

3.5.3.5 The specific compression work based on static conditions may thus be determined:

$$W_{m,\mathrm{pol}} = \int_1^2 (v\,\mathrm{d}p)_{\mathrm{pol}} = \frac{Z_{\mathrm{M}} R T_1}{m}\, \xi \left[\left(\frac{p_2}{p_1}\right)^m - 1\right]$$

NOTE — This equation is not strictly correct (see [2]) and should only be used in cases where the compressibility factor Z is substantially constant throughout the compression process.

The equation for $W_{\mathrm{m,pol}}$ given in figure D.9 can also be used.

3.5.3.6 For compression ratios $\frac{p_2}{p_1} \leqslant 4$ the correction factor ξ may be considered as equal to 1 and can therefore be neglected in the computation.

3.5.3.7 For similarity testing of compressors without intermediate cooling, a single-stage process should be adopted, determining an average polytropic exponent such as defined above.

3.5.3.8 For tests of compressors with intermediate cooling a multistage reference process should be adopted with polytropic exponents suited to each section included between two successive intercoolers.

3.5.3.9 Similarity calculations according to Schultz's method are shown schematically in figure D.7.

3.6 Definition of reference efficiency, power and losses

3.6.1 reference efficiency, η_{pol}, η_{s} or η_{T}: Ratio of the specific compression work to the total enthalpy rise in the machine (or section of the machine) within which there is no deliberate cooling and when based on static conditions as defined by the formula:

$$\eta = \frac{\int_1^2 v\,\mathrm{d}p}{h_2 - h_1} = \frac{W_m}{h_2 - h_1}$$

Unless otherwise agreed, the reference efficiency is based on total conditions and the formula becomes

$$\eta_{\mathrm{t}} = \frac{W_m + \frac{(c_2^2 - c_1^2)}{2}}{h_{\mathrm{t},2} - h_{\mathrm{t},1}}$$

Where there is deliberate cooling within the machine (or section of the machine) under consideration, this formula becomes

$$\eta_{\mathrm{t}} = \frac{W_m + \frac{(c_2^2 - c_1^2)}{2}}{h_{\mathrm{t},2} - h_{\mathrm{t},1} - Q_{1-2}}$$

where Q_{1-2} is the heat removed by the cooling within the machine (or section of the machine).

The above definition is complete only when the type of the reference process adopted is indicated by the corresponding subscript. Consequently reference efficiencies are given by the following formulae.

3.6.1.1 The polytropic efficiency

$$\eta_{\mathrm{pol}} = \frac{W_{m,\mathrm{pol}}}{h_2 - h_1 - Q_{1-2}}$$

$$\eta_{\mathrm{pol,t}} = \frac{W_{m,\mathrm{pol}} + \frac{(c_2^2 - c_1^2)}{2}}{h_{\mathrm{t},2} - h_{\mathrm{t},1} - Q_{1-2}}$$

See also 3.5.3.3.

3.6.1.2 The isentropic efficiency

$$\eta_{\mathrm{s}} = \frac{W_{m,\mathrm{s}}}{h_2 - h_1 - Q_{1-2}}$$

$$\eta_{\mathrm{s,t}} = \frac{W_{m,\mathrm{s}} + \frac{(c_2^2 - c_1^2)}{2}}{h_{\mathrm{t},2} - h_{\mathrm{t},1} - Q_{1-2}}$$

3.6.1.3 The isothermal efficiency

$$\eta_{\mathrm{T}} = \frac{W_{m,\mathrm{T}}}{h_2 - h_1 - Q_{1-2}}$$

$$\eta_{\mathrm{T,t}} = \frac{W_{m,\mathrm{T}} + \frac{(c_2^2 - c_1^2)}{2}}{h_{\mathrm{t},2} - h_{\mathrm{t},1} - Q_{1-2}}$$

3.6.2 reference power P_{pol}, P_{s} or P_{T}: Power absorbed by the gas during the reversible reference process excluding any losses. The reference power shall be defined with the subscript corresponding to the adopted reference process.

The formulae for the reference power are also different depending on the calculation system of the specific compression work.

Unless otherwise agreed the reference power is based on total conditions and is given by the following formulae.

3.6.2.1 The polytropic reference power

$$P_{\mathrm{pol,t}} = q_m \left(W_{m,\mathrm{pol}} + \frac{c_2^2 - c_1^2}{2} \right)$$

3.6.2.2 The isentropic reference power

$$P_{\mathrm{s,t}} = q_m \left(W_{m,\mathrm{s}} + \frac{c_2^2 - c_1^2}{2} \right)$$

3.6.2.3 The isothermal reference power

$$P_{\mathrm{T,t}} = q_m \left(W_{m,\mathrm{T}} + \frac{c_2^2 - c_1^2}{2} \right)$$

Where the local Mach number at the standard inlet and discharge points is less than 0,2 it is sufficiently accurate to calculate the reference power directly from total conditions using the following approximate formulae:

$$P_{\mathrm{pol,t}} \approx q_m W_{m,\mathrm{pol,t}} = q_m \int_1^2 (v\,\mathrm{d}p)_{\mathrm{t,pol}}$$

$$P_{\mathrm{s,t}} \approx q_m W_{m,\mathrm{s,t}} = q_m \int_1^2 (v\,\mathrm{d}p)_{\mathrm{t,s}}$$

$$P_{\mathrm{T,t}} \approx q_m W_{m,\mathrm{T,t}} = q_m \int_1^2 (v\,\mathrm{d}p)_{\mathrm{t,T}}$$

3.6.3 heat transmission losses, Q_α**:** Losses due to heat transmission from the area A_{Cs} of the compressor casing to the ambient atmosphere, expressed by the formula

$$Q_\alpha = \alpha A_{\mathrm{Cs}} (t_{\mathrm{M\,Cs}} - t_{\mathrm{a}})$$

For values of Q_α less than 0,02 P_{e}, an approximate value can be adopted for α, i.e.

$$\alpha = 14\ \mathrm{W/(m^2 \cdot K)}$$

3.6.4 power loss due to leakage, P_{L}**:** Losses due to leakage through external labyrinths; these can generally be calculated using the formula

$$P_{\mathrm{L}} = \Sigma\, q_{m,\mathrm{L}} \cdot \Delta h_{\mathrm{t_L}}$$

3.6.5 internal power, P_{in}**:** Power effectively absorbed by gas during the actual compression process. It is given by the formula

$$P_{\mathrm{in}} = \frac{P_{\mathrm{pol,t}}}{\eta_{\mathrm{pol,t}}} + Q_\alpha + P_{\mathrm{L}}$$

or

$$P_{\mathrm{in}} = \frac{P_{\mathrm{s,t}}}{\eta_{\mathrm{s,t}}} + Q_\alpha + P_{\mathrm{L}}$$

or

$$P_{\mathrm{in}} = \frac{P_{\mathrm{T,t}}}{\eta_{\mathrm{T,t}}} + Q_\alpha + P_{\mathrm{L}}$$

For compressors with intermediate cooling the sum of internal powers in each section between two successive intercoolers should be computed:

$$P_{\mathrm{in}} = P_{\mathrm{in_I}} + P_{\mathrm{in_{II}}} + \dots + P_{\mathrm{in}_i}$$

3.6.6 mechanical power losses, P_{f}**:** Losses due to friction in the bearings and sealing rings and in any transmission gear contractually included within the compressor.

3.6.7 effective compressor power, P_{e}**:** Power input at the coupling of the compressor or at the coupling of the transmission gear depending on the contract agreement. It is given by the formula

$$P_{\mathrm{e}} = P_{\mathrm{in}} + P_{\mathrm{f}}$$

3.6.8 power loss in driving machine, P_{Pr}**:** Power loss in the turbine or in any other driver of the compressor and the intermediate driving system.

3.6.9 total power of the unit, P_{un}**:** Power given by the formula

$$P_{\mathrm{un}} = P_{\mathrm{e}} + P_{\mathrm{Pr}}$$

3.6.10 internal efficiency, η_{in}**:** Ratio of the reference power defined in 3.6.2 to the internal power.

Its value depends on the type of adopted reference process. Internal efficiency is given by the formulae

$$\eta_{\mathrm{in,pol}} = \frac{P_{\mathrm{pol,t}}}{P_{\mathrm{in}}}$$

$$\eta_{in,s} = \frac{P_{s,t}}{P_{in}}$$

$$\eta_{in,T} = \frac{P_{T,t}}{P_{in}}$$

3.6.11 mechanical efficiency, η_f: Ratio of the internal power to the effective power at the coupling of the compressor, or at the coupling of the transmission gear, depending on the contract agreement. It is given by the formula

$$\eta_f = \frac{P_{in}}{P_e} = \frac{P_{in}}{P_{in} + P_f}$$

3.6.12 effective efficiency, η_e: Ratio of the reference power defined in 3.6.2 to the effective power at the coupling of the compressor, or at the coupling of the transmission gear, depending on the contract agreement. It is given by the formula

$$\eta_e = \frac{q_m \int_1^2 (v\,dp)_t}{P_e} = \eta_{in}\,\eta_f$$

Its value depends on the type of adopted reference process.

3.6.13 primary efficiency, η_{Pr}: Ratio of the effective power of the compressor to the power or energy input to the driver. It is given by the formula

$$\eta_{Pr} = \frac{P_e}{P_{un}} = \frac{P_e}{P_e + P_{Pr}}$$

3.6.14 overall efficiency of the unit, η_{un}: Efficiency of the unit which takes account of all energy losses of the unit, including compressor, transmission gear and driver. It is given by the formula

$$\eta_{un} = \eta_e\,\eta_{Pr}$$

3.7 Definition of tolerance, inaccuracy and uncertainty

As used in this International Standard, the terms "tolerance", "inaccuracy" and "uncertainty" have specific and clearly different meanings.

3.7.1 tolerance: Amount by which the value of a particular parameter or a quantity is permitted to deviate from a set value by the terms of the contract or other agreement.

3.7.2 inaccuracy: Extent by which the measured or computed value of a parameter or quantity deviates from the true value, resulting from the inevitable errors in measurement and computation.

3.7.3 uncertainty: Maximum likely magnitude of the inaccuracy of a particular parameter or quantity such that it can be said with at least 95 % confidence that the measured or computed value does not deviate from the true value by an amount greater than the stated uncertainty.

4 Symbols and subscripts

The symbols, subscripts and definitions used in this International Standard are in accordance with ISO 31 and ISO 1000, and are given in tables 3 and 4.

Equations used are dimensionally homogeneous.

To simplify use of this International Standard, conversion factors are given in annex C.

Table 3 — Symbols

Symbols	Quantities	Definitions and observations	Dimensions[1)]
A	Area		L^2
a	Sonic velocity	$a = \sqrt{\kappa ZRT}$	LT^{-1}
b	Outlet tip width		L
C_p	Molar specific heat at constant pressure	$C_p = Mc_p$	$ML^2T^{-2}\Theta^{-1}\,mol^{-1}$
C_V	Molar specific heat at constant volume	$C_V = Mc_V$	$ML^2T^{-2}\Theta^{-1}\,mol^{-1}$
c	Absolute velocity		LT^{-1}
c_p	Specific heat at constant pressure		$L^2T^{-2}\Theta^{-1}$
c_V	Specific heat at constant volume		$L^2T^{-2}\Theta^{-1}$
D	Reference diameter of the rotor		L
F	Torque		ML^2T^{-2}
G	Precision class		dimensionless
h	Specific enthalpy		L^2T^{-2}
h_t	Total specific enthalpy	$h_t = h + \left(\frac{c^2}{2}\right)$	L^2T^{-2}
M	Molar mass	Mass which corresponds to one mole	M
Ma	Mach number of the flow	$Ma = \frac{c}{a}$	dimensionless
Ma_t	Approximate Mach number of gas flow through area A	$Ma_t = \frac{q_m}{A\,p_t}\sqrt{\frac{ZRT_t}{\kappa}}$	dimensionless
Ma_u	Peripheral Mach number (arbitrary definition)	Refers in this International Standard to inlet conditions	dimensionless
m	Polytropic exponent in the $p-T$ diagram	$\frac{p^m}{T}$ = constant See also 3.5.3.3	dimensionless
m_i	Mass proportion of a gas component		dimensionless
N	Speed of rotation		T^{-1}
N_r	Ratio of reduced speeds	$N_r = \left(\frac{N}{\sqrt{RZ_1T_{t,1}}}\right)_{Te} \Big/ \left(\frac{N}{\sqrt{RZ_1T_{t,1}}}\right)_{Gu}$	dimensionless
n	Polytropic exponent in the $p-V$ diagram	pV^n = constant See also 3.4.1	dimensionless
P	Power		ML^2T^{-3}
p	Absolute static pressure	Force to be exerted on the unit area moving with the gas	$ML^{-1}T^{-2}$
p_a	Atmospheric pressure		$ML^{-1}T^{-2}$
p_d	Dynamic pressure	See 8.1.3	$ML^{-1}T^{-2}$
p_e	Effective (or gauge) pressure	$p_e = p - p_a$	$ML^{-1}T^{-2}$
p_{sat}	Saturation pressure	Saturation pressure at temperature of the vapour-gas mixture	$ML^{-1}T^{-2}$
p_t	Total pressure	$p_t = p + p_d$	$ML^{-1}T^{-2}$

Table 3 — Symbols *(continued)*

Symbols	Quantities	Definitions and observations	Dimensions[1)]
p_V	Partial vapour pressure		$ML^{-1}T^{-2}$
Q	Heat flow	Quantity of heat supplied or delivered per unit time	ML^2T^{-3}
Q_{co}	Corrected heat losses (equivalent)		ML^2T^{-3}
Q_{in}	Internal heat losses (equivalent)		ML^2T^{-3}
Q_{me}	Mechanical heat losses (equivalent)		ML^2T^{-3}
Q_α	Heat losses by thermal transmission from the surface		ML^2T^{-3}
q_m	Mass rate of flow		MT^{-1}
q_V	Volume rate of flow		L^3T^{-1}
R	Specific gas constant	$R = \dfrac{R_{mol}}{M}$	$L^2T^{-2}\Theta^{-1}$
Re_u	Peripheral Reynolds number (arbitrary)	$Re_u = \dfrac{ub}{\nu_{t,1}}$ Refers in this International Standard to total inlet conditions	dimensionless
R_{mol}	Universal gas constant	$R_{mol} = 8\,314$ J·kmol^{-1}·K^{-1}	$ML^2T^{-2}\Theta^{-1}mol^{-1}$
r_i	Volumetric proportion of a component		dimensionless
s	Specific entropy		$L^2T^{-2}\Theta^{-1}$
T	Absolute static temperature	Temperature on Kelvin scale	Θ
t	Usual static temperature	Temperature on Celsius scale	Θ
t_d, T_d	Dynamic temperature		Θ
t_{sat}, T_{sat}	Saturation temperature		Θ
t_t, T_t	Stagnation temperature (total)	$t_t = t + t_d$ $T_t = T + T_d$ See 8.1.4	Θ
u	Peripheral velocity	$u = \pi DN$ Peripheral velocity at reference diameter	LT^{-1}
V	Volume		L^3
V_r	Ratio of volume rate of flow ratios	$V_r = \dfrac{(q_{V,t,2}/q_{V,t,1})_{Te}}{(q_{V,t,2}/q_{V,t,1})_{Gu}}$	dimensionless
v	Specific volume	Volume per unit mass	$M^{-1}L^3$
W_m	Specific compression work	$W_m = \int v\,dp$	L^2T^{-2}
X	Isobaric deviation factor	See 3.2.4	dimensionless
x	Variable		as used
Y	Isothermal deviation factor	See 3.2.3	dimensionless
y	Molar proportion		dimensionless
Z	Compressibility factor	See 3.2.2	dimensionless
z	Number of stages considered	Indicates also number of stage groups separated by intercoolers	dimensionless
α	Heat transfer coefficient	Rate of heat flow per unit area of surface per unit temperature difference	$MT^{-3}\Theta^{-1}$
γ	Ratio of specific heats	$\gamma = \dfrac{c_p}{c_V}$	dimensionless

Table 3 — Symbols (*concluded*)

Symbols	Quantities	Definitions and observations	Dimensions[1)]
Γ	Work input coefficient	$\Gamma = \dfrac{\Delta h_t}{u^2}$	dimensionless
Δx	Absolute difference or variation of x		same as for x
$\dfrac{\Delta x}{x}$	Relative difference or variation		dimensionless
δ	Blade position	Position of adjustable guide vanes or blades	
ε_x	Absolute possible deviation of x		same as for x
$\dfrac{\varepsilon_x}{x}$	Relative deviation		dimensionless
ζ_i	Coefficient	i = 3, 4 or p (see clause 9)	dimensionless
η	Efficiency		dimensionless
κ	Isentropic exponent	$\kappa = -\dfrac{V}{p}\left(\dfrac{\partial p}{\partial V}\right)_S$	dimensionless
λ_{ik}	Coefficient	i = 1, 2, 3 ... (see annex B) k = 1, 2, 3 ...	dimensionless
μ	Dynamic viscosity		$ML^{-1}T^{-1}$
ν	Kinematic viscosity	$\nu = \dfrac{\mu}{\varrho}$	L^2T^{-1}
ξ	Correction factor	See 3.5.3.4, 3.5.3.5 and 3.5.3.6	dimensionless
ϱ	Density	Mass per unit volume	ML^{-3}
σ	Standard deviation	See 5.9.4	same as for x
τ_x	Relative measuring uncertainty or tolerance on x	See 3.7	dimensionless
Φ	Flow coefficient	$\Phi = \dfrac{q_{V,t}}{D^2 u}$	dimensionless
φ	Relative humidity	See A.4.2.1	dimensionless
χ	Humidity content	See A.4.2.1	dimensionless
Ψ	Reference process work coefficient	$\Psi_i = \dfrac{W_{m,i}}{u^2}$ where i = pol, s or T	dimensionless
ω	Acentric factor	See A.3.1	dimensionless

1) L = length; M = mass; T = time; Θ = temperature; mol = amount of substance.

Table 4 — Subscripts

Subscripts	Meaning	Observations
I, II, ..., j	Section I, Section II, ..., Section j	The Roman type figures relate to numbers of order of the compressor sections
$\mathrm{II_c}$	Cooled section	Cooled section when compressor is divided into uncooled section I and cooled section $\mathrm{II_c}$
1	Inlet	Relates to quantities measured at the standard inlet point. In combination with other subscripts denotes "inlet"
2	Discharge	Relates to quantities measured at the standard discharge point. In combination with other subscripts denotes "outlet"
a	Atmospheric	Characterizes atmospheric pressures and temperatures
adj	Additional	Additional uncertainty when inner tolerance limit is exceeded (see clause D.2)
Cd	Condensate	
Co	Converted	Relates to the quantities converted to specified conditions by similarity computation
Cr	Critical	Characterizes critical pressures and temperatures
Cs	Casing	Characterizes the quantities measured on the compressor casing
comb	Combined	When x_{comb} is combined by superposition of results of several stages
D	Rotor	
d	Dynamic	Characterizes dynamic pressures and temperatures
En	End	
Ex	Extreme	
e	Effective	Characterizes the power input at the coupling of the compressor
el	Electrical	
f	Friction	Characterizes the friction losses (mechanical losses)
fluc	Fluctuation	Additional uncertainty due to fluctuations of power input
G	Dry gas	Characterizes the quantities of dry gas
Gu	Guaranteed	Relates to the quantities specified in the contract
$\mathrm{IC_I}$, $\mathrm{IC_{II}}$, ..., IC_j	Intercooler I, II, ..., j	Relates to first, second, ..., jth intercooler
i	Component	Relates to component i of a gas mixture
in	Internal	
L	Leakage	
M	Arithmetic mean	Characterizes the arithmetical means of inlet and outlet quantities
m	Mixture	
max	Maximum	
min	Minimum	
mol	Molar	
oil	Oil	Characterizes lubricating (and sealing) oil (mechanical losses)
Pr	Primary	Characterizes the driver of the compressor
p	Isobaric	Characterizes an isobaric (constant pressure) process
pol	Polytropic	Characterizes a polytropic process
r	Reduced	Characterizes reduced pressures and temperatures
res	Resulting	When x results from combination of several variables with individual errors
s	Isentropic	Characterizes an isentropic process

Table 4 — Subscripts *(concluded)*

Subscripts	Meaning	Observations
sat	Saturation	
T	Isothermal	Characterizes an isothermal process
Te	Tested	Relates to the quantities measured during the test or fixed as test conditions
t	Stagnation (total)	
tol	Tolerated	
tot	Relative	When $\tau_{x\text{comb}}$ included additional uncertainty τ_{adj} and/or τ_{fluc}
u	Peripheral	Relates to the reference rotor diameter
un	Unit	Characterizes the unit comprising the compressor, the intermediate driving system and the prime mover
ut	Usable	
V	Vapour	
v	Isochoric	Characterizes a constant volume process
W	Water	Characterizes cooling water
x	Variable	

5 Equipment, methods and accuracy of measurements

5.1 General

5.1.1 This clause describes the instruments, equipment and methods of measurement to be used in testing compressors in accordance with this International Standard and describes the measurement accuracy.

Where no specific requirements for a particular measurement are included, the instruments and methods of measurement to be used are subject to agreement.

5.1.2 Where an International Standard is available relating to a particular measurement or type of instrument, any measurements used should be in accordance with such a standard.

5.1.3 Wherever possible, instruments with a defined and guaranteed "quality grade" (accuracy class) should be used. In this context an instrument can be said to be of quality grade "G" when the maximum error at any level in the recommended range does not exceed ± G % of the full-scale reading.

5.1.4 Instruments shall only be used in that part of their range recommended by the manufacturer or so defined in this International Standard.

5.1.5 While the instruments and measurement methods are generally to be recommended for the purpose, it is not intended to restrict the use of other equipment with the same or better accuracy.

5.1.6 Measurement stations for the determination of the condition of flowing fluids shall wherever possible be located in straight lengths of pipe where the flow is substantially uniform and steady. For the main measuring stations at the inlets to and discharges from the compressor, the arrangement shown in figure 1 is recommended.

5.2 Apparatus

5.2.1 Instruments for determining the temperature of the compressed fluid at the compressor inlets and discharges.

5.2.2 Manometers or pressure gauges for determining the pressure of the compressed fluid at appropriate points on the test stand.

5.2.3 Barometer to measure the atmospheric pressure.

5.2.4 Flow measuring device to determine the flow through the compressor.

5.2.5 Instruments to measure the power consumption of the compressor.

5.2.6 Equipment for sampling the working gas and to determine its composition.

5.2.7 A device to measure the speed of rotation.

5.2.8 Arrangements to detect the occurrence of surge (when applicable).

5.2.9 Devices for determining flow-rates and temperatures of secondary flows to enable energy balances to be established.

5.3 Measurement of temperature

5.3.1 Each temperature measuring device shall be in accordance with the appropriate national standards and shall be calibrated against an instrument certified by a recognized authority. Recommended instruments for measuring temperature are

a) liquid-in-glass thermometers;

b) thermocouples used with potentiometric-type instruments;

c) resistance thermometers.

5.3.2 The detecting element should be inserted directly into the gas stream or, where this is not feasible, thermo-wells of minimum thickness shall be used. Precautions shall be taken to minimize measurement errors due to heat conduction in stems or wells and radiation from or to parts at a temperature different from that of the gas stream.

5.3.3 For machines assembled for test with an open inlet, the inlet temperature shall be taken as the atmospheric temperature measured in a region of substantially zero velocity in the vicinity of the inlet flange.

5.3.4 The delivery temperature (or temperatures) and inlet temperature (or temperatures), when a piped inlet is used on test, shall be measured by several instruments inserted into the pipe or duct symmetrically set in one plane. The number of instruments and their precise location depends on the layout of the piping and the precision required with due regard to heat flow and radiation from the pipe surface. The fluid temperature shall be the average of the single measured values corrected to allow for the velocity recovery effect.

Except in the case of gases with a significant condensible phase the use of shielded stagnation-type probes with a very high recovery factor is recommended.

Where it can be shown that the velocity recovery effect is insignificant, it may be neglected. In no case should it be neglected if the dynamic head exceeds 0,5 % of the specific compression work. The velocity recovery factor to be used should be agreed on. In the absence of any more specific values the following may be used:

a) thermometers and thermocouples in wells : 0,65

b) bare thermocouples : 0,80

c) bare thermocouples with insulation shields : 0,97

NOTE — For more precise information see [3] and [4].

5.3.5 Each instrument should penetrate into the pipe or duct for approximately one-third of the pipe diameter.

5.3.6 Thermocouples shall have a welded hot junction and shall be calibrated together with their compensating leads over the anticipated range.

5.3.7 With liquid-in-glass thermometers, when the measured temperature differs from the ambient by more than 5 °C, an emergent stem correction shall be made according to the formula

$$t_{Te} = t + l\beta\,(t - t_M)$$

where

t_{Te} is the true temperature of the gas stream;

t is the actual thermometer reading;

t_M is the average temperature of the emergent fluid column;

l is the length of the emergent column expressed in degrees on the thermometer scale;

β is the apparent expansion coefficient of the thermometer fluid (for mercury-in-glass thermometers $\beta = 1/6\,300$).

5.3.8 When it is necessary to determine the heat extracted by intercoolers, lubricant or sealant flows or leakage flows, the accuracy of temperature and flow measurements shall permit determination of the heat extraction with due regard to the effect of such heat flows on the overall accuracy of the test.

5.4 Measurement of pressure

5.4.1 General

Each pressure measuring device shall comply with the appropriate national standards.

Each instrument shall be calibrated with the exception of vertical "U" tube manometers using a liquid of known density and dead-weight gauges.

Connecting pipes shall be leak-free, as short as possible, and of a sufficient diameter and so arranged as to avoid blockage by dirt or condensed liquid.

If the saturation temperature of the fluid is higher than ambient temperature, pressure instruments shall be situated below the tapping points and connecting pipes shall be kept full of condensate. Condensing vessels shall be installed near the tapping points to ensure constant level height.

Stations for the measurement of the pressure of flowing fluid streams shall be located in a length of straight pipe or duct of uniform cross-section in which the flow is essentially parallel to the pipe wall.

Pressure instruments shall be mounted in a position substantially free from vibration.

The effective length of the scale of pressure measuring instruments and the arrangement of the graduations shall facilitate accurate readings within ± 0,5 % of the pressure measurement.

5.4.2 Barometric pressure

Barometric pressure shall be determined by any means with a maximum error not exceeding 0,333 mbar[1]. A barometer shall be located in a stable place at the test site. Alternatively, the barometric pressure may be obtained from a local meteorological station, with a correction for any difference in altitude.

5.4.3 Static pressure

Static pressure shall be measured by a number of tappings (normally four) spaced around the pipe at the measurement station in one plane normal to the direction of flow.

The tap-holes shall be normal to and flush with the pipe wall and shall be free from burrs, countersinks or other irregularity which could disturb the flow.

The diameter of the tap holes shall be as small as possible consistent with minimizing the risk of blockage.

It shall be established that the pressure measured at any one tapping at the measurement station does not differ from the mean of all pressure measurements by more than 1 % of the absolute pressure.

Having established that this requirement is satisfied, all the tappings at one station may be connected into a common collecting ring, provided that the effective cross-sectional area of this collecting ring is not less than four times the cross-sectional area of any one tap-hole.

5.4.4 Total pressure

Total pressure shall normally be calculated from the static pressure and the calculated mean flow velocity head (see 8.1.3). Where the degree of non-uniformity of flow is significant, the characteristic parameters (e.g. total enthalpy) should be obtained from a mass-flow-weighted integration of the results of probe traverses across the measuring plane.

5.5 Measurement of flow

Fluid flow-rates shall be measured in accordance with ISO 5167-1.

When other methods of flow measurement are proposed it shall be demonstrated that they are of equal or better accuracy.

Particular care is necessary when the compressed fluid contains a condensible fraction. With most types of flow-meter it is essential that the fluid is homogeneous. In extreme cases it may be necessary to install a suitable separating device, and to measure separately the gaseous and liquid streams leaving it.

5.6 Measurement of rotational speed

When the test driver is a synchronous electric motor the rotational speed may be obtained from an accurate measurement of the supply frequency.

When the test driver is a slip-ring type of induction motor, the rotational speed may be obtained from accurate measurements of both the supply frequency and the slip frequency.

In all other cases the rotational speed shall be measured with a mechanical or electrical tachometer permanently driven from the shaft.

The preferred type is a digital pulse counter based on a crystal oscillator used in conjunction with a toothed wheel mounted on the compressor shaft.

5.7 Measurement of power

Where the performance is guaranteed in terms of the energy input to the driver, this shall be measured in accordance with the appropriate International Standards or national standards.

Where it is the power input to the compressor which is guaranteed, this shall be measured

— by performing an energy balance on the driver in accordance with the appropriate test codes for the particular type of machine,

— by measuring the torque using a cradled (swinging field) type of motor or a precision torque-meter,

or, when these methods are not possible,

— by establishing a total energy balance for the compressor, by measuring all the losses and adding them to the energy input to the compressed gas.

Torque-meters shall not be used for measurement below one-third of their rated torque. They shall be calibrated with the measuring element at the same temperature as used during the test. The calibration shall be carried out twice, once with continuously increasing load and once with continuously decreasing load, and the mean of the two sets of readings shall be used.

With both torque-meters and cradled electric motors it shall be shown that the hysteresis effect, i.e. the difference between the readings with increasing and decreasing load due to mechanical friction etc., does not exceed 0,5 % of the measured torque.

5.8 Determination of gas composition

The chemical composition of gas or gas-vapour mixtures shall be determined at regular intervals by recognized methods of analysis.

Particularly when the gas contains condensible fractions, care shall be taken to ensure that the sample analysed is truly representative of the gas being compressed. In some cases this may require that the condensible and non-condensible fractions are sampled separately as they leave a suitable separating device and that the rates of flow of the two fractions are also established separately.

1) 1 bar = 10^5 Pa

If the working fluid on test is air, its humidity shall be determined from wet- and dry-bulb temperatures or by other recognized methods such as dew-point, freezing or chemical absorption techniques.

5.9 Accuracy of measurement

5.9.1 Owing to the very nature of physical measurements, it is impossible to measure a physical quantity without error or, in fact, to determine the true error of any one particular measurement.

However, if the conditions of the measurement are sufficiently well known, it is possible to estimate or calculate a characteristic deviation of the measured value from the true value, such that it can be asserted with a certain degree of confidence that the true error is less than the said deviation.

The value of such a deviation (normally the 95 % confidence limits) constitutes a criterion of the accuracy of the particular measurement, and in this International Standard is called the uncertainty.

For a particular measurement the value of the uncertainty (95 % confidence limits) can be determined in one of the following ways.

5.9.2 Where the measurement has been made in accordance with an established International Standard or national standard which defines the accuracy of such a measurement, then the uncertainty confidence limits should be determined by reference to that standard (e.g. for flow measurement refer to ISO 5167-1).

5.9.3 Where the quality grade of the instrument has been defined and guaranteed by the manufacturer (see 5.1.3) the uncertainty for a particular measurement can be determined by reference to the quality grade. For example in the case of a pressure gauge, the uncertainty, τ_p, is given by

$$\tau_p = \pm \mathrm{G} \frac{p_{\mathrm{En}}}{p_{\mathrm{Te}}}$$

where

G is the quality grade;

p_{En} is the full-scale reading of the gauge;

p_{Te} is the actual test reading.

NOTE — Where a pressure gauge has a quality grade less than 0,2 %, use a value of 0,2 to allow for location errors.

5.9.4 If a sufficiently large number of readings of a particular variable are available, then the standard deviation of these readings can be calculated; it is accepted to assume that the uncertainty is equal to twice the standard deviation.

For example, if the values of a set of readings of one variable are

$$x_1, x_2, x_3, \ldots, x_i, \ldots, x_n$$

then the mean value x_{M} is given by

$$x_{\mathrm{M}} = \frac{1}{n} \sum_{i=1}^{n} x_i$$

and the standard deviation is given by

$$\sigma = \sqrt{\frac{1}{n-1} \sum_{i=1}^{n} (x_i - x_{\mathrm{M}})^2}$$

Hence the uncertainty can be taken as $\pm 2\sigma$.

5.9.5 In some cases the uncertainty (95 % confidence limits) can be estimated by considering the fundamental principles of the method of measurement.

For example, in the case of a simple liquid-column manometer, provided that the density of the liquid is known precisely, the accuracy depends on how precisely the levels can be determined. Without special equipment an accuracy of ± 1 mm should be easily achievable with the necessary confidence.

Hence the uncertainty, τ_H, is given by

$$\tau_H = \pm \frac{1}{H}$$

where H is the reading in millimetres.

NOTE — For differential heads in excess of 1 m assume $\tau_H = \pm 0{,}1$ %.

A further example is an electronic digital tachometer in which a precise crystal oscillator is used as the time base. In this case the uncertainty, τ_N, can be taken as

$$\tau_N = \pm \frac{S}{N}$$

where

S is the digital measuring step;

N is the speed of rotation displayed.

S and N shall be expressed in the same units.

5.9.6 Where the value of a particular quantity is derived from the measurement of two or more separate quantities, the accuracy of the result is dependent on the accuracy of the separate measurements. For example, when the shaft input power to the compressor is determined by measuring the electrical power input to the electric motor, having separately determined the motor efficiency, the uncertainty confidence limits of the shaft input power are given by

$$\tau_{P\mathrm{e}} = \pm \sqrt{\tau_{P\mathrm{el}}^2 + \tau_{\eta\mathrm{el}}^2}$$

where

$\tau_{P\mathrm{el}}$ is the uncertainty of the electrical power input;

$\tau_{\eta\mathrm{el}}$ is the uncertainty of the motor efficiency.

$\tau_{P\mathrm{el}}$ and $\tau_{\eta\mathrm{el}}$ shall both be expressed as a fraction of the measured value.

5.9.7 In the absence of sufficient information to use any of the above methods, table 5 provides a guide to the accuracy that can be expected from good quality commercially available instruments.

Table 5 — Guiding values for uncertainty of test measurements

Measurement	Uncertainty
Pressure measured with	
— liquid column	± 0,5 %
— dead-weight gauge	± 0,3 %
— Bourdon tube gauge	± 1,0 %
Temperature *)	
— suction temperature	± 0,5 °C
— discharge temperature	± 1,0 °C
— cooling water, oil, etc.	± 0,5 °C
Flow measured with standard pressure difference device (see ISO 5167-1)	± 1,5 %
Speed measured with mechanical tachometer	± 1,5 %
Power measured with	
— mechanical torque-meter	± 2 %
— cradled motor	± 1,5 %
— a.c. motor of known efficiency	± 1,5 %
— d.c. motor of known efficiency	± 1,5 %
Power determined from an energy balance for the compressor	± 1,5 % to ± 4 %

*) A guide to the accuracy achievable with a good quality liquid-in-glass thermometer is given in table 6.

Table 6 — Measuring uncertainty for calibrated liquid-in-glass thermometers

Values in degrees Celsius

Temperature range, θ	Measurement uncertainty for the following graduation intervals				
	0,1	0,2	0,5	1	2
$-50 < \theta \leqslant -5$	0,6	0,8	1,7	2	4
$-5 < \theta \leqslant 60$	0,3	0,4	1	1,4	2
$60 < \theta \leqslant 110$	0,5	0,6	1	2	3
$110 < \theta \leqslant 210$	—	1	2	3	4
$210 < \theta \leqslant 310$	—	—	3	4	6
$310 < \theta \leqslant 410$	—	—	—	5	8
$410 < \theta \leqslant 625$	—	—	—	6	12

5.10 Thermodynamic properties of fluids

Physical properties and thermodynamic data and the appropriate confidence limits shall be obtained from established International Standards. When such standards are not available it shall be agreed which data shall be used and the uncertainties which apply to such data.

In annexes A and B methods of calculating the physical properties of gas and gas-vapour mixtures are provided.

6 Preparation for the test

6.1 Items on which agreement shall be reached

6.1.1 Agreements required at the time of order

6.1.1.1 Agreement shall be reached as to the location of the test, i.e. at the manufacturer's works, on site, or elsewhere.

Particularly in the case of a site test, agreement shall be reached on the necessary safety requirements relating to the test and it shall be agreed who shall have the responsibility of ensuring compliance with these requirements.

6.1.1.2 It shall be agreed which type of test is to be performed in accordance with the following categories.

Category 1

The compressor may be tested under the conditions of the guarantee, i.e. with the specified gas under the specified intake conditions, with the guaranteed power input.

In this case, the test speed shall be as close as possible to the specified speed of the compressor.

Category 2

If the specified conditions cannot be reproduced, either because of the gas composition, or because the power available is insufficient, the compressor should be run at such a combination of speed and inlet conditions as to establish the similarity conditions specified in 7.3.

In this case, the driving unit and the lubricating and sealing devices may also be different from those on the site.

Category 2 tests can themselves be subdivided into two types:

Category 2a): Open-loop air tests

In this case, the manufacturer shall state whether or not the similarity conditions specified in 7.3 can be fulfilled.

In a case where the similarity conditions cannot be fulfilled, the customer and the manufacturer should agree on the tolerances to be applied to the test result.

Category 2b): Closed-loop tests with air or another gas

If the performance testing is carried out using a closed loop, the manufacturer is free to choose any suitable test gas and conditions, provided that the similarity requirements of this International Standard are satisfied.

6.1.1.3 Whether or not the pressure ratio or pressure rise is based on static or total conditions, the precise location at which the guaranteed inlet and discharge pressures apply shall be agreed. Unless there are good reasons otherwise, total conditions shall be assumed.

6.1.1.4 The basis for calculating the actual inlet volume flow shall be agreed. Unless otherwise agreed, the inlet volume flow shall be calculated from the measured usable mass flow and the inlet density determined from the total temperature and total pressure at the standard inlet point.

6.1.1.5 The gas composition, charts, reference tables or other data relating to the guaranteee conditions shall be agreed.

6.1.1.6 The criterion of acceptance of the test result, according to 9.3.2, shall be agreed.

6.1.1.7 In the case of a site test, a deadline for the test should be agreed upon.

6.1.1.8 A schedule for agreements to be made on the matters given in 6.1.2 should be established.

6.1.2 Agreements required before the test

6.1.2.1 In the case of a factory test, the manufacturer should supply the customer with

a) the indication of the gas if it is different from the gas handled according to the contract agreement;

b) the calculations for the determination of the similarity conditions;

c) a sketch of the installation and instrumentation.

Before testing commences, the installation should be placed at the customer's disposal, in order that the customer may check that it is in conformity with the requirements of this International Standard.

In the case of a site test, when the site layout plan is finished, this is submitted by the customer to the manufacturer in order that the latter may check that all provisions have been made to carry out testing in conformity with the requirements of this International Standard and may comment on these, if required.

Before testing commences, the installation should be placed at the manufacturer's disposal for examination of the test equipment. The test programme should be agreed upon by the customer and the manufacturer.

6.1.2.2 It shall be agreed that the calibrations of each instrument used in the test shall be according to the appropriate national code.

6.1.2.3 Usually the flow will be measured on either the inlet to or the delivery pipe from the compressor in accordance with the way it has been specified in the guarantee. Where this is not possible, however, due corrections shall be applied to compensate for gland leakage either into or out of the compressor, condensate drained from intercoolers, bleed flows, etc. It is often impossible to measure such quantities directly and in these circumstances appropriate corrections shall be agreed between the manufacturer and the purchaser. Particular care in this matter is required when a compressor is being tested under conditions far removed from those of the guarantee.

6.1.2.4 When a machine is to be tested under conditions significantly different from the guarantee conditions, the manufacturer shall propose for agreement how the mechanical losses shall be converted from test to guarantee conditions.

6.1.2.5 Where any intermediate form of power transmission, such as a gearbox, is interposed between the driving unit and the machine, the method by which its power consumption shall be taken into account shall be the subject of agreement.

When the power consumption is guaranteed in terms of the power supply to the driving unit, any variations in its performance due to the difference between test and guarantee conditions shall be allowed for, subject to agreement.

When the power consumption is guaranteed and the test has to be carried out at reduced power, errors will be introduced into the corrected power consumption because of those losses which are not strictly proportional to the total power. An agreed correction shall be made for these errors.

6.1.2.6 Where it is not feasible to test a machine under the conditions laid down in 7.3, special conditions of test and special corrections shall be agreed between purchaser and manufacturer.

6.2 Preparation of the machine

6.2.1 The compressor and associated equipment shall be in a condition that is comparable to the condition when new with respect to those aspects which can affect the performance.

6.2.2 If pipes or ducts are fitted for the purpose of by-passing any component, or if bleed-off is used for any service, any valves in such pipes or ducts shall be set so as to produce conditions specified in the guarantee.

6.2.3 All unused connections shall be blanked off.

6.2.4 Where lubricating oil, cooling water and seal-fluid consumption is significant, provision shall be made for measuring these.

6.2.5 The recommended grade of oil shall be used; otherwise corrections to the power and oil consumption shall be made.

6.3 Preparation of test equipment

6.3.1 Diagrammatic layout of test arrangement

Prior to the test a diagrammatic layout of the test arrangement shall be established showing the agreed position of all measuring points and containing the reference numbers and letters used in the test report.

6.3.2 Installation of test equipment

See clause 5.

Inlet and discharge pressures shall be measured at agreed points. A typical arrangement is shown in figure 1, taken from [1].

Where necessary, provision shall be made for determining the humidity, chemical composition, density or viscosity of the gas at the appropriate and agreed times during the test.

6.3.3 Calibration of instruments

Initial calibration, where possible, of the instruments shall be available prior to the test.

Recalibration, where possible, after the test shall be made for those instruments of primary importance which are liable to variation in their calibration as a result of use during the test. Any change in the instrument calibrations which will create a variation exceeding that of the class of the instrument may be a cause for rejecting the part of the test which it invalidates.

7 Tests

7.1 Preliminary tests

Preliminary tests may be run for the purpose of

a) determining whether the compressor and associated system is in a suitable condition to conduct an acceptance test;

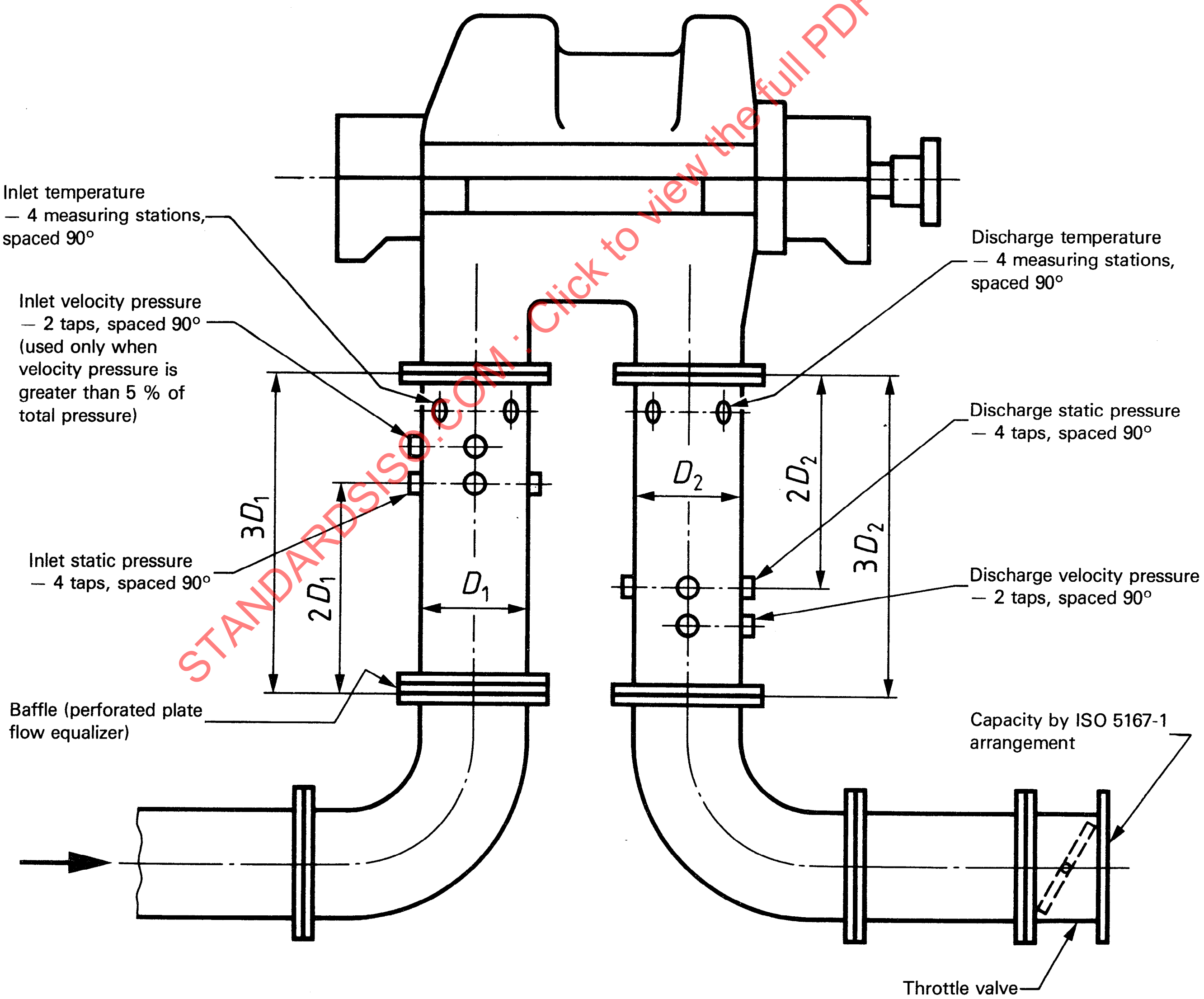


Figure 1 — Test arrangement

b) checking of instruments;

c) training of staff.

After a preliminary test has been made, this may, by agreement, be considered to be the acceptance test, provided that all requirements for acceptance have been met.

7.2 General rules for setting up the compressor for the acceptance test

7.2.1 During the test all measurements that have any bearing on the performance shall be made. In the following, the determination of the capacity and the power consumption of the compressor will be treated in detail.

7.2.2 The measurements shall be carried out by competent persons with measuring equipment according to clause 5.

7.2.3 The test conditions shall be as close as is reasonably possible to the guaranteed conditions: deviations from these shall not exceed the limits specified in 7.3.

7.2.4 During the test, the lubricant, and the adjustment of lubricating pumps, lubricators or other lubricating means shall comply with the operating instructions.

7.2.5 During the test adjustments other than those required to maintain the test conditions, and those required for normal operation as given in the instruction manual, shall not be made.

7.2.6 Before readings begin, the compressor shall be run long enough to ensure that steady state conditions are reached, so that no systematic changes occur in the instrument readings during the test.

Should, however, the conditions be such that changes cannot be avoided, a distinction may be made between systematic changes (drift) and fluctuations of high frequency.

In the case of a drift an adjustment to maintain the allowable deviations shall be made only if the operational conditions exceed the limits given in 7.3.

During the fluctuations, the number of readings shall be increased, but the results may still be accepted provided that the fluctuations are within the limits given in table 7.

7.2.7 For each guaranteed point a number of test points shall be obtained (not less than two) embracing the specified point. The range covered by these test points shall be not less than the range defined in figure 2, i.e. from the condition in which the reference process work coefficient on test is equal to that for the guaranteed point to the condition in which the flow coefficient on test is equal to that for the guaranteed point.

The reference process selected for this purpose depends on the type of compressor. Generally, for uncooled machines an isentropic or polytropic process should be selected and for cooled machines an isothermal process.

7.2.8 For each test point the compressor shall be run for a sufficient time and a sufficient number of sets of readings shall be taken to indicate that steady conditions have been reached, to enable meaningful averages to be taken and to ensure that the requirements of 7.4.1 have been met. Not less than three sets of readings shall be taken. If a computerized data acquisition system is used then one representative printout of the test results should be taken. As far as possible all readings in one set shall be taken simultaneously at a given signal.

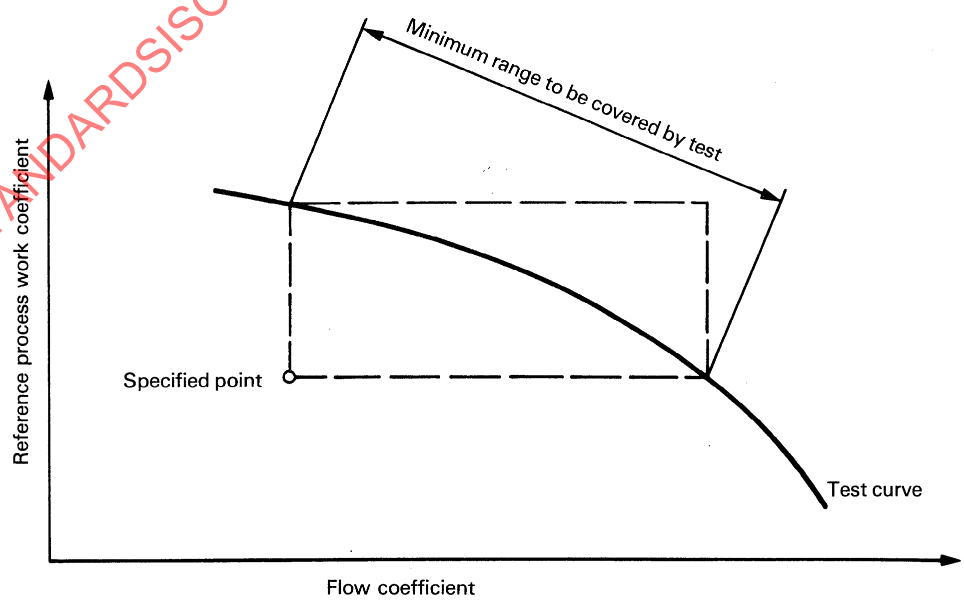


Figure 2 — Minimum range to be covered by the test

7.2.9 Where the usable stable range of the compressor is to be proved, the capacity at which surge occurs is determined by slowly raising the pressure ratio at constant speed until unstable or pulsating flow appears. A test point shall be set as close to the surge as possible. The precision of the determination shall be demonstrated by repeating the setting.

For variable speed and/or variable geometric compressors, surge points may be determined at different speeds or settings to cover the anticipated range of operation.

It shall be understood that the surge point determined in a shop test may not accurately represent the surge condition with other pipe configurations and other gas densities.

7.2.10 After the test the compressor plant and the measuring equipment shall be inspected. Should any faults be found that may affect the test results then a further test shall be run after these faults have been corrected.

7.3 Particular rules for setting up the compressor for the acceptance test

7.3.1 Guiding principle

Before the beginning of a test run, the settings of the compressor shall be fixed so as to achieve the best possible approximation to similarity of flow for test and specified conditions.

The conditions to be fulfilled in order to achieve similarity of flow are dealt with in clause D.1.

In practice it is often impossible to fulfill all the above conditions. Admissible limits for the deviation from similarity are considered in the following subclauses.

7.3.2 Allowed deviation from specified rate of flow

The magnitude of the permissible deviation of V_r from 1 and the additional uncertainty depend upon the characteristic curve being flat or steep.

A characteristic curve is steep if, at the point being considered,

$$\left| \frac{\mathrm{d}\left(\dfrac{p_2}{p_1}\right)_\mathrm{t}}{\mathrm{d}q_{V,\mathrm{t},1}} \right| \times \frac{q_{V,\mathrm{t},1}}{\left(\dfrac{p_2}{p_1}\right)_\mathrm{t}} > 1$$

If a flat characteristic curve is assumed according to a calculated performance diagram supplied by the manufacturer, this assumption shall be confirmed by the test result. Otherwise, the settings shall be corrected to satisfy the requirements of this International Standard for a steep characteristic curve.

The range of values of the reduced speeds corresponding to the permissible deviation of V_r and any additional resulting uncertainty can be determined in accordance with annex D.

7.3.3 Permissible variation from specified Reynolds number

It is necessary to check the deviation of the test Reynolds number from the specified conditions. The influence of this deviation on the performance of turbocompressors is taken into account by appropriate correction methods which should be used only in a certain range of application.

The range of application of the correction formula and the selection of a suitable test Reynolds number are influenced by two factors, i.e.

— the accuracy of the correction formula at different Reynolds numbers, and

— the reliability of tests carried out at reduced suction pressures or low driving power.

For centrifugal compressors, the well-proven method of the International Compressed Air and Allied Machinery Committee (ICAAMC) for correcting the performance for variations in Reynolds number shall be used (see annex E).

The limit of application of the ICAAMC formula is given in annex D, figure D.5, and annex E.

For axial compressors, the appropriate method of Reynolds number corrections depends on the blade characteristics used by the manufacturer. Therefore the method should be agreed by the manufacturer and the customer, as well as the range of its application.

7.3.4 Determination of the settings

The formulae required for the determination of the settings are evident from the flow charts given in clause D.5.

These charts, one for perfect and near-perfect gases (see figure D.6), the other for real gases (see figure D.7), will be followed according to the conditions prevailing. Each of them includes the following cases:

a) uncooled compressors;

b) compressors with intercooling.

Both have two possibilities for the isentropic exponent κ_{Te} for the test conditions, i.e.

a) same as for specified conditions, or

b) different.

Furthermore, three alternatives are considered for the determination of the isentropic exponent κ_{Te}:

a) constant (perfect and near-perfect gases only);

b) known from characteristic equations as a function of pressure and temperature;

c) calculated from conditions of state referred to the critical pressure and temperature of the test gas, following Schultz.

The input values for the above flow charts are to be found in D.5.1.

The relationships shown in the flow charts for determining the limits of the permissible setting up conditions for the compressor on test are shown in figure D.3 for $\kappa_{Te} = \kappa_{Gu}$, and in figure D.4 for $\kappa_{Te} \neq \kappa_{Gu}$.

These are referred to in clause 8.

The charts giving the generalized compressibility factors necessary to calculate Z, X and Y as a function of reduced pressure and temperature are to be found in clause A.3.

7.4 Evaluation of the readings

7.4.1 Before final calculations are undertaken, the recorded data shall be scrutinized for consistency of the operating conditions. The fluctuation of readings during one test shall not exceed the limits given in table 7.

Table 7 — Maximum allowable fluctuation of readings from average during one test run

Variable	Maximum allowable fluctuation of readings from average during one test run
Absolute inlet pressure	± 1 %
Absolute inlet temperature (of each section in the case of a cooled compressor)	± 1 %
Speed	± 0,5 %
Ratio $\left(\frac{p_2 - p_1}{p_1}\right)_t$	± 2,0 %
Ratio*) $\frac{\Delta p}{p}$	± 2,0 %

*) Where Δp is the pressure drop across a flow measuring device, p is the absolute pressure at this instrument.

NOTE — During the test the gas composition should not vary by an amount greater than that equivalent to a variation in either R or κ of ± 1 %.

If these limits are exceeded, the test is invalid except by special agreement, in which case such an agreement is to include consideration of the appropriate tolerance.

7.4.2 All accepted sets of readings from any test run shall be consecutive.

7.4.3 Sets of readings showing excessive fluctuation may be discarded but only at the beginning or the end of a test run. All readings in any set should be taken as nearly as possible simultaneously.

7.4.4 Test results shall be calculated from the arithmetic average values of the accepted readings.

8 Calculation and adjustment of test results

8.1 Computation of results

8.1.1 Purpose of evaluation

For the acceptance test only those values which are essential for the verification of the specified data are calculated and compiled.

From the average values of the readings, determined according to clauses 5 to 7, the results may be computed according to the following subclauses.

8.1.2 Mass rate of flow and inlet volumetric flow

The actual inlet capacity is obtained by converting the gas flow measured by the measuring device from the condition there to the total condition at the standard inlet point, due consideration being paid to any leakage flow and condensate flow, i.e.

$$q_{V,t,1} = \frac{q_{m,ut}}{\varrho_{t,1}} = \frac{q_{m,ut} Z_1 R T_{t,1}}{p_{t,1}}$$

where

$q_{m,ut}$ is the measured usable mass rate of flow;

$\varrho_{t,1}$ is the density based on total conditions;

R is the gas constant (see annex A);

$p_{t,1}$ is the total pressure;

$T_{t,1}$ is the total temperature;

Z_1 is the compressibility factor.

8.1.3 Dynamic pressure (velocity pressure)

In most cases the static pressure p can be directly measured at the standard inlet and discharge points.

If the velocity distribution across the area A is uniform, the dynamic pressure, p_d, can be derived from the continuity equation and the measured mass rate of flow:

$$\frac{p_t}{p} = 1 + \frac{p_d}{p} = \left\{1 - \frac{1}{(\kappa - 1)\, Ma_t^2}\left[(\kappa - 1)\, Ma_t^2 - \sqrt{1 + 2(\kappa - 1)\, Ma_t^2} + 1\right]\right\}^{\frac{\kappa}{1-\kappa}}$$

where

$$Ma_t = \frac{c}{a_t} \approx \frac{q_m}{Ap}\sqrt{\frac{ZRT_t}{\kappa}}$$

For low gas velocities an approximate simplified formula is

$$\frac{p_t}{p} \approx 1 + \frac{\kappa}{2} Ma_t^2$$

This approximation may be used provided that the value of $\frac{p_t}{p}$ so calculated does not exceed 1,05.

If the velocity distribution across the area A is significantly non-uniform, the dynamic pressure shall be found from a mass-flow-weighted integration over the area (see 5.4.4).

8.1.4 Total temperature

Normally the pressure ratio and temperature ratio are based on total conditions (see 6.1.1.3).

In most cases the total temperature may be measured at normal aspiration and exhalation points.

If static conditions are specified in a particular case, the static temperature shall be obtained from the following relation:

$$\frac{T_t}{T} = \frac{1}{1 - \frac{(\kappa - 1)}{2} Ma_t^2}$$

where

$$Ma_t = \frac{q_m}{Ap} \sqrt{\frac{ZRT_t}{\kappa}}$$

8.1.5 Reference power for compression

The reference power consumption shall be computed according to 3.6.2 after the appropriate choice of the reference process has been decided upon.

8.1.6 Actual power consumption

The actual power consumption can be computed in three ways:

a) from direct measurement of the input torque using a torque-meter or a dynamometer;

b) from an energy balance for the prime mover using the formula

$$P_{in} = P_{un} - P_{Pr} - P_f$$

c) from an energy balance for the compressor itself using the formulae given in 3.6.5.

Methods b) and c) should not be used unless the energy balance can be established with sufficient accuracy.

8.2 Conversion of test results to specified conditions

8.2.1 General

8.2.1.1 Purpose of conversion

The test results can only be compared directly with the guaranteed values when, during the acceptance tests, the measurements of the performance of the compressor are taken under precisely the specified conditions. If the operating conditions prevailing during the tests differ from those on which the guarantee is based, the test results shall be converted to the conditions specified for the guarantee. These converted results may then be compared with the specified figures.

8.2.1.2 Individual results to be converted

The particular results with which the conversion is concerned are

a) the actual inlet volume flow $q_{V,t,1}$,

b) the pressure ratio $(p_2/p_1)_t$, and

c) the power consumption at the coupling P_e.

The power consumption at the coupling is made up of the internal compression power P_{in} and the mechanical power loss P_f which will be converted separately.

8.2.1.3 Theoretical basis of conversion

The principle of the theory of similarity of flow upon which the following conversion rules are based, together with the permissible approximations to this theory are covered in annex D.

8.2.2 Conversion of mechanical losses

The sum of all mechanical losses may vary considerably with the particular test and specified conditions especially with regard to speed, power input, thrust and viscosity of the lubricant.

Mechanical losses occur mainly in the bearings, lubricant pumps, gearing belonging to the compressor, liquid seals, sliding-ring seals, and so on.

These losses are normally determined by measuring the rise in temperature of the oil, etc., or from the physical dimensions and test results, using recognized formulae. The various individual losses have to be converted separately, their sum giving the desired figure of mechanical losses P_f with the aid of which P_e can be calculated, as follows:

$$P_e = P_{in} + P_f$$

The formula for the conversion of the mechanical losses shall be agreed upon before the test (see 6.1.2.4).

8.2.3 Conversion of test results for uncooled compressors

8.2.3.1 In the case of all uncooled compressors, whether the isentropic exponents are equal or not and whether the speed or inlet temperature is adjustable or not, the test results can be converted to the specified conditions using the formulae given in figure D.8.

8.2.3.2 Where $\kappa_{Te} = \kappa_{Gu}$, the conversion is as follows.

If the speed or intake temperature is adjustable, the requirement of equal peripheral Mach numbers or that $(N/\sqrt{Z_1RT_{t,1}})_{Te}$ should be equal to $(N/\sqrt{Z_1RT_{t,1}})_{Gu}$ can easily be complied with within the inner tolerance limit [see figure D.3a)] and conversion may be carried out in accordance with the formulae given in figure D.8.

In this case no additional uncertainty is introduced.

If the speed or intake temperature cannot be adjusted with sufficient accuracy, or not at all, deviation of the ratio

$$(N/\sqrt{Z_1RT_{t,1}})_{Te} \Big/ (N/\sqrt{Z_1RT_{t,1}})_{Gu}$$

up to the outer tolerance limit is permissible.

In this case an additional uncertainty is introduced (according to instructions given in clause D.2).

Outside the outer tolerance limit there is no point in converting, unless special agreements have been reached.

8.2.3.3 Where $\kappa_{Te} \neq \kappa_{Gu}$, the conversion is as follows.

Regardless of whether the speed or intake temperature is adjustable or not, check with the aid of graphs (see figure D.4), whether the ratio

$$(N/\sqrt{Z_1RT_{t,1}})_{Te} \Big/ (N/\sqrt{Z_1RT_{t,1}})_{Gu}$$

is within the inner or outer tolerance limit. In the latter case an additional uncertainty will be introduced according to instructions given in clause D.2.

If this ratio falls outside the outer tolerance limit, the test shall be invalid except by special agreement. The conversion formulae are given in figure D.8.

8.2.3.4 If the test results are to be computed according to Schultz's method of polytropic analysis, the conversion formulae given in figure D.9 shall be used, after checking whether the test is valid (as outlined in 8.2.3.3).

8.2.4 Conversion of test results for cooled compressors

8.2.4.1 In the case of cooled compressors in which the ratios of the absolute temperatures after the intercoolers to the compressor inlet temperature are the same under test and specified conditions, $\kappa_{Te} = \kappa_{Gu}$, the conversion may be as follows.

If the speed and the temperature at the inlet to individual stages are adjustable by varying the ratio of flow of cooling water through the intercoolers, the condition of equal Mach numbers, or $(N/\sqrt{Z_1RT_{t,1}})_{Te} = (N/\sqrt{Z_1RT_{t,1}})_{Gu}$ should easily be met at each stage, to within the inner tolerance limit [see figure D.3a)], so that the conversion to specified conditions can be carried out using the formulae in figure D.10. No additional uncertainty is incurred.

The same can be applied for $\kappa_{Te} \neq \kappa_{Gu}$ provided that the ratio

$$(N/\sqrt{Z_1RT_{t,1}})_{Te} \Big/ (N/\sqrt{Z_1RT_{t,1}})_{Gu}$$

for any stage between two intercoolers does not exceed the limits (see figure D.4). If these ratios are within the outer but outside the inner tolerance limit, an additional uncertainty as given in clause D.2 is introduced. Outside the outer tolerance limit, the test is invalid except by special agreement.

If the speed is not adjustable, but the temperature at the inlet to individual stages can be varied by regulating the flow of cooling water to the intercoolers, such that the ratio of these absolute inlet temperatures to one another and to the temperature at the compressor inlet on test are the same as laid down in the specified conditions, the formulae in figure D.10 may also be used for conversion, provided that the ratio

$$(N/\sqrt{Z_1RT_{t,1}})_{Te} \Big/ (N/\sqrt{Z_1RT_{t,1}})_{Gu}$$

is within the outer tolerances [see figures D.3a) and D.3b)]. In this case an additional uncertainty as given by clause D.2 is introduced. Outside the outer tolerance limit, there is no point in converting.

8.2.4.2 In the case of cooled compressors in which the ratios of the absolute temperature after the intercoolers to one another are the same under test and specified conditions but the ratios of these temperatures to the compressor inlet temperature are not the same, and when $\kappa_{Te} = \kappa_{Gu}$, the conversion may be carried out as follows.

Whether the speed is adjustable or not, and if the ratio of the absolute temperatures after intercoolers during the test is the same as in the guarantee, but the ratio of these temperatures to the temperature at the compressor inlet is not, the results obtained for the uncooled part I and the cooled part II of the compressor may be converted separately to specified conditions according to the formulae given in figure D.3 provided that the ratios

$$(N/\sqrt{Z_1RT_{t,1}})_{Te} \Big/ (N/\sqrt{Z_1RT_{t,1}})_{Gu}$$

are within the outer tolerance limit given in figure D.8 for part I and for any stage of part II between intercoolers.

If $\kappa_{Te} \neq \kappa_{Gu}$ the above ratios shall lie within the respective tolerance limits given in figure D.4.

When the compressor is considered in two separate parts in this way, the discharge conditions for the uncooled part (part I) should be taken as the temperature and pressure before the first intercooler. The inlet conditions for the cooled part (part II) should be taken as the pressure before the first intercooler (i.e. the discharge pressure of part I) together with the temperature after the first intercooler.

Where it is not possible to obtain a representative measurement of the temperature after the first intercooler, this temperature may be estimated from the cooling water temperature and the design terminal temperature difference in the intercooler making due allowance for any difference in heat transfer in the intercooler between test and specified conditions.

The absolute temperatures after the intercoolers may be assumed to be in the same ratio to one another under test and specified conditions if the cooling water temperatures are in the same ratio and provided that the mass flow of gas through the intercoolers under test and specified conditions is the same within 5 %.

The measured power consumption of the compressor shall be separated into part I for the uncooled section and part II for the cooled section. In general, this separation will be possible by providing the proper measuring devices. If this is impossible, the division may be in relation to the stage heads according to the design data.

8.2.4.3 In the case of cooled compressors in which neither the ratios of the absolute temperatures after the intercoolers to one another nor the ratios of these temperatures to the compressor inlet temperature are the same under test and specified conditions, the conversion may be made as follows.

If the conditions of constant relationship between the gas temperatures after the intercoolers are not fulfilled, the values shall be converted separately for all cooled stages and the results combined.

If the test results are computed according to Schultz's method of polytropic analysis, the conversion formulae for a stage between intercoolers are given in figure D.9.

Where, in accordance with the above rules the compressor is considered in two or more parts, the converted results shall then be put together in order to compare them with the specified values. An additional uncertainty as described in 9.2.9 may be introduced.

8.2.5 Special note

If in the intercoolers of the compressor, some of the working fluid is lost due to condensation, either on test or under specified conditions, this loss shall be taken into account in working out the power consumption and the discharge flow. It must be borne in mind that the quantity actually lost in such an intercooler is generally smaller than the figure calculated thermodynamically (efficiency of separation less than 1). See A.4.2.8.

Moreover, during the process of compression it is possible for chemical combination to take place, which will change the gas parameters, especially density and temperature.

If a compressor has intermediate inputs and/or extractions, the corresponding incoming or outgoing volume flow and the temperatures of any incoming streams shall have the same relationships to the main flow and temperatures under test as under specified conditions. Otherwise each section of the compressor between intermediate inputs or extractions shall be considered separately.

If the compressor is run at different pressures on test and under specified conditions, the change in external leakage, if any, shall be allowed for in the effective volume flow and power consumption.

In the case of multiple or series-connected compressors it may be advantageous to consider each casing separately.

9 Comparison with guaranteed values and tolerances

9.1 General

9.1.1 The test results, converted to the specified operating conditions in accordance with clause 8, shall be compared with the guaranteed or specified performance.

9.1.2 The comparison shall include

a) comparison of the converted power consumption (specific power consumption, fuel consumption or efficiency depending on the terms of the guarantee) with the guaranteed power consumption (specific power consumption, fuel consumption or efficiency);

b) comparison of the converted capacity with the guaranteed capacity. This may be defined as either the pressure rise (or pressure ratio, or specific compression work) at the specified flow-rate or the flow-rate at the specified pressure rise (or pressure ratio, or specific compression work);

c) comparison of the acceptable operating range with the guaranteed range where this has been stated.

9.1.3 It is recommended that in most cases the comparison should be presented graphically (see 9.3).

9.1.4 Unless otherwise agreed, the comparison of the measured and corrected performance with the guaranteed performance shall be made

a) in the case of a compressor with a steep characteristic, by comparison at the guaranteed pressure rise (or pressure ratio or specific compression work);

b) in the case of a compressor with a flat characteristic, by comparison at the guaranteed actual inlet volume flow-rate.

NOTE — For the definition of the flat or steep characteristic see 7.3.2.

9.1.5 In making the comparison the following shall be taken into account:

a) measuring uncertainties (see clause 5);

b) errors due to the inaccuracy or uncertainty in the thermodynamic properties of the gases used;

c) errors due to the inaccuracy of the methods used to convert the test results to the guaranteed operating conditions;

d) errors due to non-steady conditions during the test;

e) any tolerance in the performance of the compressor permitted by the terms of the guarantee. (See 5.9.)

The errors listed above [9.1.5a) to d)] should be combined together to determine the total test uncertainty (limits of confidence). This and the manufacturing tolerance [9.1.5e)] should be clearly and separately stated or illustrated in the presentation of the comparison.

9.1.6 In the presentation of the comparison, a conclusion should be included stating whether the results of the test indicate that the compressor does or does not meet the terms of the guarantee.

9.2 Total test uncertainty (95 % confidence limit)

9.2.1 Measuring uncertainties

The confidence limits associated with the various measurements are determined using one of the methods described in 5.9.

9.2.2 Thermodynamic data uncertainties

The confidence limits associated with the thermodynamic properties of the gases used should be obtained from the same source as the values used, or otherwise agreed separately.

9.2.3 Conversion uncertainties

Where the pattern of flow on test cannot be made the same as the pattern of flow under the guaranteed conditions at all points throughout the compressor, then the methods of converting the test results defined in clause 8 are only approximate. If the deviations from true similarity of flow are significant, then additional errors are introduced by the conversion. The magnitude of any such additional errors can be determined from the charts given in annex D.

9.2.4 Errors due to non-steady test conditions

If the operating conditions during a test run are not steady, then additional errors may occur. The maximum fluctuations in test conditions which are normally allowable are specified in 7.4.1.

In some cases, however, if the test is conducted on site it may not be possible to maintain steady conditions within these limits. In such cases, by mutual agreement the test may be considered valid but an additional error is introduced, the likely maximum value of which (confidence limit) can be obtained from table 8.

Table 8 — Uncertainties due to the instability of test conditions

Values in per cent

Fluctuation in power input about the mean value	Uncertainty (confidence limit)
2	0
3	0,5
4	1,0
5	1,5

These additional uncertainties should be added in accordance with 9.2.9 to the value obtained from combining the measuring error, thermodynamic data error and conversion errors.

9.2.5 Total uncertainty in flow-rate

The total uncertainty in the volumetric flow-rate is obtained from

$$\tau_{\text{res},q_V} = \pm\sqrt{\tau^2_{q_m,\text{Te}} + \tau^2_{N,\text{Te}} + \tau^2_{p_1,\text{Te}} + \tau^2_{T_1,\text{Te}} + \tau^2_{Z_1,\text{Te}}}$$

where $\tau_{q_m,\text{Te}}$ is the uncertainty in the mass flow-rate according to ISO 5167-1.

9.2.6 Total uncertainty in pressure ratio

The total uncertainty in the pressure ratio is

$$\tau_{\text{res},p_2/p_1} = \pm\frac{1}{N_\text{r}^2}\left\{[\ln(p_2/p_1)]^2_\text{Te}\left(4\tau^2_{N,\text{Te}} + \tau^2_{T_1,\text{Te}} + \tau^2_{R,\text{Te}} + \tau^2_{Z_1,\text{Te}}\right) + \tau^2_{p_1,\text{Te}} + \tau^2_{p_2,\text{Te}}\right\}^{1/2}$$

where

$$N_\text{r}^2 = \frac{\left(\dfrac{N^2}{RZ_1T_{\text{t},1}}\right)_\text{Te}}{\left(\dfrac{N^2}{RZ_1T_{\text{t},1}}\right)_\text{Gu}}$$

9.2.7 Total uncertainty in power consumption, specific power consumption, specific fuel consumption or efficiency

The total uncertainty in power consumption (or fuel consumption) is obtained by calculating the square root of the sum of the squares of the appropriate individual uncertainties each multiplied by an appropriate factor.

The relevant individual uncertainties and the appropriate factors by which each should be multiplied for each type of compressor and each type of test are given in table 9.

Similarly the individual uncertainties and their appropriate multiplying factors to determine the total uncertainty in specific power consumption, specific fuel consumption or efficiency are given in table 10.

EXAMPLE

In the case of a cooled compressor in accordance with 8.2.4.2 in which the power consumption was obtained by measuring the shaft torque and speed, the total uncertainty in power consumption is given by

$$\tau_{\text{res},P_e} = \pm \left\{ \left(\frac{1}{1+\zeta_4} \tau_{P_f,\text{Te}} \right)^2 + \left(\frac{1}{1+1/\zeta_4} \tau_{F,\text{Te}} \right)^2 + \right.$$

$$+ \left[2\,\zeta_p \ln(p_2/p_1)_{t,I,\text{Co}}\, \tau_{N,\text{Te}} \right]^2 +$$

$$+ \left(\left[1 - \frac{1}{\ln(p_2/p_1)_t} \right] \tau_{p_1} \right)^2_{\text{Te}} +$$

$$+ \left[\zeta_p \ln(p_2/p_1)_{t,I,\text{Co}}\, \tau_{T_1,\text{Te}} \right]^2 +$$

$$+ \left[\zeta_p \ln(p_2/p_1)_{t,I,\text{Co}}\, \tau_{Z_1,\text{Te}} \right]^2 +$$

$$+ \left[\frac{1}{\ln(p_2/p_1)_t} \tau_{p_{t,2}} \right]^2_{\text{Te}} + \left(\zeta_p\, \tau_{T_{t,1},\text{IIc}} \right)^2_{\text{Te}} +$$

$$\left. + \left(\zeta_p\, \tau_{Z_1,\text{IIc}} \right)^2_{\text{Te}} + \left(\frac{z-2}{z-1} \tau_{T_{t,1,j}} \right)^2_{\text{Te}} \right\}^{1/2}$$

where

z is the number of stage groups separated by intercoolers;

j is the number of the stage group.

9.2.8 Total uncertainties in the performance of a multistage compressor obtained by combining the results of separate tests on the individual stages or stage groups

In this case the overall uncertainties can be computed as follows.

9.2.8.1 Volumetric flow

$$\tau_{\text{comb},q_V} = \frac{1}{z} (1 + 0{,}2\sqrt{z-1}) \sum_{j=1}^{z} \tau_{q_{Vj}}$$

9.2.8.2 Pressure ratio

$$\tau_{\text{comb},p_2/p_1} = (1 + 0{,}2\sqrt{z-1}) \sum_{j=1}^{z} \tau_{p_2/p_1,j} \frac{W_{m,j}}{\sum W_{m,j}}$$

9.2.8.3 Power consumption

$$\tau_{\text{comb},P} = (1 + 0{,}2\sqrt{z-1}) \sum_{j=1}^{z} \tau_{P,j} \frac{P_j}{\sum P_j}$$

The factor $(1 + 0{,}2\sqrt{z-1})$ is included in each of the above formulae to allow for the unavoidable inaccuracies involved in measuring each stage or stage group separately and combining the results.

9.2.9 Additional uncertainties

Where additional uncertainties due to the conversion from the test conditions to the guaranteed conditions (see 9.2.3) or to non-steady test conditions (see 9.2.4) are appropriate, these should be added to the total uncertainty statistically:

$$\tau_{\text{tot}} = \pm \sqrt{\tau^2_{\text{res}} + \tau^2_{\text{adj}} + \tau^2_{\text{fluc}}}$$

9.3 Graphical comparison

9.3.1 Normal case

Normally (see 7.2.7) at least two test points will be obtained close to and bridging the guarantee point or each of several guarantee points. The comparison is made as follows.

A chart is constructed in which the parameter to be compared (specific compression work, pressure ratio, pressure rise, power consumption, specific power consumption, efficiency, etc.) is plotted as the ordinate against flow-rate as the abscissa (see figure 3). Each test point is plotted on this chart and around each point is constructed an ellipse, the axes of which indicate the magnitude of the uncertainties in the relevant parameters. Upper and lower curves are drawn tangential to these ellipses. It can be stated, with at least 95 % confidence, that the true performance of the compressor lies within this test band.

The comparison is made by plotting the guaranteed operating point or points on the graph.

9.3.2 Criterion of acceptance

The criterion of acceptance of the test result shall be subject to specific agreement between manufacturer and customer taking account of the test uncertainty and the agreed manufacturing tolerance.

Where no such agreement has been reached the following criterion shall apply.

The guarantee shall be deemed to have been satisfied if the guarantee point lies within the test band or if the extent by which it lies outside the test band (measured either horizontally or vertically depending on how the guarantee was stated) does not exceed the agreed manufacturing tolerance.

9.3.3 Special cases

Where it is not possible to satisfy the requirement of 7.2.7, a single test point may be compared with a guaranteed performance curve (see figure 4). In this case, the guaranteed curve is drawn on the chart together with the agreed manufacturing tolerance giving a guaranteed band. The single test result is plotted together with the ellipse indicating the magnitude of the test uncertainty.

In this case, unless otherwise agreed, the guarantee shall be considered to have been met provided that some part of the test ellipse touches or cuts the guaranteed band.

9.4 Non-graphical comparison

In the exceptional case that only one test point is possible and this has to be compared with one guaranteed point, the measured power requirement which has been converted to guarantee conditions $P_{\mathrm{e,Co}}$ is further reduced to guarantee values using the formula

$$P_{\mathrm{e,Co,Gu}} = \frac{q_{V,\mathrm{t,1,Gu}} \cdot W_{m,\mathrm{Gu}}}{q_{V,\mathrm{t,1,Co}} \cdot W_{m,\mathrm{Co}}} \cdot P_{\mathrm{e,Co}}$$

assuming a constant efficiency, and is compared with the guaranteed power requirement $P_{\mathrm{e,Gu}}$. The applicability of this method is limited by the admissibility of the constant efficiency assumption.

10 Test report

After completion of the acceptance test, a test report shall be drawn up recording all the necessary information as to the procedure and results of the test. It shall contain the following items:

a) data and place of the test and the names of the supervisor and other participants;

b) technical data as follows

— compressor: owner, site and purpose of installation, manufacturer, type and serial number, year of manufacture, a short technical description giving operational data, auxiliaries and their drive, and any other special features (intercooling and lubricating system, etc.),

— driving unit: generally the same items as for the compressor, but in particular those which are essential for establishing the specified performance;

c) conditions and scope of the guarantees according to the contract;

d) programme of the procedure and diagram of the test arrangement indicating location of measuring points, type of instruments used and their calibration records;

e) a record of the test run together with a table of the average values of the important readings and the time they were taken; if possible, a record of the maximum and minimum readings; copies of the log sheets and of any readouts from an automatic recorder, as well as duplicates of the gas analysis, etc.;

f) an indication of any unscheduled occurrences which were noted during the test;

g) the formulae used for the calculation of the results, with due regard to the propagation of the mean uncertainties as they influence the final results;

h) a statement of the method used for converting the test results to specified conditions with reference to the tables and charts used (8.2); a clear definition of the reference process chosen;

i) a comparison of the actual performance with the guaranteed values or data; a statement of whether the contract values have been met or not.

The test report shall be signed by representatives of the manufacturer and customer. The original log sheets shall remain in the custody of the engineer in charge of the acceptance test.

Table 9 — Individual uncertainties and appropriate multiplying factors[1)] to be used to calculate the total uncertainty in power or fuel consumption[2)]

Individual uncertainty	Uncooled compressor (see 8.2.3) Method of power measurement[3)] 1	2	3	Cooled compressor (see 8.2.4.1) Method of power measurement[3)] 1	2	3	Cooled compressor (see 8.2.4.2) Method of power measurement[3)] 1	2	3
$\tau_{P_\mathrm{e},\mathrm{Te}}$	0	$\frac{1}{1+1/\zeta_4}$	0	0	$\frac{1}{1+1/\zeta_4}$	0	0	$\frac{1}{1+1/\zeta_4}$	0
$\tau_{P_\mathrm{in},\mathrm{Te}}$	$\frac{P_\mathrm{in,Co}}{P_\mathrm{e,Co}}$	0	0	$\frac{P_\mathrm{in,Co}}{P_\mathrm{e,Co}}$	0	0	$\frac{P_\mathrm{in,Co}}{P_\mathrm{e,Co}}$	0	0
$\tau_{P_\mathrm{f},\mathrm{Te}}$	$\frac{P_\mathrm{f,Co}}{P_\mathrm{e,Co}}$	$\frac{1}{1+\zeta_4}$	$\frac{1}{1+\zeta_4}$	$\frac{P_\mathrm{f,Co}}{P_\mathrm{e,Co}}$	$\frac{1}{1+\zeta_4}$	$\frac{1}{1+\zeta_4}$	$\frac{P_\mathrm{f,Co}}{P_\mathrm{e,Co}}$	$\frac{1}{1+\zeta_4}$	$\frac{1}{1+\zeta_4}$
$\tau_{q_V,\mathrm{Te}}$	0	0	0	0	0	0	0	0	0
$\tau_{F,\mathrm{Te}}$	0	0	$\frac{1}{1+1/\zeta_4}$	0	0	$\frac{1}{1+1/\zeta_4}$	0	0	$\frac{1}{1+1/\zeta_4}$
$\tau_{N,\mathrm{Te}}$	1	1	0	1	1	0	$1+2\zeta_p\ln\left(\frac{p_2}{p_1}\right)_\mathrm{I,Co}$		$2\zeta_p\ln\left(\frac{p_2}{p_1}\right)_\mathrm{I,Co}$
$\tau_{p_1,\mathrm{Te}}$	$1-\frac{1}{\ln(p_2/p_1)_\mathrm{Te}}$								
$\tau_{T_1,\mathrm{Te}}$	0	0	0	0	0	0	$\zeta_p\ln\left(\frac{p_2}{p_1}\right)_\mathrm{I,Co}$		
$\tau_{Z_1,\mathrm{Te}}$	0	0	0	0	0	0	$\zeta_p\ln\left(\frac{p_2}{p_1}\right)_\mathrm{I,Co}$		
$\tau_{p_2,\mathrm{Te}}$ $\tau_{R,\mathrm{Te}}$	$\frac{1}{\ln(p_2/p_1)_\mathrm{Te}}$								
$\tau_{\kappa,\mathrm{Te}}$	ζ_3	ζ_3	ζ_3	0	0	0	0	0	0
$\tau_{T_1,\mathrm{II},\mathrm{Te}}$	0	0	0	0	0	0	ζ_p	ζ_p	ζ_p
$\tau_{Z_1,\mathrm{II},\mathrm{Te}}$	0	0	0	0	0	0	ζ_p	ζ_p	ζ_p
$\tau_{T_1,j,\mathrm{Te}}$	0	0	0	$\frac{z-1}{z}$	$\frac{z-1}{z}$	$\frac{z-1}{z}$	$\frac{z-2}{z-1}$	$\frac{z-2}{z-1}$	$\frac{z-2}{z-1}$

1) Correction factors are given by the formulae

$$\zeta_3 = \frac{1}{(\kappa-1)_\mathrm{Te}} - \frac{1}{\kappa_\mathrm{Te}} \frac{\ln(p_2/p_1)_\mathrm{Te}}{1-(p_2/p_1)_\mathrm{Te}^{(1-\kappa_\mathrm{Te})/\kappa_\mathrm{Te}}}$$

$$\zeta_4 = \frac{P_\mathrm{e,Te}}{P_\mathrm{f,Co}\frac{P_\mathrm{in,Te}}{P_\mathrm{in,Co}} - P_\mathrm{f,Te}}$$

$$\zeta_p = \frac{P_\mathrm{in,II,Co}}{P_\mathrm{in,Co}}$$

2) z = number of stage groups separated by intermediate coolers.

3) Methods of power measurement:

Method 1: by measuring the internal power P_in and the mechanical losses P_f.

Method 2: by measurement at the drive unit.

Method 3: by measuring the torque F and the speed N.

Table 10 — Individual uncertainties and appropriate multiplying factors[1] to be used to calculate the total uncertainty in specific power consumption, specific fuel consumption or efficiency[2]

Individual uncertainty	Uncooled compressor (see 8.2.3)			Cooled compressor (see 8.2.4.1)			Cooled compressor (see 8.2.4.2)		
	Method of power measurement[3]			Method of power measurement[3]			Method of power measurement[3]		
	1	2	3	1	2	3	1	2	3
$\tau_{P_{e},Te}$	0	$\frac{1}{1+1/\zeta_4}$	0	0	$\frac{1}{1+1/\zeta_4}$	0	0	$\frac{1}{1+1/\zeta_4}$	0
$\tau_{P_{in},Te}$	$\frac{P_{in,Co}}{P_{e,Co}}$	0	0	$\frac{P_{in,Co}}{P_{e,Co}}$	0	0	$\frac{P_{in,Co}}{P_{e,Co}}$	0	0
$\tau_{P_{f},Te}$	$\frac{P_{f,Co}}{P_{e,Co}}$	$\frac{1}{1+\zeta_4}$	$\frac{1}{1+\zeta_4}$	$\frac{P_{f,Co}}{P_{e,Co}}$	$\frac{1}{1+\zeta_4}$	$\frac{1}{1+\zeta_4}$	$\frac{P_{f,Co}}{P_{e,Co}}$	$\frac{1}{1+\zeta_4}$	$\frac{1}{1+\zeta_4}$
$\tau_{F,Te}$	0	0	$\frac{1}{1+1/\zeta_4}$	0	0	$\frac{1}{1+1/\zeta_4}$	0	0	$\frac{1}{1+1/\zeta_4}$
$\tau_{q_V,Te}$	1	1	1	1	1	1	1	1	1
$\tau_{N,Te}$	0	0	1	0	0	1	$2\zeta_p \ln\left(\frac{p_2}{p_1}\right)_{I,Co}$		$2\zeta_p \ln\left(\frac{p_2}{p_1}\right)_{I,Co}$
$\tau_{p_1,Te}$	$1 - \frac{1}{\ln (p_2/p_1)_{Te}}$								
$\tau_{T_1,Te}$ $\tau_{R,Te}$	1	1	1	1	1	1	$1 + \zeta_p \ln\left(\frac{p_2}{p_1}\right)_{I,Co}$		
$\tau_{Z_1,Te}$	0	0	0	0	0	0	$1 + \zeta_p \ln\left(\frac{p_2}{p_1}\right)_{I,Co}$		
$\tau_{p_2,Te}$	$\frac{1}{\ln (p_2/p_1)_{Te}}$								
$\tau_{\kappa,Te}$	ζ_3	ζ_3	ζ_3	0	0	0	0	0	0
$\tau_{T_1,II,Te}$	0	0	0	0	0	0	ζ_p	ζ_p	ζ_p
$\tau_{Z_1,II,Te}$	0	0	0	0	0	0	ζ_p	ζ_p	ζ_p
$\tau_{T_1,j,Te}$	0	0	0	$\frac{z-1}{z}$	$\frac{z-1}{z}$	$\frac{z-1}{z}$	$\frac{z-2}{z-1}$	$\frac{z-2}{z-1}$	$\frac{z-2}{z-1}$

1) Correction factors are given by the formulae

$$\zeta_3 = \frac{1}{(\kappa-1)_{Te}} - \frac{1}{\kappa_{Te}} \frac{\ln (p_2/p_1)_{Te}}{1-(p_2/p_1)_{Te}^{(1-\kappa_{Te})/\kappa_{Te}}}$$

$$\zeta_4 = \frac{P_{e,Te}}{P_{f,Co}\dfrac{P_{in,Te}}{P_{in,Co}} - P_{f,Te}}$$

$$\zeta_p = \frac{P_{in,II,Co}}{P_{in,Co}}$$

2) z = number of stage groups separated by intermediate coolers.

3) Methods of power measurement:

Method 1: by measuring the internal power P_{in} and the mechanical losses P_f.

Method 2: by measurement at the drive unit.

Method 3: by measuring the torque F and the speed N.

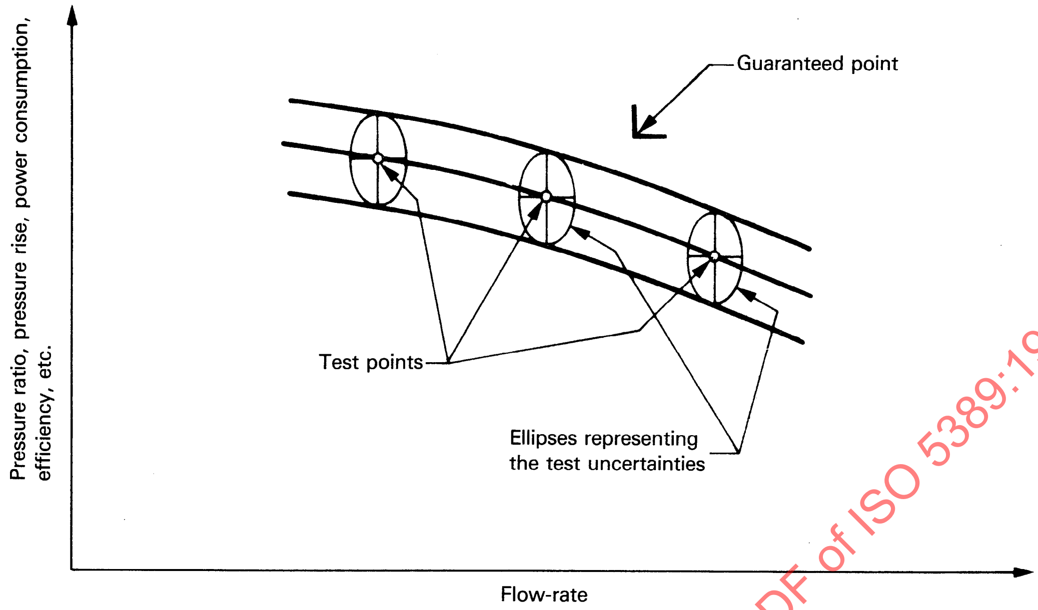


Figure 3 — Comparison of a single guaranteed point with a test curve

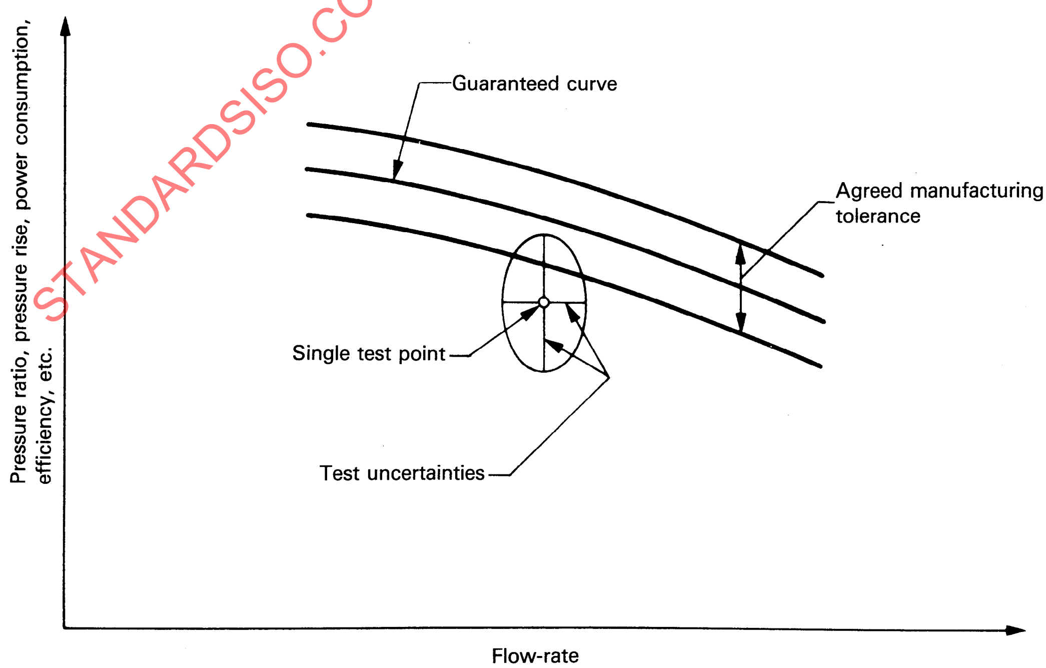


Figure 4 — Comparison of a single test point with a guaranteed curve

Annex A
(normative)

Properties of gases and gas mixtures — Recommendations relating to the compressibility factor and to derivative compressibility factors

A.1 General recommendations

A.1.1 The following general recommendations relating to thermodynamic data for gases and gas mixtures shall be used in connection with this International Standard.

These recommendations should not be considered to be exclusive, but to represent the most up-to-date information. More data are becoming available continually and in any specific case the user of this International Standard should refer to the latest published information.

A.1.2 Where suitable coefficients are available to be used in conjunction with an established equation of state (e.g. B.W.R.S., see [5]) for the particular gas or gas mixture in question, then these shall be used.

A.1.3 In the case of gas mixtures where suitable coefficients for use with an equation of state are available for the separate components of the mixture but not for the actual mixture in question, then the coefficients for the mixture can be obtained from the coefficients for the components by using the mixing rules appropriate to the equation of state being used. Suitable mixing rules are presented in [6].

A.1.4 When suitable coefficients are available neither for the particular gas or gas mixture in question nor, in the case of a mixture, for the component gases, then the gas properties may be obtained by using generalized compressibility functions based on the concept of reduced temperature and reduced pressure.

It is recommended that the method employing the concept of the acentric factor (see [7]) be used rather than the simpler two-parameters charts given in [8].

Tables and charts, from which the compressibility factors $Z^{(0)}$ and $Z^{(1)}$ for use with the method can be obtained, are presented in [9][1]. The isothermal derivative factors $Z_T^{(0)}$ and $Z_T^{(1)}$ and the isobaric derivative factors $Z_p^{(0)}$ and $Z_p^{(1)}$ can be obtained from the method given in [10], chapter 26.

Where the conditions of the gas in question are close to the critical conditions (i.e. within the shaded area shown in the charts for $Z^{(0)}$ published in [9]), it is essential to use the method which includes the acentric factor, as the simpler method specified in [8] is subject to particular inaccuracy in this area.

A.1.5 In the case of gas mixtures, except in the case of mixtures containing polar gases and particularly

— water vapour,

— ammonia,

— hydrogen sulfide, and

— halogenated hydrocarbon refrigerant mixed with other gases of a very different molecular structure,

simple linear mixing rules may be used to determine the pseudo-critical conditions of the mixture, i.e.

$$\bar{p}_{Cr} = \Sigma r_i p_{Cr}$$

and

$$\bar{T}_{Cr} = \Sigma r_i T_{Cr}$$

$$\omega = \Sigma r_i \omega_i$$

provided that the following conditions are satisfied:

$$\frac{T_{Cr\ max}}{T_{Cr\ min}} < 2$$

and either

$$\frac{p_{Cr\ max}}{p_{Cr\ min}} \text{ or } \frac{(v_{Cr}M)_{max}}{(v_{Cr}M)_{min}}$$

is close to unity, where $T_{Cr\ max}$, $T_{Cr\ min}$, $p_{Cr\ max}$, $p_{Cr\ min}$, $v_{Cr\ max}$ and $v_{Cr\ min}$ are respectively the maximum and minimum values of the critical properties of the components of the mixtures.

1) Other tables based on the method of calculation of Lee and Kesler have been published in [6]. The deviation-function table of Lee and Kesler (see [6]) differs somewhat from those of Pitzer (see [7]) and Curl (see [6]), but it seems according to R.C. Reid and T.K. Sherwood [6] that extensive testing indicates that this new table is the more accurate.

A.2 Specific recommendations for the sources of thermodynamic data and suitable equations of state and the coefficients to be used therewith for some of the more common gases

These recommendations should not be considered to be exclusive nor necessarily to be the best data available. They are mainly based on the results of a survey among compressor manufacturers and users.

More data are becoming available continually and the user of this International Standard is recommended to refer to the latest published literature.

A.2.1 Air

The data currently in use are those given in [11] and [12].

These are believed to be adequate for most compressor purposes. However, later and probably more accurate data, particularly at extreme temperatures and pressures, are available (see [13], [14] and [15]).

A.2.2 Refrigerants (halogenated hydrocarbons)

Data are available in [16], [17] [18], [19], [20] and [21].

A.2.3 Ammonia

The data in use are those published in [16]. However, these data are based on experiments made 50 years ago and published then (see [22]); they are inaccurate at high pressures, close to saturation conditions, and near the critical point.

More data are available in [12].

A new equation of state taking account of recent work, particularly that by Frank and Baehr, has been developed by L. Haar and is published in [23] to replace [22].

A.2.4 Pure hydrocarbons

The data in use are those published by Edmister [24] and Starling [5].

For saturated hydrocarbons the Edmister data, or other data based on [25], are considered to be satisfactory for most compressor purposes. However, new data are currently being produced by the National Bureau of Standards (see [26]).

For unsaturated hydrocarbons the situation is more confused and many data in common use are known to be inaccurate by reference to more recent experiments.

Specifically for ethylene and propylene, more accurate data are available in [5], [27] [28] and [29].

Where it is desired to use an equation of state, the equations given by Benedict, Webb and Rubin [30] and by Redlich and Kwong [31] or the slightly less accurate but simpler Beattie — Bridgeman [32] equation (see [24] and [10]) may be used provided that the compression process under consideration does not extend into the mixed-phase region.

Where saturated or mixed-phase conditions are involved, the use of one of the "extended B.W.R.S." equations is recommended (see [5]). However, a great many experiments are required to determine the values of the many constants used in these equations.

The values derived from the selected equation should be compared with the data recommended above over the field of interest.

A.2.5 Carbon dioxide

See [33].

A.2.6 Helium

See [34].

A.3 Compressibility factors of hydrocarbon gases and isentropic coefficient

A.3.1 Compressibility factor of pure hydrocarbon gases

The following equation can be used to predict the compressibility factor, Z, of pure hydrocarbon gases:

$$Z = Z^{(0)} + \omega Z^{(1)}$$

where

$Z^{(0)}$ is the compressibility factor for the simple fluid, which is tabulated as a function of T_r and p_r in table A.1;

$Z^{(1)}$ is the correction term for molecular acentricity, which is tabulated as a function of T_r and p_r in table A.2;

(see also [6], tables 3.1 and 3.2)

ω is the acentric factor, which is defined as

$$\omega = -(\log p_{r,\,sat} + 1)$$

at

$$T_{r,\,sat} = \frac{T_{sat}}{T_{Cr}} = 0{,}7$$

or, more conveniently,

$$\omega = \frac{3}{7}\left[\frac{\lg p_{Cr}}{(T_{Cr}/T_{sat}) - 1}\right] - 1$$

where

p_{Cr} is the critical pressure in atmospheres (1 atm = 760 mmHg = 1,013 bar);

T_{Cr} is the critical temperature, in kelvins;

T_{sat} is the boiling temperature, in kelvins;

p_{sat} is the saturation pressure at a temperature T_{sat} = 0,7 × T_{Cr};

$$p_{r,\ sat} = \frac{p_{sat}}{p_{Cr}}$$

A.3.2 Derivative compressibility factors

(see figures A.1 and A.2)

The derivative compressibility factors, Z_T and Z_p, are given by the equations

$$Z_T = Z_T^{(0)} + \omega\, Z_T^{(1)}$$

and

$$Z_p = Z_p^{(0)} + \omega\, Z_p^{(1)}$$

where the values of $Z_T^{(0)}$, $Z_T^{(1)}$, $Z_p^{(0)}$ and $Z_p^{(1)}$ are given in figures A.1 and A.2. (See also [10].)

The correlation between the Edmister derivative compressibility factors and the Schultz coefficients X and Y is given by the formulae

$$X = \frac{Z_T}{Z} - 1$$

$$Y = \frac{Z_p}{Z}$$

A.3.3 Isentropic exponent

The isentropic exponent κ of a real gas differs from the value c_p/c_V which it assumes for an ideal gas. The calculation of the specific heat capacity and of the isentropic exponent for a real gas shall take into account the effect of pressure.

The definition of the isentropic exponent for a real gas is given in table 3.

Table A.1 — Values of compressibility factor $Z^{(0)}$

T_r	$Z^{(0)}$						
	p_r = 0,010	p_r = 0,050	p_r = 0,100	p_r = 0,200	p_r = 0,400	p_r = 0,600	p_r = 0,800
0,30	0,002 9	0,014 5	0,029 0	0,057 9	0,115 8	0,173 7	0,231 5
0,35	0,002 6	0,013 0	0,026 1	0,052 2	0,104 3	0,156 4	0,208 4
0,40	0,002 4	0,011 9	0,023 9	0,047 7	0,095 3	0,142 9	0,190 4
0,45	0,002 2	0,011 0	0,022 1	0,044 2	0,088 2	0,132 2	0,176 2
0,50	0,002 1	0,010 3	0,020 7	0,041 3	0,082 5	0,123 6	0,164 7
0,55	0,980 4	0,009 8	0,019 5	0,039 0	0,077 8	0,116 6	0,155 3
0,60	0,984 9	0,009 3	0,018 6	0,037 1	0,074 1	0,110 9	0,147 6
0,65	0,988 1	0,937 7	0,017 8	0,035 6	0,071 0	0,106 3	0,141 5
0,70	0,990 4	0,950 4	0,895 8	0,034 4	0,068 7	0,102 7	0,136 6
0,75	0,992 2	0,959 8	0,916 5	0,033 6	0,067 0	0,100 1	0,133 0
0,80	0,993 5	0,966 9	0,931 9	0,853 9	0,066 1	0,098 5	0,130 7
0,85	0,994 6	0,972 5	0,943 6	0,881 0	0,066 1	0,098 3	0,130 1
0,90	0,995 4	0,976 8	0,952 8	0,901 5	0,780 0	0,100 6	0,132 1
0,93	0,995 9	0,979 0	0,957 3	0,911 5	0,805 9	0,663 5	0,135 9
0,95	0,996 1	0,980 3	0,960 0	0,917 4	0,820 6	0,696 7	0,141 0
0,97	0,996 3	0,981 5	0,962 5	0,922 7	0,833 8	0,724 0	0,558 0
0,98	0,996 5	0,982 1	0,963 7	0,925 3	0,839 8	0,736 0	0,588 7
0,99	0,996 6	0,982 6	0,964 8	0,927 7	0,845 5	0,747 1	0,613 8
1,00	0,996 7	0,983 2	0,965 9	0,930 0	0,850 9	0,757 4	0,635 3
1,01	0,996 8	0,983 7	0,966 9	0,932 2	0,856 1	0,767 1	0,654 2
1,02	0,996 9	0,984 2	0,967 9	0,934 3	0,861 0	0,776 1	0,671 0
1,05	0,997 1	0,985 5	0,970 7	0,940 1	0,874 3	0,800 2	0,713 0
1,10	0,997 5	0,987 4	0,974 7	0,948 5	0,893 0	0,832 3	0,764 9
1,15	0,997 8	0,989 1	0,978 0	0,955 4	0,908 1	0,857 6	0,803 2
1,20	0,998 1	0,990 4	0,980 8	0,961 1	0,920 5	0,877 9	0,833 0
1,30	0,998 5	0,992 6	0,985 2	0,970 2	0,939 6	0,908 3	0,876 4
1,40	0,998 8	0,994 2	0,988 4	0,976 8	0,953 4	0,929 8	0,906 2
1,50	0,999 1	0,995 4	0,990 9	0,981 8	0,963 6	0,945 6	0,927 8
1,60	0,999 3	0,996 4	0,992 8	0,985 6	0,971 4	0,957 5	0,943 9
1,70	0,999 4	0,997 1	0,994 3	0,988 6	0,977 5	0,966 7	0,956 3
1,80	0,999 5	0,997 7	0,995 5	0,991 0	0,982 3	0,973 9	0,965 9
1,90	0,999 6	0,998 2	0,996 4	0,992 9	0,986 1	0,979 6	0,973 5
2,00	0,999 7	0,998 6	0,997 2	0,994 4	0,989 2	0,984 2	0,979 6
2,20	0,999 8	0,999 2	0,998 3	0,996 7	0,993 7	0,991 0	0,988 6
2,40	0,999 9	0,999 6	0,999 1	0,998 3	0,996 9	0,995 7	0,994 8
2,60	1,000 0	0,999 8	0,999 7	0,999 4	0,999 1	0,999 0	0,999 0
2,80	1,000 0	1,000 0	1,000 1	1,000 2	1,000 7	1,001 3	1,002 1
3,00	1,000 0	1,000 2	1,000 4	1,000 8	1,001 8	1,003 0	1,004 3
3,50	1,000 1	1,000 4	1,000 8	1,001 7	1,003 5	1,005 5	1,007 5
4,00	1,000 1	1,000 5	1,001 0	1,002 1	1,004 3	1,006 6	1,009 0

Table A.1 — Values of compressibility factor $Z^{(0)}$ (*concluded*)

T_r	$Z^{(0)}$							
	p_r = 1,000	p_r = 1,200	p_r = 1,500	p_r = 2,000	p_r = 3,000	p_r = 5,000	p_r = 7,000	p_r = 10,000
0,30	0,289 2	0,347 0	0,433 5	0,577 5	0,864 8	1,436 6	2,004 8	2,850 7
0,35	0,260 4	0,312 3	0,390 1	0,519 5	0,777 5	1,290 2	1,798 7	2,553 9
0,40	0,237 9	0,285 3	0,356 3	0,474 4	0,709 5	1,175 8	1,637 3	2,321 1
0,45	0,220 0	0,263 8	0,329 4	0,438 4	0,655 1	1,084 1	1,507 7	2,133 8
0,50	0,205 6	0,246 5	0,307 7	0,409 2	0,611 0	1,009 4	1,401 7	1,980 1
0,55	0,193 9	0,232 3	0,289 9	0,385 3	0,574 7	0,947 5	1,313 7	1,852 0
0,60	0,184 2	0,220 7	0,275 3	0,365 7	0,544 6	0,895 9	1,239 8	1,744 0
0,65	0,176 5	0,211 3	0,263 4	0,349 5	0,519 7	0,852 6	1,177 3	1,651 9
0,70	0,170 3	0,203 8	0,253 8	0,336 4	0,499 1	0,816 1	1,124 1	1,572 9
0,75	0,165 6	0,198 1	0,246 4	0,326 0	0,482 3	0,785 4	1,078 7	1,504 7
0,80	0,162 6	0,194 2	0,241 1	0,318 2	0,469 0	0,759 8	1,040 0	1,445 6
0,85	0,161 4	0,192 4	0,238 2	0,313 2	0,459 1	0,738 8	1,007 1	1,394 3
0,90	0,163 0	0,193 5	0,238 3	0,311 4	0,452 7	0,722 0	0,979 3	1,349 6
0,93	0,166 4	0,196 3	0,240 5	0,312 2	0,450 7	0,713 8	0,964 8	1,325 7
0,95	0,170 5	0,199 8	0,243 2	0,313 8	0,450 1	0,709 2	0,956 1	1,310 8
0,97	0,177 9	0,205 5	0,247 4	0,316 4	0,450 4	0,705 2	0,948 0	1,298 8
0,98	0,184 4	0,209 7	0,250 3	0,318 2	0,450 8	0,703 5	0,944 2	1,290 1
0,99	0,195 9	0,215 4	0,253 8	0,320 4	0,451 4	0,701 8	0,940 6	1,283 5
1,00	0,290 1	0,223 7	0,258 3	0,322 9	0,452 2	0,700 4	0,937 2	1,277 2
1,01	0,464 8	0,237 0	0,264 0	0,326 0	0,453 3	0,699 1	0,933 9	1,271 0
1,02	0,514 6	0,262 9	0,271 5	0,329 7	0,454 7	0,698 0	0,930 7	1,265 0
1,05	0,602 6	0,443 7	0,313 1	0,345 2	0,460 4	0,695 6	0,922 2	1,248 1
1,10	0,688 0	0,598 4	0,458 0	0,395 3	0,477 0	0,695 0	0,911 0	1,223 2
1,15	0,744 3	0,680 3	0,579 8	0,476 0	0,504 2	0,698 7	0,903 3	1,202 1
1,20	0,785 8	0,736 3	0,660 5	0,560 5	0,542 5	0,706 9	0,899 0	1,184 4
1,30	0,843 8	0,811 1	0,752 4	0,690 8	0,634 4	0,735 8	0,899 8	1,158 0
1,40	0,882 7	0,859 5	0,825 6	0,775 3	0,720 2	0,776 1	0,911 2	1,141 9
1,50	0,910 3	0,893 3	0,868 9	0,832 8	0,788 7	0,820 0	0,929 7	1,133 9
1,60	0,930 8	0,918 0	0,900 0	0,873 8	0,841 0	0,861 7	0,951 8	1,132 0
1,70	0,946 3	0,936 7	0,923 4	0,904 3	0,880 9	0,898 4	0,974 5	1,134 3
1,80	0,958 3	0,951 1	0,941 3	0,927 5	0,911 8	0,929 7	0,996 1	1,139 1
1,90	0,967 8	0,962 4	0,955 2	0,945 6	0,935 9	0,955 7	1,015 7	1,145 2
2,00	0,975 4	0,971 5	0,966 4	0,959 9	0,955 0	0,977 2	1,032 8	1,151 6
2,20	0,986 5	0,984 7	0,982 6	0,980 6	0,982 7	1,009 4	1,060 0	1,163 5
2,40	0,994 1	0,993 6	0,993 5	0,994 5	1,001 1	1,031 3	1,079 3	1,172 8
2,60	0,999 3	0,999 8	1,001 0	1,004 0	1,013 7	1,046 3	1,092 6	1,179 2
2,80	1,003 1	1,004 2	1,006 3	1,010 6	1,022 3	1,056 5	1,101 6	1,183 0
3,00	1,005 7	1,007 4	1,010 1	1,015 3	1,028 4	1,063 5	1,107 5	1,184 8
3,50	1,009 7	1,012 0	1,015 6	1,022 1	1,036 8	1,072 3	1,113 8	1,183 4
4,00	1,011 5	1,014 0	1,017 9	1,024 9	1,040 1	1,074 7	1,113 6	1,177 3

Table A.2 — Values of correction term $Z^{(1)}$

T_r	$Z^{(1)}$ $p_r = 0{,}010$	$p_r = 0{,}050$	$p_r = 0{,}100$	$p_r = 0{,}200$	$p_r = 0{,}400$	$p_r = 0{,}600$	$p_r = 0{,}800$
0,30	−0,000 8	−0,004 0	−0,008 1	−0,016 1	−0,032 3	−0,048 4	−0,064 5
0,35	−0,000 9	−0,004 6	−0,009 3	−0,018 5	−0,037 0	−0,055 4	−0,073 8
0,40	−0,001 0	−0,004 8	−0,009 5	−0,019 0	−0,038 0	−0,057 0	−0,075 8
0,45	−0,000 9	−0,004 7	−0,009 4	−0,018 7	−0,037 4	−0,056 0	−0,074 5
0,50	−0,000 9	−0,004 5	−0,009 0	−0,018 1	−0,036 0	−0,053 9	−0,071 6
0,55	−0,031 4	−0,004 3	−0,008 6	−0,017 2	−0,034 3	−0,051 3	−0,068 2
0,60	−0,020 5	−0,004 1	−0,008 2	−0,016 4	−0,032 6	−0,048 7	−0,064 6
0,65	−0,013 7	−0,077 2	−0,007 8	−0,015 6	−0,030 9	−0,046 1	−0,061 1
0,70	−0,009 3	−0,050 7	−0,116 1	−0,014 8	−0,029 4	−0,043 8	−0,057 9
0,75	−0,006 4	−0,033 9	−0,074 4	−0,014 3	−0,028 2	−0,041 7	−0,055 0
0,80	−0,004 4	−0,022 8	−0,048 7	−0,116 0	−0,027 2	−0,040 1	−0,052 6
0,85	−0,002 9	−0,015 2	−0,031 9	−0,071 5	−0,026 8	−0,039 1	−0,050 9
0,90	−0,001 9	−0,009 9	−0,020 5	−0,044 2	−0,111 8	−0,039 6	−0,050 3
0,93	−0,001 5	−0,007 5	−0,015 4	−0,032 6	−0,076 3	−0,156 2	−0,051 4
0,95	−0,001 2	−0,006 2	−0,012 6	−0,026 2	−0,058 9	−0,111 0	−0,054 0
0,97	−0,001 0	−0,005 0	−0,010 1	−0,020 8	−0,045 0	−0,077 0	−0,164 7
0,98	−0,000 9	−0,004 4	−0,009 0	−0,018 4	−0,039 0	−0,064 1	−0,110 0
0,99	−0,000 8	−0,003 9	−0,007 9	−0,016 1	−0,033 5	−0,053 1	−0,079 6
1,00	−0,000 7	−0,003 4	−0,006 9	−0,014 0	−0,028 5	−0,043 5	−0,058 8
1,01	−0,000 6	−0,003 0	−0,006 0	−0,012 0	−0,024 0	−0,035 1	−0,042 9
1,02	−0,000 5	−0,002 6	−0,005 1	−0,010 2	−0,019 8	−0,027 7	−0,030 3
1,05	−0,000 3	−0,001 5	−0,002 9	−0,005 4	−0,009 2	−0,009 7	−0,003 2
1,10	−0,000 0	0,000 0	0,000 1	0,000 7	0,003 8	0,010 6	0,023 6
1,15	0,000 2	0,001 1	0,002 3	0,005 2	0,012 7	0,023 7	0,039 6
1,20	0,000 4	0,001 9	0,003 9	0,008 4	0,019 0	0,032 6	0,049 9
1,30	0,000 6	0,003 0	0,006 1	0,012 5	0,026 7	0,042 9	0,061 2
1,40	0,000 7	0,003 6	0,007 2	0,014 7	0,030 6	0,047 7	0,066 1
1,50	0,000 8	0,003 9	0,007 8	0,015 8	0,032 3	0,049 7	0,067 7
1,60	0,000 8	0,004 0	0,008 0	0,016 2	0,033 0	0,050 1	0,067 7
1,70	0,000 8	0,004 0	0,008 1	0,016 3	0,032 9	0,049 7	0,066 7
1,80	0,000 8	0,004 0	0,008 1	0,016 2	0,032 5	0,048 8	0,065 2
1,90	0,000 8	0,004 0	0,007 9	0,015 9	0,031 8	0,047 7	0,063 5
2,00	0,000 8	0,003 9	0,007 8	0,015 5	0,031 0	0,046 4	0,061 7
2,20	0,000 7	0,003 7	0,007 4	0,014 7	0,029 3	0,043 7	0,057 9
2,40	0,000 7	0,003 5	0,007 0	0,013 9	0,027 6	0,041 1	0,054 4
2,60	0,000 7	0,003 3	0,006 6	0,013 1	0,026 0	0,038 7	0,051 2
2,80	0,000 6	0,003 1	0,006 2	0,012 4	0,024 5	0,036 5	0,048 3
3,00	0,000 6	0,002 9	0,005 9	0,011 7	0,023 2	0,034 5	0,045 6
3,50	0,000 5	0,002 6	0,005 2	0,010 3	0,020 4	0,030 3	0,040 1
4,00	0,000 5	0,002 3	0,004 6	0,009 1	0,018 2	0,027 0	0,035 7

Table A.2 — Values of correction term $Z^{(1)}$ (*concluded*)

T_r	$Z^{(1)}$							
	p_r = 1,000	p_r = 1,200	p_r = 1,500	p_r = 2,000	p_r = 3,000	p_r = 5,000	p_r = 7,000	p_r = 10,000
0,30	−0,080 6	−0,096 6	−0,120 7	−0,160 8	−0,240 7	−0,399 6	−0,557 2	−0,791 5
0,35	−0,092 1	−0,110 5	−0,137 9	−0,183 4	−0,273 8	−0,452 3	−0,627 9	−0,886 3
0,40	−0,094 6	−0,113 4	−0,141 4	−0,187 9	−0,279 9	−0,460 3	−0,636 5	−0,893 6
0,45	−0,092 9	−0,111 3	−0,138 7	−0,184 0	−0,273 4	−0,447 5	−0,616 2	−0,860 6
0,50	−0,089 3	−0,106 9	−0,133 0	−0,176 2	−0,261 1	−0,425 3	−0,583 1	−0,809 9
0,55	−0,084 9	−0,101 5	−0,126 3	−0,166 9	−0,246 5	−0,399 1	−0,544 6	−0,752 1
0,60	−0,080 3	−0,096 0	−0,119 2	−0,157 2	−0,231 2	−0,371 8	−0,504 7	−0,692 8
0,65	−0,075 9	−0,090 6	−0,112 2	−0,147 6	−0,216 0	−0,344 7	−0,465 3	−0,634 6
0,70	−0,071 8	−0,085 5	−0,105 7	−0,138 5	−0,201 3	−0,318 4	−0,427 0	−0,578 5
0,75	−0,068 1	−0,080 8	−0,099 6	−0,129 8	−0,187 2	−0,292 9	−0,390 1	−0,525 0
0,80	−0,064 8	−0,076 7	−0,094 0	−0,121 7	−0,173 6	−0,268 2	−0,354 5	−0,474 0
0,85	−0,062 2	−0,073 1	−0,088 8	−0,113 8	−0,160 2	−0,243 9	−0,320 1	−0,425 4
0,90	−0,060 4	−0,070 1	−0,084 0	−0,105 9	−0,146 3	−0,219 5	−0,286 2	−0,378 8
0,93	−0,060 2	−0,068 7	−0,081 0	−0,100 7	−0,137 4	−0,204 5	−0,266 1	−0,351 6
0,95	−0,060 7	−0,067 8	−0,078 8	−0,096 7	−0,131 0	−0,194 3	−0,252 6	−0,333 9
0,97	−0,062 3	−0,066 9	−0,075 9	−0,092 1	−0,124 0	−0,183 7	−0,239 1	−0,316 3
0,98	−0,064 1	−0,066 1	−0,074 0	−0,089 3	−0,120 2	−0,178 3	−0,232 2	−0,307 5
0,99	−0,068 0	−0,064 6	−0,071 5	−0,086 1	−0,116 2	−0,172 8	−0,225 4	−0,298 9
1,00	−0,087 9	−0,060 9	−0,067 8	−0,082 4	−0,111 8	−0,167 2	−0,218 5	−0,290 2
1,01	−0,022 3	−0,047 3	−0,062 1	−0,077 8	−0,107 2	−0,161 5	−0,211 6	−0,281 6
1,02	−0,006 2	0,022 7	−0,052 4	−0,072 2	−0,102 1	−0,155 6	−0,204 7	−0,273 1
1,05	0,022 0	0,105 9	0,045 1	−0,043 2	−0,083 8	−0,137 0	−0,183 5	−0,247 6
1,10	0,047 6	0,089 7	0,163 0	0,069 8	−0,037 3	−0,102 1	−0,146 9	−0,205 6
1,15	0,062 5	0,094 3	0,154 8	0,166 7	0,033 2	−0,061 1	−0,108 4	−0,164 2
1,20	0,071 9	0,099 1	0,147 7	0,199 0	0,109 5	−0,014 1	−0,067 8	−0,123 1
1,30	0,081 9	0,104 8	0,142 0	0,199 1	0,207 9	0,087 5	0,017 6	−0,042 3
1,40	0,085 7	0,106 3	0,138 3	0,189 4	0,239 7	0,173 7	0,100 8	0,035 0
1,50	0,086 4	0,105 5	0,134 5	0,180 6	0,243 3	0,230 9	0,171 7	0,105 8
1,60	0,085 5	0,103 5	0,130 3	0,172 9	0,238 1	0,263 1	0,225 5	0,167 3
1,70	0,083 8	0,100 8	0,125 9	0,165 8	0,230 5	0,278 8	0,262 8	0,217 9
1,80	0,081 6	0,097 8	0,121 6	0,159 3	0,222 4	0,284 6	0,287 1	0,257 6
1,90	0,079 2	0,094 7	0,117 3	0,153 2	0,214 4	0,284 8	0,301 7	0,287 6
2,00	0,076 7	0,091 6	0,113 3	0,147 6	0,206 9	0,281 9	0,309 7	0,309 6
2,20	0,071 9	0,085 7	0,105 7	0,137 4	0,193 2	0,272 0	0,313 5	0,335 5
2,40	0,067 5	0,080 3	0,098 9	0,128 5	0,181 2	0,260 2	0,308 9	0,345 9
2,60	0,063 4	0,075 4	0,092 9	0,120 7	0,170 6	0,248 4	0,300 9	0,347 5
2,80	0,059 8	0,071 1	0,087 6	0,113 8	0,161 3	0,237 2	0,291 5	0,344 3
3,00	0,056 5	0,067 2	0,082 8	0,107 6	0,152 9	0,226 8	0,281 7	0,338 5
3,50	0,049 7	0,059 1	0,072 8	0,094 9	0,135 6	0,204 2	0,258 4	0,319 4
4,00	0,044 3	0,052 7	0,065 1	0,084 9	0,121 9	0,185 7	0,237 8	0,299 4

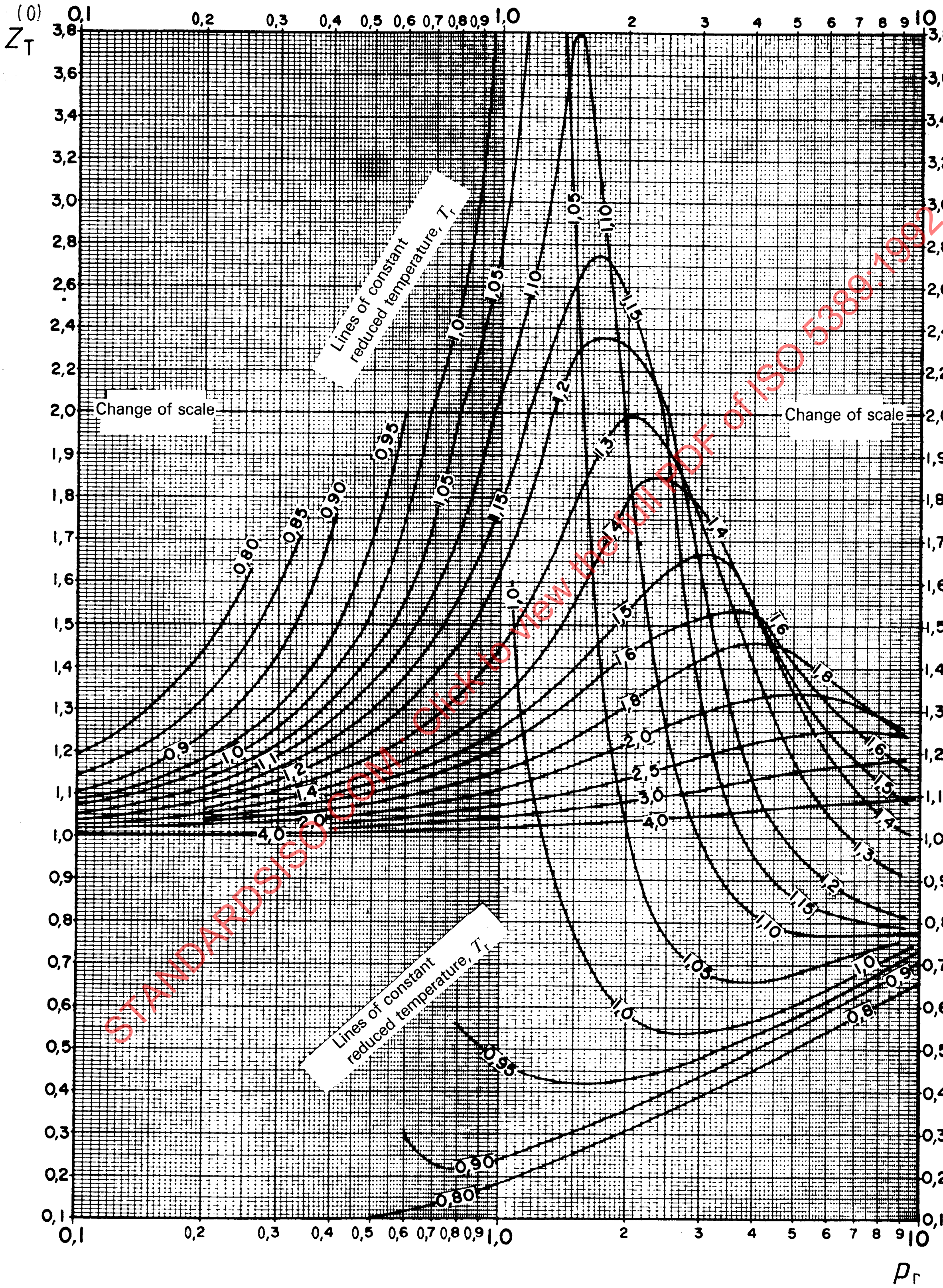


Figure A.1a) — Values of $Z_T^{(0)}$ as a function of p_r

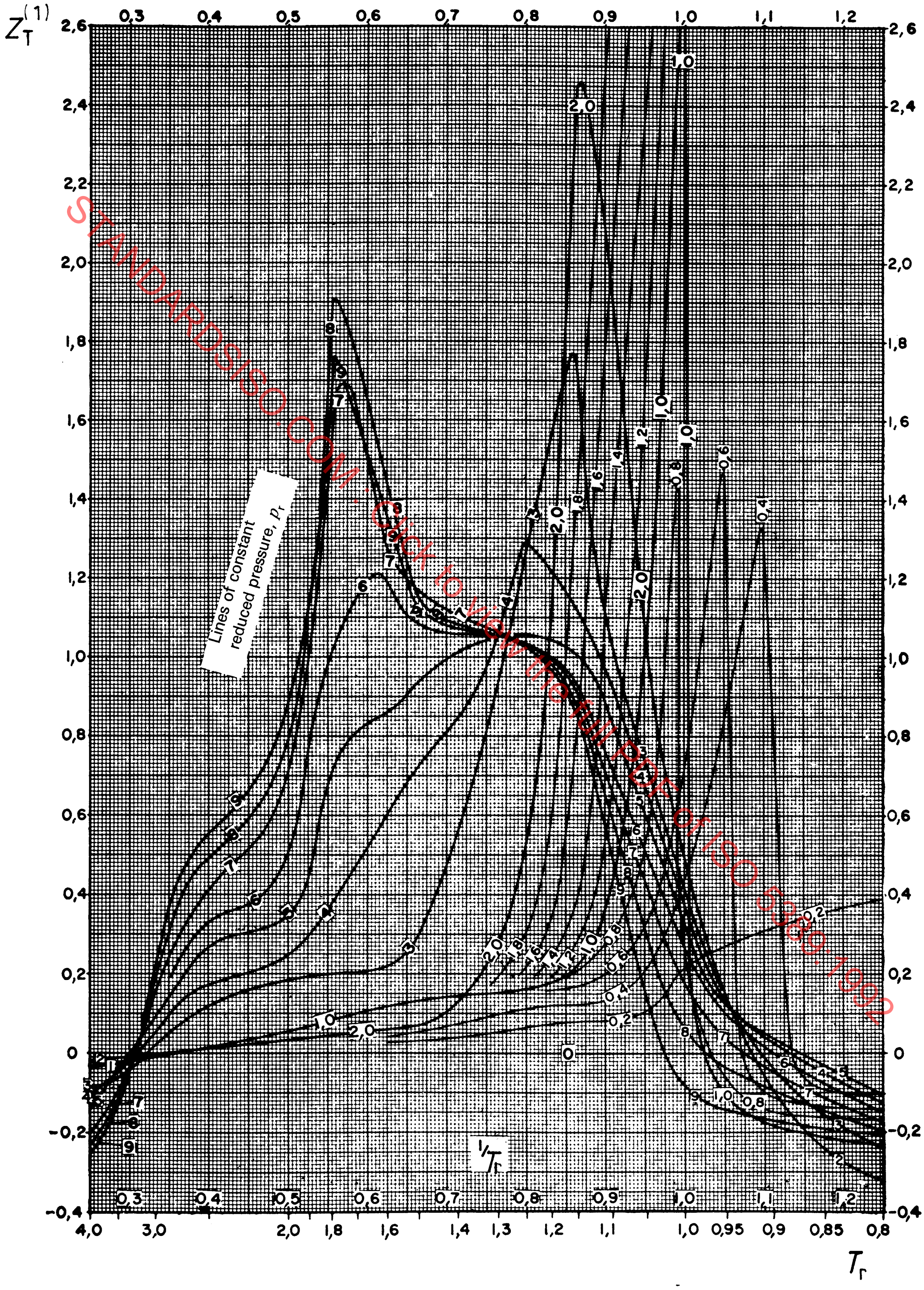


Figure A.1b) — Values of $Z_T^{(1)}$ as a function of T_r

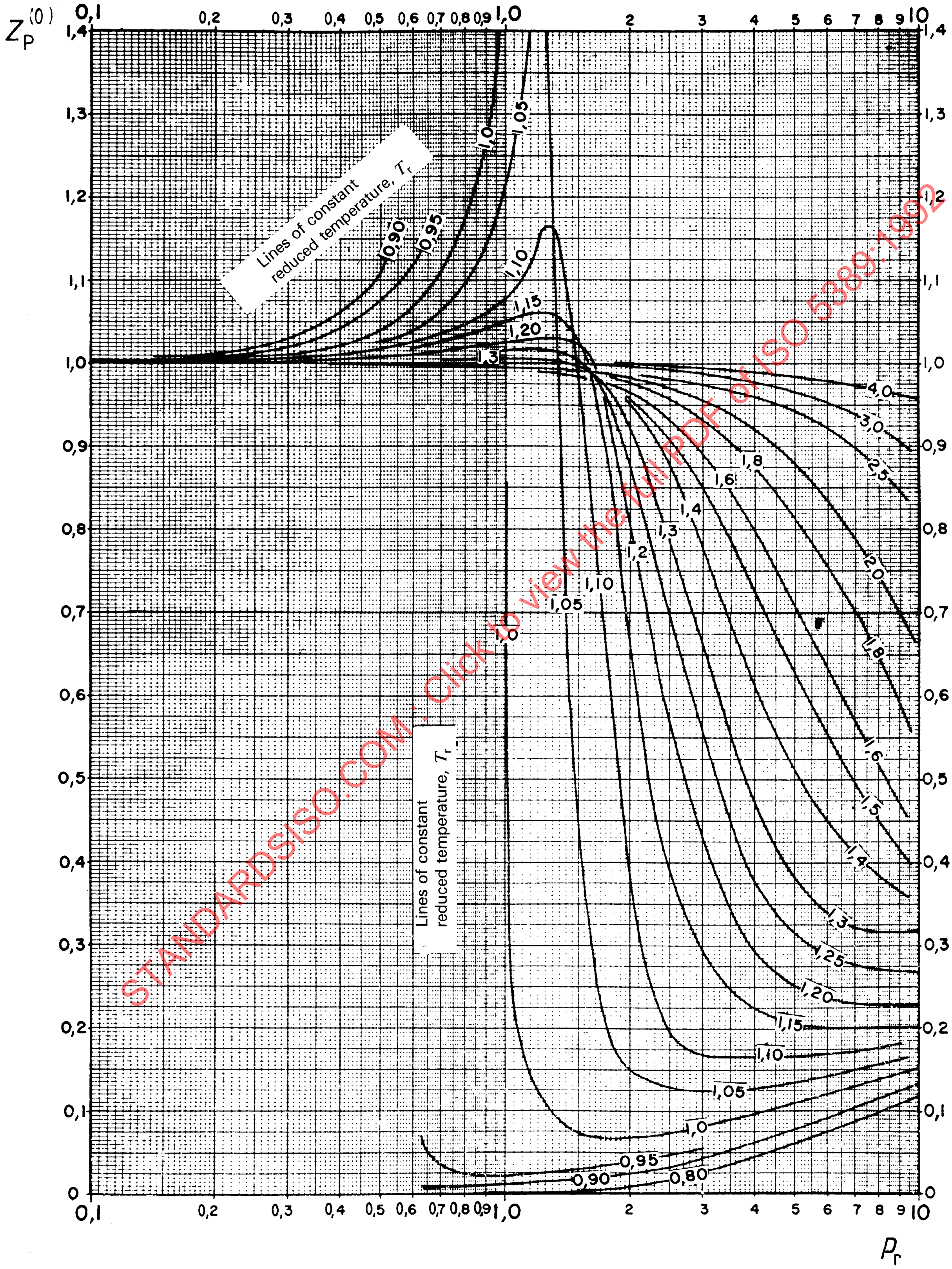


Figure A.2a) — Values of $Z_p^{(0)}$ as a function of p_r

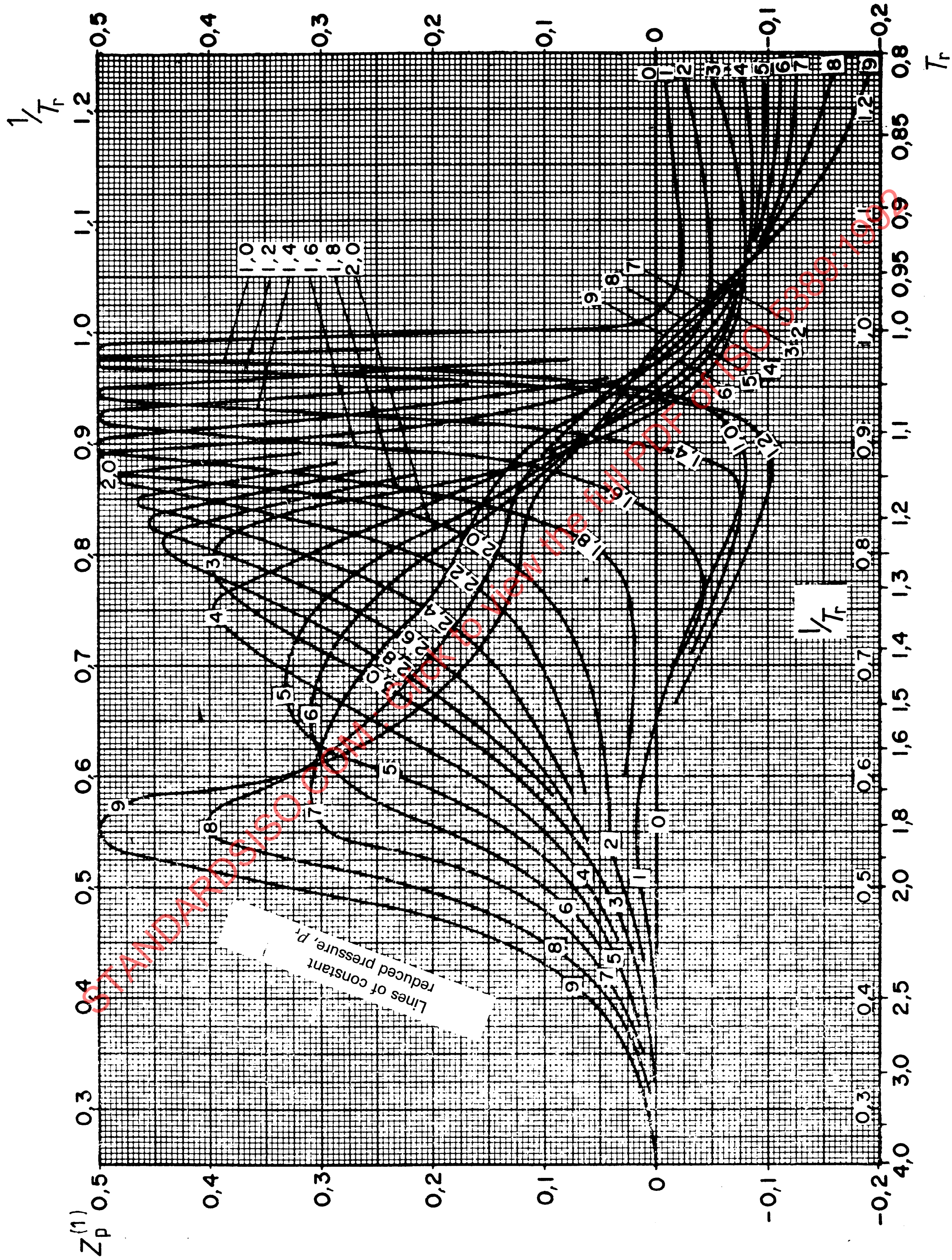


Figure A.2b) — Values of $Z_p^{(1)}$ as a function of T_r

A.4 Properties of gas mixtures and gas-vapour mixtures, considered as perfect gases

A.4.1 Gas mixtures

A.4.1.1 The ratio of the components is given by the formula

$$m_i = \frac{\varrho_i V_i}{\varrho V} = \frac{q_{m,i}}{q_m}$$

$$\sum m_i = 1$$

A.4.1.2 The molar proportion is given by the formula

$$Y_i = \frac{\text{mole}_i}{\sum \text{mole}}$$

A.4.1.3 The volume ratio of the components is given by the formula

$$r_i = \frac{V_i}{V} = \frac{q_{V,i}}{q_V}$$

$$\sum r_i = 1$$

NOTE — For mixtures of perfect gases r_i represents the molar proportion.

A.4.1.4 The partial pressures p_i of the components are given by the formula

$$\sum p_i = p$$

A.4.1.5 The mean molecular mass of the mixture is determined as follows:

$$M_\mathrm{M} = \sum r_i M_i = \frac{1}{\sum \frac{m_i}{M_i}} = \sum \frac{p_i}{p} M_i$$

A.4.1.6 The mean gas constant of the mixture is determined as follows:

$$R_\mathrm{M} = \frac{R_\mathrm{mol}}{M} = \frac{1}{\sum \frac{r_i}{R_i}}$$

A.4.1.7 The following relationships exist:

$$m_i = \frac{r_i M_i}{\sum r_i M_i}$$

$$r_i = \frac{\frac{m_i}{M_i}}{\sum \frac{m_i}{M_i}}$$

$$p_i = r_i p$$

A.4.1.8 The mean molar specific heat of the mixture is determined as follows:

$$C_{p\mathrm{M}} = \sum r_i C_{pi}$$

$$C_{V\mathrm{M}} = \sum r_i C_{Vi}$$

A.4.1.9 The mean specific heat of the mixture is determined as follows:

$$c_p = \sum m_i c_{pi}$$

$$c_V = c_p - R$$

A.4.1.10 The mean isentropic exponent is determined as follows:

$$\kappa_\mathrm{M} = \frac{\sum m_i c_{pi}}{\sum m_i c_{Vi}} = 1 \Bigg/ \left\{ 1 - \frac{1}{\sum m_i [\kappa_i / (\kappa_i - 1)]} \right\}$$

A.4.2 Gas-vapour mixtures

A.4.2.1 The vapour content is defined by one of the following quantities.

A.4.2.1.1 The relative humidity, φ, which is the ratio between the partial vapour pressure and the saturation pressure at the temperature of the mixture is given by the formula

$$\varphi = \frac{p_\mathrm{V}}{p_\mathrm{sat}} < 1$$

For $\varphi > 1$ there is a formation of condensate.

A.4.2.1.2 The humidity content, χ, which is the vapour mass related to the mass of dry gas is given by the formula

$$\chi = \frac{q_V v_\mathrm{G}}{q_V v_\mathrm{V}} = \frac{R_\mathrm{G}}{R_\mathrm{V}} \times \frac{p_\mathrm{V}}{p - p_\mathrm{V}} = \frac{R_\mathrm{G}}{R_\mathrm{V}} \times \frac{\varphi p_\mathrm{sat}}{p - \varphi p_\mathrm{sat}}$$

In the particular case of humid air

$$\chi = 0{,}622\,1 \frac{\varphi p_\mathrm{sat}}{p - \varphi p_\mathrm{sat}}$$

A.4.2.1.3 The mass ratio of vapour, m_V, is given by the formula

$$m_\mathrm{V} = \frac{q_V v}{q_V v_\mathrm{V}} = \frac{q_V v_\mathrm{G}}{q_V v_\mathrm{V} + q_V v_\mathrm{G}} = \frac{\chi}{1 + \chi}$$

A.4.2.1.4 The volume ratio of vapour, r_V, is given by the formula

$$r_V = \frac{q_m m_V R_V}{q_m R} = m_V \frac{R_V}{R} = \frac{p_V}{p} = \frac{\varphi p_{sat}}{p}$$

A.4.2.2 The mean molecular mass of the mixture, M_M, is determined as follows:

$$M_M = M_V r_V + M_G (1 - r_V) = M_G + \frac{\varphi\, p_{sat}}{p} (M_V - M_G)$$

A.4.2.3 The mean gas constant, R_M, of the mixture is determined as follows:

$$R_M = m_V R_V + (1 - m_V) R_G = R_G \left[1 + \frac{\chi}{1+\chi} \left(\frac{R_V}{R_G} - 1\right)\right]$$

For the particular case of humid air

$$R_M = R_G \left(1 + 0{,}607\,5 \frac{\chi}{1+\chi}\right)$$

A.4.2.4 The mean molar specific heat is determined as indicated in A.4.1.8.

A.4.2.5 The mean isentropic exponent is determined as indicated in A.4.1.10.

A.4.2.6 The viscosity of a gas-vapour mixture is determined in the same manner as for a gas mixture by using the methods indicated in annex B.

A.4.2.7 The volume flow-rate of a gas-vapour mixture can be defined in two different ways, according to the contract specifications, as given in A.4.2.7.1 and A.4.2.7.2.

A.4.2.7.1 It is defined as the flow, q_V, of a gas-vapour mixture actually handled by the compressor and resulting directly from the flow-rate measurement.

A.4.2.7.2 It is defined as a flow-rate of dry gas q_{VG}.

A.4.2.7.3 These two flow-rates are linked together by the following relation:

$$q_V = \frac{q_{VG}}{1 - r_V} = q_{VG} \times \frac{p}{p - \varphi\, p_{sat}}$$

A.4.2.8 The condensate flow-rate separated in an intercooler can be calculated as follows.

If between stage I and stage II of a compressor the gas is cooled such that the relative humidity at the outlet of the intercooler becomes greater than 1, a mass flow-rate q_{mCd} can be extracted at this point of the circuit.

In this case it is assumed that after the extraction of the condensate the gas-vapour mixture is still saturated, i.e. that $\varphi = 1$.

The condensate flow-rate is determined as follows:

$$q_{mCd} = q_{mV,I} - q_{mV,II} = q_{mG} (\chi_I - \chi_{sat,II})$$

where

q_{mG} is the mass flow-rate of dry gas;

χ_I is the humidity content at the inlet of the intercooler, determined according to A.4.2.1.2;

$\chi_{sat,II}$ is the humidity content of the saturated mixture at the outlet conditions of the intercooler.

Annex B
(normative)

Viscosity of gases and gas mixtures

B.1 Dynamic viscosity

B.1.1 Dynamic viscosity at atmospheric pressure or at a pressure close to atmospheric pressure

B.1.1.1 Pure substances

Information is contained in [35], [36] and [37].

If the dynamic viscosity of a pure substance is to be assessed at atmospheric pressure or at a pressure close to atmospheric pressure, three methods can be used as follows:

a) the theory of Chapman — Enskog;

b) the law of corresponding states;

c) the method of VDI-Wärmeatlas.

B.1.1.1.1 The theory of Chapman — Enskog

As a general rule the equation giving the value of dynamic viscosity, μ, expressed in pascal seconds, is as follows:

$$\mu = 26\,693 \times 10^{-3} \frac{\sqrt{MT}}{\sigma^2 \Omega_v}$$

where

M is the molar mass, in grams;

T is the temperature, in kelvins;

σ is the collision diameter, in angstroms;

Ω_v is the reduced integral of collision.

In the equation above, σ and Ω_v are dependent on the potential of interaction of the two molecules present.

The expressions for potential as given by Lennard — Jones for non-polar substances and by Stockmayer for polar substances are adopted for the assessment of quantities σ and Ω_v.

These quantities may also have been determined experimentally.

Numerical values for these quantities may be found, for instance, in [6] or [38].

If none can be found, it is advised for non-polar substances to calculate the corresponding values by application of the equations in [39]:

$$\sigma \left(\frac{p_{Cr}}{T_{Cr}}\right)^{1/3} = 2{,}355\,1 - 0{,}087\,\omega$$

$$\frac{\varepsilon}{kT_{Cr}} = 0{,}791\,5 + 0{,}169\,3\,\omega$$

where

k is the Boltzmann constant ($k = 1{,}380\,5 \times 10^7$ J/K);

ε is the depth of potential energy minimum;

σ is the collision diameter, in angstroms;

ω is the acentric factor.

In the case of polar gases, the calculation made and published in [40] can be used.

In the case of substances for which σ and $T^* = \dfrac{kT_{Cr}}{\varepsilon}$ are not given, it is recommended that the following equations be used (see [41]):

$$\sigma = \left(\frac{1{,}585\,V_b}{1 + 1{,}3\,\delta^2}\right)^{1/3}$$

$$\frac{\varepsilon}{k} = 1{,}18\,(1 + 1{,}3\,\delta^2)\,T_b$$

$$\delta = \frac{1{,}94 \times 10^3 \mu_p^2}{V_b T_b}$$

where subscript b indicates conditions at boiling point, with μ_p expressed in debyes[1)] and V_b in cubic centimetres per mole gram.

B.1.1.1.2 Method of the corresponding states

In the case of non-polar substances, the following relationships have been established (see [43]):

$$\mu\xi = 4{,}610\,T_r^{0{,}618} - 2{,}04\,e^{-0{,}449}\,T_r + 1{,}94\,e^{-4{,}058}\,T_r + 0{,}1$$

where

$$\xi = T_{Cr}^{1/6} M^{-1/2} p_{Cr}^{-2/3}$$

and

p_{Cr} is expressed in atmospheres[2)].

1) 1 debye $= 10^{-18}\,(\text{dyn} \cdot \text{cm}^4)^{1/2}$
$= 10^{-25}\,(10\ \text{N} \cdot \text{m}^4)^{1/2}$
$= 3{,}162 \times 10^{-25}\,(\text{N} \cdot \text{m}^4)^{1/2}$

2) 1 atm = 101 325 Pa

The most accurate method is the application of the theory of Chapman — Enskog using the reduced integral of collision as defined previously.

In the case of polar substances with or without hydrogen bonding, the utmost caution is necessary.

B.1.1.1.3 Method of VDI-Wärmeatlas

See [37].

B.1.1.2 Mixtures

For mixtures with n components, the kinetic theory of gases gives the following equation, neglecting second-order effects:

$$\mu_m = \sum_{i=1}^{n} \mu_i \Big/ \left[1 + \sum_{j=1}^{n} \phi_{ij} \, (y_i / y_j)^{-1} \right]$$

where

μ_i are the viscosities of pure gases;

μ_m is the viscosity of the mixture;

y_i and y_j are mole fractions;

$$\phi_{ij} = \frac{[1 + (\mu_i / \mu_j)^{1/2} \, (M_i / M_j)^{-1/4}]^2}{2\sqrt{2} \, [1 + (M_i / M_j)]^{1/2}}$$

$$\phi_{ji} = \phi_{ij} \left(\frac{\mu_j}{\mu_i} \right) \left(\frac{M_i}{M_j} \right)$$

NOTE — Among the various methods for assessing ϕ_{ij} and ϕ_{ji}, that given in [44] is currently preferred, even in the case of polar compounds. However, the error which is usually less than 2 % may become very significant if M_i is markedly greater than M_j and μ_i much greater than μ_j.

When hydrogen or helium are present in a proportion exceeding 35 % by volume, the more accurate, though more complex, method of Reichenberg as given in [45] may be considered.

The diagrams given in figures B.1 and B.2 may be used in determining the viscosity of gas mixtures; these diagrams are based on the Bromley and Wilke method.

B.1.2 Dynamic viscosity at high pressure

B.1.2.1 Pure substances

A diagram in which the effect of pressure is shown has been established in [46].

For any reduced pressure or temperature it can be determined whether the gas is "dilute" or dense (see figure B.3). The boundary is such that the required correction is no greater than 1 %.

With regard to the method of the corresponding states, either the charts given in [47], or the diagrams given in [48] or in [49], or the following equation taken from [50], may be used:

$$\frac{\mu}{\mu_o} = 1 + (1 - 0{,}45\, q) \, \frac{A p_r^{1/5}}{B p_r + (1 + C p_r^D)^{-1}}$$

where

$$q = \frac{668 \, (\mu_p)^2 \, p_{Cr}}{T_{Cr}^2}$$

$$A = \frac{\alpha_1}{T_r} \exp \alpha_2 \, T_r^{-\alpha_3}$$

$$B = A \, (\beta_1 \, T_r - \beta_2)$$

$$C = \frac{\gamma_1}{T_r} \exp \gamma_2 \, T_r^{-\gamma_3}$$

$$D = \frac{\delta_1}{T_r} \exp \delta_2 \, T_r^{-\delta_3}$$

and

$\alpha_1 = 1{,}982\,4 \times 10^{-3}$

$\alpha_2 = 5{,}268\,3$

$\alpha_3 = 0{,}576\,7$

$\beta_1 = 1{,}655\,2$

$\beta_2 = 1{,}276\,0$

$\gamma_1 = 0{,}131\,9$

$\gamma_2 = 3{,}703\,5$

$\gamma_3 = 79{,}867\,8$

$\delta_1 = 2{,}949\,6$

$\delta_2 = 2{,}919\,0$

$\delta_3 = 16{,}616\,9$

In the formulae above, μ_p is expressed in debyes, p_{Cr} in atmospheres and T_{Cr} in kelvins, which leads to an error of less than 10 %.

The reduced viscosity as recommended in [51] may also be used, which leads to $\mu \, \xi = 7{,}7$.

The result may be obtained rapidly but is unfortunately approximate.

Among the most recent data those published in [37] and [52] may be mentioned.

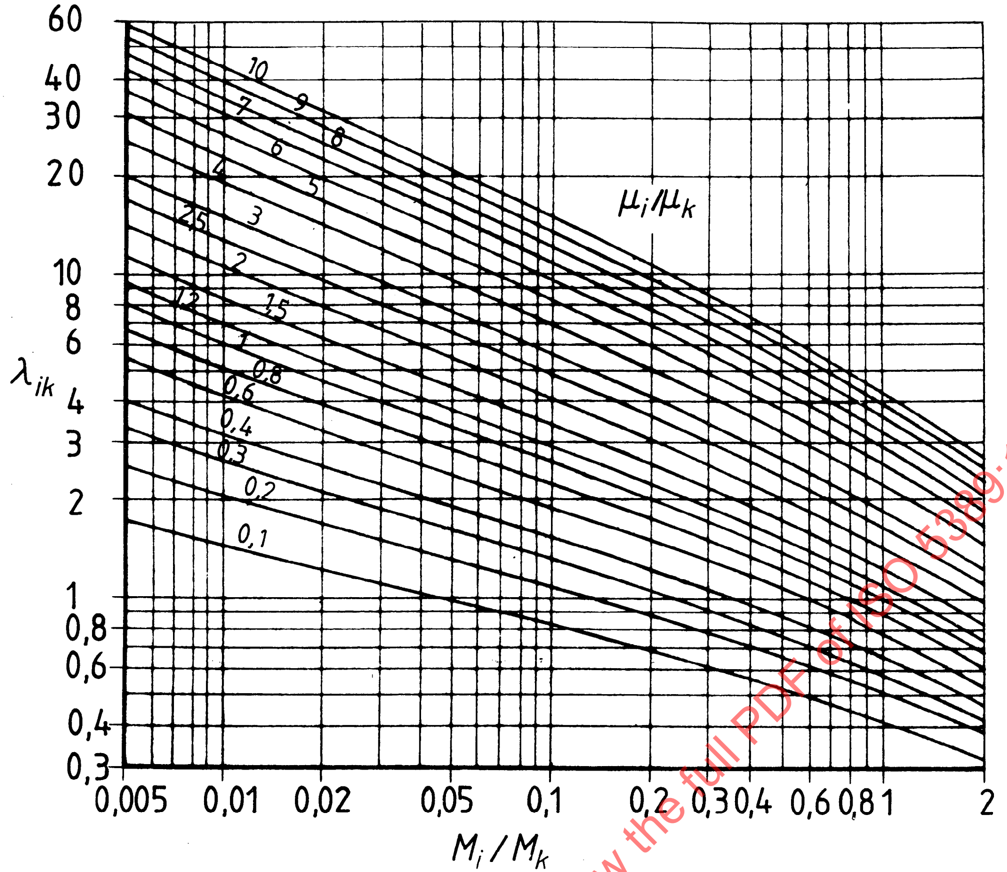


Figure B.1 — Diagram for the determination of the viscosity of gas mixtures

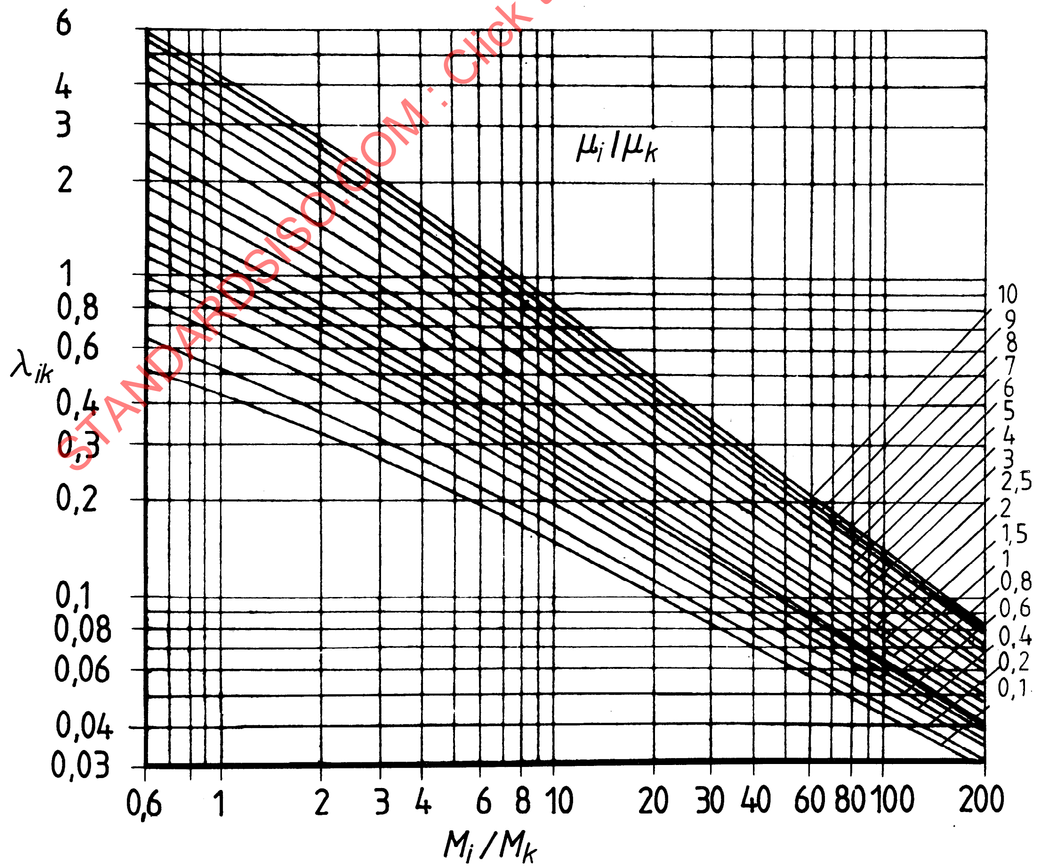


Figure B.2 — Diagram for the determination of the viscosity of gas mixtures

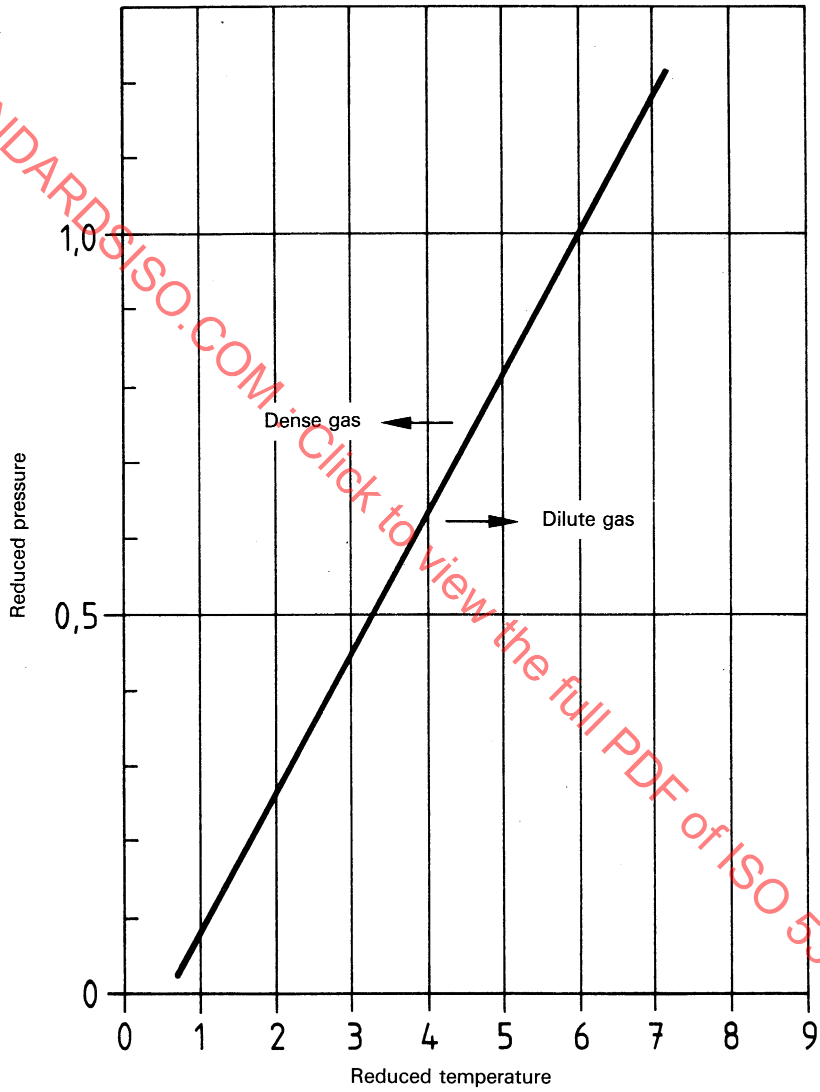


Figure B.3 — Diagram showing the effect of pressure
(see [46])

B.1.2.2 Mixtures

Although its accuracy is questionable, the Dean and Stiel method is the most frequently recommended:

$$(\mu_{m} - \mu_{m}^{o})\,\xi_{m} = 1{,}08\,[\exp(1{,}439\,\varrho_{rm}) - \exp(1{,}111\,\varrho_{rm}^{\;1{,}858})]$$

where

μ_{m} is the viscosity of the mixture at high pressure, in micropoises[1];

μ_{m}^{o} is the viscosity of the mixture at low pressure, in micropoises[1];

ϱ_{rm} is the pseudo-reduced density of the mixture

$$\varrho_{rm} = \varrho_{m}/\varrho_{Crm}$$

where

ϱ_{m} is the density of the mixture, in gram moles per cubic centimetre;

ϱ_{Crm} is the pseudo-critical density of the mixture, in gram moles per cubic centimetre ($\varrho_{Crm} = p_{Crm}/Z_{Crm}\,R\,T_{Crm}$);

$$\xi_{m} = T_{Crm}^{1/6}\,M_{m}^{-1/2}\,p_{Crm}^{-2/3}$$

the calculation of the mixture pseudo-critical parameter being made according to the amended rules of Prausnitz and Gunn, i.e.

$$T_{Crm} = \sum_{i} y_i\,T_{Cri}$$

$$Z_{Crm} = \sum_{i} y_i\,Z_{Cri}$$

$$V_{Crm} = \sum_{i} y_i\,V_{Cri}$$

$$p_{Crm} = \frac{Z_{Crm}\,R\,T_{Crm}}{V_{Crm}}$$

from which ϱ_{Crm} and ξ_{m} may be calculated.

In the case of non-polar mixtures with low molar masses, the accuracy may be within 5 %.

In the case of non-polar mixtures with high molar masses or of polar mixtures, the Dean and Stiel method may still be used but with lesser accuracy.

Another general method is described in [37].

B.2 Kinematic viscosity

The kinematic viscosity of a pure substance or of a mixture is determined using the formula

$$\nu = \frac{\mu}{\varrho}$$

where

μ is the dynamic viscosity;

$$\varrho = \frac{p}{Z\,R\,T}$$

1) 1 µP = 10^{-7} Pa·s

Annex C
(normative)

Conversion factors from non-SI units to SI units

Measurement No.	Measurement	Term and symbol for non-SI unit	SI unit symbol	Multiplication factor
C.1	length	inch, in (″) foot, ft yard, yd	mm m m	25,4 (exact) 0,304 8 (exact) 0,914 4 (exact)
C.2	speed	foot per minute, ft/min	m/s	0,005 08 (exact)
C.3	area	square inch, in^2 square foot, ft^2 square yard, yd^2	cm^2 m^2 m^2	6,451 6 (exact) 0,092 903 06 (exact) 0,836 127
C.4	volume	cubic inch, in^3 cubic foot, ft^3 cubic yard, yd^3 gallon, gal (UK) gallon, gal (US)	ml dm^3 m^3 dm^3 dm^3	16,387 064 (exact) 28,316 8 0,764 555 4,546 09 3,785 41
C.5	density	cubic foot per pound, ft^3/lb	m^3/kg	0,062 428
C.6	flow	cubic foot per minute, ft^3/min	l/s	0,471 95
C.7	mass	grain, gr ounce, oz pound, lb slug hundredweight, cwt (UK) ton (UK) ton (US)	mg g kg kg kg kg kg	64,798 91 (exact) 28,349 5 0,453 592 37 (exact) 14,593 9 50,802 3 1 016,05 907,185
C.8	density	pound per cubic foot, lb/ft^3	kg/m^3	16,018 5
C.9	force	kilogram-force, kgf pound-force, lbf sthène, st	N N N	9,806 65 (exact) 4,448 22 10^3 (exact)
C.10	pressure	bar kilogram-force per square metre, kgf/m^2 Torr (mmHg 0 °C) pièze, pz pound-force per square inch, lbf/in^2 technical atmosphere, at (= kgf/cm^2) physical atmosphere, atm	Pa Pa Pa Pa Pa Pa Pa	10^5 (exact) 9,806 65 (exact) 1,333 22 × 10^2 10^3 (exact) 6,894 76 × 10^3 0,980 665 × 10^5 (exact) 1,013 25 × 10^5
C.11	work	kilowatt-hour, kW·h kilogram-force metre, kgf·m foot pound-force, ft·lbf	J J J	3,6 × 10^6 (exact) 9,806 65 (exact) 1,355 82
C.12	quantity of heat	15 °C calories, cal_{15} I.T. calorie, cal_{IT} British thermal unit, Btu metric horsepower hour, ch·h horsepower hour, hp·h thermie, th	J J J MJ MJ MJ	4,185 5 (exact) 4,186 8 (exact) 1,055 06 × 10^3 2,647 80 2,684 52 4,185 5
C.13	internal energy density	British thermal unit per pound, Btu/lb	kJ/kg	2,326 01 (exact)
C.14	internal energy volume	metric horsepower minute per cubic metre, ch·min/m^3 horsepower per hundred cubic feet per minute, hp/100 cfm	J/l J/l	44,129 9 15,800 5
C.15	power	kilogram-force metre per second, kgf·m/s metric horsepower, ch horsepower, hp foot poundal per second, ft·pdl/s	W kW kW W	9,806 65 (exact) 0,735 499 0,745 700 0,042 140 1

Measurement No.	Measurement	Term and symbol for non-SI unit	SI unit symbol	Multiplication factor
C.16	heat flow	I.T. kilocalorie per hour, $kcal_{IT}/h$ British thermal unit per hour, Btu/h	W W	1,163 (exact) 0,293 072
C.17	dynamic viscosity	poise, P pound-force second per square foot, $lbf \cdot s/ft^2$	Pa·s Pa·s	0,1 (exact) 47,880 3
C.18	kinematic viscosity	stokes, St square foot per second, ft^2/s	m^2/s m^2/s	10^{-4} (exact) 0,092 903 0
C.19	heat transfer	I.T. kilocalorie per hour metre kelvin, $kcal_{IT}/(h \cdot m \cdot K)$ British thermal unit per hour square foot degree Fahrenheit, $Btu/(h \cdot ft^2 \cdot {}^\circ F)$ British thermal unit per hour foot degree Fahrenheit, $Btu/(h \cdot ft \cdot {}^\circ F)$	W/(m·K) W/(m·K) W/(m·K)	1,163 (exact) 0,144 228 1,730 73
C.20	heat transfer coefficient	I.T. kilocalorie per hour square metre kelvin, $kcal_{IT}/(h \cdot m^2 \cdot K)$ British thermal unit per hour square foot degree Fahrenheit, $Btu/(h \cdot ft^2 \cdot {}^\circ F)$	$W/(m^2 \cdot K)$ $W/(m^2 \cdot K)$	1,163 (exact) 1,678 25

C.21 Temperature

The Celsius temperature t of a system is given by the expression

$$t = T - T_0$$

where

T is the absolute thermodynamic temperature, in kelvins;

$T_0 = 273,15$ K;

(in accordance with the International Temperature Scale of 1968).

The Fahrenheit temperature t_F of a system is given by the expression

$$t_F = T_R - T_0$$

where

T_R is the thermodynamic temperature, in Rankine degrees;

$T_0 = 459,67$ °R

NOTES

1 °R should not be mistaken for an abbreviation for the abandoned degree Réaumur.

2 1 °R = 5/9 K

If t in degrees Celsius, t_F in degrees Fahrenheit, T in kelvins and T_R in Rankine degrees refer to one and the same physical state, then the numerical values t, t_F, T and T_R are related by the following formulae:

$$t = 5/9\,(t_F - 32)$$

$$t = T - 273,15$$

$$t = 5/9\,T_R - 273,15$$

C.22 Multiples of SI units

To allow words and symbols of multiples and submultiples of SI units to be formed, the following SI prefixes are used.

Prefix	Symbol	Factor
tera	T	10^{12}
giga	G	10^9
mega	M	10^6
kilo	k	10^3
hecto	h	10^2
deca	da	10
deci	d	10^{-1}
centi	c	10^{-2}
milli	m	10^{-3}
micro	µ	10^{-6}
nano	n	10^{-9}
pico	p	10^{-12}

Annex D
(normative)

Similarity of flow
(Similar velocity triangles)

D.1 General

The following theory strictly applies only to uncooled compressors or to sections of cooled compressors between intercoolers. However, it may be applied to cooled compressors, provided that the conditions for similarity of flow are satisfied simultaneously in all sections of the compressor.

The conversion of test results from the test conditions to the specified conditions, assuming constant polytropic efficiency, is generally only possible when similarity of flow is assured for the conversion of a test point to specified conditions, i.e. when the following main requirements are fulfilled:

a) geometrical similarity;

b) equal isentropic exponents;

c) equal enthalpy rise coefficients and flow coefficients;

d) equal Mach numbers;

e) equal Reynolds numbers.

The above similarity requirements refer only to flow in the compressor, and not to any intercoolers which may possibly be fitted nor to the mechanical losses. Furthermore the heat transfer from the compressed medium to the casing of uncooled compressors, the conduction of heat by the casing and the radiation of heat by the casing to the surrounding atmosphere are not considered.

For proof of compliance with the guarantee, these factors shall be dealt with separately, unless their effect can be shown to be small enough to be neglected.

D.1.1 Essential conditions

D.1.1.1 Geometrical similarity

In most cases the machine for which the guarantee is given is the same as the one tested, so that the condition of geometrical similarity is automatically fulfilled. However, if the tests are carried out with a model compressor, strict attention shall be paid to the geometrical similarity, including the relative roughness of swept surfaces. If the compressor contains adjustable means of influencing the flow (for example, variable stator or rotor blades), conversion can only be carried out when the position of these devices is unchanged.

D.1.1.2 Equal isentropic exponents

The change of state of the medium handled can only be kept the same at all points within the compressor under test and specified conditions when the isentropic exponents are equal.

D.1.1.3 Equal enthalpy rise and flow coefficients and equal Mach numbers

Strict similarity of flow at each point within the compressor can only be achieved when the enthalpy rise and flow coefficients and the peripheral Mach numbers are equal, i.e. when

$$\Gamma = \frac{\Delta h_t}{u^2}$$

$$\Phi = \frac{q_{V,t,1}}{D^2 u}$$

$$Ma_u = \frac{u}{\sqrt{\kappa_1 R Z_1 T_{t,1}}}$$

are the same under both test and specified conditions.

However, it is quite sufficient for two of these three parameters to be equal under both test and specified conditions. In that case the third is bound to be equal too. From the measuring point of view, the simplest and most accurate parameters to maintain are the enthalpy rise coefficient, by adjusting the discharge pressure of the compressor, and the peripheral Mach number, by adjusting the speed or the intake temperature.

Fulfilment of these conditions then ensures that the Mach number and the ratios of density, velocity and temperature at any point, as related to a reference point, are the same under specified and test conditions.

If the process of compression is interrupted by one or more intercoolers, the equality of the above dimensionless coefficients shall be fulfilled for each of the groups of stages situated between successive coolers. This can be done by adjusting the speed and by controlling the intercoolers to produce equal temperature ratios on the gas side, under test and under specified conditions.

D.1.1.4 Equal Reynolds numbers

In order that the relative thickness of the boundary layer and the flow pattern influenced by it remain the same, the Reynolds number and relative roughness shall be the same under specified and test conditions.

D.1.2 Permissible approximations to the theory of similarity

D.1.2.1 In most cases all the conditions specified in D.1.1 cannot be complied with at the same time. Therefore, equality of those parameters which are only of secondary importance is dispensed with, the magnitude of their influence on the efficiency having been established under test and specified conditions.

D.1.2.2 The influence of the Reynolds number on efficiency is relatively larger at low values of Reynolds number and smaller at higher values.

The influence of a deviation in the test Reynolds number from the specified conditions is taken into account by an appropriate correction method which shall be used in a certain range of application (see 7.3.3).

D.1.2.3 The stipulation of equality for the enthalpy rise and flow coefficients and for the Mach numbers and isentropic exponents can generally be complied with merely by maintaining equality of the flow coefficients $\Phi_{\mathrm{Te}} = \Phi_{\mathrm{Gu}}$, provided that the deviation of the isentropic exponent κ or the polytropic exponent n and the factor $\dfrac{N}{\sqrt{Z_1 R T_{\mathrm{t},1}}}$ do not exceed the limits given in figures D.3 and D.4.

These limits are specified in such a manner that when $\Phi_{1\mathrm{Te}} = \Phi_{1\mathrm{Gu}}$, or

$$\left(\frac{q_{V,\mathrm{t},1}}{D^2 u}\right)_{1\,\mathrm{Te}} = \left(\frac{q_{V,\mathrm{t},1}}{D^2 u}\right)_{1\,\mathrm{Gu}}$$

the velocity relationship and thus the volume flow relationships in the compressor may differ up to a certain percentage during the test, and under the conditions on which the guarantees are based, without the efficiency η_{pol} (uncooled compressors) or η_{T} (cooled compressors) and the work input coefficient Γ being noticeably affected.

The above limits are based on the fact that the maximum deviation of the ratio of volume rate of flows $(q_{V,x}/q_{V,1})$ at any point within the compressor is not more than 1 % (inner tolerance limit) or 2,5 % or 5 % (outer tolerance limits for compressors with steep or flat characteristic respectively).

D.1.2.4 When $\kappa_{\mathrm{Te}} = \kappa_{\mathrm{Gu}}$, the largest variations in volume flow always occur at the end of the compression process. Consequently the permissible deviation is defined in respect to the ratio $q_{V,x}/q_{V,1}$ (outlet to inlet) and can be represented through the permissible deviation of $\dfrac{N}{\sqrt{Z_1 R T_{\mathrm{t},1}}}$ (see figure D.3).

In this case the ratio

$$\left(\frac{N}{\sqrt{Z_1 R T_{\mathrm{t},1}}}\right)_{\mathrm{Te}} \Bigg/ \left(\frac{N}{\sqrt{Z_1 R T_{\mathrm{t},1}}}\right)_{\mathrm{Gu}}$$

is equal to the ratio of the peripheral Mach numbers.

D.1.2.5 When $\kappa_{\mathrm{Te}} \neq \kappa_{\mathrm{Gu}}$, the requirement of equal ratio of volume rate of flows can only be fulfilled approximately, because the maximum difference in volume flows may occur somewhere inside the compressor, the change of state following a different course.

A series of calculations has been made in which the limiting values of the ratio of reduced speeds N_{r} have been established for the variation in the ratio of volume rate of flow not to exceed limits of 1 %, 2,5 % and 5 %. The results of these calculations for various polytropic exponents and various pressure ratios are presented in figure D.4.

When the isentropic exponents differ, the equal peripheral Mach numbers no longer lead to an equal ratio of volume rate of flows; they are therefore abandoned, except for cases where the performance may be directly affected by the local Mach number.

D.1.2.6 If the test conditions extend beyond the inner tolerance limit, an additional uncertainty will be introduced (see figure D.2) for the conversion to specified conditions.

If the outer tolerance limit is still not sufficient, it will be necessary to check whether there is any point in carrying out the test under such circumstances. At any rate a higher additional uncertainty shall then be agreed for the conversion.

In the case of compressors operating with local velocities close to the velocity of sound, it is necessary to check whether the deviation of Mach numbers on test from those at specified conditions is still permissible.

For this of course the local Mach number (ratio of local velocity of flow to local velocity of sound) shall be taken into account instead of the peripheral Mach number.

For cooled compressors it may be necessary to check the limits of tolerance for the conversion separately for the cooled and the uncooled sections.

D.2 Additional uncertainties resulting from deviations in the ratio of volume flows

Under test conditions the ratio of volume flows $(q_{V,\mathrm{t},x}/q_{V,\mathrm{t},1})_{\mathrm{Te}}$ may differ from the specified value $(q_{V,\mathrm{t},2}/q_{V,\mathrm{t},1})_{\mathrm{Gu}}$ by up to 1 % without any additional uncertainty being applicable to the result of the conversion to specified conditions (inner tolerance limit, see D.1.2.3).

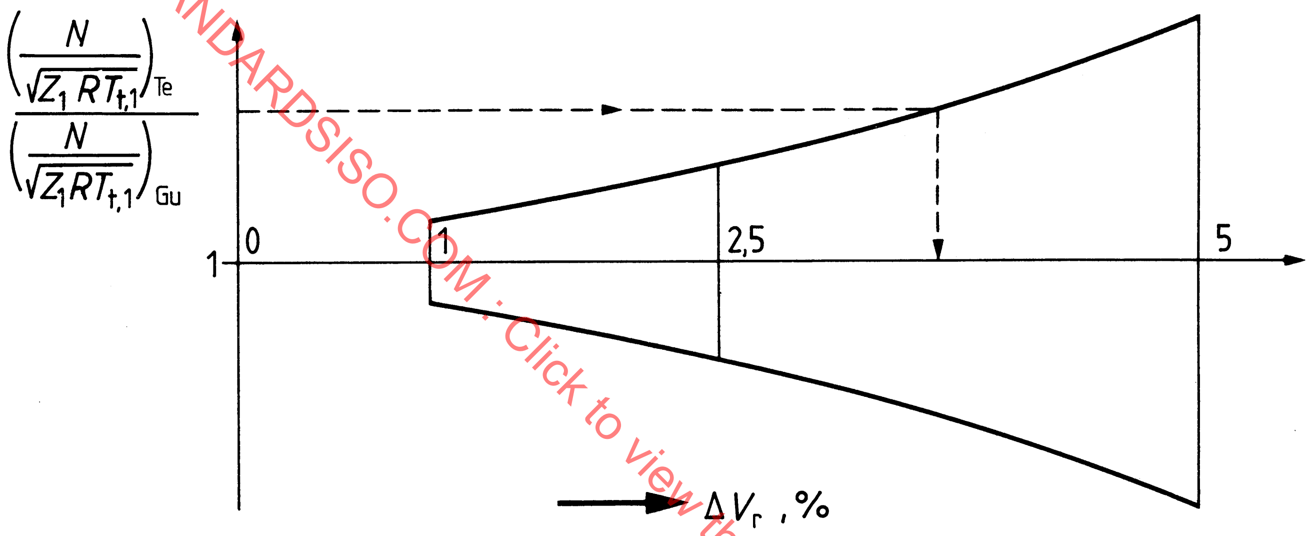
If the conditions of the test do not allow the ratio of reduced speeds

$$N_r = \left(\frac{N}{\sqrt{Z_1 R T_{t,1}}}\right)_{Te} \Big/ \left(\frac{N}{\sqrt{Z_1 R T_{t,1}}}\right)_{Gu}$$

to be chosen within the inner tolerance limit (see figures D.3 and D.4), an additional uncertainty shall be applied to account for the inaccuracies introduced when converting to specified conditions.

This additional uncertainty is determined as follows.

For the given pressure ratio and polytropic exponents $n_{M,Te}$ and $n_{M,Gu}$, read the upper and lower limits for the admissible ratio of reduced speeds N_r from the corresponding charts in clause D.3, at the inner and outer tolerance limits. When the characteristic curve is flat (see 7.3.2), allowing a deviation of 5 % in the ratio of volume flows at the outer tolerance limit, the admissible values of N_r should be determined for the three tolerance limits ± 1 %, ± 2,5 % and ± 5 %. These values are plotted on a diagram as shown in figure D.1. This diagram, which has to be drawn for each occasion, gives the deviation ΔV_r from the specified ratio of volume flows corresponding to the test value of N_r.



NOTE — With the value of ΔV_r obtained, the additional uncertainty, τ_{adj}, on the volume flow, energy consumption and pressure ratio can be taken from figure D.2.

Figure D.1 — Deviation, ΔV_r, as a function of N_r

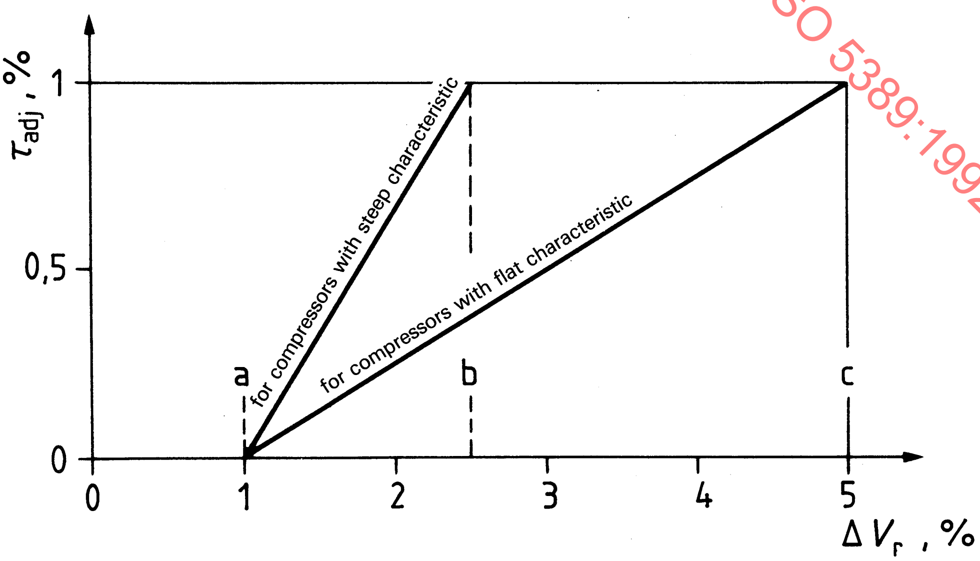


Figure D.2 — Additional uncertainty, τ_{adj}

D.3 Permissible limits for the conversion from test to guarantee conditions

NOTE — When test conditions are such that the range of the diagrams in figures D.4 and D.5 is exceeded, use the flow charts in figures D.6 and D.7 to calculate $N_{r,\,tol}$.

Figures D.3 and D.4 are taken from [53].

Reynolds number corrections are not incorporated in figures D.3 and D.4.

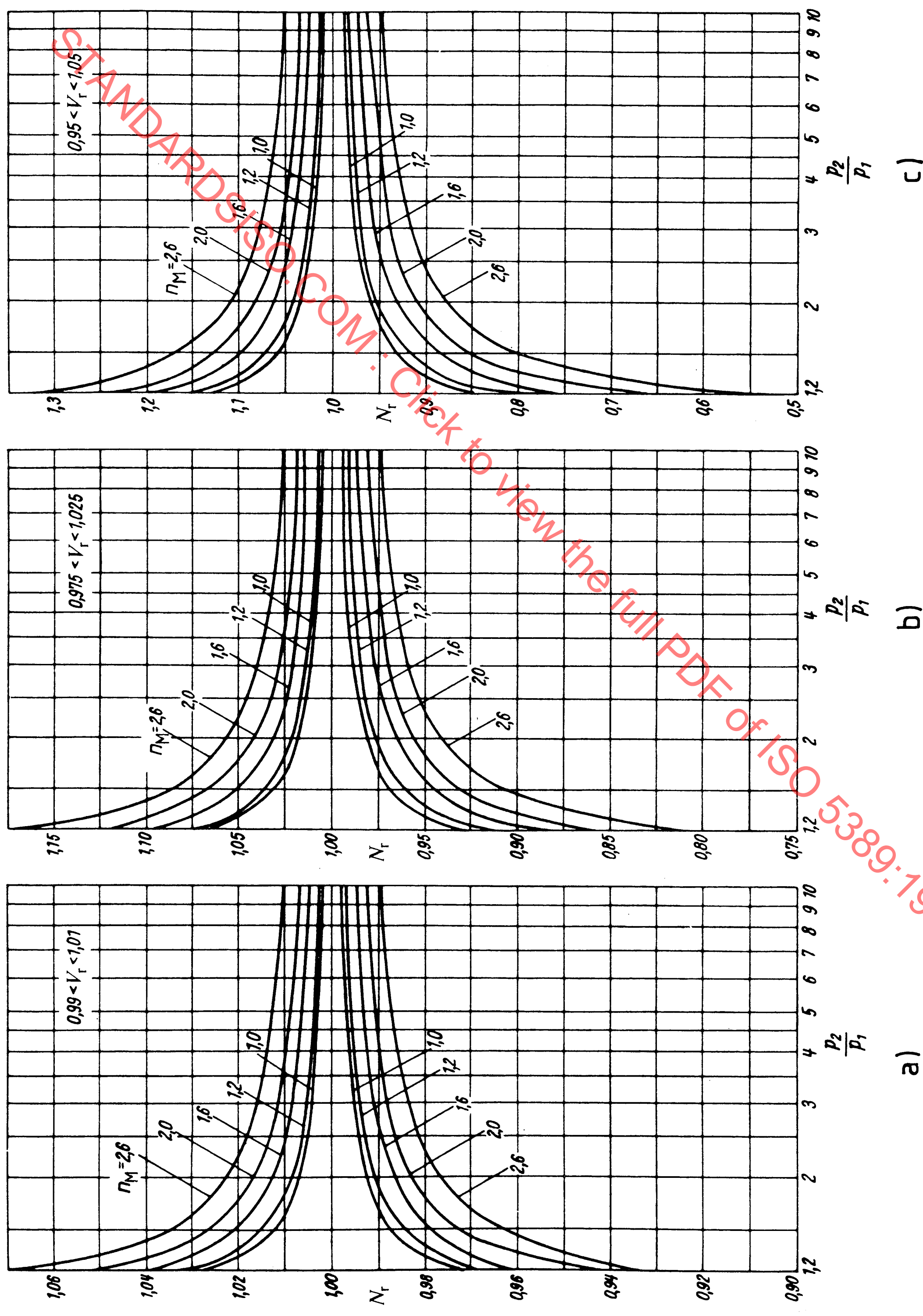


Figure D.3 — Permissible limits for the conversion from test to guarantee conditions ($\kappa_{Te} = \kappa_{Gu}$)

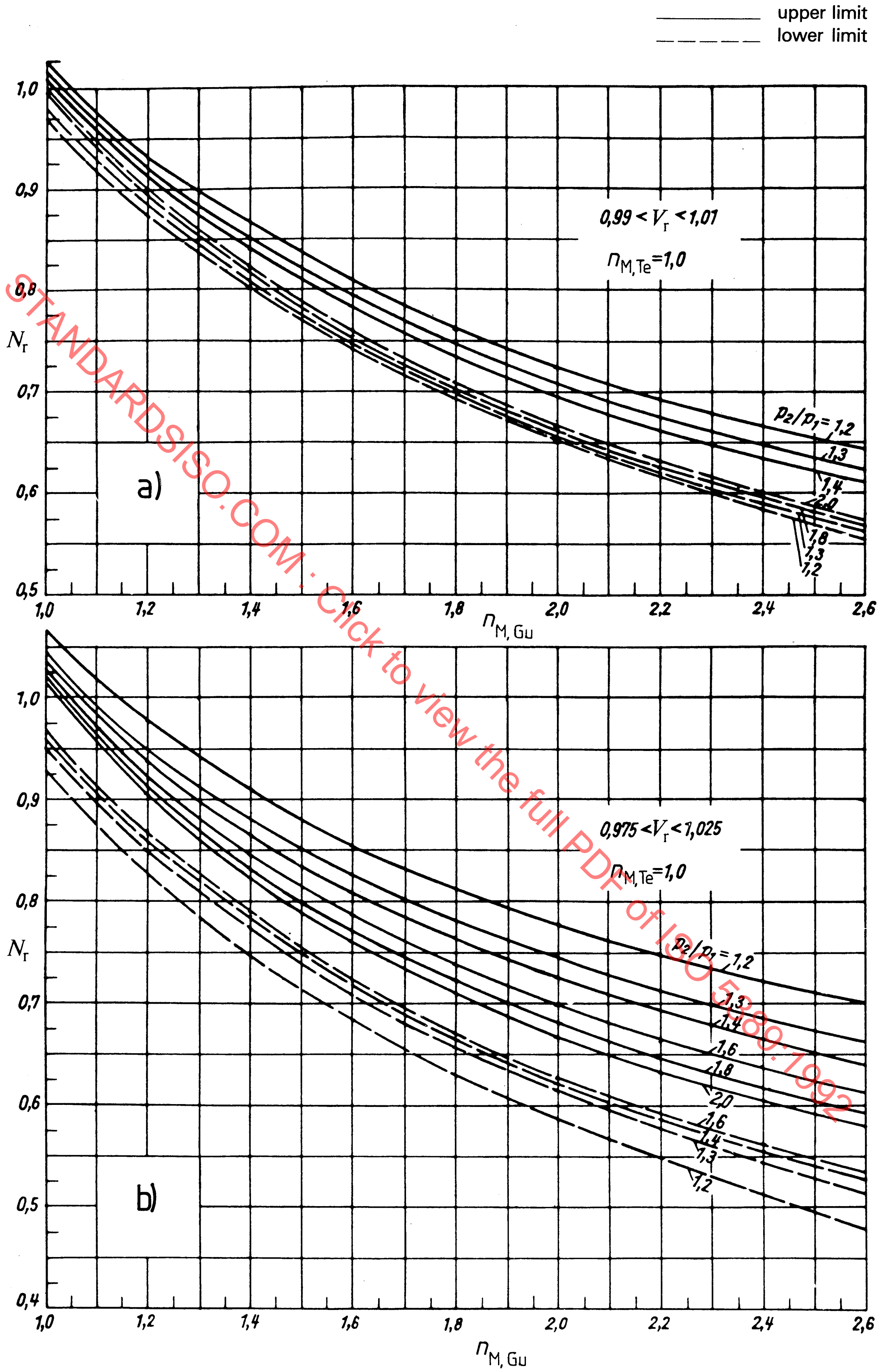


Figure D.4 — Permissible limits for the conversion from test to guarantee conditions ($\kappa_{Te} \neq \kappa_{Gu}$)

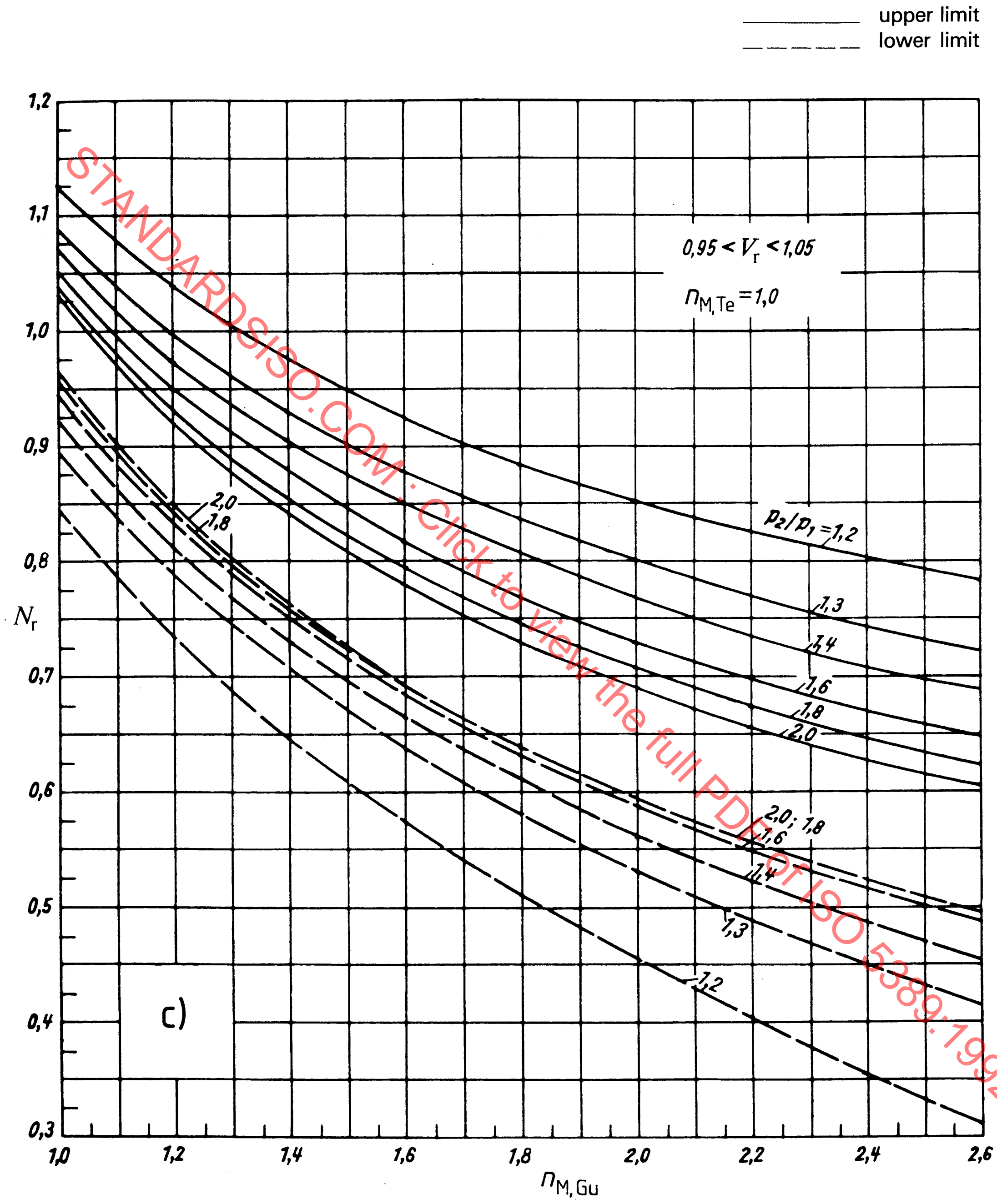


Figure D.4 — Permissible limits for the conversion from test to guarantee conditions ($\kappa_{Te} \neq \kappa_{Gu}$) (*continued*)

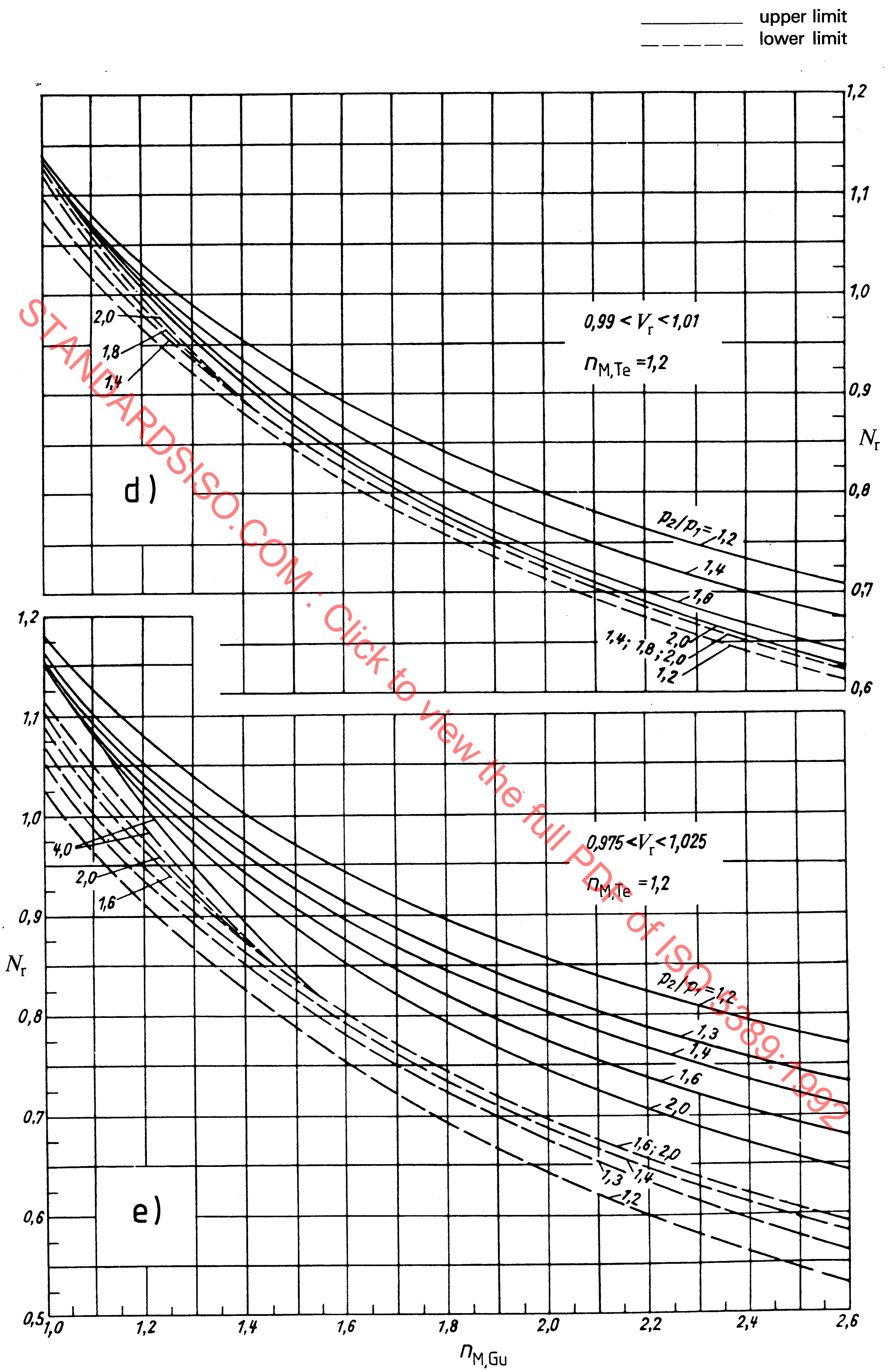


Figure D.4 — Permissible limits for the conversion from test to guarantee conditions ($\kappa_{Te} \neq \kappa_{Gu}$) (*continued*)

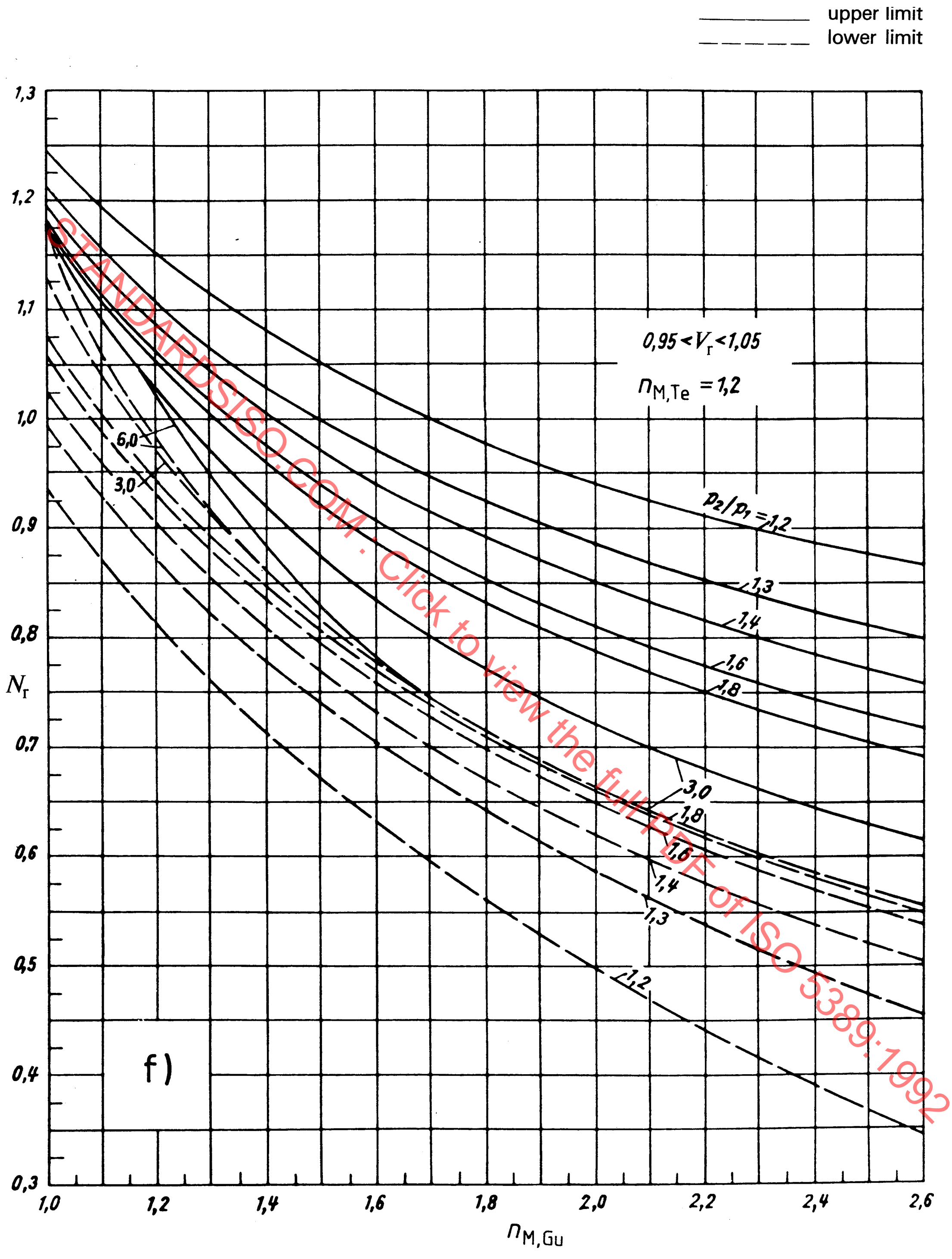


Figure D.4 — Permissible limits for the conversion from test to guarantee conditions ($\kappa_{Te} \neq \kappa_{Gu}$) (*continued*)

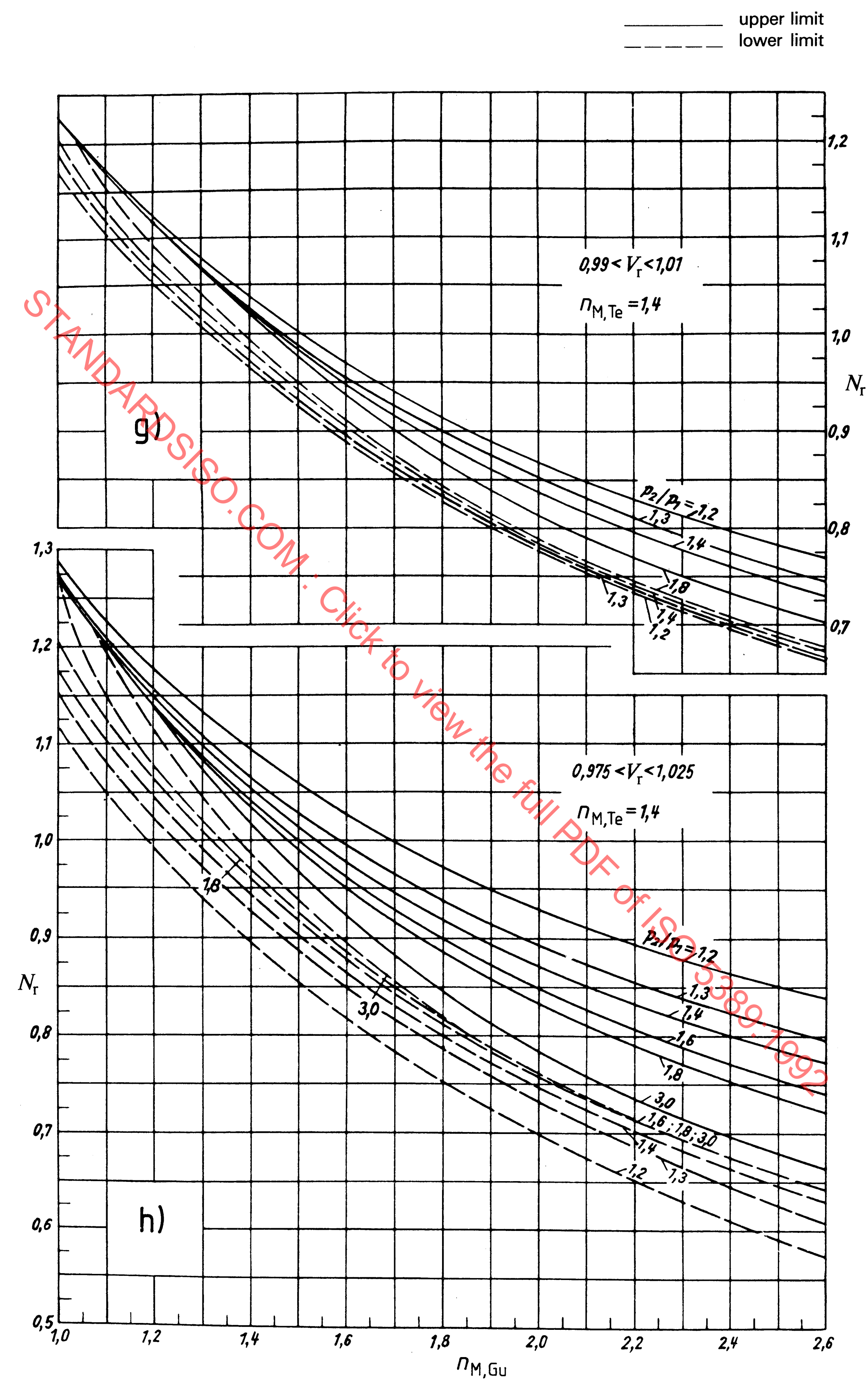


Figure D.4 — Permissible limits for the conversion from test to guarantee conditions ($\kappa_{Te} \neq \kappa_{Gu}$) *(continued)*

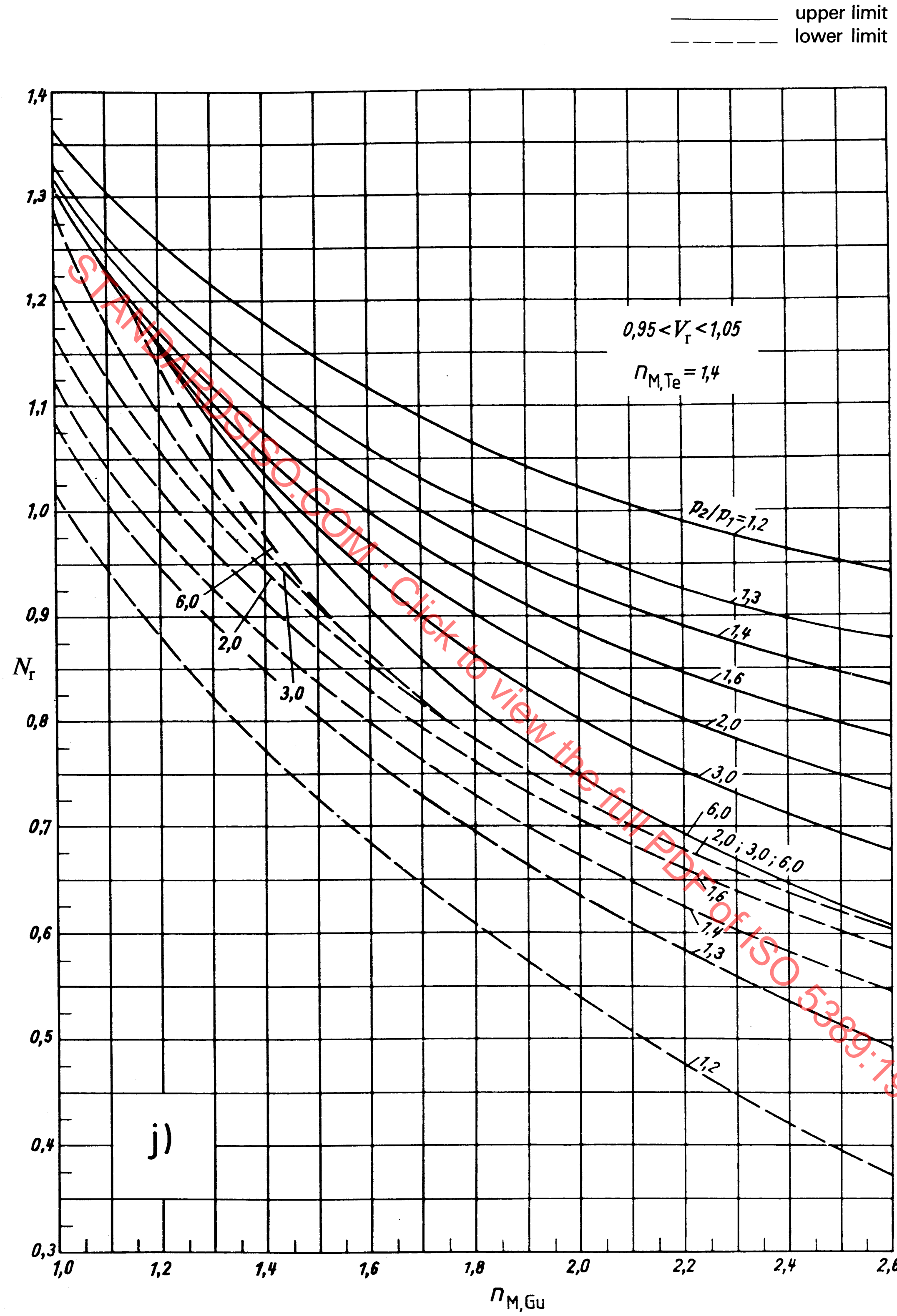


Figure D.4 — Permissible limits for the conversion from test to guarantee conditions ($\kappa_{Te} \neq \kappa_{Gu}$) (*continued*)

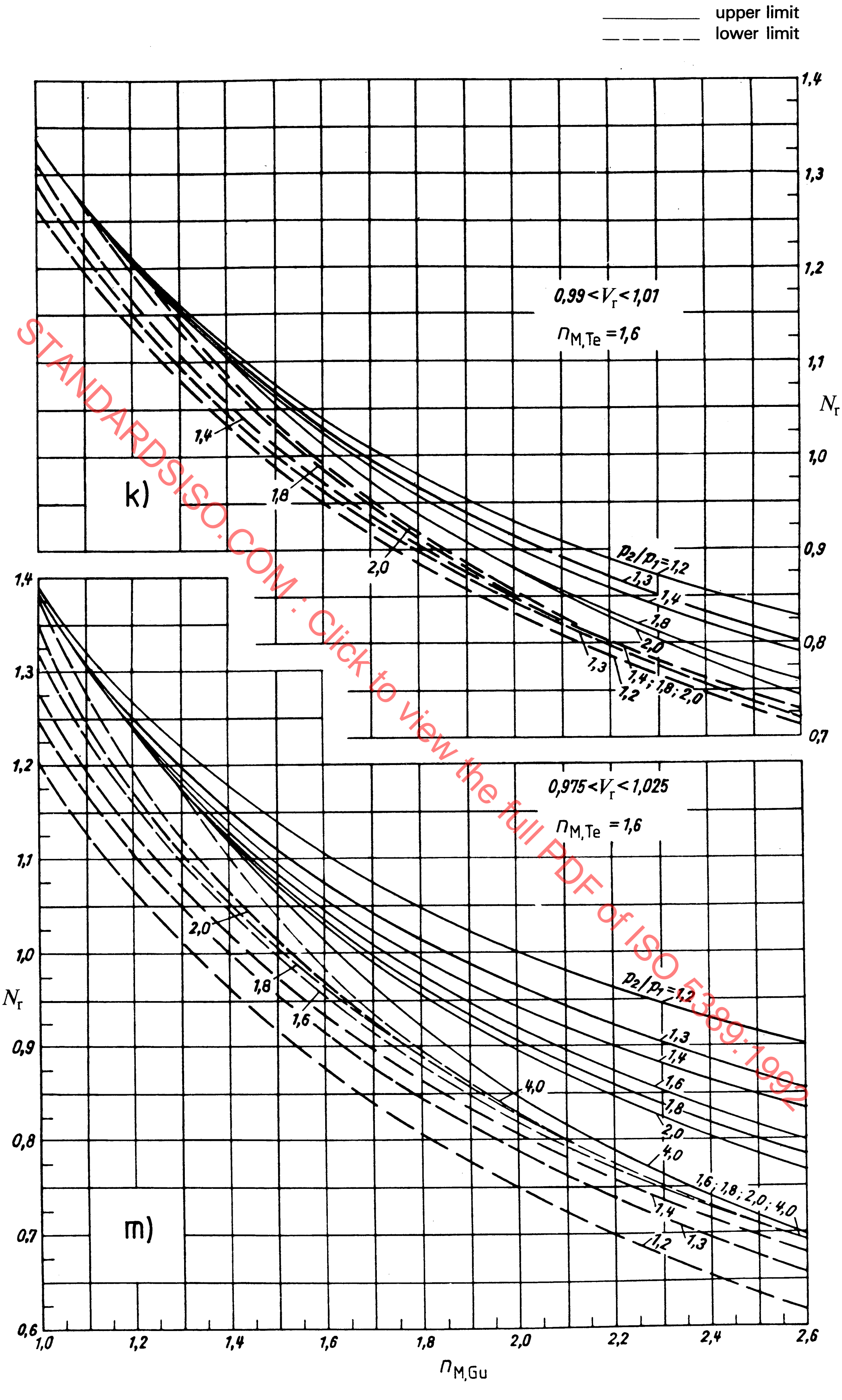


Figure D.4 — Permissible limits for the conversion from test to guarantee conditions ($\kappa_{Te} \neq \kappa_{Gu}$) *(continued)*

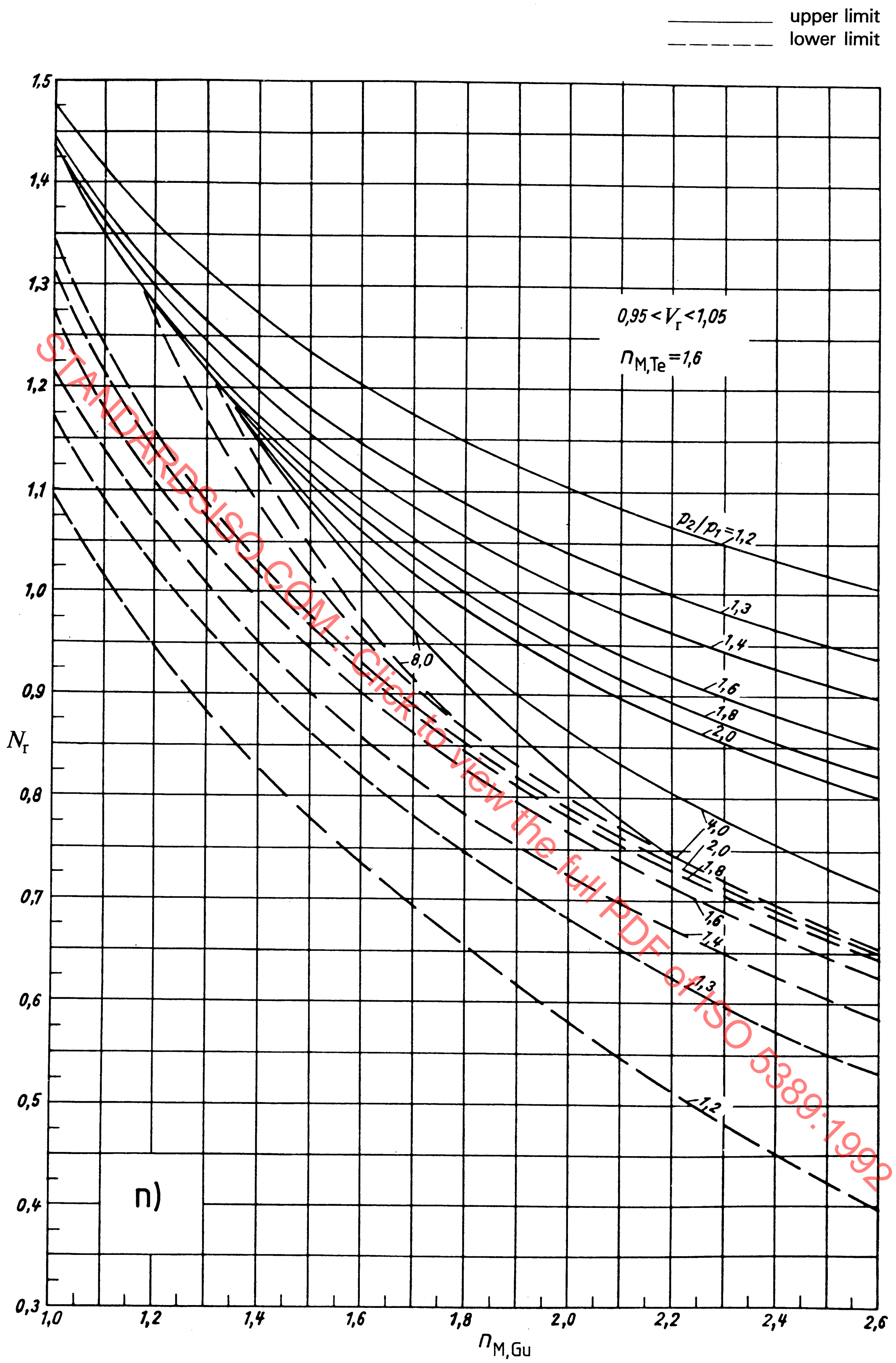


Figure D.4 — Permissible limits for the conversion from test to guarantee conditions ($\kappa_{Te} \neq \kappa_{Gu}$) (*continued*)

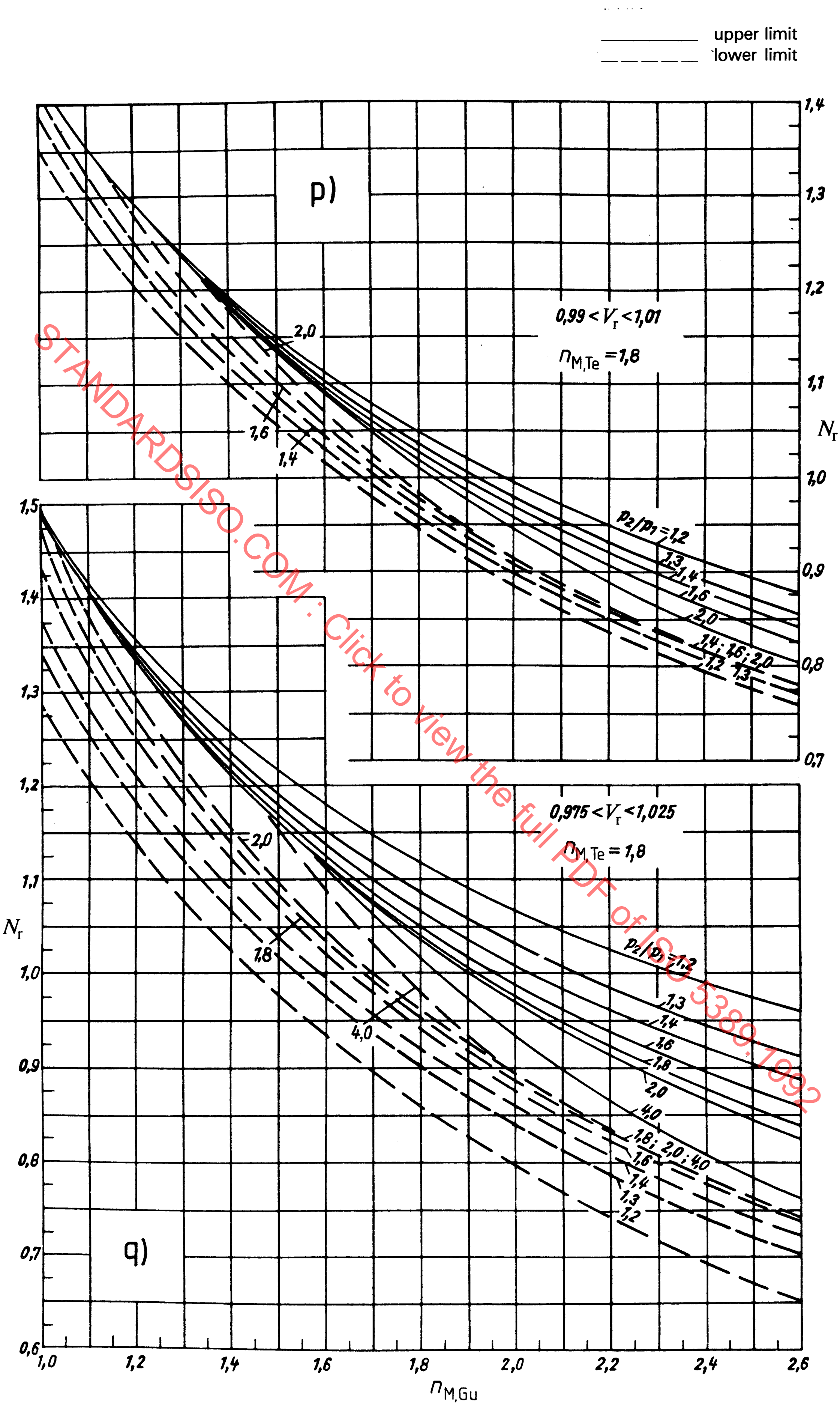


Figure D.4 — Permissible limits for the conversion from test to guarantee conditions ($\kappa_{Te} \neq \kappa_{Gu}$) (*continued*)

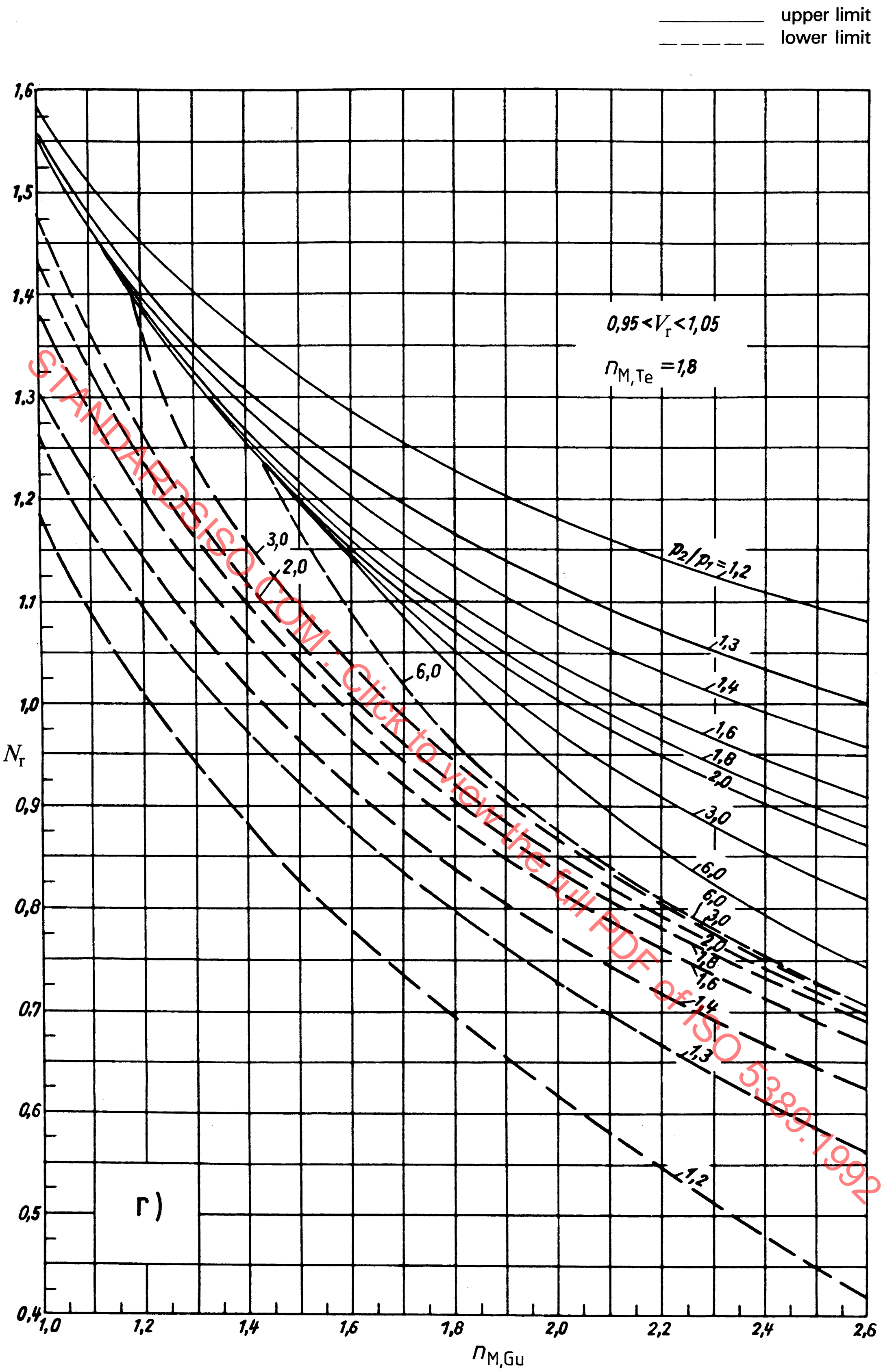


Figure D.4 — Permissible limits for the conversion from test to guarantee conditions ($\kappa_{Te} \neq \kappa_{Gu}$) (*continued*)

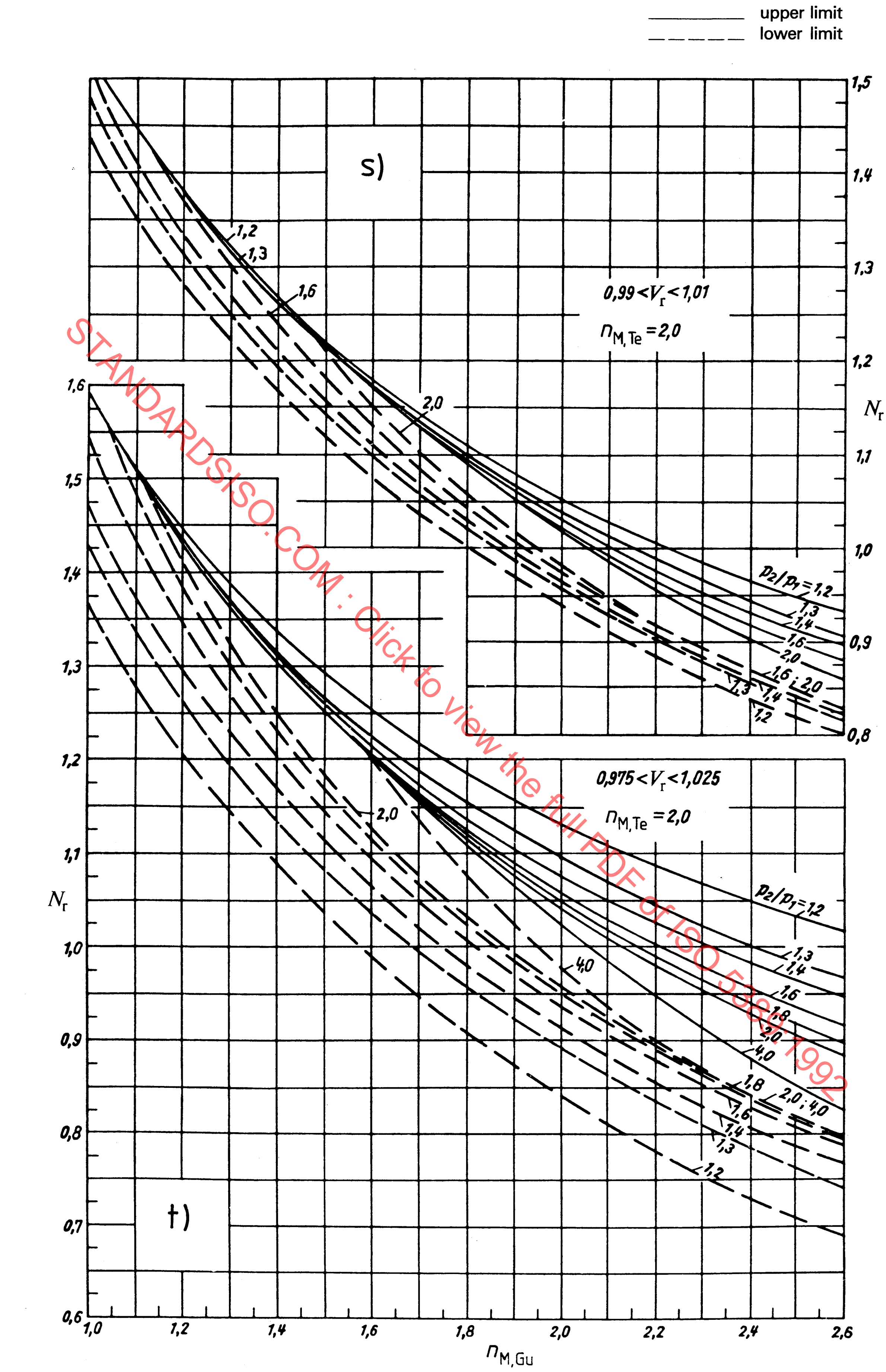


Figure D.4 — Permissible limits for the conversion from test to guarantee conditions ($\kappa_{Te} \neq \kappa_{Gu}$) (*continued*)

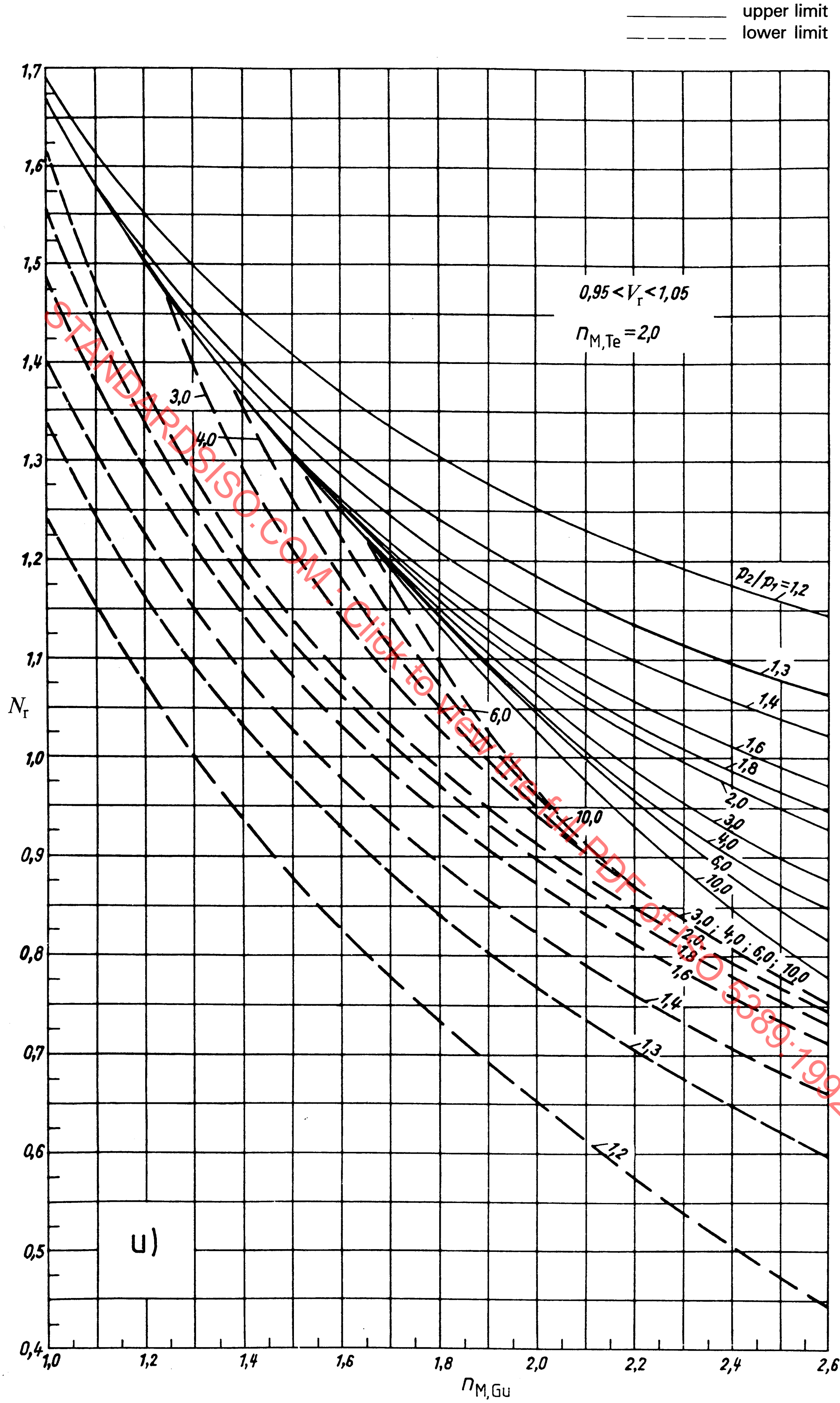


Figure D.4 — Permissible limits for the conversion from test to guarantee conditions ($\kappa_{Te} \neq \kappa_{Gu}$) *(continued)*

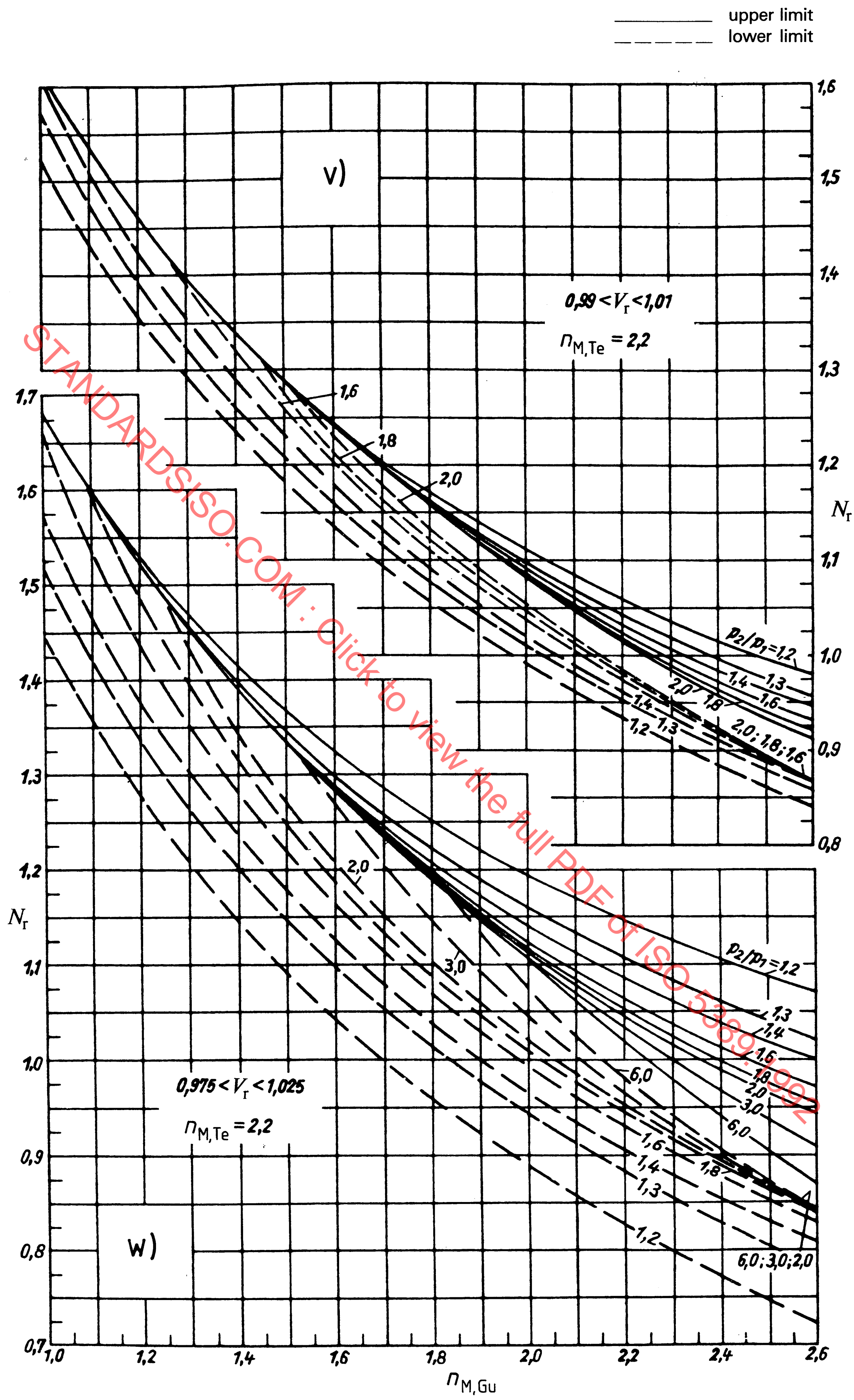


Figure D.4 — Permissible limits for the conversion from test to guarantee conditions ($\kappa_{Te} \neq \kappa_{Gu}$) *(continued)*

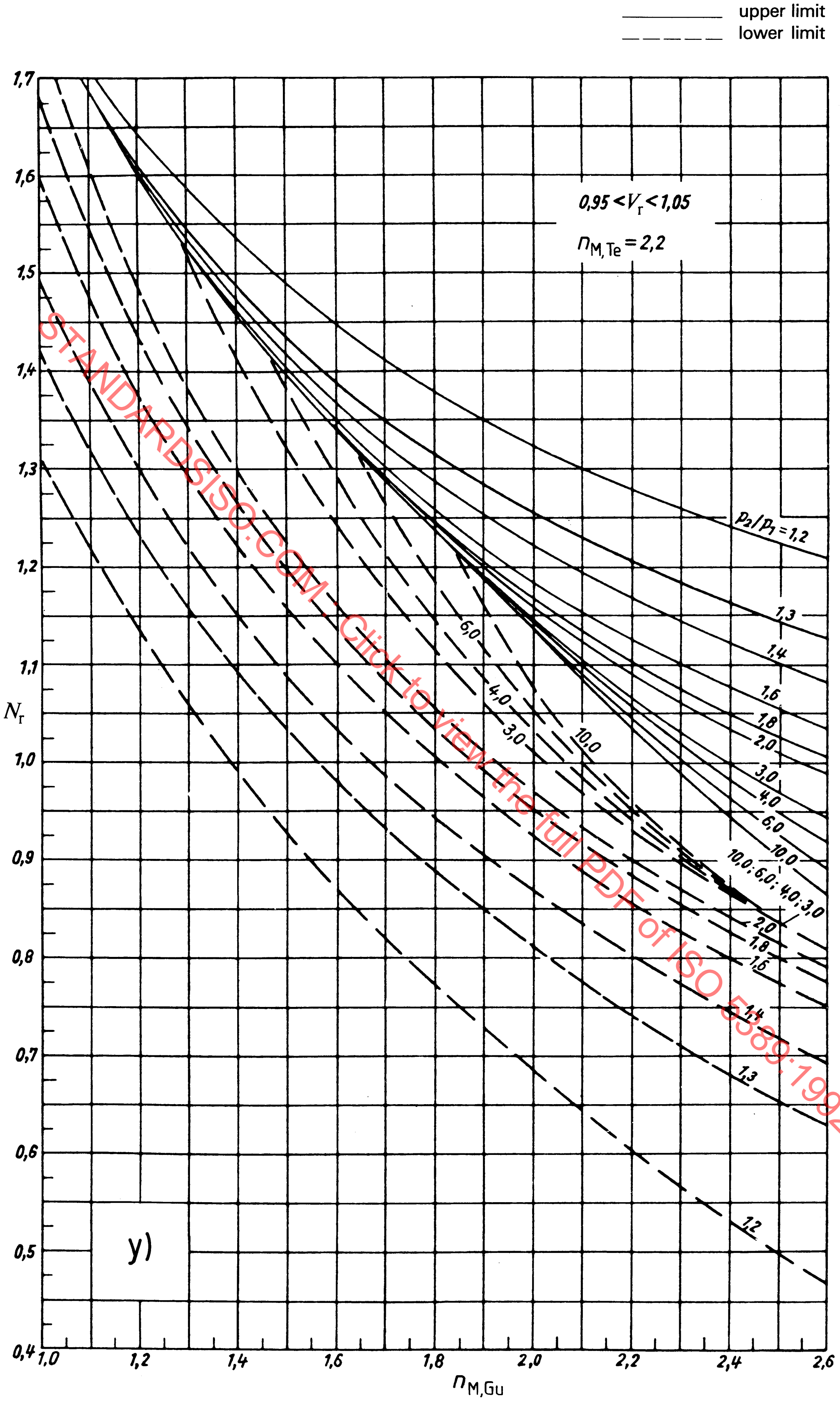


Figure D.4 — Permissible limits for the conversion from test to guarantee conditions ($\kappa_{Te} \neq \kappa_{Gu}$) (*continued*)

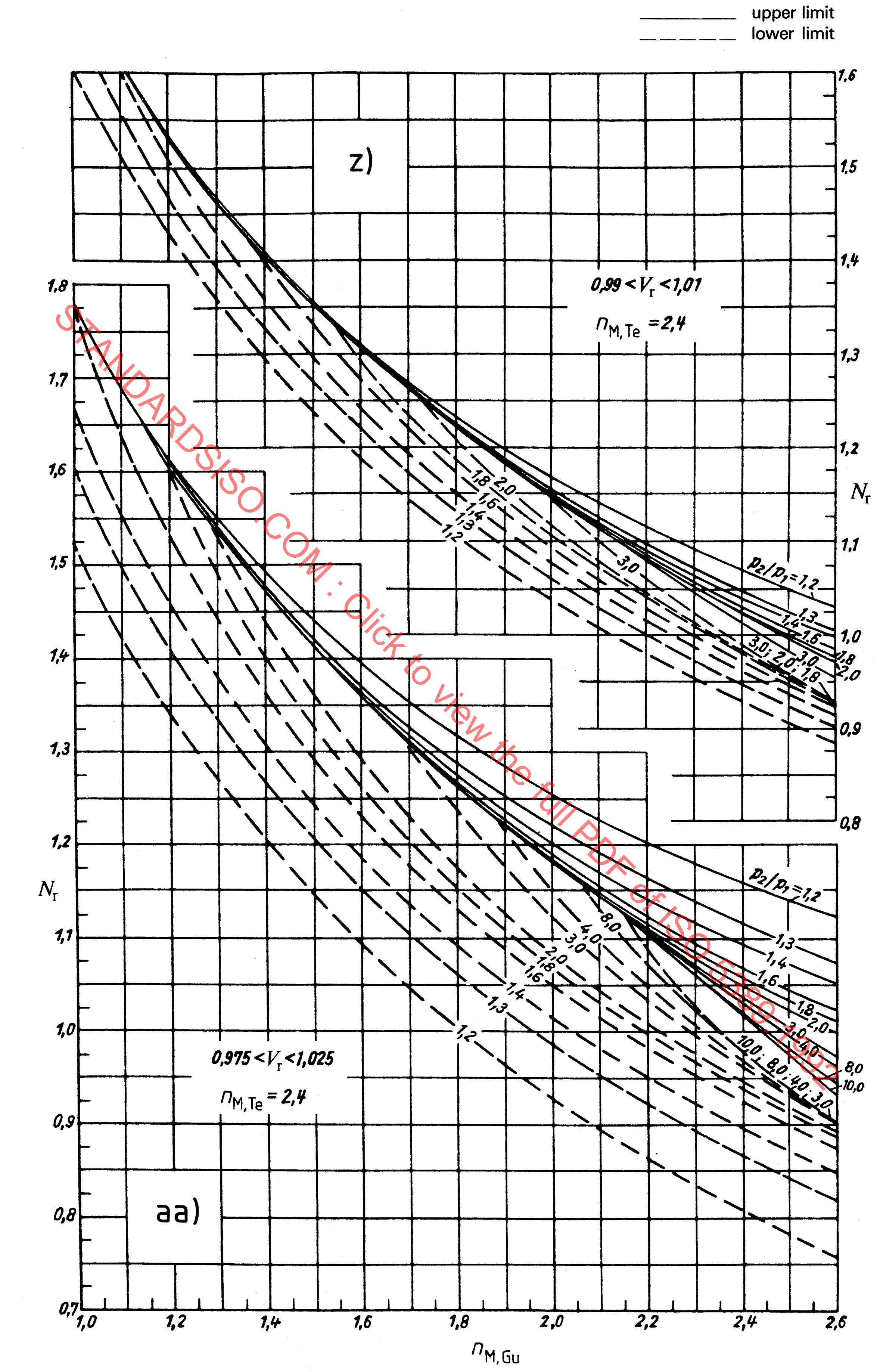


Figure D.4 — Permissible limits for the conversion from test to guarantee conditions ($\kappa_{Te} \neq \kappa_{Gu}$) *(continued)*

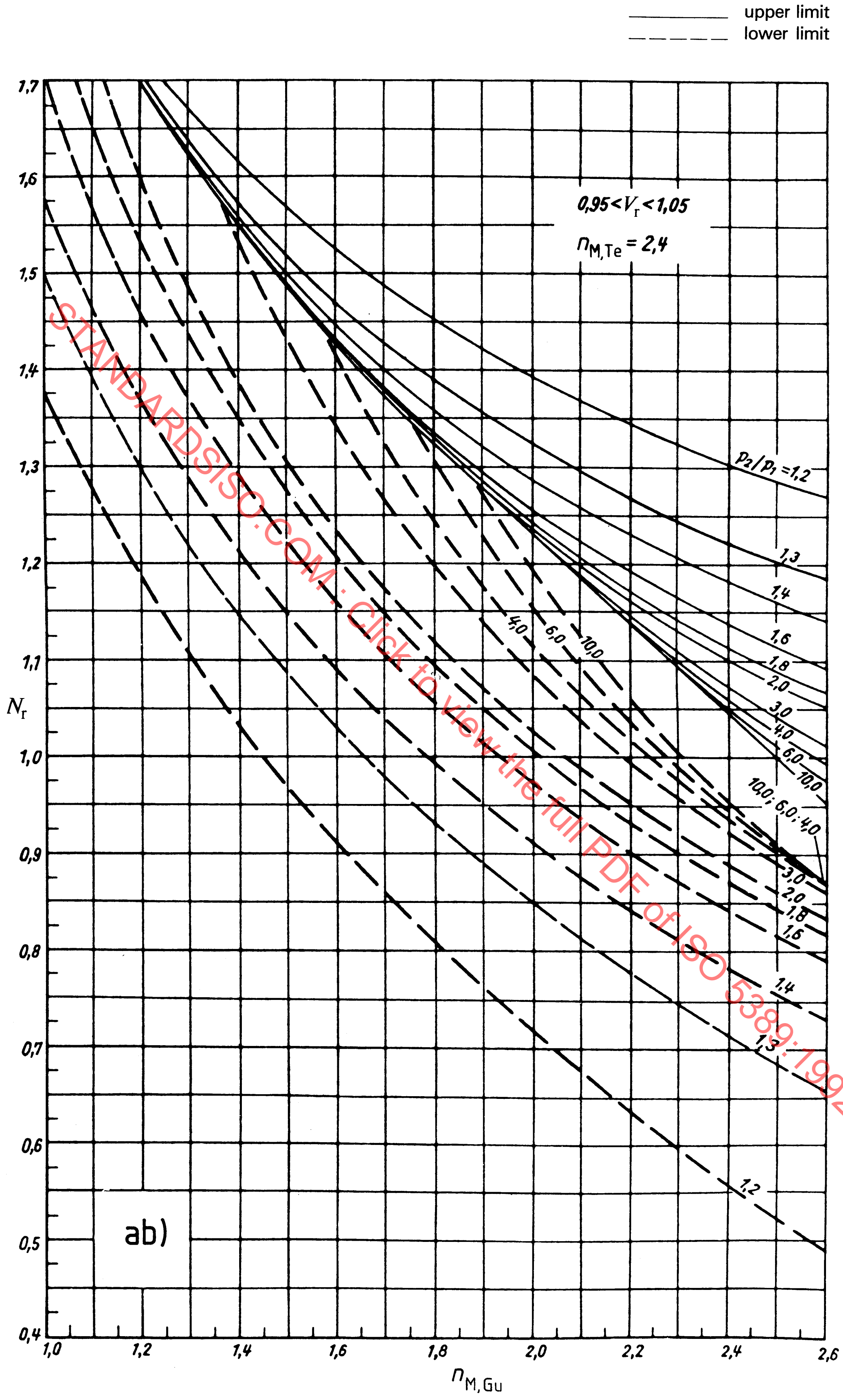


Figure D.4 — Permissible limits for the conversion from test to guarantee conditions ($\kappa_{Te} \neq \kappa_{Gu}$) (*continued*)

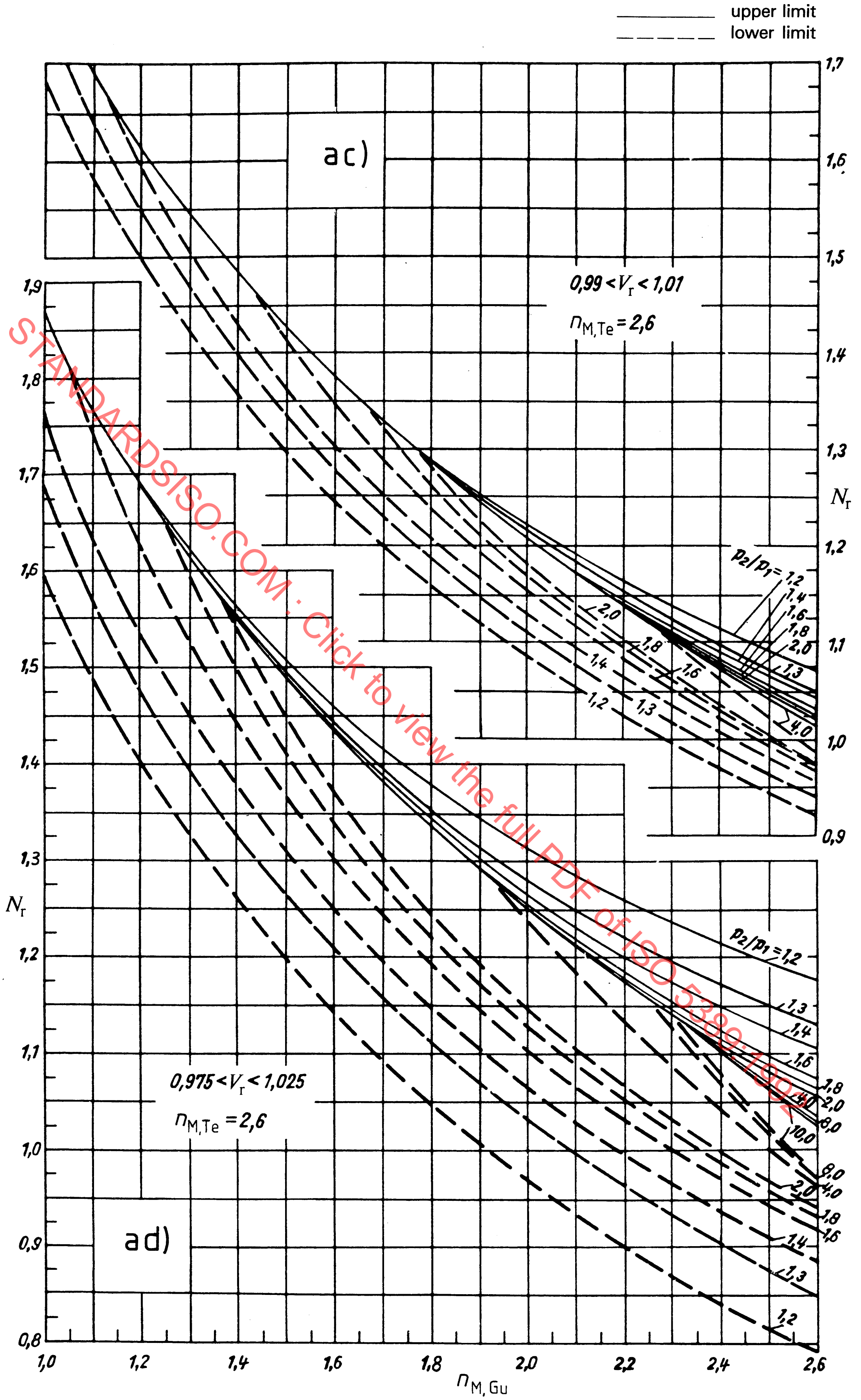


Figure D.4 — Permissible limits for the conversion from test to guarantee conditions ($\kappa_{Te} \neq \kappa_{Gu}$) (*continued*)

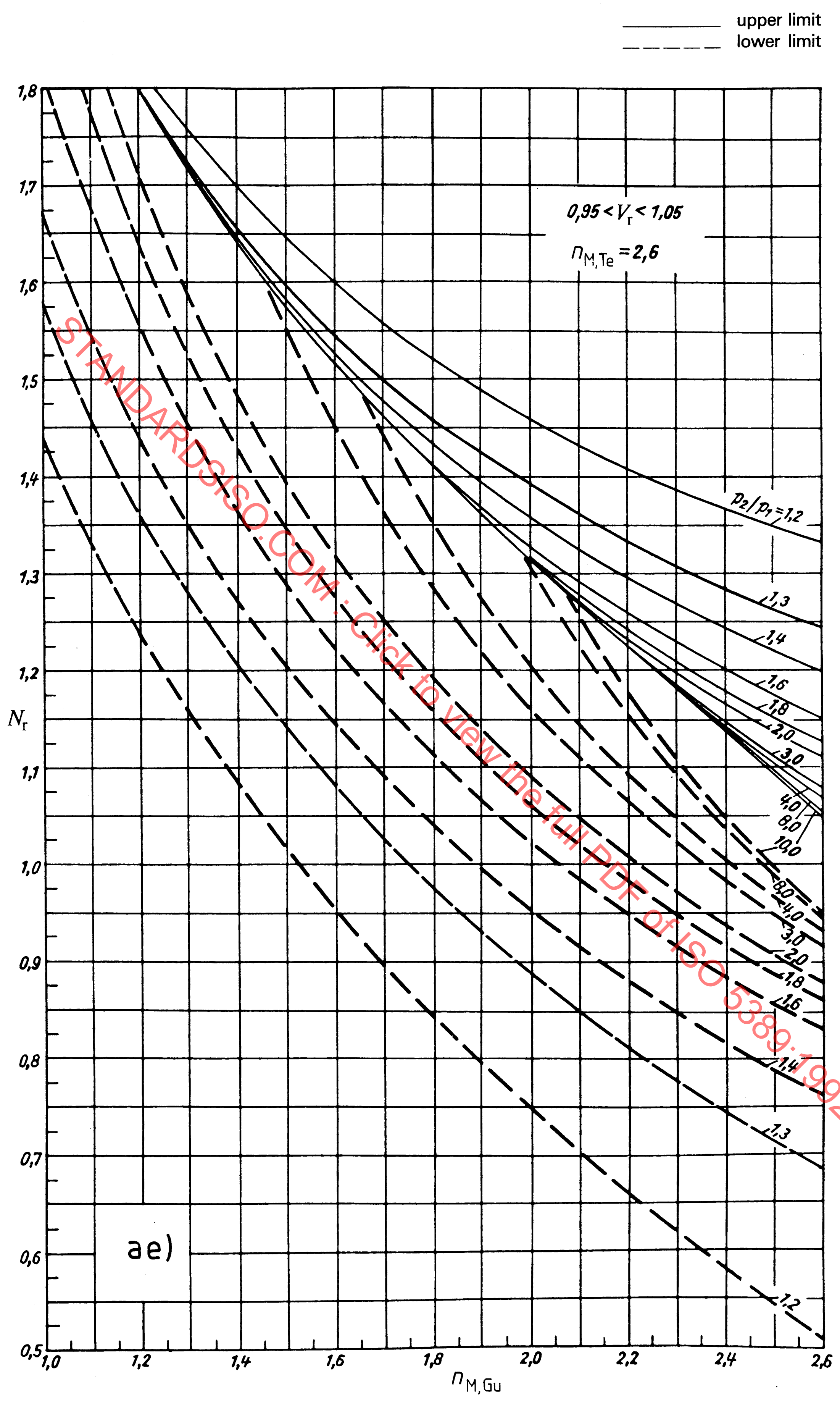


Figure D.4 — Permissible limits for the conversion from test to guarantee conditions ($\kappa_{Te} \neq \kappa_{Gu}$) (*concluded*)

D.4 Permissible deviation of the peripheral Reynolds number

Figure D.5 (taken from [54]) shows the permissible deviations in the peripheral Reynolds number of centrifugal compressors for the application of the correction method (see annex E).

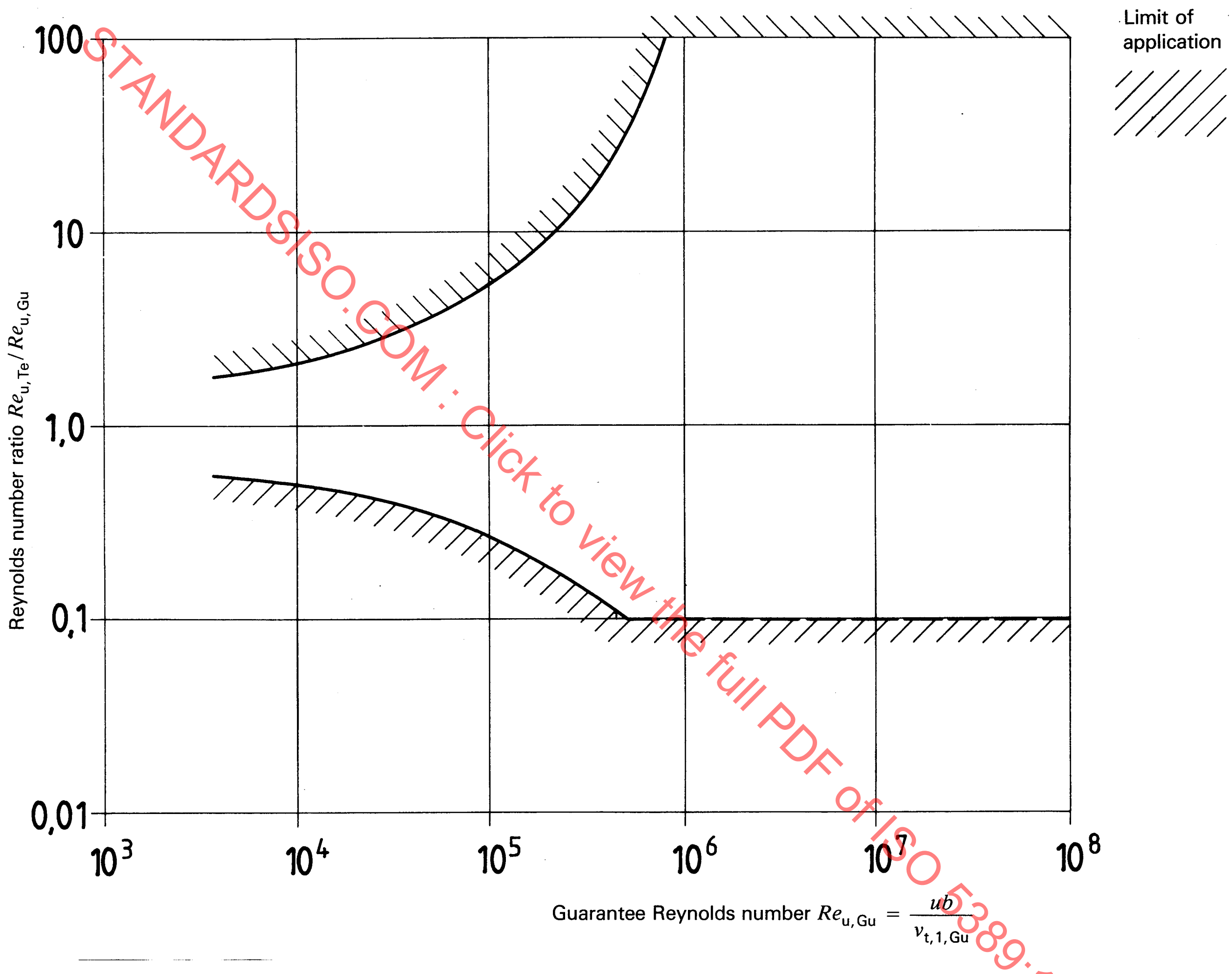


Figure D.5 — Allowable range of application of the correction method

D.5 Determination of settings

D.5.1 Input values for flow charts in figures D.6 and D.7

D.5.1.1 General

Unless otherwise agreed between manufacturer and customer all pressures and temperatures refer to total conditions (see 6.1.1.3).

NOTE — For simplicity, the subscript t has been omitted when referring to total pressures and temperatures in figures D.6 to D.11.

b is the rotor reference outlet tip width.

$\Delta V_{r,x,\mathrm{tol}}$ is the permissible deviation of the ratio of the volume rates of flow from 1.

$\left(\dfrac{Re_{u,Te}}{Re_{u,Gu}}\right)_{\mathrm{tol}} = f(Re_{u,Gu})$ (for centrifugal compressors) (see figure D.5), or any other agreement between manufacturer and customer.

For real gases,

$X = f(p_r, T_r)$ compressibility function

$Y = f(p_r, T_r)$ compressibility function

(See clause A.3.)

Reynolds number corrections are not incorporated in the flow charts and formulae.

D.5.1.2 Test data

At compressor inlet:

$p_{t,1,Te}$	pressure
$T_{t,1,Te}$	temperature
$\kappa_{1,Te}$	isentropic exponent
R_{Te}	gas constant
$\varphi_{1,Te}$	relative humidity
$\mu_{Te} = f(p_{Te}, T_{Te})$	dynamic viscosity
N_{max}	maximum adjustable speed
N_{min}	minimum adjustable speed

For stages following intercoolers:

$T_{t,1,Te}$	temperature at stage inlet

Additional input for real gases:

$p_{Cr,Te}$	critical pressure of test gas
$T_{Cr,Te}$	critical temperature of test gas

and, if available,

$Z_{Te} = f(p, T)$	compressibility factor
$\left.\begin{matrix} X_{M,Te} \\ Y_{M,Te} \end{matrix}\right\}$	compressibility functions
$\kappa_{Te} = f(p, T)$	isentropic exponent

D.5.1.3 Specified values

Input for all stage groups:

$p_{t,1,Gu}$	inlet pressure
$T_{t,1,Gu}$	inlet temperature
$p_{t,2,Gu}$	discharge pressure
$T_{t,2,Gu}$	discharge temperature
$\kappa_{1,Gu}$	isentropic exponent at inlet conditions
$\varphi_{1,Gu}$	relative humidity, inlet conditions
R_{Gu}	gas constant
N_{Gu}	speed

Additional input for real gases:

$Z_{M,Gu}$	compressibility factor
$\left.\begin{matrix} X_{M,Gu} \\ Y_{M,Gu} \end{matrix}\right\}$	compressibility functions
$\kappa_{M,Gu}$	isentropic exponent

or

$p_{Cr,Gu}$	critical pressure of specified gas
$T_{Cr,Gu}$	critical temperature of specified gas

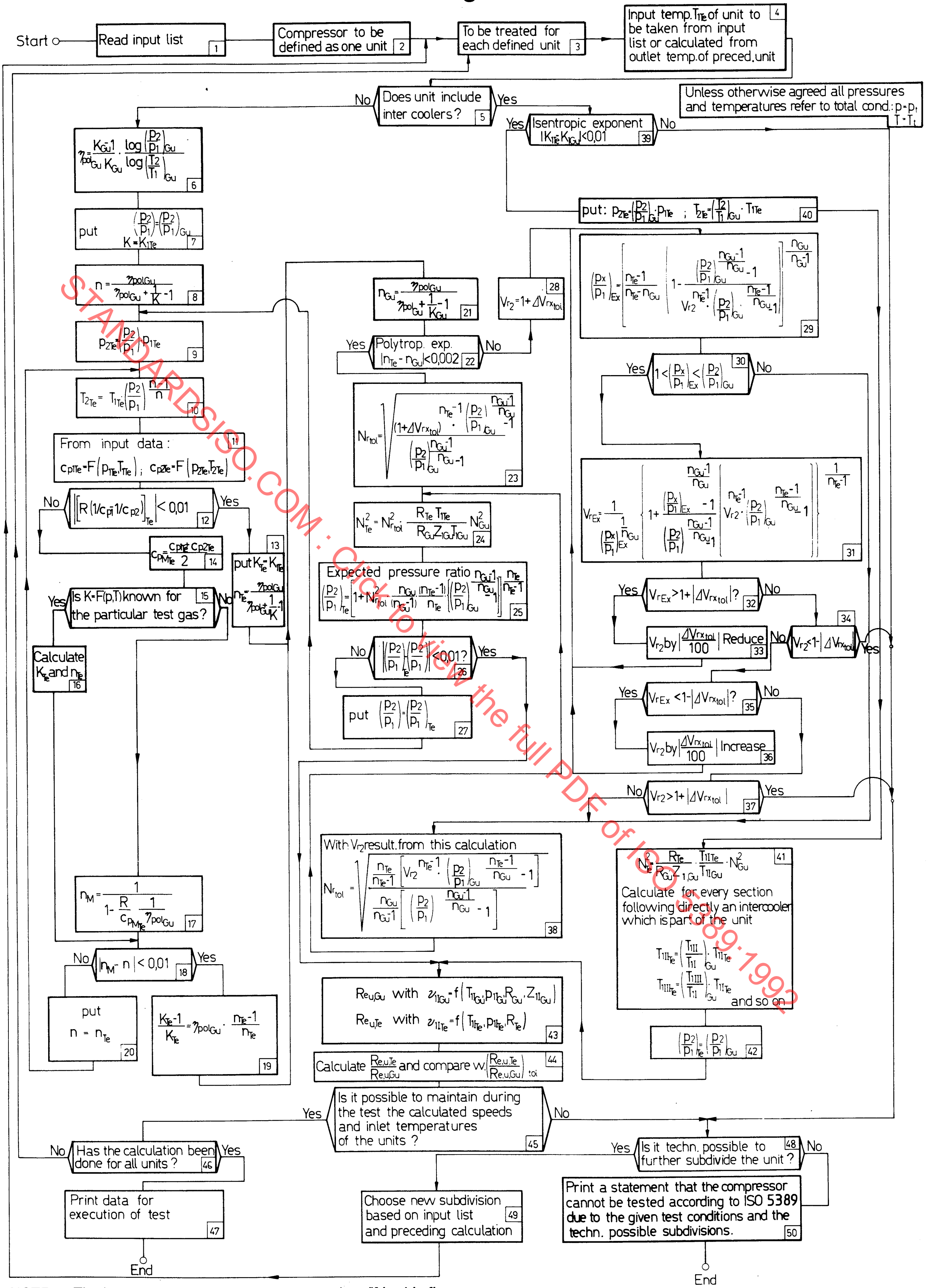
D.5.2 Perfect or near-perfect gases

NOTE — See table 2 for suggested limits within which the gas may be considered to be near perfect.

D.5.2.1 Test gas

The compressibility factor $Z_{1,Te} = Z_{2,Te} = 1$.

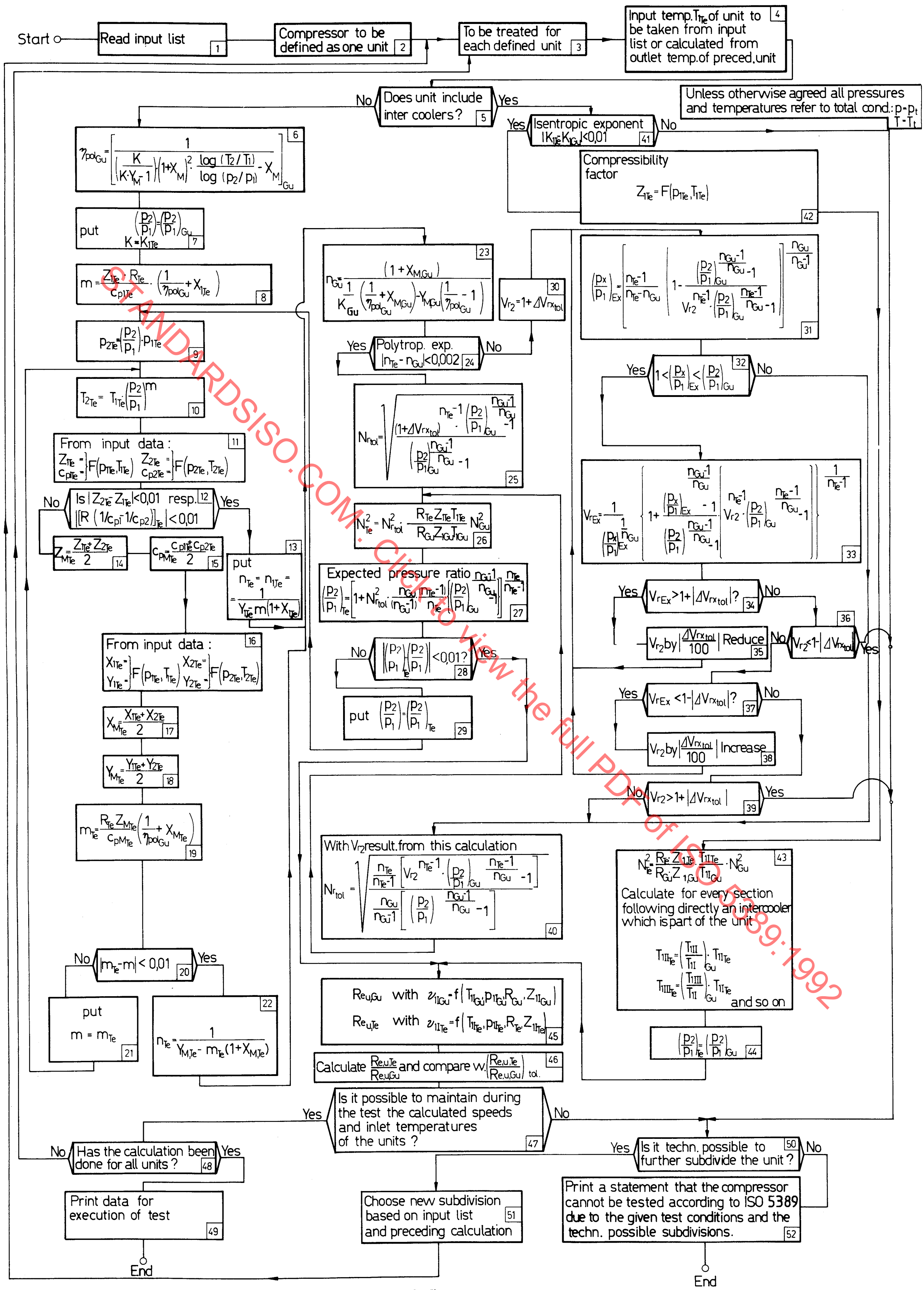
Determination of the settings : Perfect and near-perfect gases



NOTE — The isentropic exponent κ is represented as K in this figure.

Figure D.6 — Flow chart for the determination of settings in the case of perfect and near-perfect gases

Determination of the settings : Real gases



NOTE — The isentropic exponent κ is represented as K in this figure.

Figure D.7 — Flow chart for the determination of settings in the case of real gases

In many cases the isentropic exponent can be considered to be constant, i.e.

$$\kappa_{\mathrm{Te}} = \kappa_{1,\mathrm{Te}} = \kappa_{2,\mathrm{Te}}$$

The settings may be determined by following the flow chart given in figure D.6.

D.5.2.2 Specified gas

If the specified gas is a perfect or near-perfect gas and the discharge temperature $T_{2,\mathrm{Gu}}$ is not specified, the polytropic efficiency can be calculated from the internal power figure according to the following equation:

$$\eta_{\mathrm{pol,Gu}} = \frac{(\kappa - 1)_{\mathrm{M,Gu}}}{\kappa_{\mathrm{M,Gu}}} \times \frac{\log (p_2/p_1)_{\mathrm{Gu}}}{\log \left\{ 1 + \dfrac{(p_2/p_1)_{\mathrm{Gu}}^{[(\kappa-1)/\kappa]_{\mathrm{M,Gu}}} - 1}{\eta_{\mathrm{s,Gu}}} \right\}}$$

where $\eta_{\mathrm{s,Gu}}$ is the isentropic efficiency given by the formula

$$\eta_{\mathrm{s,Gu}} = \left(\frac{q_{m,\mathrm{ut}} W_{m,\mathrm{s}}}{P_{\mathrm{in}}} \right)_{\mathrm{Gu}}$$

The above applies to step 6 in the flow chart given in figure D.6.

D.5.3 Real gases

The test gas is a real gas.

The settings may be determined by following the flow chart given in figure D.7.

D.6 Conversion to guarantee conditions for uncooled compressors

Reynolds number corrections are not incorporated in the formulae.

D.6.1 Perfect or near-perfect gases

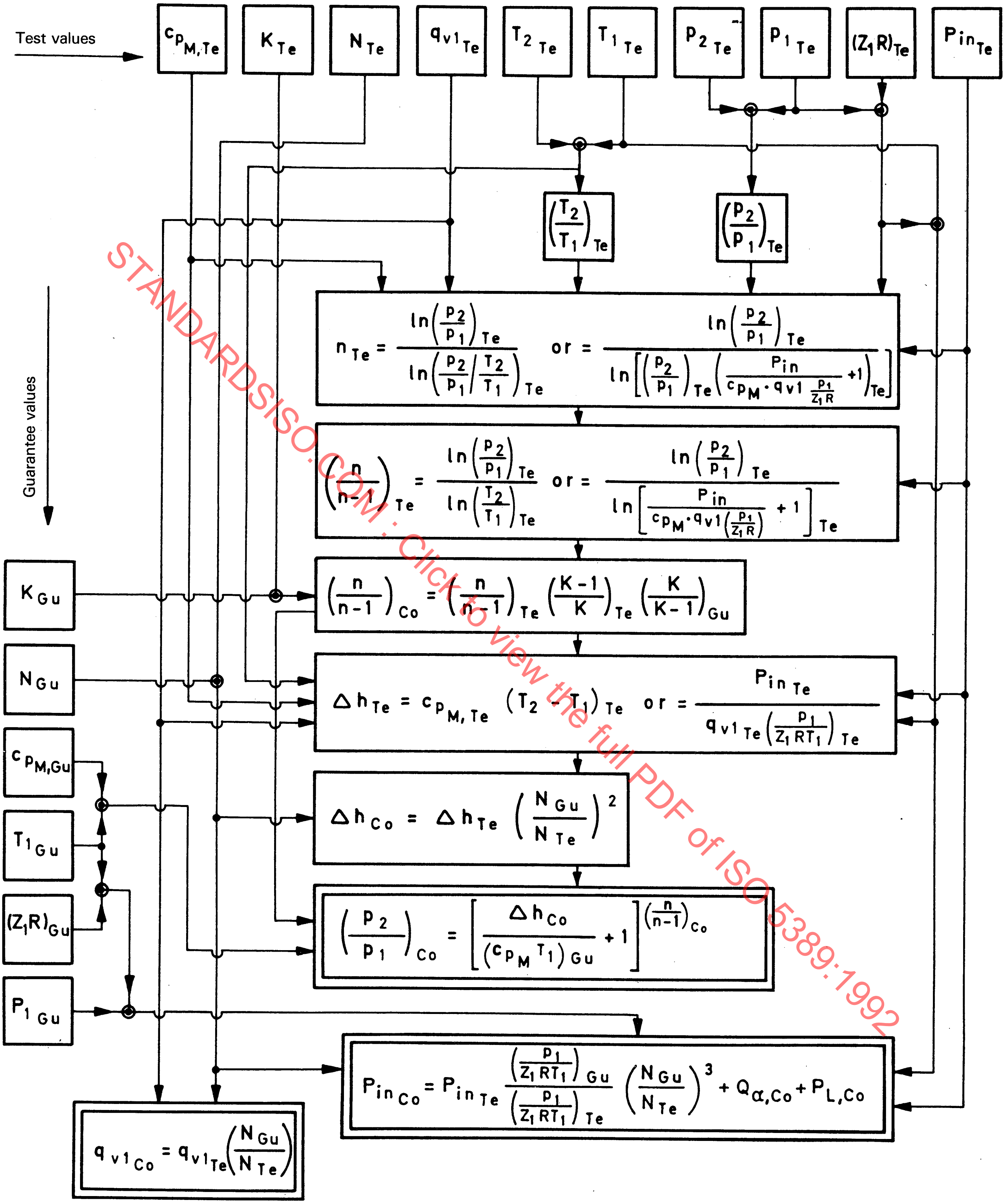
The conversion formulae are given in figure D.8.

NOTE — See table 2 for suggested limits within which the gas may be considered to be near perfect.

D.6.2 Real gases

The conversion method in figure D.9 is used for Schultz's method of polytropic analysis (see [1]).

The same formulae apply when gas properties are determined from tables and/or charts and equations of state rather than from generalized compressibility functions as given in annex A.

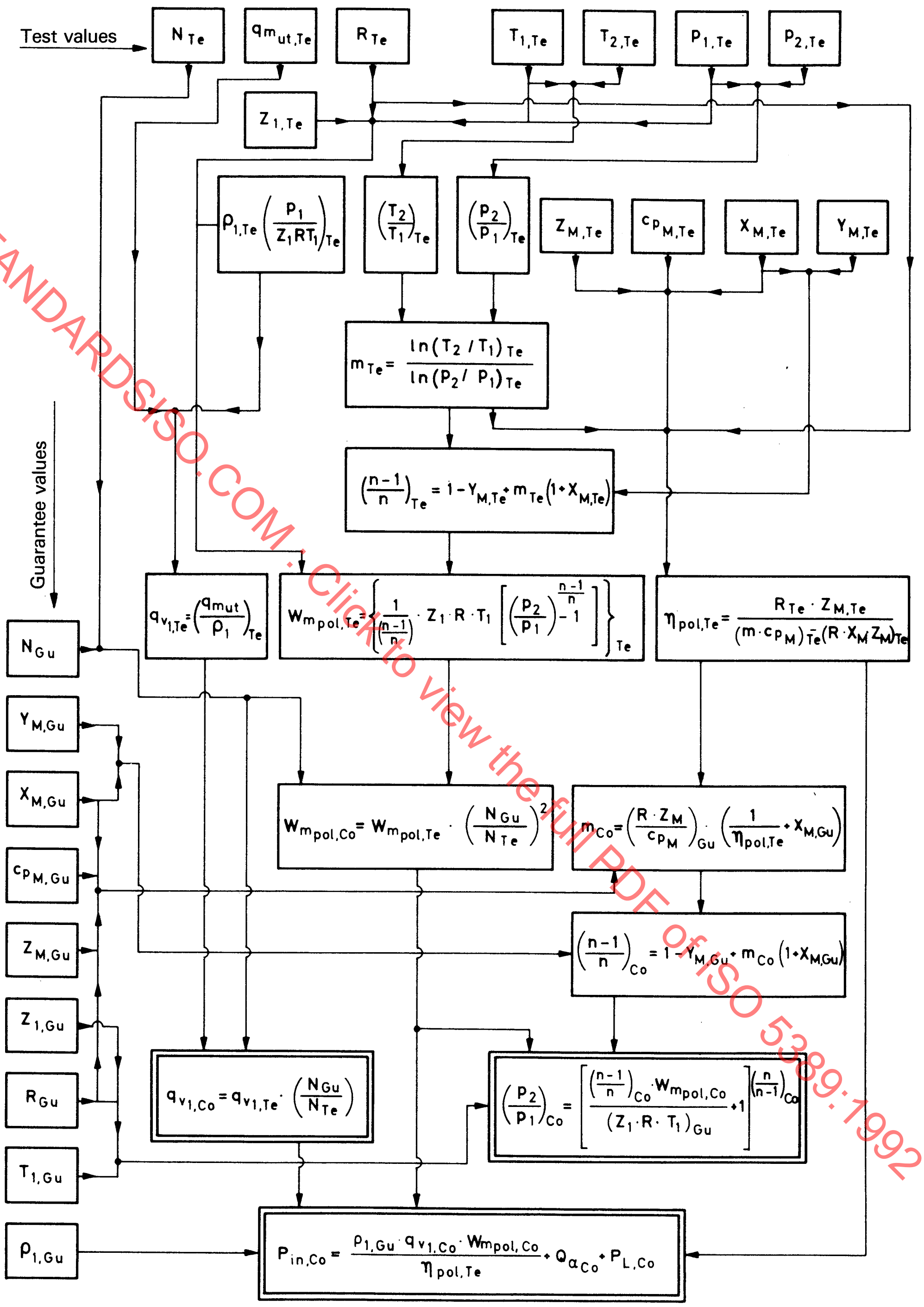


NOTES

1 Unless otherwise agreed all pressures and temperatures refer to total conditions ($p = p_t$ and $T = T_t$).

2 The isentropic exponent κ is represented as K in this figure.

Figure D.8 — Conversion of test results for perfect or near-perfect gases in the case of uncooled compressors



NOTE — Unless otherwise agreed all pressures and temperatures refer to total conditions ($p = p_t$ and $T = T_t$).

Figure D.9 — Conversion of test results for real gases in the case of uncooled compressors

D.7 Conversion to guarantee conditions in the case of cooled compressors

D.7.1 Perfect or near-perfect gases

NOTE — See table 2 for suggested limits within which the gas may be considered to be near perfect.

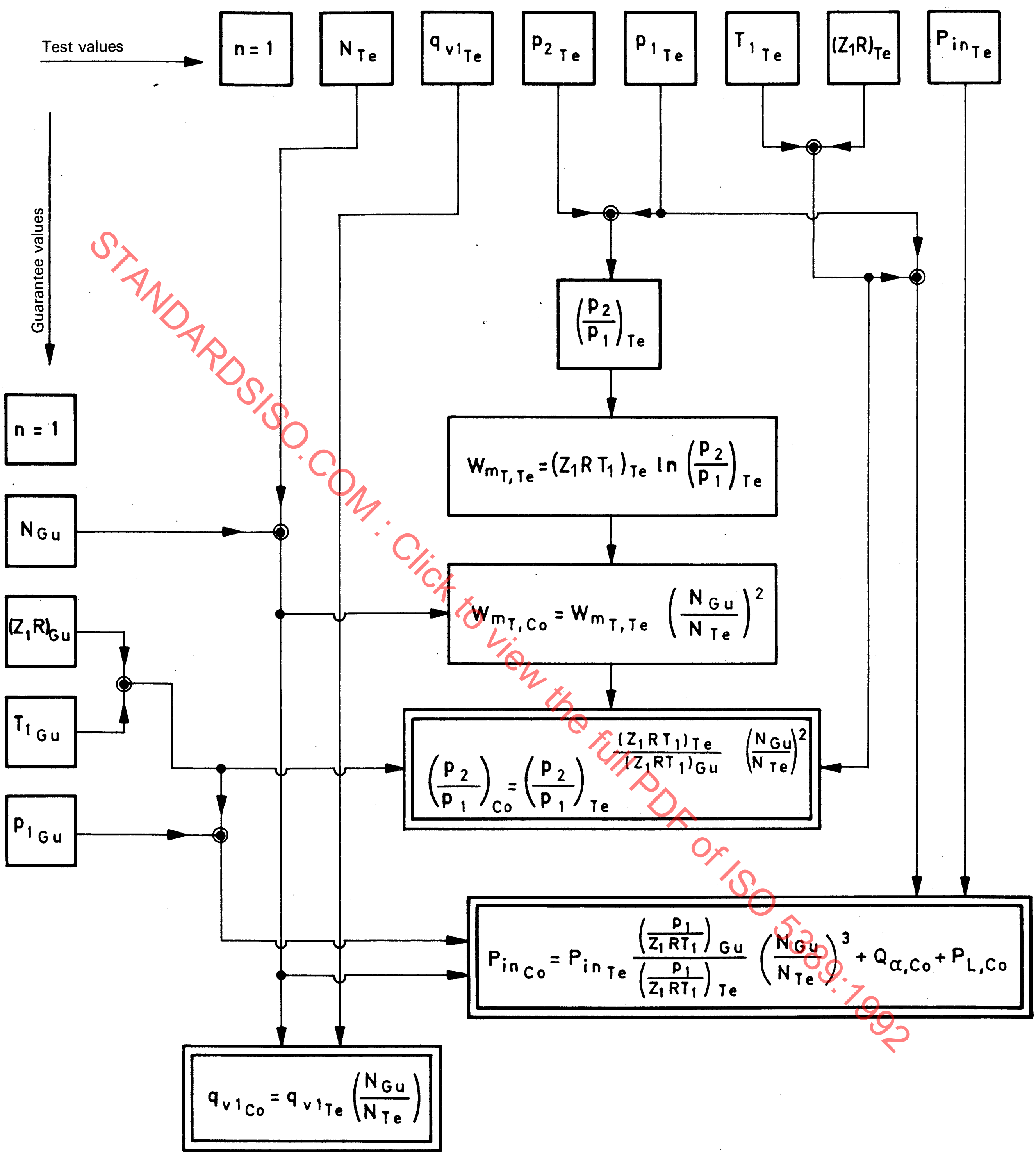
D.7.1.1 For cooled compressors according to 8.2.4.1, where the ratio of the gas inlet temperature of the stages or stage groups following intercoolers to the inlet temperature of the compressor is the same under test and guarantee conditions, the conversion formulae are given in figure D.10.

D.7.1.2 For cooled compressors according to 8.2.4.2, where the ratio of the gas inlet temperature of the stages or stage groups following intercoolers to the inlet temperature of the compressor is different under test and guarantee conditions, the conversion formulae are given in figure D.11.

D.7.2 Real gases

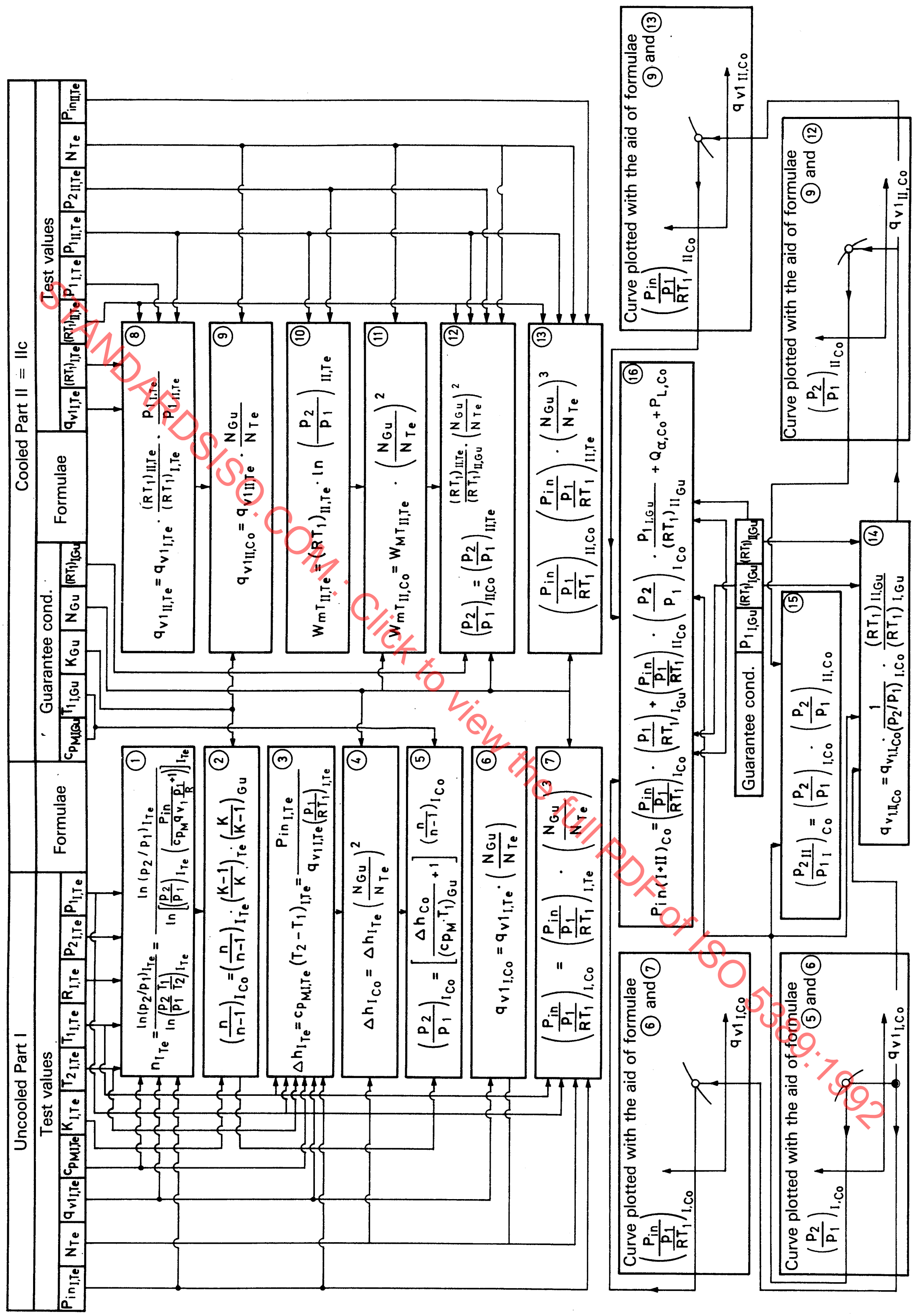
In accordance with 3.5.3.8, the compressor sections between intercoolers must be converted separately (see clause D.6) and the results combined.

For this combination, figure D.11 may be used where applicable (steps 14, 15 and 16).



NOTE — Unless otherwise agreed all pressures and temperatures refer to total conditions ($p = p_t$ and $T = T_t$).

Figure D.10 — Conversion of test results for perfect or near-perfect gases in the case of compressors cooled in accordance with 8.2.4.1 (see D.7.1.1)



NOTES

1 Unless otherwise agreed all pressures and temperatures refer to total conditions ($p = p_t$ and $T = T_t$).

2 The isentropic exponent κ is represented as K in this figure.

Figure D.11 — Conversion of test results for perfect or near-perfect gases in the case of compressors cooled in accordance with 8.2.4.2 (see D.7.1.2)

Annex E
(normative)

Correction method for the influence of the Reynolds number on the performance of centrifugal compressors

(according to the International Compressed Air and Allied Machinery Committee)

E.0 Introduction

The method provides formulae for the efficiency, specific compression work and flow corrections necessitated by differences in Reynolds number between the workshop test conditions and the guarantee conditions.

The total losses are divided into two portions. In the region of the best efficiency point, the portion of the losses independent of the Reynolds number is represented by a constant fraction of 0,3. The losses due to friction are assumed to be dependent on a representative value of the friction coefficient Λ according to the Moody diagram for pipe friction. The representative value of Λ is related to a reference Reynolds number and to a reference relative roughness of the compressor. These corrections can be applied in an allowable range, taking into account the inherent limitations for accurate testing at low Reynolds numbers.

The method is the result of an investigation of a Working Group of the Process Compressor Sub-committee of the International Compressed Air and Allied Machinery Committee (ICAAMC), issued on June 25, 1984. A brief description of the application of this method is given herein.

The correction method applies only to the internal hydraulic losses and, therefore, balancing piston losses and mechanical losses must be accounted for separately.

E.1 Definitions

For the purposes of this annex, the following definitions apply.

E.1.1 Reynolds number Re: A dimensionless parameter expressing the ratio between the inertia and the viscous forces in a liquid. In this International Standard, the representative value of the peripheral Reynolds number for the first stage of a stage group is given by

$$Re_u = \frac{ub}{\nu_{t,1}} \qquad \text{... (E.1)}$$

where $\nu_{t,1}$ is the kinematic viscosity at the inlet total condition.

In multistage compressors, the representative value of the Reynolds number of the first stage is taken to represent each stage group.

E.1.2 roughness R_a**; centre-line average (CLA)** /GB/; **arithmetical average (AA)** /USA/: Roughness average from the centre-line of the peaks. It is given by

$$R_a = \frac{1}{l}\int_0^l |y|\,dx \qquad \text{... (E.2)}$$

The reference roughness R_a is the average roughness for the impeller and its diffusor and can be either measured or taken from the manufacturer's drawing (with agreement between manufacturer and customer). The roughness values are measured inside the impeller on one blade, on the disc and on the shroud near the outer diameter. The values for the diffusor are measured on the side walls and in the middle of one blade near the inlet diameter.

The representative relative roughness of the stage is given by

$$\frac{R_a}{b}$$

In multistage compressors, the representative value of the relative roughness of the first stage is taken to represent each stage group.

E.2 Symbols and subscripts

Table E.1 — Symbols

Symbols	SI unit symbol	Definitions and observations
b	m	Outlet width of the first impeller of the stage group
D	m	Outlet diameter of the first impeller of the stage group
e	m	Deviation of the surface from the line of the mean surface height
Δh_t	J/kg	Enthalpy rise of the stage group
l	m	Length of the line of the mean surface
$q_{V,t,1}$	m^3/s	Inlet volume rate of flow of the stage group
R_a	m	Average roughness
u	m/s	Peripheral velocity at reference diameter, i.e. the outlet diameter of the first impeller of the stage group
$W_{m,pol}$	J/kg	Polytropic specific compression work of the stage group
$\Gamma = \Delta h_t/u^2$	dimensionless	Work input coefficient of the stage group
η_{pol}	dimensionless	Polytropic efficiency of the stage group
Λ	dimensionless	Pipe flow friction factor
$\nu_{t,1}$	m^2/s	Kinematic viscosity at the inlet total condition
$\Phi = q_{V,t,1}/(D^2\, u)$	dimensionless	Flow coefficient of the stage group
$\Psi_{pol} = W_{m,\,pol}/u^2$	dimensionless	Polytropic work coefficient of the stage group

Table E.2 — Subscripts

Subscripts	Definitions
Gu	Guarantee conditions
Te	Test conditions
∞	At Reynolds number tending to infinity

E.3 Formula for efficiency correction

The formula for efficiency correction in the region of the best efficiency point is given by

$$\frac{1-\eta_{pol,Gu}}{1-\eta_{pol,Te}} = \frac{0{,}3 + 0{,}7\,\Lambda_{Gu}/\Lambda_\infty}{0{,}3 + 0{,}7\,\Lambda_{Te}/\Lambda_\infty} \quad \ldots \text{(E.3)}$$

For calculation of the Λ values, the following generally accepted equations are used:

from von Karman

$$\frac{1}{\sqrt{\Lambda_\infty}} = 1{,}74 - 2\log_{10}\left(2\frac{R_a}{b}\right) \quad \ldots \text{(E.4)}$$

for guarantee conditions, from Colebrook,

$$\frac{1}{\sqrt{\Lambda_{Gu}}} = 1{,}74 - 2\log_{10}\left(2\frac{R_a}{b} + \frac{18{,}7}{Re_{u,Gu}\sqrt{\Lambda_{Gu}}}\right) \ldots \text{(E.5)}$$

for test conditions, from Colebrook,

$$\frac{1}{\sqrt{\Lambda_{Te}}} = 1{,}74 - 2\log_{10}\left(2\frac{R_a}{b} + \frac{18{,}7}{Re_{u,Te}\sqrt{\Lambda_{Te}}}\right) \ldots \text{(E.6)}$$

For rough estimations, the Moody diagram (figure E.1) can be used.

E.4 Formulae for specific compression work, work input and flow correction

There is a definite increase in both work coefficient and flow coefficient with an increase in Reynolds number.

In the neighbourhood of the test efficiency point, roughly half the increase in the efficiency appears as an increase in specific compression work, and this leads to the following formula:

$$\frac{\Psi_{pol,Gu}}{\Psi_{pol,Te}} = 0{,}5 + 0{,}5\frac{\eta_{pol,Gu}}{\eta_{pol,Te}} \quad \ldots \text{(E.7)}$$

With a knowledge of the efficiency and work coefficient corrections, the correction for the work input can be calculated from the relationship $\Gamma_{in} = \Psi_{pol}/\eta_{pol}$, which gives the following formula:

$$\frac{\Gamma_{Gu}}{\Gamma_{Te}} = 0,5 + 0,5 \frac{\eta_{pol,Te}}{\eta_{pol,Gu}} \qquad \text{... (E.8)}$$

The change in flow with increasing Reynolds number can be approximated by

$$\frac{\Phi_{Gu}}{\Phi_{Te}} = \left(\frac{\Psi_{pol,Gu}}{\Psi_{pol,Te}}\right)^{1/2} \qquad \text{... (E.9)}$$

E.5 Application of formulae to test data

The equations given in clause E.4 define the influence of the change in the best efficiency point on the performance characteristics. The change in other points can then be calculated by noting that the shape of the characteristic curve remains essentially the same.

The full correction procedure is illustrated in figure E.2 and is summarized below.

a) At the best efficiency point on the test characteristic the ratio

$$\frac{1 - \eta_{pol,Gu}}{1 - \eta_{pol,Te}}$$

is calculated according to equation (E.3).

b) From this, the ratio $\eta_{pol,Gu}/\eta_{pol,Te}$ is found.

c) Equation (E.7) is used to calculate the ratio $\Psi_{pol,Gu}/\Psi_{pol,Te}$, equation (E.9) to calculate the ratio Φ_{Gu}/Φ_{Te}, and equation (E.8) to calculate the ratio Γ_{Gu}/Γ_{Te} at the best efficiency point.

d) The ratios $\eta_{pol,Gu}/\eta_{pol,Te}$, $\Psi_{pol,Gu}/\Psi_{pol,Te}$, and Γ_{Gu}/Γ_{Te} calculated at the best efficiency point are taken to be the same at all points of the measured test characteristic.

e) The measured test points are now converted to the corrected characteristic using these fixed ratios [d)].

E.6 Allowable range

The range of application of the correction formulae and the selection of a suitable test Reynolds number are influenced by two factors, i.e.

— the accuracy of the correction formulae at different Reynolds numbers,

— the reliability of tests carried out at reduced suction pressures or low driving power.

The limit of application of the ICAAMC formula is shown in figure D.5.

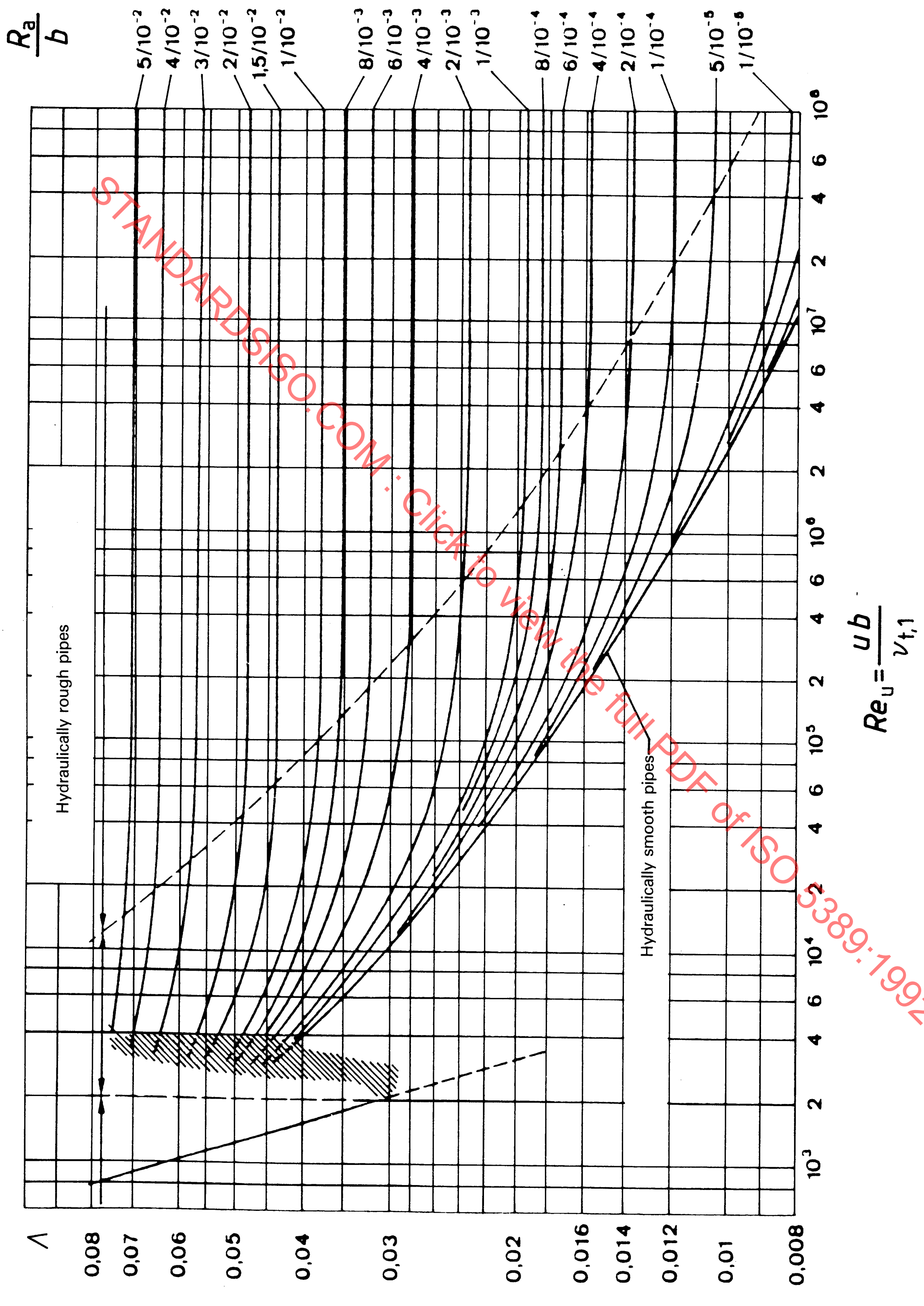


Figure E.1 — Friction factor for turbulent flow in rough pipes

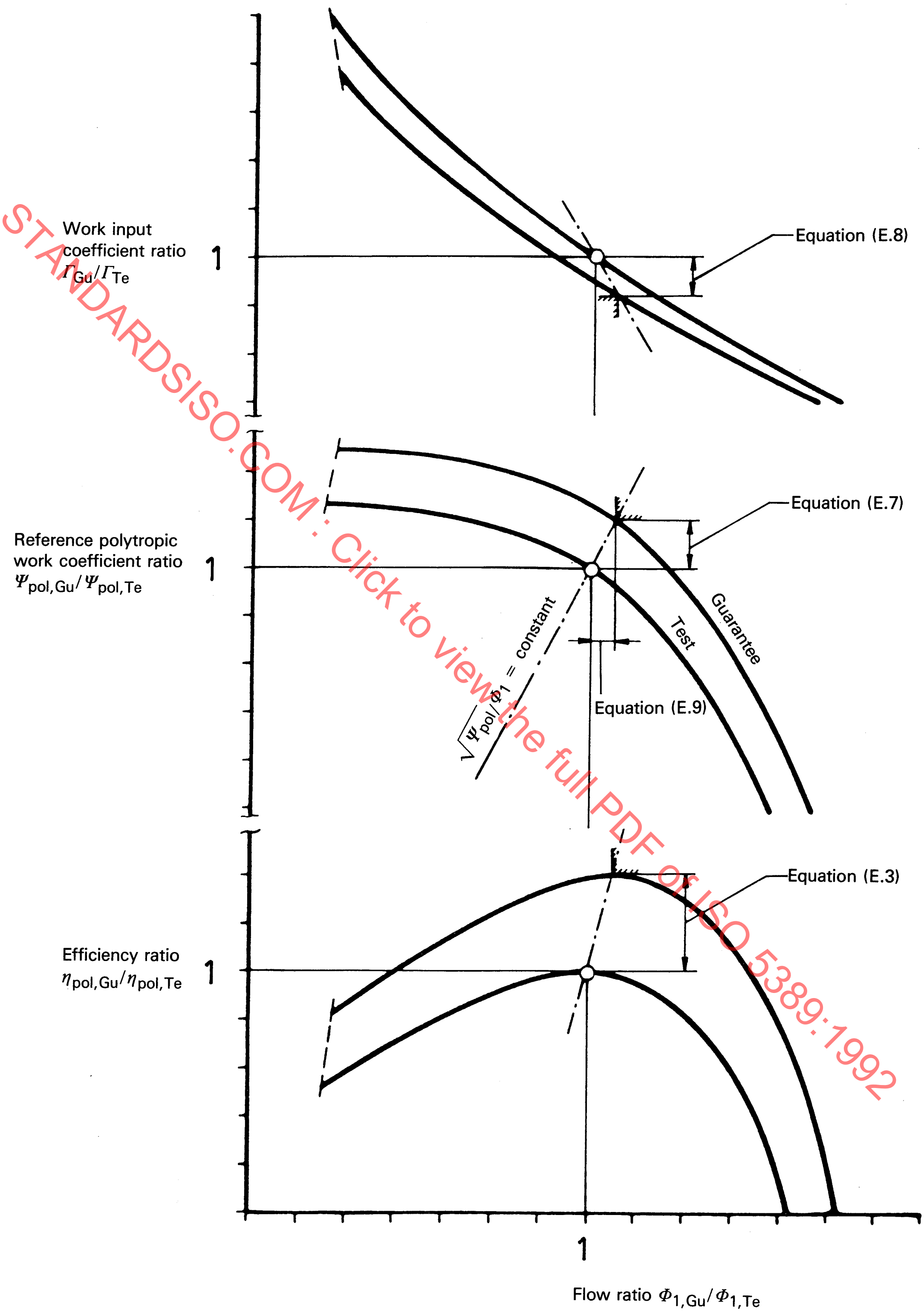


Figure E.2 — Illustration of correction procedure

Annex F
(informative)

Examples

F.0 Introduction

F.0.1 General

This annex gives examples to demonstrate the determination of test settings and the conversion of test results to guarantee conditions in typical cases.

NOTE — The examples given in this annex do not incorporate the Reynolds number corrections because the ICAAMC method was not available at the time when the examples were incorporated in this International Standard.

F.0.2 Test conditions

Table F.1 summarizes the test conditions covered by the different examples regarding gas properties (perfect gas or real gas) and the approximation to strict similarity of flow.

F.0.3 Computation of specific compression work

In all examples the specific compression work is computed on the basis of total pressures and temperatures. The velocity heads at the standard inlet and discharge points therefore do not appear as a separate term, which is the case when the computation is carried out according to the correct formulae based on static pressures and temperatures (see 3.4).

This method of calculation gives sufficient accuracy, because the gas velocity in the inlet and discharge nozzles of turbocompressors is usually low ($Ma < 0{,}2$).

In order to verify the validity of this approximation, examples 1 and 4 have also been calculated using the correct method of accounting separately for the dynamic pressure. The following comparison of the results indicates the magnitude of the error introduced and shows that the approximation in these two particular cases is acceptable.

Approximate method

$$W_{m,\mathrm{t}} = \int_1^2 (v\,\mathrm{d}p)_\mathrm{t}$$

Correct method

$$W_{m,\mathrm{t}} = W_m + \frac{c_2^2 - c_1^2}{2}$$

Example 1:

Converted internal power at guaranteed pressure ratio:

Approximate method	Correct method
320,4 kW	320,5 kW

NOTE 1 — Local Mach numbers at inlet and discharge are 0,043 and 0,071 respectively.

Example 4:

Converted internal power at guaranteed pressure ratio:

Approximate method	Correct method
7 993,3 kW	7 992,8 kW

NOTE 2 — Local Mach numbers at inlet and discharge are 0,058 and 0,119 respectively.

F.0.4 Accuracy of computation

Re-calculation of the numerical results of the examples may reveal differences in the last digits depending on the accuracy and storage capacity of the computer used.

Table F.1 — Synopsis of test conditions

Test conditions		Example 1	2	3	4	5	6
Similarity of flow	strict	x			x		
	approximate		x	x		x	x
Isentropic exponent	$\kappa_{\mathrm{Te}} = \kappa_{\mathrm{Gu}}$	x	x		x	x	
	$\kappa_{\mathrm{Te}} \neq \kappa_{\mathrm{Gu}}$			x			x
	constant	x	x	x	x	x	
	not constant						x
Compressibility factor	constant = 1	x	x	x	x	x	
	not constant						x
Test gas	air	x	x	x	x	x	
	gas mixture						x
Speed	adjustable	x				x	x
	not adjustable		x	x	x		
Intercooling	no	x	x	x			
	yes				x	x	x

F.1 Example 1 — Uncooled turbocompressor, isentropic exponent $\kappa_{Te} = \kappa_{Gu}$, speed adjustable

F.1.1 General

The conditions at the inlet on test deviate from the guarantee conditions. The same peripheral Mach number as given in the guarantee conditions can be achieved by changing the speed.

The guarantee conditions, guaranteed performance and other design values are given in tables F.2 to F.4.

F.1.2 Purpose of tests

The purpose of the tests is to prove the guaranteed power requirement for the guarantee point.

F.1.3 Design of installation

The installation consists of a three-stage turbocompressor for a biatomic gas mixture, driven by a back-pressure turbine.

Table F.2 — Guarantee conditions

Designation	Symbol	Numerical value	Unit
Inlet pressure (total)	$p_{t,1,Gu}$	0,952 4	bar
Inlet temperature (total)	$t_{t,1,Gu}$	30,16	°C
Gas constant	R_{Gu}	764,9	N·m/(kg·K)
Isentropic exponent (Gas mixture, perfect gas Z_{Gu} = 1)	κ_{Gu}	1,4	—
Kinematic viscosity	$\nu_{1,Gu}$	$3,27 \times 10^{-5}$	m²/s

Table F.3 — Guaranteed performance

Designation	Symbol	Numerical value	Unit
Volumetric flow at inlet	$q_{V,t,1,Gu}$	9,715	m³/s
Discharge pressure	$p_{t,2,Gu}$	1,220 3	bar
Specific power requirement at compressor coupling	$\dfrac{P_{e,Gu}}{q_{V,t,Gu}}$	0,009 297	kWh/m³

Table F.4 — Other design values

Designation	Symbol	Numerical value	Unit
Compressor speed (due to variable-speed drive not guaranteed)1)	N_{Gu}	4 700	r/min
Outside diameter of Stage 1 impeller	D_I	0,9	m
Cross-section of inlet nozzle2)	A_1	0,39	m²
Cross-section of discharge nozzle2)	A_2	0,192 8	m²

1) The subscript Gu designates the design speed.

2) It is important that these flow cross-sections be given because of low pressure increase in this instance. (Thus the static and dynamic components of the pressures can be determined.)

F.1.4 Test set-up

Since it is impossible to test the machine set with gas on site, the tests are carried out on the supplier's test bed using air, the compressor being driven by a variable-speed cradled electric dynamometer, in such a way that the compressor shaft torque can be measured directly.

The test set-up is illustrated in figure F.1, and the test conditions are given in table F.5.

F.1.5 Setting conditions

The test conditions are set to achieve flow similarity in accordance with 8.2.3.2, i.e.

$$N_r = \frac{\left(\dfrac{N}{\sqrt{RZ_1T_{t,1}}}\right)_{Te}}{\left(\dfrac{N}{\sqrt{RZ_1T_{t,1}}}\right)_{Gu}} = 1$$

$$N_{Te} = 4\,700\sqrt{\frac{289{,}3 \times 301{,}31}{764{,}9 \times 303{,}31}} = 2\,881 \text{ r/min}$$

Since figure F.1 is not required in fulfilling this condition, $\eta_{pol,Gu}$ and η_{Gu} do not need to be calculated.

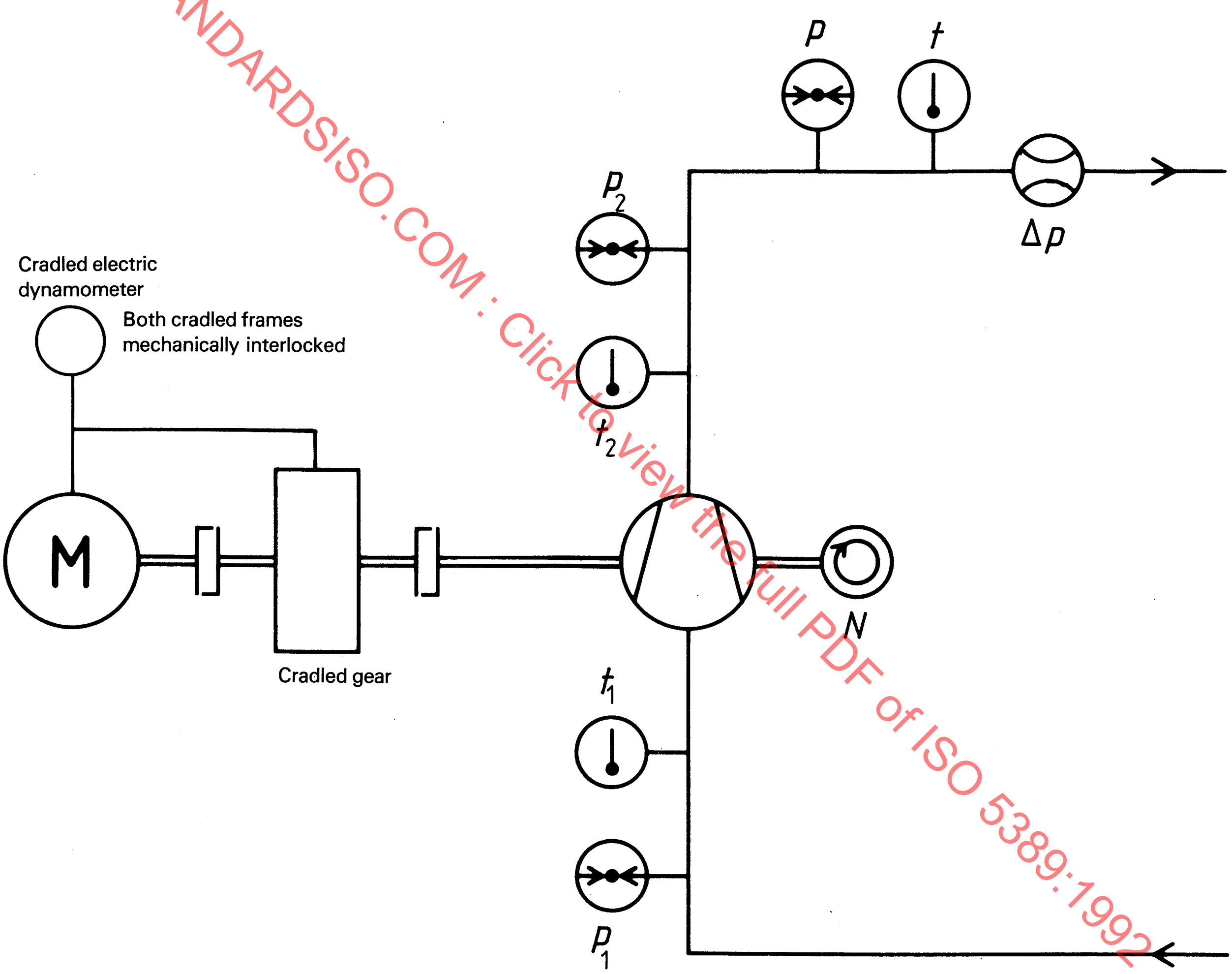


Figure F.1 — Test set-up for example 1

Table F.5 — Test conditions

Designation	Symbol	Numerical value	Unit
Isentropic exponent (air)	κ_{Te}	1,4	—
Mean inlet temperature	$t_{t,1,Te}$	28,16	°C
Gas constant	R_{Te}	289,30	N·m/(kg·K)
Test gas assumed perfect, therefore $Z_{Te} = 1$			

Alternatively the settings could be determined as follows by using the flow chart given in figure D.6. The calculation sequence is as follows (the numbers given are those designating individual boxes in the flow chart):

1 — 2 — 3 — 4 — 5 — 6 — 7 — 8 — 9 — 10 — 11 — 12 — 13 — 21 — 22 — 23 — 24

Since strict similarity can be guaranteed here with $\kappa_{Te} = \kappa_{Gu}$ and with variable speed, $\Delta V_{r,tol} = 0$ is inserted into 23.

A check as to whether the assumption made in 7 is true is carried out in 25 — 26.

Since in this particular case Z and c_p do not vary within the machine, the condition of 26 is fulfilled.

Pass 43 — 44 (see the note to F.0.1).

A check as to whether the setting conditions determined can be maintained in the test is made in 45. Since this is the case in this instance, the calculation ends with 46 — 47.

The test was carried out maintaining the above test conditions.

F.1.6 Test results and conversion

See tables F.6 to F.8.

Table F.6 — Test results

Designation	Symbol	Numerical value	Unit
Test number	—	1	—
Date of test	—	1964-09-28	—
Time of test	—	12,50	h
Atmospheric pressure	p_a	1,013	bar
Gas constant	R_{Te}	289,30	N·m/(kg·K)
Speed	N_{Te}	2 881	r/min
Mass flow*)	$q_{m,Te}$	6,752 78	kg/s
Inlet pressure (total)	$p_{t,1,Te}$	1,00	bar
Inlet temperature (total)	$t_{t,1,Te}$	28	°C
Density	$\rho_{t,1,Te}$	1,150	kg/m^3
Volumetric flow at inlet (also usable inlet volume flow)	$q_{V,t,1,Te}$	5,8	m^3/s
Discharge temperature	$t_{t,2,Te}$	59,68	°C
Discharge pressure	$p_{t,2,Te}$	1,297 1	bar

*) Computed from the mass flow which is measured in accordance with ISO 5167-1.

Table F.7 — Calculation results

Designation	Symbol	Numerical value	Unit
Pressure ratio	$(p_2/p_1)_{t,Te}$	1,292 6	—
Specific isentropic compression work	$W_{m,s,t,Te}$	23 220	N·m/kg
Isentropic compression power	$P_{s,t,Te}$	156,8	kW
Power at compressor coupling	$P_{e,Te}$	214,7	kW
Mechanical compressor power loss	$P_{f,Te}$	3,0	kW
Internal power	$P_{in,Te}$	211,7	kW
Internal isentropic efficiency (from $P_{in,Te}$)	$\eta_{s,in,Te}$	74,07	%
Internal polytropic efficiency (from $t_{t,2,Te}$)	$\eta_{pol,in,Te}$	75,05	%
Polytropic exponent	n_{Te}	1,630	—
Specific polytropic compression work	$W_{m,pol,t,Te}$	23 528	N·m/kg

Table F.8 — Conversion to guarantee conditions (see figure D.8)

Designation	Symbol	Numerical value	Unit
Design speed	N_{Gu}	4,700	r/min
Test speed	N_{Te}	2 881	r/min
Converted volumetric flow at inlet	$q_{V,t,1,Co}$	9,572 8	m³/s
Converted specific compression work	$W_{m,pol,t,Co}$	62 617	N·m/kg
Converted pressure ratio calculated with $n_{Co} = n_{Te}$ because $\kappa_{Gu} = \kappa_{Te}$	$(p_2/p_1)_{t,Co}$	1,292 7	—
Converted internal power	$P_{in,Co}$	327,87	kW
Converted mechanical power losses	$P_{f,Co}$	6,3	kW
Converted power at coupling	$P_{e,Co}$	334,17	kW

NOTE — The conversion of mechanical losses is carried out according to the formula

$$P_{f,Co} = P_{f,Te}\left(\frac{N_{Gu}}{N_{Te}}\right)^{1,5}$$

F.1.7 Test uncertainty and comparison with guarantee

Table F.9 — Comparison with the guarantee (see 9.4)

Designation	Symbol	Numerical value	Unit
Guaranteed specific polytropic compression work	$W_{m,pol,t,Gu}$	60,351	N·m/kg
Conversion factor	$\dfrac{q_{V,t,Gu} W_{m,pol,t,Gu}}{q_{V,t,Co} W_{m,pol,t,Co}}$	0,978 1	—
Internal power absorption related to $(p_2/p_1)_{t,Gu}$	$P_{in,Co}$	320,7	kW
Power at coupling related to $(p_2/p_1)_{t,Gu}$	$P_{e,Co}$	327,0	kW
Specific power at coupling related to $(p_2/p_1)_{t,Gu}$	$\dfrac{P_{e,Co}}{q_{V,t,1,Co}}$	0,009 35	kWh/m³
Guaranteed specific power at coupling	$\dfrac{P_{e,Gu}}{q_{V,t,1,Gu}}$	0,009 297	kWh/m³
Deviation in specific power at coupling from guarantee value		+ 0,57	%

The specific power at the coupling is exceeded by 0,57 % (see table F.9) but, since this is within the overall test uncertainty, the guarantees are met accordingly.

The measuring errors are calculated in accordance with 9.2.

The test uncertainty for example 1 is shown in table F.10.

Table F.10 — Resulting test uncertainty

Designation	Symbol	Numerical value	Comments
Uncertainty in volumetric flow at inlet	$\tau_{q_V,\mathrm{Te}}$	1,1 %	According to ISO 5167-1
	$\tau_{N,\mathrm{Te}}$	0,35 %	Quality grade 0,2; final scale value 5 000
	$\tau_{p_1,\mathrm{Te}}$	0,13 %	1 mmHg on absolute pressure
	$\tau_{T_1,\mathrm{Te}}$	0,33 %	1 ° on absolute temperature
	τ_{Z_1}	—	Not applicable, in accordance with 5.9
	τ_{res,q_V}	1,207 %	See 9.2.5
Uncertainty in pressure ratio	N_r	1,0	
	$\ln(p_2/p_1)$	0,254	
	$\tau_{p_2,\mathrm{Te}}$	0,43 %	1 mmHg on absolute pressure
	$\tau_{\mathrm{res},p_2/p_1}$	0,49 %	See 9.2.6
Uncertainty in specific power at coupling	ζ_4	201,1	
	τ_{P_f}	10 %	Estimated
	τ_F	1,34 %	One scale division as assumed error
	$\tau_{\mathrm{res},P_\mathrm{e}}$	1,740 %	See 9.2.7 and table 10

F.1.8 Comparison of different calculation methods

A comparison of different calculation methods is given in table F.11.

Table F.11 — Comparison of different calculation methods

Example 1	Total (p_t, t_t)	Total $\left(W_m = W_{m,\text{static}} + \frac{c_2^2 - c_1^2}{2}\right)$	Suction side: total Discharge side: static	Static
Guarantee conditions				
Inlet pressure	$p_{t,1,Gu}$ = 0,952 4 bar	0,952 4 bar	0,952 4 bar	$p_{1,Gu}$ = 0,951 2 bar
Inlet temperature	$t_{t,1,Gu}$ = 30,16 °C	30,16 °C	30,16 °C	$t_{1,Gu}$ = 30 °C
Gas constant	R_{Gu} = 764,9 N·m/(kg·K)			
Isentropic exponent	κ_{Gu} = 1,4			
Compressibility factor	Z_{Gu} = 1,0			
Kinematic viscosity	ν_{Gu} = 3,267 8 × 10^{-5} m²/s			
Discharge pressure	$p_{t,2,Gu}$ = 1,220 3 bar	$p_{t,2,Gu}$ = 1,220 3 bar	$p_{2,Gu}$ = 1,216 0 bar	$p_{2,Gu}$ = 1,216 0 bar
Volumetric flow at inlet	$q_{V,t,1,Gu}$ = 9,715 m³/s	$q_{V,t,1,Gu}$ = 9,715 m³/s	$q_{V,t,1,Gu}$ = 9,715 m³/s	$q_{V,1,Gu}$ = 9,722 2 m³/s
Specific power required at compressor coupling	$\frac{P_{e,Gu}}{q_{V,t,1}}$ = 0,009 297 kWh/m³	$\frac{P_{e,Gu}}{q_{V,t,1}}$ = 0,009 297 kWh/m³	$\frac{P_{e,Gu}}{q_{V,t,1}}$ = 0,009 297 kWh/m³	$\frac{P_{e,Gu}}{q_{V,t,1}}$ = 0,009 297 kWh/m³
Expected compressor speed	N_{Gu} = 4 700 r/min			
Outside diameter of 1st impeller	D = 0,9 m			
Area of inlet nozzle	A_1 = 0,39 m²			
Area of discharge nozzle	A_2 = 0,192 8 m²			
Test conditions				
Test gas	Air			Air
Isentropic exponent	κ_{Te} = 1,4			κ_{Te} = 1,4
Inlet temperature	$t_{t,1,Te}$ = 28,16 °C			$t_{1,Te}$ = 28 °C
Gas constant	R_{Te} = 289,30 N·m/(kg·K)			R_{Te} = 289,30 N·m/(kg·K)
Compressibility factor	Z_{Te} = 1,0			Z_{Te} = 1,0
Kinematic viscosity	ν_{Te} = 1,640 1 × 10^{-5} m²/s			ν_{Te} = 1,640 1 × 10^{-5} m²/s

Table F.11 — Comparison of different calculation methods *(continued)*

Example 1	Total (p_t, t_t)	Total $\left(W_m = W_{m,\text{static}} + \frac{c_2^2 - c_1^2}{2}\right)$	Suction side: total Discharge side: static	Static
Test conditions *(continued)*				
Determination of test settings	$N_{Te} = N_{Gu}\sqrt{\frac{R_{Te}T_{t,1,Te}}{R_{Gu}T_{t,1,Gu}}}$ $= 4\ 700\sqrt{\frac{289{,}3 \times 301{,}31}{764{,}9 \times 303{,}31}}$ $N_{Te} = 2\ 881$ r/min $\frac{R_{e,Te}}{R_{e,Gu}} = \frac{N_{Te}\nu_{Gu}}{N_{Gu}\nu_{Te}}$ $= \frac{2\ 881 \times 3{,}267\ 8 \times 10^{-3}}{4\ 700 \times 1{,}640\ 1 \times 10^{-5}}$ $\frac{R_{e,Te}}{R_{e,Gu}} = 1{,}221\ 3$			$N_{Te} = N_{Gu}\sqrt{\frac{R_{Te}T_{t,1,Te}}{R_{Gu}T_{t,1,Gu}}}$ $= 4\ 700\sqrt{\frac{289{,}3 \times 301{,}31}{764{,}9 \times 303{,}31}}$ $N_{Te} = 2\ 881$ r/min $\frac{R_{e,Te}}{R_{e,Gu}} = \frac{N_{Te}\nu_{Gu}}{N_{Gu}\nu_{Te}}$ $= \frac{2\ 881 \times 3{,}267\ 8 \times 10^{-3}}{4\ 700 \times 1{,}640\ 1 \times 10^{-5}}$ $\frac{R_{e,Te}}{R_{e,Gu}} = 1{,}221\ 3$
Test readings				
Test No.	1	1	1	1
Date of test	1964-09-28	1964-09-28	1964-09-28	1964-09-28
Time of test	12,30	12,30	12,30	12,30
Atmospheric pressure	$p_a = 1{,}013$ bar	$p_a = 1{,}013$ bar	$p_a = 1{,}013$ bar	$p_a = 1{,}013$ bar
Gas constant	$R_{Te} = 289{,}30$ N·m/(kg·K)	$R_{Te} = 289{,}30$ N·m/(kg·K)	$R_{Te} = 289{,}30$ N·m/(kg·K)	$R_{Te} = 289{,}30$ N·m/(kg·K)
Speed	$N_{Te} = 2\ 881$ r/min	$N_{Te} = 2\ 881$ r/min	$N_{Te} = 2\ 881$ r/min	$N_{Te} = 2\ 881$ r/min
Mass flow	$q_{m,Te} = 6{,}752\ 78$ kg/s	$q_{m,Te} = 6{,}752\ 78$ kg/s	$q_{m,Te} = 6{,}752\ 78$ kg/s	$q_{m,Te} = 6{,}752\ 78$ kg/s
Inlet pressure	$p_{t,1,Te} = 1{,}003\ 5$ bar	$p_{t,1,Te} = 1{,}003\ 5$ bar	$p_{t,1,Te} = 1{,}003\ 5$ bar	$p_{t,1,Te} = 1{,}003\ 5$ bar
Inlet temperature	$t_{t,1,Te} = 28{,}26$ °C	$t_{t,1,Te} = 28{,}26$ °C	$t_{t,1,Te} = 28{,}26$ °C	$t_{t,1,Te} = 28{,}26$ °C
Discharge pressure	$p_{t,2,Te} = 1{,}297\ 1$ bar	$p_{t,2,Te} = 1{,}297\ 1$ bar	$p_{t,2,Te} = 1{,}297\ 1$ bar	$p_{t,2,Te} = 1{,}297\ 1$ bar
Discharge temperature	$t_{t,2,Te} = 59{,}68$ °C	$t_{t,2,Te} = 59{,}68$ °C	$t_{t,2,Te} = 59{,}68$ °C	$t_{t,2,Te} = 59{,}68$ °C
Input torque at compressor shaft	711,56 N·m	711,56 N·m	711,56 N·m	711,56 N·m

Table F.11 — Comparison of different calculation methods (*continued*)

Example 1	Total (p_t, t_t)	Total $\left(W_m = W_{m,\text{static}} + \frac{c_2^2 - c_1^2}{2}\right)$	Suction side : total Discharge side : static	Static
Test calculations Inlet gas density	$\varrho_{t,1,Te} = \frac{p_{t,1,Te}}{Z_{1,Te} R_{Te} T_{1,Te}}$ $= \frac{1{,}003\,5 \times 10^5}{1{,}0 \times 289{,}3 \times 301{,}41}$ $= 1{,}150\,8$ kg/m³	$\varrho_{t,1,Te} = \frac{p_{t,1,Te}}{Z_{1,Te} R_{Te} T_{1,Te}}$ $= \frac{1{,}003\,5 \times 10^5}{1{,}0 \times 289{,}3 \times 301{,}41}$ $= 1{,}150\,8$ kg/m³	$\varrho_{t,1,Te} = 1{,}150\,8$ kg/m³	$\varrho_{1,Te} = \frac{p_{1,Te}}{Z_{1,Te} R_{Te} T_{1,Te}}$ $= \frac{1{,}002\,2 \times 10^5}{1{,}0 \times 289{,}30 \times 301{,}25}$ $= 1{,}499\,9$ kg/m³
Volumetric flow at inlet	$q_{V,t,1,Te} = \frac{q_{m,Te}}{\varrho_{t,1,Te}}$ $= \frac{6{,}752\,78}{1{,}150\,8}$ $= 5{,}867\,9$ m³/s	$q_{V,t,1,Te} = \frac{q_{m,Te}}{\varrho_{t,1,Te}}$ $= \frac{6{,}752\,78}{1{,}150\,8}$ $= 5{,}867\,9$ m³/s	$q_{V,t,1,Te} = \frac{q_{m,Te}}{\varrho_{t,1,Te}}$ $= \frac{6{,}752\,78}{1{,}150\,8}$ $= 5{,}867\,9$ m³/s	$q_{V,1,Te} = \frac{q_{m,Te}}{\varrho_{1,Te}}$ $= \frac{6{,}752\,78}{1{,}149\,9}$ $= 5{,}872\,5$ m³/s
Pressure ratio	$\frac{p_{t,2,Te}}{p_{t,1,Te}} = \frac{1{,}297\,1}{1{,}003\,5}$ $= 1{,}292\,6$	$\frac{p_{2,Te}}{p_{1,Te}} = \frac{1{,}292\,5}{1{,}002\,2}$ $= 1{,}289\,7$	$\frac{p_{2,Te}}{p_{t,1,Te}} = \frac{1{,}292\,5}{1{,}003\,5}$ $= 1{,}288\,80$	$\frac{p_{2,Te}}{p_{1,Te}} = \frac{1{,}292\,5}{1{,}002\,2}$ $= 1{,}289\,7$
Specific isentropic compression work	$W_{m,s,t,Te} =$ $\frac{\kappa_{Te}}{\kappa_{Te}-1} \frac{p_{t,1,Te}}{\varrho_{t,1,Te}} \left[\left(\frac{p_{t,2,Te}}{p_{t,1,Te}}\right)^{\frac{\kappa_{Te}-1}{\kappa_{Te}}} - 1\right]$ $= \frac{1{,}4}{0{,}4} \times \frac{10^5 \times 1{,}003\,5}{1{,}150\,8} \times$ $\times \left[(1{,}292\,6)^{0{,}285\,7} - 1\right]$ $= 23\,220$ N·m/kg	$W_{m,s,t,Te} =$ $\frac{\kappa_{Te}}{\kappa_{Te}-1} \frac{p_{1,Te}}{\varrho_{1,Te}} \left[\left(\frac{p_{2,Te}}{p_{1,Te}}\right)^{\frac{\kappa_{Te}-1}{\kappa_{Te}}} - 1\right] +$ $+ \frac{c_2^2 - c_1^2}{2}$ $= \frac{1{,}4}{0{,}4} \times \frac{10^5 \times 1{,}002\,2}{1{,}149\,9} \times$ $\times \left[(1{,}289\,7)^{0{,}285\,7} - 1\right] +$ $+ \frac{(26{,}04^2 - 15{,}056\,97^2)}{2}$ $= 23\,223$ N·m/kg	$W_{m,s,Te} =$ $\frac{\kappa_{Te}}{\kappa_{Te}-1} \frac{p_{t,1,Te}}{\varrho_{t,1,Te}} \left[\left(\frac{p_{2,Te}}{p_{1,t,Te}}\right)^{\frac{\kappa_{Te}-1}{\kappa_{Te}}} - 1\right]$ $= \frac{1{,}4}{0{,}4} \times \frac{10^5 \times 1{,}003\,5}{1{,}150\,8} \times$ $\times \left[(1{,}288\,0)^{0{,}285\,7} - 1\right]$ $= 22\,885$ N·m/kg	$W_{m,s,Te} =$ $\frac{\kappa_{Te}}{\kappa_{Te}-1} \frac{p_{1,Te}}{\varrho_{1,Te}} \left[\left(\frac{p_{2,Te}}{p_{1,Te}}\right)^{\frac{\kappa_{Te}-1}{\kappa_{Te}}} - 1\right]$ $= \frac{1{,}4}{0{,}4} \times \frac{10^5 \times 1{,}002\,2}{1{,}149\,9} \times$ $\times \left[(1{,}289\,7)^{0{,}285\,7} - 1\right]$ $= 22\,998$ N·m/kg
Isentropic compression power	$P_{s,t,Te} = W_{m,s,t,Te}\, q_{m,Te}$ $= 23\,220 \times 6{,}752\,78 \times 10^{-3}$ $= 156{,}8$ kW	$P_{s,t,Te} = W_{m,s,t,Te}\, q_{m,Te}$ $= 23\,223 \times 6{,}752\,78 \times 10^{-3}$ $= 156{,}8$ kW	$P_{s,Te} = W_{m,s,Te}\, q_{m,Te}$ $= 22\,885 \times 6{,}752\,8 \times 10^{-3}$ $= 154{,}53$ kW	$P_{s,Te} = W_{m,s,t,Te}\, q_{m,Te}$ $= 22\,998 \times 6{,}752\,78 \times 10^{-3}$ $= 155{,}3$ kW
Power at compressor coupling	$P_{e,Te} = \text{Torque} \cdot 2\,\pi \cdot N_{Te}$ $= 10^{-3} \times 711{,}56 \times 2 \times \pi \times \frac{2\,881}{60}$ $= 214{,}7$ kW	$P_{e,Te} = \text{Torque} \cdot 2\,\pi \cdot N_{Te}$ $= 10^{-3} \times 711{,}56 \times 2 \times \pi \times \frac{2\,881}{60}$ $= 214{,}7$ kW	$P_{e,Te} = 214{,}7$ kW	$P_{e,Te} = \text{Torque} \cdot 2\,\pi \cdot N_{Te}$ $= 10^{-3} \times 711{,}56 \times 2 \times \pi \times \frac{2\,881}{60}$ $= 214{,}7$ kW

Table F.11 — Comparison of different calculation methods *(continued)*

Example 1	Total (p_t, t_t)	Total $\left(W_m = W_{m,\text{static}} + \frac{c_2^2 - c_1^2}{2}\right)$	Suction side : total Discharge side : static	Static
Test calculations *(continued)*				
Estimated mechanical losses	$P_{f,Te} = 3{,}0$ kW	$P_{f,Te} = 3{,}0$ kW	$P_{f,Te} = 3{,}0$ kW	$P_{f,Te} = 3{,}0$ kW
Internal power	$P_{in,Te} = P_{e,Te} - P_{f,Te}$ $= 214{,}7 - 3 = 211{,}7$ kW	$P_{in,Te} = P_{e,Te} - P_{f,Te}$ $= 214{,}7 - 3 = 211{,}7$ kW	$P_{in,Te} = 211{,}7$ kW	$P_{in,Te} = P_{e,Te} - P_{f,Te}$ $= 214{,}7 - 3 = 211{,}7$ kW
Internal isentropic efficiency	$\eta_{s,in,Te} = \frac{P_{s,t,Te}}{P_{in,Te}} = \frac{156{,}8}{211{,}7}$ $= 74{,}07$ %	$\eta_{s,in,Te} = \frac{P_{s,t,Te}}{P_{in,Te}} = \frac{156{,}8}{211{,}7}$ $= 74{,}07$ %	$\eta_{s,in,Te} = \frac{P_{s,Te}}{P_{in,Te}} = \frac{154{,}53}{211{,}7}$ $= 72{,}99$ %	$\eta_{s,in,Te} = \frac{P_{s,Te}}{P_{in,Te}} = \frac{155{,}3}{211{,}7}$ $= 73{,}4$ %
Polytropic exponent	$n_{Te} = \frac{\ln\left(\frac{p_{2,t,Te}}{p_{1,t,Te}}\right)}{\ln\left(\frac{p_{t,2,Te}T_{t,1,Te}}{p_{t,1,Te}T_{t,2,Te}}\right)}$ $= \frac{\ln\left(\frac{1{,}297\ 1}{1{,}003\ 5}\right)}{\ln\left(\frac{1{,}297\ 1 \times 301{,}41}{1{,}003\ 5 \times 332{,}83}\right)}$ $= 1{,}63$	$n_{Te} = \frac{\ln\left(\frac{p_{2,Te}}{p_{1,Te}}\right)}{\ln\left(\frac{p_{2,Te}T_{1,Te}}{p_{1,Te}T_{2,Te}}\right)}$ $= \frac{\ln\left(\frac{1{,}292\ 5}{1{,}002\ 2}\right)}{\ln\left(\frac{1{,}292\ 5 \times 301{,}25}{1{,}002\ 2 \times 332{,}35}\right)}$ $= 1{,}629$	$n_{Te} = \frac{\ln\left(\frac{p_{2,Te}}{p_{1,t,Te}}\right)}{\ln\left(\frac{p_{2,Te}T_{1,Te}}{p_{t,1,Te}T_{2,Te}}\right)}$ $= \frac{\ln\left(\frac{1{,}292\ 5}{1{,}003\ 5}\right)}{\ln\left(\frac{1{,}292\ 5 \times 301{,}41}{1{,}003\ 5 \times 332{,}35}\right)}$ $= 1{,}632\ 4$	$n_{Te} = \frac{\ln\left(\frac{p_{2,Te}}{p_{1,Te}}\right)}{\ln\left(\frac{p_{2,Te}T_{1,Te}}{p_{1,Te}T_{2,Te}}\right)}$ $= \frac{\ln\left(\frac{1{,}292\ 5}{1{,}002\ 2}\right)}{\ln\left(\frac{1{,}292\ 5 \times 301{,}25}{1{,}002\ 2 \times 332{,}35}\right)}$ $= 1{,}629$
Specific polytropic compression work	$W_{m,pol,t,Te} =$ $\frac{n_{Te}}{n_{Te}-1}\frac{p_{t,1,Te}}{\varrho_{t,Te}}\left[\left(\frac{p_{t,2,Te}}{p_{t,1,Te}}\right)^{\left(\frac{n-1}{n}\right)_{Te}} - 1\right]$ $= \frac{1{,}63}{0{,}63} \times \frac{10^5 \times 1{,}003\ 5}{1{,}150\ 8} \times$ $\times \left[(1{,}292\ 6)^{\frac{0{,}63}{1{,}63}} - 1\right]$ $= 23\ 528$ N·m/kg	$W_{m,pol,t,Te} =$ $\frac{n_{Te}}{n_{Te}-1}\frac{p_{1,Te}}{\varrho_{t,Te}}\left[\left(\frac{p_2}{p_1}\right)^{\left(\frac{n-1}{n}\right)_{Te}} - 1\right]$ $+ \frac{c_2^2 - c_1^2}{2}$ $= \frac{1{,}629}{0{,}629} \times \frac{10^5 \times 1{,}002\ 2}{1{,}149\ 9} \times$ $\times \left[(1{,}289\ 7)^{\frac{0{,}629}{1{,}629}} - 1\right] +$ $+ \frac{26{,}04^2 - 15{,}056\ 97^2}{2}$ $= 23\ 524$ N·m/kg	$W_{m,pol,Te} =$ $\frac{n_{Te}}{n_{Te}-1}\frac{p_{t,1,Te}}{\varrho_{t,1,Te}}\left[\left(\frac{p_2}{p_{t,1}}\right)^{\left(\frac{n-1}{n}\right)_{Te}} - 1\right]$ $= \frac{1{,}632\ 4}{0{,}632\ 4} \times \frac{10^5 \times 1{,}003\ 5}{1{,}150\ 8} \times$ $\times \left[\left(\frac{1{,}292\ 5}{1{,}003\ 5}\right)^{\frac{0{,}632\ 4}{1{,}632\ 4}} - 1\right]$ $= 23\ 187$ N·m/kg	$W_{m,pol,Te} =$ $\frac{n_{Te}}{n_{Te}-1}\frac{p_{1,Te}}{\varrho_{t,Te}}\left[\left(\frac{p_2}{p_1}\right)^{\left(\frac{n-1}{n}\right)_{Te}} - 1\right]$ $= \frac{1{,}629}{0{,}629} \times \frac{10^5 \times 1{,}002\ 2}{1{,}149\ 9} \times$ $\times \left[(1{,}289\ 7)^{\frac{0{,}629}{1{,}629}} - 1\right]$ $= 23\ 299$ N·m/kg
Internal polytropic efficiency	$\eta_{pol,in,Te} = \frac{W_{m,pol,t,Te}q_{m,Te}}{P_{in,Te}}$ $= \frac{23\ 528 \times 6{,}752\ 78 \times 10^{-3}}{211{,}7}$ $= 75{,}05$ %	$\eta_{pol,in,Te} = \frac{W_{m,pol,t,Te}q_{m,Te}}{P_{in,Te}}$ $= \frac{23\ 524 \times 6{,}752\ 78 \times 10^{-3}}{211{,}7}$ $= 75{,}05$ %	$\eta_{pol,in,Te} = \frac{W_{m,pol,Te}q_{m,Te}}{P_{in,Te}}$ $= \frac{23\ 187 \times 6{,}752\ 78 \times 10^{-3}}{211{,}7}$ $= 73{,}96$ %	$\eta_{pol,in,Te} = \frac{W_{m,pol,Te}q_{m,Te}}{P_{in,Te}}$ $= \frac{23\ 299 \times 6{,}752\ 78 \times 10^{-3}}{211{,}7}$ $= 74{,}32$ %

Table F.11 — Comparison of different calculation methods (*continued*)

Example 1	Total (p_t, t_t)	Total $\left(W_m = W_{m,\text{static}} + \frac{c_2^2 - c_1^2}{2}\right)$	Suction side: total Discharge side: static	Static
Conversion of test results to guarantee conditions				
Converted volumetric flow at inlet	$q_{V,t,1,Co} = q_{V,1,Te} \frac{N_{Gu}}{N_{Te}}$ $= 5{,}867\,9 \times \frac{4\,700}{2\,881}$ $= 9{,}572\,8$ m³/s	$q_{V,t,1,Co} = q_{V,1,Te} \frac{N_{Gu}}{N_{Te}}$ $= 5{,}867\,9 \times \frac{4\,700}{2\,881}$ $= 9{,}572\,8$ m³/s	$q_{V,t,1,Co} = 9{,}572\,8$ m³/s	$q_{V,t,1,Co} = q_{V,1,Te} \frac{N_{Gu}}{N_{Te}}$ $= 5{,}872\,5 \times \frac{4\,700}{2\,881}$ $= 9{,}580\,3$ m³/s
Converted specific polytropic compression work	$W_{m,pol,t,Co} = W_{m,pol,t,Te} \left(\frac{N_{Gu}}{N_{Te}}\right)^2$ $= 23\,528 \left(\frac{4\,700}{2\,881}\right)^2$ $= 62\,617$ N·m/kg	$W_{m,pol,t,Co} = W_{m,pol,t,Te} \left(\frac{N_{Gu}}{N_{Te}}\right)^2$ $= 23\,524 \left(\frac{4\,700}{2\,881}\right)^2$ $= 62\,607$ N·m/kg	$W_{m,pol,Co} = W_{m,pol,Te} \left(\frac{N_{Gu}}{N_{Te}}\right)^2$ $= 23\,187 \left(\frac{4\,700}{2\,881}\right)^2$ $= 62\,710$ N·m/kg	$W_{m,pol,Co} = W_{m,pol,Te} \left(\frac{N_{Gu}}{N_{Te}}\right)^2$ $= 23\,299 \left(\frac{4\,700}{2\,881}\right)^2$ $= 62\,008$ N·m/kg
Gas density at inlet under guarantee conditions	$\varrho_{t,1,Gu} = \frac{p_{t,1,Gu}}{Z_{Gu} R_{Gu} T_{t,1,Gu}}$ $= \frac{0{,}952\,4 \times 10^5}{1{,}0 \times 764{,}9 \times 303{,}31}$ $= 0{,}410\,5$ kg/m³	$\varrho_{t,1,Gu} = \frac{p_{t,1,Gu}}{Z_{Gu} R_{Gu} T_{t,1,Gu}}$ $= \frac{0{,}952\,4 \times 10^5}{1{,}0 \times 764{,}9 \times 303{,}31}$ $= 0{,}410\,5$ kg/m³	$\varrho_{t,1,Gu} = \frac{p_{t,1,Gu}}{Z_{Gu} R_{Gu} T_{t,1,Gu}}$ $= \frac{0{,}952\,4 \times 10^5}{1{,}0 \times 764{,}9 \times 303{,}31}$ $= 0{,}410\,5$ kg/m³	$\varrho_{1,Gu} = \frac{p_{1,Gu}}{Z_{Gu} R_{Gu} T_{1,Gu}}$ $= \frac{0{,}951\,2 \times 10^5}{1{,}0 \times 764{,}9 \times 303{,}15}$ $= 0{,}410\,2$ kg/m³
Converted pressure ratio ($n_{Co} = n_{Te}$ because $\kappa_{Gu} = \kappa_{Te}$)	$\left(\frac{p_2}{p_1}\right)_{Co} = \left[\left(\frac{n-1}{n}\right)_{Co} \times \frac{W_{m,pol,t,Co}}{Z_{Gu} R_{Gu} T_{1,Gu}} + 1\right]^{\left(\frac{n}{n-1}\right)_{Co}}$ $= \left(\frac{0{,}63 \times 62\,617}{1{,}63 \times 1{,}0 \times 764{,}9 \times 303{,}31} + 1\right)^{\frac{1{,}63}{0{,}63}}$ $= 1{,}29\,269$	$\left(\frac{p_2}{p_1}\right)_{Co} = \left[\left(\frac{n-1}{n}\right)_{Co} \times \frac{W_{m,pol,t,Co}}{Z_{Gu} R_{Gu} T_{1,Gu}} + 1\right]^{\left(\frac{n}{n-1}\right)_{Co}}$ $= \left(\frac{0{,}629 \times 62\,607}{1{,}629 \times 1{,}0 \times 764{,}9 \times 303{,}31} + 1\right)^{\frac{1{,}63}{0{,}63}}$ $= 1{,}29\,266$	$\left(\frac{p_2}{p_1}\right)_{Co} = \left[\left(\frac{n-1}{n}\right)_{Co} \times \frac{W_{m,pol,Co}}{Z_{Gu} R_{Gu} T_{1,t,Gu}} + 1\right]^{\left(\frac{n}{n-1}\right)_{Co}}$ $= \left(\frac{0{,}632\,4 \times 61\,710}{1{,}632\,4 \times 1{,}0 \times 764{,}9 \times 303{,}31} + 1\right)^{\frac{1{,}632\,4}{0{,}632\,4}}$ $= 1{,}288$	$\left(\frac{p_2}{p_1}\right)_{Co} = \left[\left(\frac{n-1}{n}\right)_{Co} \times \frac{W_{m,pol,Co}}{Z_{Gu} R_{Gu} T_{1,Gu}} + 1\right]^{\left(\frac{n}{n-1}\right)_{Co}}$ $= \left(\frac{0{,}629 \times 62\,008}{1{,}629 \times 1{,}0 \times 764{,}9 \times 303{,}15} + 1\right)^{\frac{1{,}629}{0{,}629}}$ $= 1{,}289\,8$
Converted internal power	$P_{in,Co} = P_{in,Te} \frac{\varrho_{t,1,Gu}}{\varrho_{t,1,Te}} \left(\frac{N_{Gu}}{N_{Te}}\right)^3$ $= 211{,}7 \times \frac{0{,}410\,5}{1{,}150\,8} \times \left(\frac{4\,700}{2\,881}\right)^3$ $= 327{,}87$ kW	$P_{in,Co} = P_{in,Te} \frac{\varrho_{t,1,Gu}}{\varrho_{t,1,Te}} \left(\frac{N_{Gu}}{N_{Te}}\right)^3$ $= 211{,}7 \times \frac{0{,}410\,5}{1{,}150\,8} \times \left(\frac{4\,700}{2\,881}\right)^3$ $= 327{,}87$ kW	$P_{in,Co} = P_{in,Te} \frac{\varrho_{t,1,Gu}}{\varrho_{t,1,Te}} \left(\frac{N_{Gu}}{N_{Te}}\right)^3$ $= 211{,}7 \times \frac{0{,}410\,5}{1{,}150\,8} \times \left(\frac{4\,700}{2\,881}\right)^3$ $= 327{,}87$ kW	$P_{in,Co} = P_{in,Te} \frac{\varrho_{1,Gu}}{\varrho_{1,Te}} \left(\frac{N_{Gu}}{N_{Te}}\right)^3$ $= 211{,}7 \times \frac{0{,}410\,2}{1{,}249\,9} \times \left(\frac{4\,700}{2\,881}\right)^3$ $= 327{,}88$ kW

Table F.11 — Comparison of different calculation methods *(concluded)*

Example 1	Total (p_t, t_t)	Total $\left(W_m = W_{m,\text{static}} + \frac{c_2^2 - c_1^2}{2}\right)$	Suction side : total Discharge side : static	Static
Conversion of test results to guarantee conditions (*continued*)				
Estimated mechanical losses	6,3 kW	6,3 kW	6,3 kW	6,3 kW
Converted power at compressor coupling	334,17 kW	334,17 kW	334,17 kW	334,18 kW
Specific polytropic compression work at guarantee conditions assuming the same polytropic efficiency as on test $n_{Co} = n_{Te}$	$W_{m,\text{pol,t,Gu}} =$ $\left(\frac{n}{n-1}\right)_{Co} Z_{Gu} R_{Gu} T_{t,1,Gu} \times$ $\times \left[\left(\frac{p_{t,2}}{p_{t,1}}\right)^{\left(\frac{n-1}{n}\right)_{Co}} - 1\right]$ $= \frac{1,63}{0,63} \times 764,9 \times 303,31 \times$ $\times \left[(1,281\,3)^{\frac{0,63}{1,63}} - 1\right]$ $= 60\,351$ N·m/kg	$W_{m,\text{pol,t,Gu}} =$ $\left(\frac{n}{n-1}\right)_{Co} Z_{Gu} R_{Gu} T_{1,Gu} \times$ $\times \left[\left(\frac{p_2}{p_1}\right)^{\left(\frac{n-1}{n}\right)_{Co}} - 1\right] + \frac{c_2^2 - c_1^2}{2}$ $= \frac{1,629}{0,629} \times 764,9 \times 303,15 \times$ $\times \left[(1,278\,4)^{\frac{0,629}{1,629}} - 1\right] +$ $+ \frac{42,481^2 - 26,04^2}{2}$ $= 60\,303$ N·m/kg	$W_{m,\text{pol,Gu}} =$ $\left(\frac{n}{n-1}\right)_{Co} Z_{Gu} R_{Gu} T_{1,Gu} \times$ $\times \left[\left(\frac{p_{2,Gu}}{p_{1,t,Gu}}\right)^{\left(\frac{n-1}{n}\right)_{Co}} - 1\right]$ $= \frac{1,632\,4}{0,632\,4} \times 764,9 \times 303,31 \times$ $\times \left[\left(\frac{1,216\,0}{0,952\,4}\right)^{\frac{0,632\,4}{1,632\,4}} - 1\right]$ $= 59\,456$ N·m/kg	$W_{m,\text{pol,Gu}} =$ $\left(\frac{n}{n-1}\right)_{Co} Z_{Gu} R_{Gu} T_{1,Gu} \times$ $\times \left[\left(\frac{p_2}{p_1}\right)_{Gu}^{\left(\frac{n-1}{n}\right)_{Co}} - 1\right]$ $= \frac{1,629}{0,629} \times 764,9 \times 303,15 \times$ $\times \left[(1,278\,4)^{\frac{0,629}{1,629}} - 1\right]$ $= 59\,740$ N·m/kg
Corrected internal power at guarantee pressure ratio	$P_{e,Co,Gu} =$ $P_{in,Co} \frac{q_{V,t,1,Gu} W_{m,\text{pol,t,Gu}}}{q_{V,t,1,Co} W_{m,\text{pol,t,Co}}}$ $= 327,87 \frac{9,715 \times 60\,351}{9,572\,8 \times 62\,617}$ $= 320,4$ kW	$P_{e,Co,Gu} =$ $P_{in,Co} \frac{q_{V,t,1,Gu} W_{m,\text{pol,t,Gu}}}{q_{V,t,1,Co} W_{m,\text{pol,t,Co}}}$ $= 327,87 \frac{9,715 \times 60\,303}{9,572\,8 \times 62\,607}$ $= 320,5$ kW	$P_{e,Co,Gu} =$ $P_{in,Co} \frac{q_{V,t,1,Gu} W_{m,\text{pol,Gu}}}{q_{V,t,1,Co} W_{m,\text{pol,Co}}}$ $= 327,87 \frac{9,715 \times 59\,456}{9,572\,8 \times 61\,710}$ $= 320,59$ kW	$P_{e,Co,Gu} =$ $P_{in,Co} \frac{q_{V,1,Gu} W_{m,\text{pol,Gu}}}{q_{V,1,Co} W_{m,\text{pol,Co}}}$ $= 327,88 \frac{9,722\,22 \times 59\,740}{9,580\,3 \times 62\,008}$ $= 320,6$ kW
Corrected power at coupling	320,7 6,3 327,0 kW	320,5 6,3 326,8 kW	320,59 6,3 326,89 kW	320,6 6,3 326,9 kW
Corrected specific power at coupling	0,009 35 kWh/m³	0,009 344 kWh/m³	0,009 346 kWh/m³	0,009 34 kWh/m³
Deviation from guarantee	0,57 %	0,51 %	0,527 %	0,538 %

Example 1

$$q_{V,1,\mathrm{Te}} = 5{,}872\ 22\ \mathrm{m^3/s}$$

$$A_1 = 0{,}39\ \mathrm{m^2}$$

$$c_{1,\mathrm{Te}} = \frac{5{,}872\ 22}{0{,}39} = 15{,}056\ 97\ \mathrm{m/s}$$

$$c_{1,\mathrm{Gu}} = 15{,}056\ 97\ \frac{N_{\mathrm{Gu}}}{N_{\mathrm{Te}}} = 24{,}563\ 6\ \mathrm{m/s}$$

$$q_{V,2,\mathrm{Te}} = \frac{6{,}752\ 78}{1{,}344\ 3} = 5{,}023\ 3$$

$$A_2 = 0{,}192\ 8\ \mathrm{m^2}$$

$$c_{2,\mathrm{Te}} = \frac{5{,}021}{0{,}192\ 8} = 26{,}04\ \mathrm{m/s}$$

$$c_{2,\mathrm{Gu}} = 26{,}04\ \frac{N_{\mathrm{Gu}}}{N_{\mathrm{Te}}} = 42{,}481\ \mathrm{m/s}$$

$$Ma_{\mathrm{t},1,\mathrm{Te}} = \frac{q_m}{A_1 p_{1,\mathrm{Te}}} \sqrt{\frac{ZRT_{\mathrm{t},1}}{\kappa}}$$

$$= \frac{6{,}752\ 78}{0{,}39 \times 1{,}002\ 2 \times 10^5} \times \sqrt{\frac{1 \times 289{,}3 \times 301{,}3}{1{,}4}}$$

$$= 0{,}043\ 11$$

$$p_{\mathrm{t},1,\mathrm{Te}} = p_{1,\mathrm{Te}} \left(1 + \frac{\kappa}{2} Ma^2_{\mathrm{t},1,\mathrm{Te}}\right)$$

$$= 1{,}002\ 2 \left(1 + \frac{1{,}4}{2} \times 0{,}043\ 11^2\right)$$

$$= 1{,}003\ 5\ \mathrm{bar}$$

$$T_{\mathrm{t},1,\mathrm{Te}} = T_{1,\mathrm{Te}} \frac{1}{1 - \frac{\kappa - 1}{\kappa} Ma^2_{\mathrm{t,Te}}}$$

$$= 301{,}25\ \frac{1}{1 - 0{,}285\ 7 \times 0{,}431\ 1^2}$$

$$= 301{,}41\ \mathrm{K}$$

$$Ma_{\mathrm{t},2,\mathrm{Te}} = \frac{q_m}{A_2 p_{2,\mathrm{Te}}} \sqrt{\frac{ZRT_{\mathrm{t},2,\mathrm{Te}}}{\kappa}}$$

$$= \frac{6{,}752\ 78}{0{,}192\ 8 \times 1{,}292\ 5 \times 10^5} \times \sqrt{\frac{1 \times 289{,}3 \times 332{,}35}{1{,}4}}$$

$$= 0{,}071$$

$$p_{\mathrm{t},2,\mathrm{Te}} = p_{2,\mathrm{Te}} \left(1 + \frac{\kappa}{2} Ma^2_{\mathrm{t},2,\mathrm{Te}}\right)$$

$$= 1{,}292\ 5 \times \left(1 + \frac{1{,}4}{2} \times 0{,}071^2\right)$$

$$= 1{,}297\ 1\ \mathrm{bar}$$

$$T_{\mathrm{t},2,\mathrm{Te}} = T_{2,\mathrm{Te}} \left(\frac{1}{1 - \frac{\kappa - 1}{\kappa} Ma^2_{\mathrm{t,Te}}}\right)$$

$$= 332{,}35 \times \left(\frac{1}{1 - 0{,}285\ 7 \times 0{,}071^2}\right)$$

$$= 332{,}83\ \mathrm{K}$$

F.2 Example 2 — Uncooled turbocompressor, isentropic exponent $\kappa_{\mathrm{Te}} = \kappa_{\mathrm{Gu}}$, speed not variable

F.2.1 General

The test conditions under inlet conditions deviate from the guarantee conditions. The driver has a fixed speed so that similar flow conditions can only be approximated using figure D.3.

The guarantee conditions, guaranteed performance and other design values are given in tables F.12 to F.14.

F.2.2 Purpose of test

The purpose of the test is to prove the guaranteed power requirement for the guarantee point and to determine the pressure-volume curve at a constant speed.

F.2.3 Design of installation

The installation consists of a three-stage, uncooled turbocompressor for air, driven by an electric motor.

Table F.12 — Guarantee conditions

Designation	Symbol	Numerical value	Unit
Inlet pressure (total)	$p_{\mathrm{t},1,\mathrm{Gu}}$	0,980 7	bar
Inlet temperature (total)	$t_{\mathrm{t},1,\mathrm{Gu}}$	30	°C
Gas constant	R_{Gu}	288,32	N·m/(kg·K)
Isentropic exponent	κ_{Gu}	1,4	—
Compressor speed	N_{Gu}	9 500	r/min

Table F.13 — Guaranteed performance

Designation	Symbol	Numerical value	Unit
Volumetric flow at inlet	$q_{V,t,1,Gu}$	9,166 67	m^3/s
Discharge pressure	$p_{t,2,Gu}$	3,903 1	bar
Power requirement at compressor coupling	$P_{e,Gu}$	2 065	kW

Table F.14 — Other design values

Designation	Symbol	Numerical value	Unit
Outside diameter of Stage 1 impeller	D_I	0,63	m

F.2.4 Test set-up

The test is carried out at the compressor site under atmospheric conditions.

The test set-up is shown in figure F.2, and the test conditions are given in table F.15.

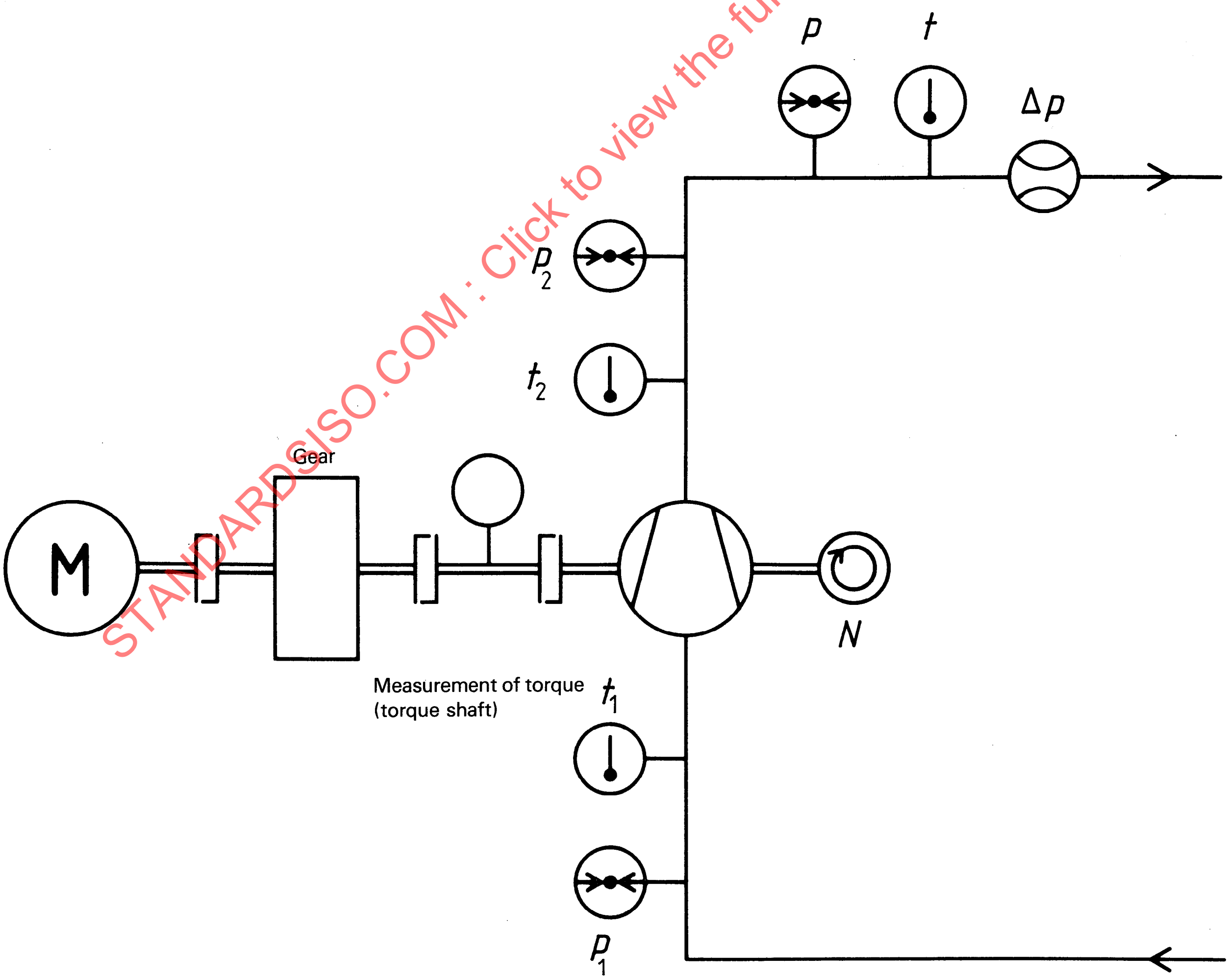


Figure F.2 — Test set-up for example 2

F.2.5 Setting conditions (see 8.2.3.2)

Since the speed cannot be varied, it is impossible to maintain precisely the flow conditions on which the guarantee was based.

$$\frac{\left(\dfrac{N}{\sqrt{RZ_1T_{t,1}}}\right)_{\mathrm{Te}}}{\left(\dfrac{N}{\sqrt{RZ_1T_{t,1}}}\right)_{\mathrm{Gu}}} = \frac{\dfrac{9\,500}{\sqrt{288{,}3 \times 1{,}0 \times 283{,}2}}}{\dfrac{9\,500}{\sqrt{288{,}3 \times 1{,}0 \times 303{,}2}}} = 1{,}035$$

To determine the permissible deviations from figure D.3, it is necessary to calculate the polytropic exponent from guarantee conditions and guaranteed performance.

The deviation in the volumetric flow ratios shown in figure D.3 is less than 5 % with the calculated ratio 1,035, a pressure ratio $(p_2/p_1)_{t,\mathrm{Gu}}$ = 3,98, and a polytropic exponent n_{Gu} = 1,564.

The setting conditions are given in table F.16.

Since a centrifugal compressor with flat characteristic is involved, a maximum deviation in the volumetric flow ratios of ± 5 % is acceptable in the optimum efficiency range [see figure D.3c)] and, therefore, conversion of the test results to guarantee conditions is still possible. In accordance with figure D.2, a supplementary tolerance of 1 % for the measured power requirement can be used.

In determining the setting conditions in accordance with the flow chart in figure D.6 the following calculation sequence is used (the numbers given are those designating individual boxes in the flow chart):

1 — 2 — 3 — 4 — 5 — 6 — 7 — 8 — 9 — 10 — 11 — 12 — 13 — 21 — 22 — 23 — 24

$\Delta V_{r,\mathrm{tol}}$ = + 0,05 is inserted in 23 in the first instance and $\Delta V_{r,\mathrm{tol}}$ = − 0,05 in the next sequence to determine the upper and lower tolerance limits. The calculation ends with 25 — 26 — (43 — 44, but see the note to F.0.1) — 45 — 46 — 47.

Had the test shown in 45 that the setting conditions determined could not be maintained in the test, the compressor would have had to be split up into several units and the calculations recommenced at 3 through 48 — 49.

The deviation is within the permitted range. The test was carried out maintaining the above test conditions.

Table F.15 — Test conditions

Designation	Symbol	Numerical value	Unit
Isentropic exponent	κ_{Te}	1,4	—
Mean inlet temperature (total)	$t_{t,1,\mathrm{Te}}$	10	°C
Gas constant	R_{Te}	288,3	N·m/(kg·K)
Speed (not variable)	N_{Te}	9 500	r/min

Table F.16 — Setting conditions

Designation	Symbol	Numerical value	Unit
Mechanical efficiency	η_{f}	0,985 (estimated)	—
Internal power	$P_{\mathrm{in,Gu}} = \eta_{\mathrm{f}} P_{\mathrm{e}}$	2 034	kW
Specific isentropic compression work	$W_{m,\mathrm{s,Gu}}$	148 024	N·m/kg
Isentropic compression power	$P_{\mathrm{s,Gu}}$	1 521	kW
Isentropic efficiency	$\eta_{\mathrm{s,in,Gu}} = \dfrac{P_{\mathrm{s}}}{P_{\mathrm{in,Gu}}}$	0,748	—
Pressure ratio	$\dfrac{p_{t,2}}{p_{t,1,\mathrm{Gu}}}$	3,98	—
Polytropic efficiency	$\eta_{\mathrm{pol,Gu}}$	0,792	—
Polytropic exponent	n_{Gu}	1,564	—

F.2.6 Test results and conversion

See tables F.17 to F.19.

Table F.17 — Test results

Designation	Symbol	Numerical value Test No. 1	Test No. 2	Test No. 3	Unit
Date of test	—	1965-04-30	1965-04-30	1965-04-30	—
Time of test	—	11,30	12,30	14,00	h
Atmospheric pressure	p_a	1,009 1	1,009 1	1,009 1	bar
Gas constant	R_{Te}	288,32	288,32	288,32	N·m/(kg·K)
Speed	N_{Te}	9 500	9 500	9 500	r/min
Mass flow *)	$q_{m,Te}$	11,566 7	11,341 7	10,202 8	kg/s
Inlet pressure	$p_{t,1,Te}$	0,996 4	0,997 3	0,999 3	bar
Inlet temperature	$t_{t,1,Te}$	9,8	10,0	10,3	°C
	$T_{t,1,Te}$	283,0	283,2	283,5	K
Density	$\varrho_{t,1,Te}$	1,221	1,221	1,223	kg/m³
Volumetric flow at inlet (also usable volumetric flow at inlet)	$q_{V,t,1,Te}$	9,47	9,289	8,342	m³/s
Discharge pressure	$p_{t,2,Te}$	3,765 8	4,148 2	4,903 3	bar

*) Calculated from the mass flow measured in accordance with ISO 5167-1.

Table F.18 — Calculation results

Designation	Symbol	Numerical value Test No. 1	Test No. 2	Test No. 3	Unit
Pressure ratio	$(p_2/p_1)_{t,Te}$	3,780	4,159	4,907	—
Specific isentropic compression work	$W_{m,s,Te}$	131 985	143 640	164 596	N·m/kg
Isentropic compression power	$P_{s,Te}$	1 527	1 628	1 679	kW
Power at compressor coupling	$P_{e,Te}$	2 107	2 208	2 234	kW
Mechanical compressor power dissipation	$P_{f,Te}$	31	32	32	kW
Internal power	$P_{in,Te}$	2 076	2 176	2 202	kW
Internal isentropic efficiency	$\eta_{s,in,Te}$	0,734	0,748	0,763	—
Internal polytropic efficiency	$\eta_{pol,in,Te}$	0,779	0,792	0,808	—
Polytropic exponent	n_{Te}	1 579	1 564	1 547	—
Specific polytropic compression work	$W_{m,pol,Te}$	139 880	152 150	174 462	N·m/kg

Table F.19 — Conversion to guarantee condition (see figure D.8)

Designation	Symbol	Numerical value Test No. 1	Test No. 2	Test No. 3	Unit
Converted specific compression work	$W_{m,pol,Co}$	139 880	152 150	174 462	N·m/kg
Converted pressure ratio	$(p_2/p_1)_{t,Co}$	3,521	3,860	4,529	—
Converted discharge pressure	$p_{t,2,Co}$	3,453	3,786	4,44	bar
Converted volumetric flow at inlet	$q_{V,t,1,Co}$	9,473	9,289	8,342	m³/s
Converted internal power	$P_{in,Co}$	1 907	1 998	2 020	kW
Converted mechanical power dissipation	$P_{f,Co}$	31	32	32	kW
Converted power at coupling	$P_{e,Co}$	1 938	2 030	2 052	kW

F.2.7 Test uncertainty and comparison with guarantee

The test uncertainties are calculated according to 9.2.

As can be seen in figures F.3 and F.4, the guarantees are met within the test uncertainty (see also 9.4).

The test uncertainty for example 2 is shown in table F.20.

Table F.20 — Resulting test uncertainty

Designation	Symbol	Numerical value	Comments
Uncertainty in volumetric flow at inlet	$\tau_{q_m,\mathrm{Te}}$	1,2 %	In accordance with ISO 5167-1
	$\tau_{N,\mathrm{Te}}$	0,53 %	Quality grade 0,5, final scale value 10 000
	$\tau_{p_1,\mathrm{Te}}$	0,13 %	1 mmHg on absolute pressure
	$\tau_{T_1,\mathrm{Te}}$	0,33 %	1 ° on absolute temperature
	τ_{adj}	1,0 %	
	$\tau_{\mathrm{tot},q_V,1}$	$\sqrt{1{,}36^2 + 1{,}0^2} = 1{,}69$ %	See 9.2.5
Uncertainty in pressure ratio	N_r	1,035	
	$\ln(p_2/p_1)$	1,33	
	$\tau_{p_2,\mathrm{Te}}$	0,83 %	Quality grade 0,6; final pressure gauge value 4 atm gauge (relative)
	τ_{adj}	1,0 %	
	$\tau_{\mathrm{tot},p_2/p_1}$	$\sqrt{1{,}586^2 + 1{,}0^2} = 1{,}87$ %	See 9.2.6
Uncertainty in power at coupling	ζ_4	780	Current transformer 0,5; voltage transformer 0,5; wattmeter 0,2 (estimated)
	τ_{P_e}	0,75 %	
	τ_{P_f}	10 %	
	τ_{adj}	1,0 %	
	$\tau_{\mathrm{tot},P_\mathrm{e}}$	$\sqrt{0{,}975^2 + 1{,}0^2} = 1{,}40$ %	See table 9

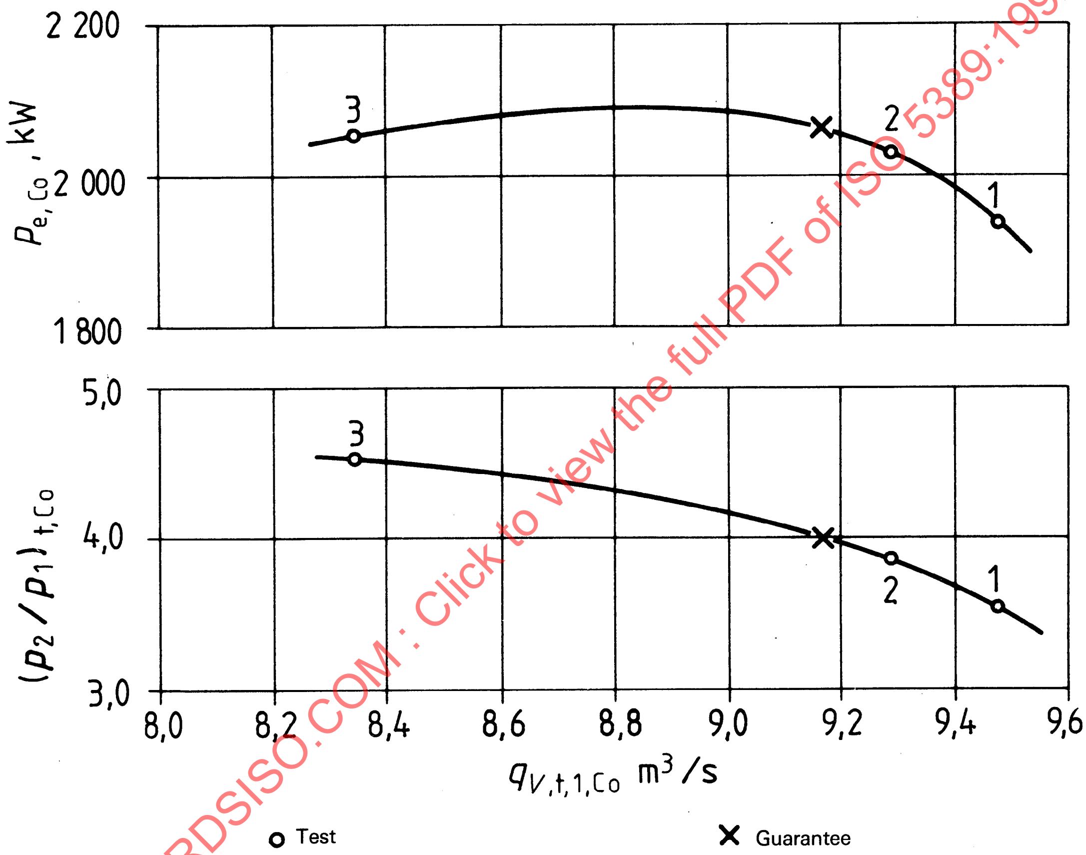


Figure F.3 — Graphs for comparison with guarantee

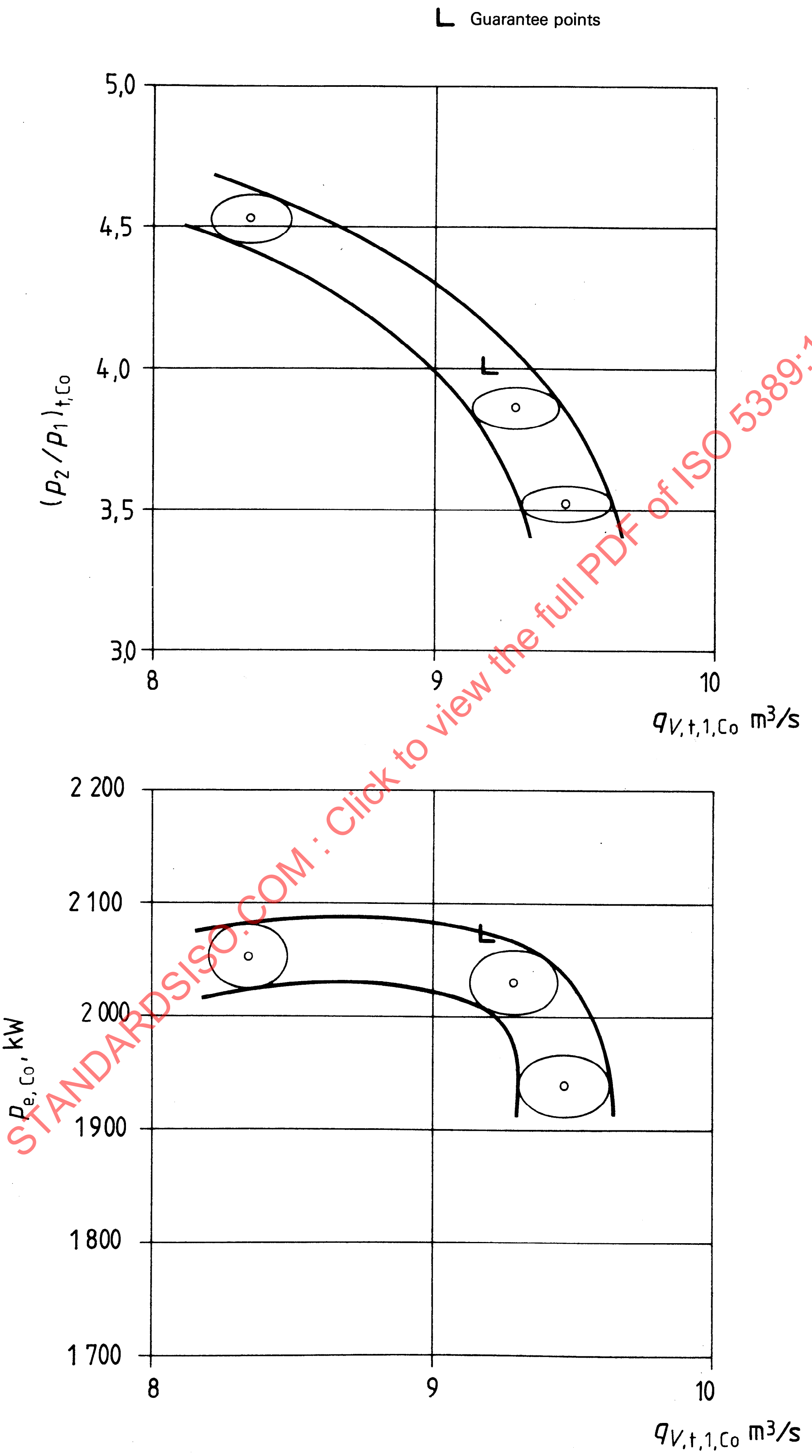


Figure F.4 — Graphs for comparison with guarantee

F.3 Example 3 — Uncooled turbocompressor, isentropic exponent $\kappa_{Te} \neq \kappa_{Gu}$, speed not variable

F.3.1 General

No similar flow conditions can be set in accordance with the similarity considerations given in 8.2.3.2. Approximation solutions are sought on the basis of figure D.4.

The guarantee conditions, guaranteed performance and other design values are given in tables F.21 to F.23.

F.3.2 Purpose of tests

The purpose of the tests is to prove the guaranteed power requirement for two guarantee points and to determine the pressure-volume curve at constant speed.

F.3.3 Design of installation

The installation consists of a two-stage, uncooled turbocompressor for ethylene, driven by an electric motor.

Table F.21 — Guarantee conditions

Designation	Symbol	Numerical value	Unit
Inlet pressure	$p_{t,1,Gu}$	0,980	bar
Inlet temperature	$t_{t,1,Gu}$	32	°C
Gas constant	R_{Gu}	296,65	N·m/(kg·K)
Isentropic exponent	κ_{Gu}	1,25	—
Compressor speed	N_{Gu}	12 700	r/min

Table F.22 — Guaranteed performance

Designation	Symbol	Numerical value		Unit
		Guarantee point (a)	Guarantee point (b)	
Volumetric flow at inlet	$q_{V,t,1,Gu}$	15 000	12 500	m^3/h
Discharge pressure	$p_{t,2,Gu}$	1,765	1,863	bar
Power requirement at compressor coupling	$P_{e,Gu}$	335	310	kW

Table F.23 — Other design values

Designation	Symbol	Numerical value	Unit
Outside diameter of Stage 1 impeller	D_I	0,4	m

F.3.4 Test set-up

Since it is not possible to carry out the acceptance test using ethylene, a test is carried out on site using air.

The test set-up is illustrated in figure F.5, and the test conditions are given in table F.24. The power at the coupling is determined by the motor rating and transmission losses.

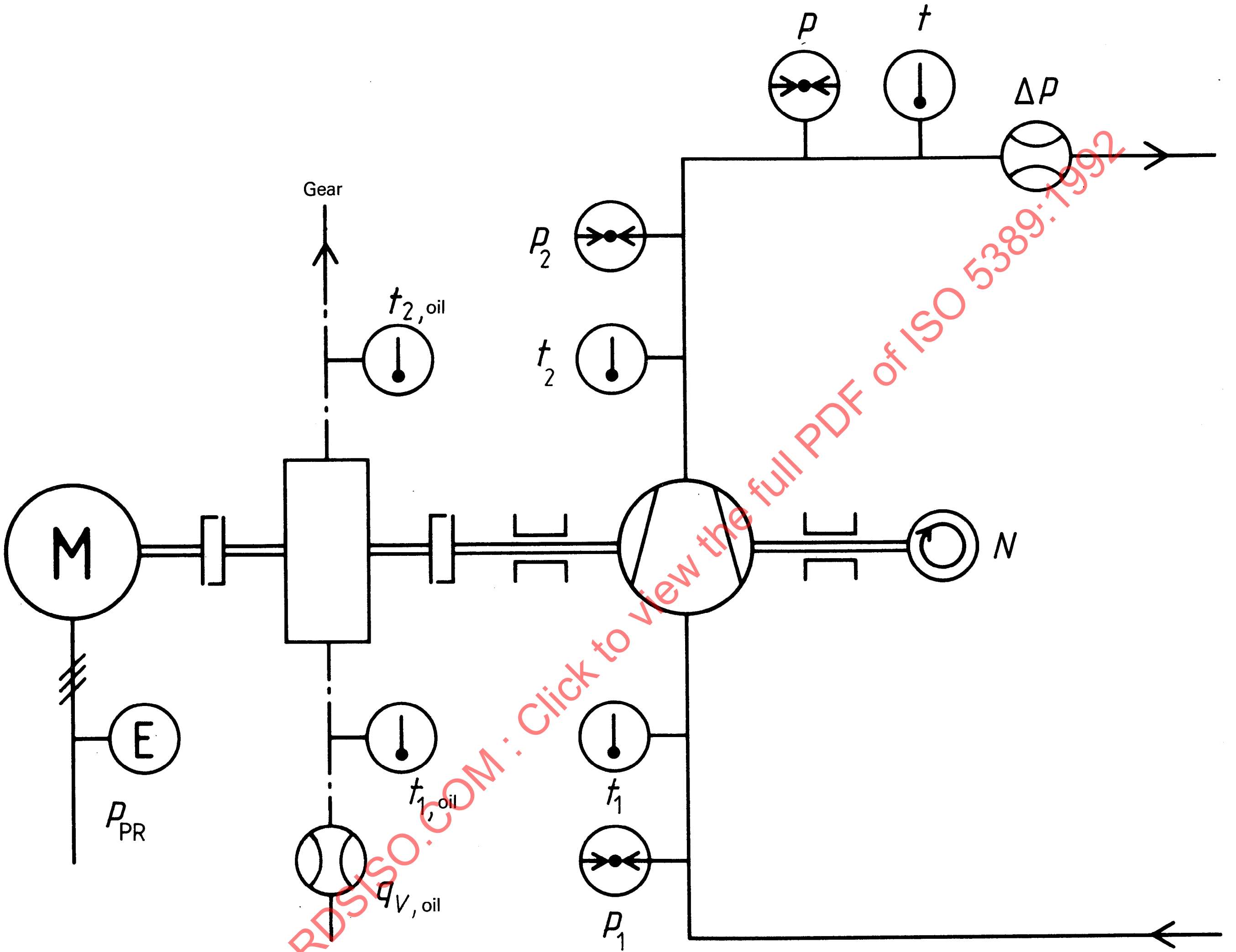


Figure F.5 — Test set-up for example 3

Table F.24 — Test conditions

Designation	Symbol	Numerical value	Unit
Isentropic exponent (air)	κ_{Te}	1,4	—
Mean inlet temperature	$t_{t,1,Te}$	5	°C
Gas constant	R_{Te}	287,3	N·m/(kg·K)

F.3.5 Setting conditions (see 8.2.3.3)

The speed cannot be varied.

Because the speed cannot be varied, it is impossible to maintain the flow ratios presupposed on giving the guarantee.

$$\frac{\left(\dfrac{N}{\sqrt{RZ_1T_{t,1}}}\right)_{Te}}{\left(\dfrac{N}{\sqrt{RZ_1T_{t,1}}}\right)_{Gu}} = \frac{\dfrac{12\,700}{\sqrt{287{,}3 \times 1{,}0 \times 278{,}2}}}{\dfrac{12\,700}{\sqrt{296{,}65 \times 1{,}0 \times 305{,}2}}} = 1{,}064$$

In order to determine whether the calculated ratio is within the permitted range (see 8.2.3.2), the polytropic exponents are required. The internal isentropic efficiency η_s required to determine $\eta_{pol,Gu}$ is determined here from the isentropic compression power formed by the guarantee values and from the guaranteed power at the coupling assuming a mechanical efficiency $\eta_f = 0{,}98$.

According to figures D.4g) and D.4k), the deviation in the volumetric flow ratio is so high that it is impossible to conduct the test within the inner tolerance limits (0,99 to 1,01).

Consequently, a check must be made to find out whether the test is possible within the outer tolerance limits (0,95 to 1,05) which can be used in the case of the anticipated flat characteristic of a centrifugal compressor in the range of the guarantee points (see figure D.2).

It is expedient to draw an auxiliary diagram (see figures F.6 and F.7) for each of the two guarantee points from figures D.4j), D.4m) and D.4q) for better interpolation. Using these diagrams, it was found that a test could be carried out on the present compressor for conversion within the outer tolerance limits $0{,}95 < V_r < 1{,}05$, in the case of guarantee point (a) ($q_{V,t,1,Gu} = 15\,000$ m³/h) with

$$1{,}033 \leqslant \frac{\left(\dfrac{N}{\sqrt{RZ_1T_{t,1}}}\right)_{Te}}{\left(\dfrac{N}{\sqrt{RZ_1T_{t,1}}}\right)_{Gu}} \leqslant 1{,}175$$

and in the case of guarantee point (b) ($q_{V,t,1,Gu} = 12\,500$ m³/h) with

$$1{,}043 \leqslant \frac{\left(\dfrac{N}{\sqrt{RZ_1T_{t,1}}}\right)_{Te}}{\left(\dfrac{N}{\sqrt{RZ_1T_{t,1}}}\right)_{Gu}} \leqslant 1{,}180$$

The value determined at 1,064 in the above example for the two guarantee points is within these limits.

The setting conditions are given in table F.25.

Table F.25 — Setting conditions

Designation	Symbol	Numerical value Guarantee point (a)	Numerical value Guarantee point (b)	Unit
Specific isentropic compression work	$W_{m,s,Gu}$	56	6	N·m/kg
Density	$\varrho_{t,1,Gu}$	1,083	1,083	kg/m³
Isentropic compression power	$P_{s,Gu}$	255	233	kW
Power at the compressor coupling	$P_{e,Gu}$ ($\eta_f = 0{,}98$)	335	310	kW
Isentropic efficiency	$\eta_{s,in,Gu}$	0,777	0,767	—
Polytropic efficiency	$\eta_{pol,in,Gu}$	0,789	0,781 5	—
Specific polytropic compression work	$W_{m,pol,Gu}$	57 448	63 214	N·m/kg
Converted guaranteed polytropic exponent	n_{Gu}	1,34	1,344	—
Polytropic exponent calculated using $\eta_{pol,in,Gu}$ in the test	n_{Te}*)	1,567	1,567	—

*) Since there are no other details available than those given in the guarantee, n_{Te} is formed using $\eta_{pol,in,Gu}$.

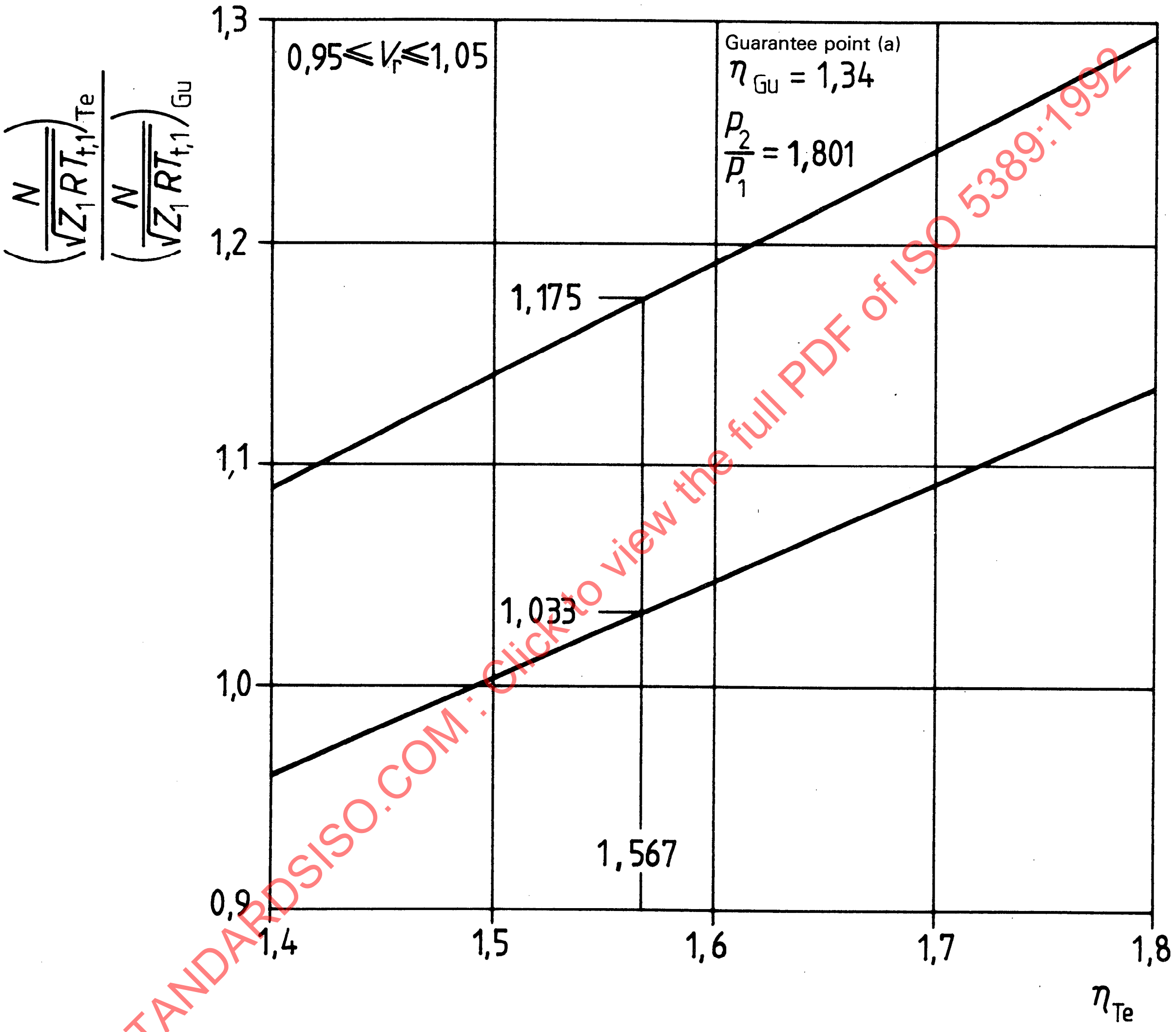


Figure F.6 — Graph for comparison with guarantee

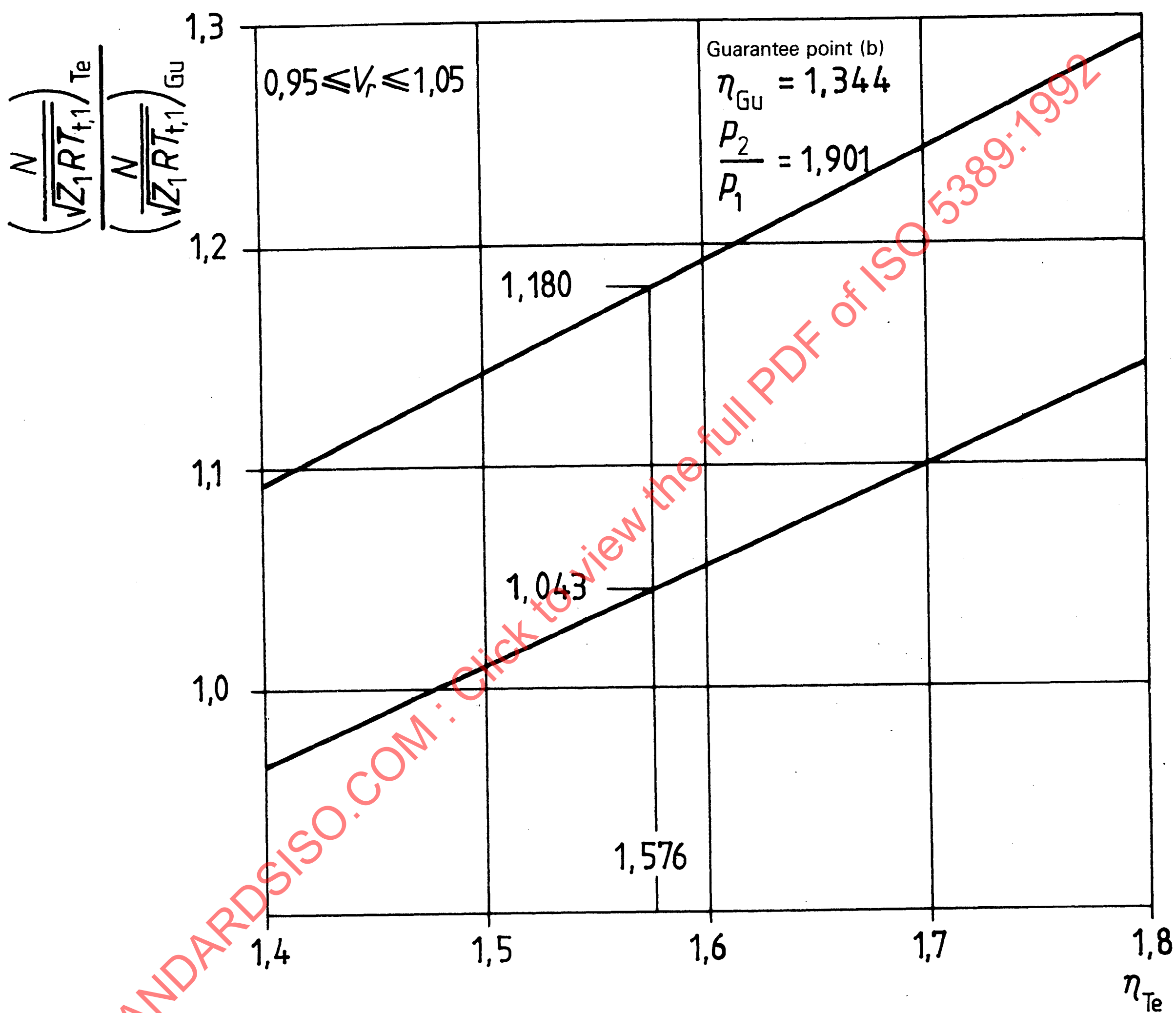


Figure F.7 — Graph for comparison with guarantee

In determining the setting conditions in accordance with the flow chart in figure D.6, the following calculation sequence is used (the numbers given are those designating individual boxes in the flow chart):

1 — 2 — 3 — 4 — 5 — 6 — 7 — 8 — 9 — 10 — 11 — 12 — 13 — 21 — 22 — 28 — 29

$\Delta V_{r,tol} = +0{,}05$ or $\Delta V_{r,tol} = -0{,}05$ is inserted into 28 to determine the upper and lower tolerance limits. In the present instance, the pressure ratio $(p_x/p_1)_{Ex}$ determined in 29 at which the maximum deviation $\Delta V_{r,Ex} = \Delta V_{r,tol}$ occurs is outside the range 1 to $(p_2/p_1)_{Gu}$.

Consequently, the formula sequence continues through 30 — 38 — 24 — 25 — 26 — (43 — 44, but see the note to F.0.1) — 45 — 46 — 47.

Had the calculated pressure ratio $(p_x/p_1)_{Ex}$ been within the range of the compression sequence, the extreme value of the deviation $\Delta V_{r,Ex}$ could have been determined by 31 and the limit $V_{r,2}$ would have had to be adapted by 32 — 37 in such a way that at no point of the compression process $\Delta V_{r,x} > \Delta V_{r,tol}$.

The test was carried out maintaining the above test conditions.

F.3.6 Test results and conversion

See tables F.26 to F.28.

It was established after the tests that all 3 test points are within the extreme tolerance limits to figure D.8.

Table F.26 — Test results

Designation	Symbol	Numerical value Test No. 1	Test No. 2	Test No. 3	Unit
Date of test	—	1965-03-12	1965-03-12	1965-03-12	—
Time of test	—	16,30	18,45	19,00	h
Atmospheric pressure	p_a	1,005 1	1,005	1,005	bar
Gas constant	R_{Te}	287,33	287,33	287,33	N·m/(kg·K)
Compressor speed	N_{Te}	12 700	12 700	12 700	r/min
Mass flow*) (also usable mass flow)	$q_{m,Te}$	19 985	17 304	14 720	kg/h
Inlet pressure	$p_{t,1,Te}$	0,995 4	0,997 3	0,999 3	bar
Inlet temperature	$t_{t,1,Te}$ $T_{t,1,Te}$	4,7 277,9	5,1 278,3	5,3 278,5	°C K
Density	$\varrho_{t,1,Te}$	1,246 6	1,247 2	1,248 8	kg/m³
Volumetric flow at inlet (also usable volumetric flow at inlet)	$q_{V,t,1,Te}$	16 032	13 874	11 787	m³/s
Discharge pressure	$p_{t,2,Te}$	1,857	1,974	2,041	bar

*) In accordance with ISO 5167-1.

Table F.27 — Calculation results

Designation	Symbol	Numerical value Test No. 1	Test No. 2	Test No. 3	Unit
Pressure ratio	$(p_2/p_1)_t$	1,866	1,979	2,042	—
Specific compression work	$W_{m,s,Te}$	54 524	60 269	65 373	N·m/kg
Isentropic compression power	$P_{s,Te}$	302,5	289,8	259,1	kW
Power at compressor coupling	$P_{e,Te}$	421,8	385,5	348,5	kW
Mechanical compressor power dissipation	$P_{f,Te}$	8	8	8	kW
Internal power	$P_{in,Te}$	413,8	377,5	340,5	kW
Internal isentropic efficiency	$\eta_{s,in,Te}$	0,731	0,768	0,761	—
Internal polytropic efficiency	$\eta_{pol,in,Te}$	0,754	0,788	0,784	—
Polytropic exponent	n_{Te}	1,611	1,566	1,573	—
Specific polytropic compression work	$W_{m,pol,Te}$	56 195	61 932	65 246	N·m/kg

Table F.28 — Conversion to guarantee conditions (see figure D.8)

Designation	Symbol	Numerical value Test No. 1	Test No. 2	Test No. 3	Unit
Converted specific polytropic compression work	$W_{m,\text{pol,Co}}$	56 195	61 932	65 246	N.m/kg
Converted polytropic exponent	n_{Co}	1,361	1,340	1,342	—
Converted pressure ratio	$(p_2/p_1)_{\text{t,Co}}$	1,777	1,879	1,937	—
Converted inlet volume	$q_{V,\text{t,1,Co}} = q_{V,\text{t,1,Te}}$	16 032	13 874	11 787	m^3/h
Converted internal power requirement	$P_{\text{in,Co}}$	359,6	327,9	295,3	kW
Converted mechanical power dissipation	$P_{\text{f,Co}}$	8	8	8	kW
Converted power at coupling	$P_{\text{e,Co}}$	367,6	335,9	303,3	kW

F.3.7 Test uncertainty and comparison with guarantee

The test uncertainties are calculated in accordance with 9.2.

The compressor in question having a flat performance curve, the comparison will be made at the guaranteed inlet volume.

For this purpose the converted test results, i.e. specific polytropic compression work, pressure ratio, internal polytropic efficiency and power input at coupling, are plotted versus the converted usable inlet volume (see figure F.8). Taking into account the total resulting inaccuracies, the conclusions are as follows (see figure F.9).

— Guarantee point (b), $q_{V,\text{t,1,Gu}} = 12\ 500\ m^3/h$: the requirements of the guarantee are met for both pressure ratio and power consumption.

— Guarantee point (a), $q_{V,\text{t,1,Gu}} = 15\ 000\ m^3/h$: the pressure ratio is higher than guaranteed, and accordingly, so is the power input. For better comparison it is necessary to apply the procedure outlined in 9.4. The converted test results are read from the curves plotted in figure F.8 and defined by the three test points, at the guaranteed inlet volume flow $q_{V,\text{t,1,Gu}} = 15\ 000\ m^3/h$, i.e.

$W_{m,\text{pol,Co}} = 59\ 470$ N·m/kg

$P_{\text{e,Co}} = 353{,}2$ kW

— the power at coupling reduced to guarantee conditions is

$$P_{\text{e,Co,Gu}} = P_{\text{e,Co}} \times \frac{W_{m,\text{pol,Gu}}}{W_{m,\text{pol,Co}}}$$

$$= 353{,}2 \times \frac{57\ 448}{59\ 470} = 341{,}2 \text{ kW}$$

and the guarantee value is 335 kW.

The converted power is thus slightly in excess, even if the resulting total test uncertainty is taken into account.

The test uncertainty for example 3 is shown in table F.29.

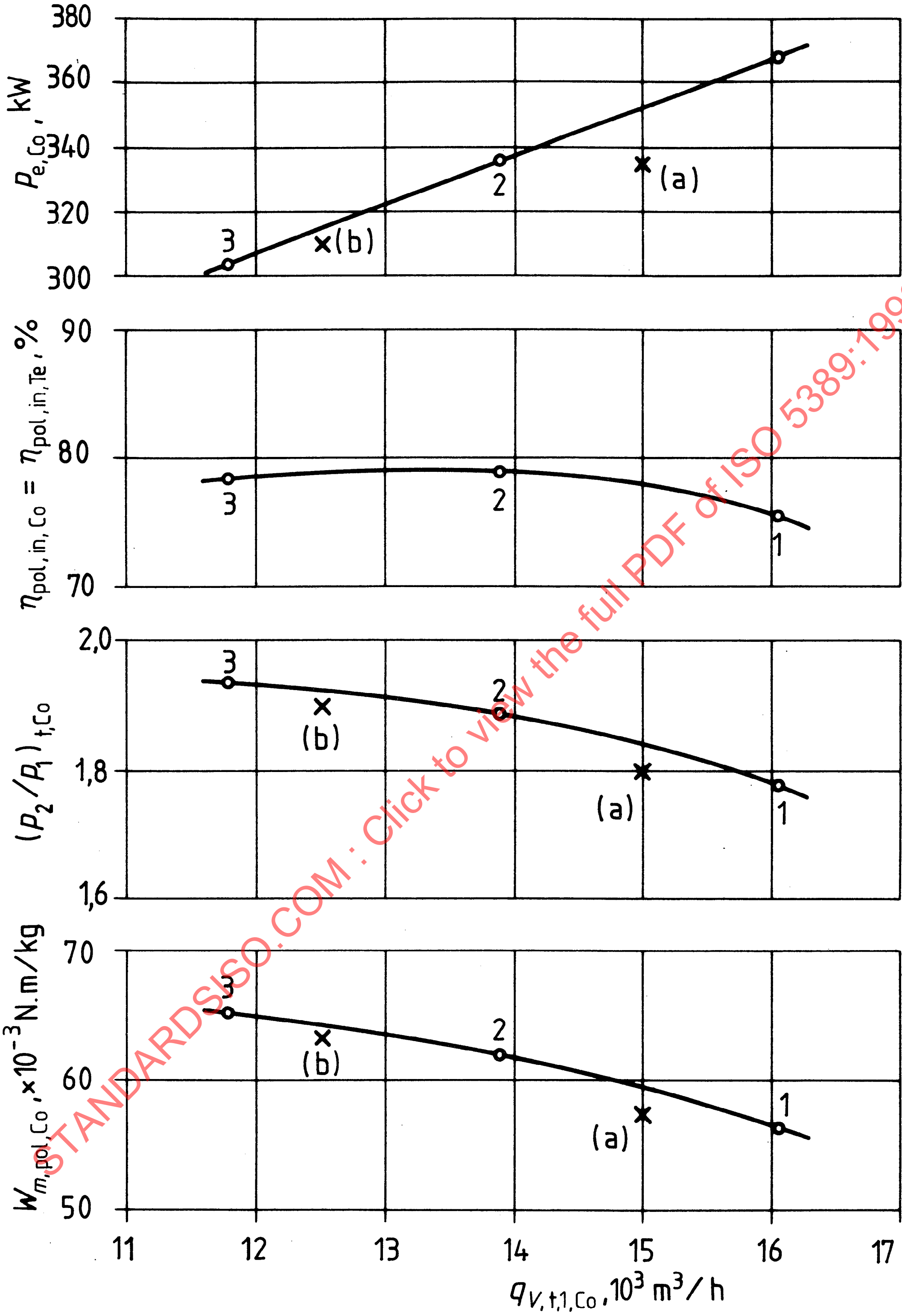


Figure F.8 — Graphs for comparison with guarantee

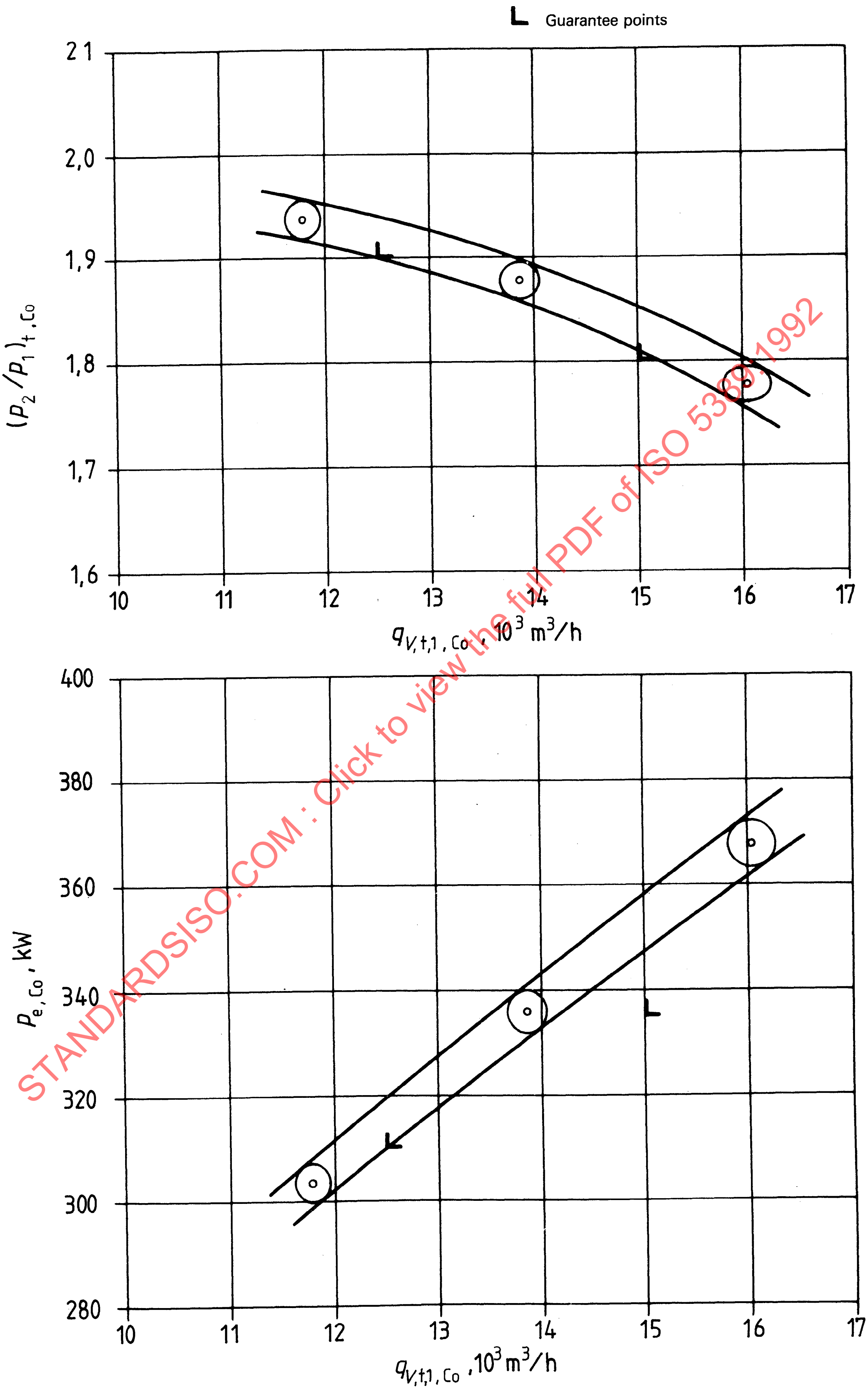


Figure F.9 — Graphs for comparison with guarantee

Table F.29 — Resulting test uncertainty

Designation	Symbol	Numerical value	Comments
Uncertainty in volumetric flow at inlet	$\tau_{q_m,\mathrm{Te}}$	1,0 %	In accordance with ISO 5167-1
	$\tau_{N,\mathrm{Te}}$	0,08 %	Digital measuring instrument — 1 rev on indicator (see 5.9)
	$\tau_{p_1,\mathrm{Te}}$	0,13 %	1 mmHg on absolute pressure (see 5.9)
	$\tau_{T_1,\mathrm{Te}}$	0,36 %	1 ° on absolute temperature (see 5.9)
	τ_{adj}	0,9 %	
	$\tau_{\mathrm{tot},q_V,1}$	$\sqrt{1{,}075^2 + 0{,}9^2} = 1{,}40$ %	See 9.2.5
Uncertainty in pressure ratio	N_r	1,064	
	$\ln(p_2/p_1)$	0,623	
	$\tau_{p_2,\mathrm{Te}}$	0,16 %	1 mmHg on absolute pressure (see 5.9)
	τ_{adj}	0,9 %	
	$\tau_{\mathrm{tot},p_2/p_1}$	$\sqrt{0{,}283^2 + 0{,}9^2} = 0{,}94$ %	See 9.2.6
Uncertainty in power at coupling	ζ_3	2,191	
	ζ_4	351,5	
	τ_{P_e}	0,75 %	Current transformer 0,5; voltage transformer 0,5; wattmeter 0,2 (estimated)
	τ_{P_f}	10 %	
	τ_{adj}	0,9 %	
	$\tau_{\mathrm{tot},P_\mathrm{e}}$	$\sqrt{0{,}79^2 + 0{,}9^2} = 1{,}20$ %	See 9.2.7 and table 9

F.4 Example 4 — Cooled turbocompressor, isentropic exponent $\kappa_{Te} = \kappa_{Gu}$, variable speed and temperature ratios

F.4.1 General

The speed and temperature ratios can be varied so that similar flow conditions can be used.

The guarantee conditions, guaranteed performance and other design values are given in tables F.30 to F.32.

F.4.2 Purpose of tests

The purpose of the tests is to prove the guaranteed power requirement for the guarantee point.

F.4.3 Design of installation

The installation consists of a five-stage turbocompressor of centrifugal design for air with intermediate cooling after each stage, driven by steam turbine.

Table F.30 — Guarantee conditions

Designation	Symbol	Numerical value	Unit
Pressure	$p_{t,1,Gu}$	0,983 5	bar
Temperature	$t_{t,1,Gu}$	25,31	°C
Relative humidity	φ_{Gu}	70	%
Gas constant	R_{Gu}	289,5	N·m/(kg·K)
Isentropic exponent	κ_{Gu}	1,4	—
Volumetric flow of cooling water	$q_{V,W,Gu}$	800	m³/h
Inlet temperature of cooling water	$t_{W,1,Gu}$	27	°C

Table F.31 — Guaranteed performance

Designation	Symbol	Numerical value	Unit
Volumetric flow at inlet	$q_{V,t,1,Gu}$	114 826	m³/h
Discharge pressure	$p_{t,2,Gu}$	6,532 1	bar
Specific power requirement at compressor coupling	$\left(\frac{P_e}{q_{V,t,1}}\right)_{Gu}$	0,071 06	kWh/m³

Table F.32 — Other design values

Designation	Symbol	Numerical value	Unit
Speed (Not a guarantee condition for variable-speed driver 1))	N_{Gu}	4 650	r/min
Outside diameter of Stage 1 impeller	D_I	1,12	m
Inlet temperature of air ahead of			
Stage II	$t_{t,1,II,Gu}$	40	°C
Stage III	$t_{t,1,III,Gu}$	42	°C
Stage IV	$t_{t,1,IV,Gu}$	45	°C
Stage V	$t_{t,1,V,Gu}$	48	°C
1) The subscript Gu designates the design speed.			

F.4.4 Test set-up

The compressor is subjected to trials on the test bed, driven by a variable-speed electric motor. Since the rating of the test bed motor is insufficient for operation at atmospheric intake pressure, the compressor shall be run at diminished intake pressure. Running in this way, it is not possible to judge the intercoolers as a reduced flow of cooling water must be admitted because of the reduction in the pressure level.

The test set-up is illustrated in figure F.10, and the test conditions are given in table F.33.

F.4.5 Setting conditions (see 8.2.4)

By means of the flow of cooling water, the inlet temperatures of the air into the individual stages (see table F.34) can be set so that the temperature relationships

$$T_{t,1,j}/T_{t,1,I}$$

where j = II, III, IV, V, agree with the design value.

To guarantee similar flow conditions, i.e.

$$\frac{\left(\dfrac{N}{\sqrt{RZ_1T_{t,1}}}\right)_{Te}}{\left(\dfrac{N}{\sqrt{RZ_1T_{t,1}}}\right)_{Gu}} = 1$$

the test shall be conducted at a speed reduced in the proportion

$$\frac{N_{Te}}{N_{Gu}} = \sqrt{\frac{288{,}3 \times 288{,}23}{289{,}5 \times 298{,}46}} = 0{,}981$$

The result is N_{Te} = 4 560 r/min.

In determining the setting conditions in accordance with the flow chart in figure D.6, the following calculation sequence is used (the numbers given are those designating individual boxes in the flow chart):

1 — 2 — 3 — 4 — 5 — 39 — 40 — 41 — 42 — (43 — 44, but see the note to F.0.1) — 45

Since the speeds and the inlet temperatures for the stages are variable, the compressor can fulfil the setting conditions as a unit. Accordingly, the calculation ends with 46 — 47.

The test was carried out maintaining the above test conditions.

Table F.33 — Test conditions

Designation	Symbol	Numerical value	Unit
Inlet pressure	$p_{t,1,Te}$	0,240 9	bar
Inlet temperature	$t_{t,1,Te}$	15,08	°C
Humidity of air	φ_{Te}	70	%
Gas constant	R_{Te}	288,3	N·m/(kg·K)
Inlet temperature of cooling water	$t_{W,1,Te}$	24,8	°C

Table F.34 — Inlet air temperatures at different stages

Values in degrees Celsius

Designation	Design value	Value to be set during tests
$t_{t,1,I}$	—	15
$t_{t,1,II}$	40	30
$t_{t,1,III}$	42	32
$t_{t,1,IV}$	45	34
$t_{t,1,V}$	48	37

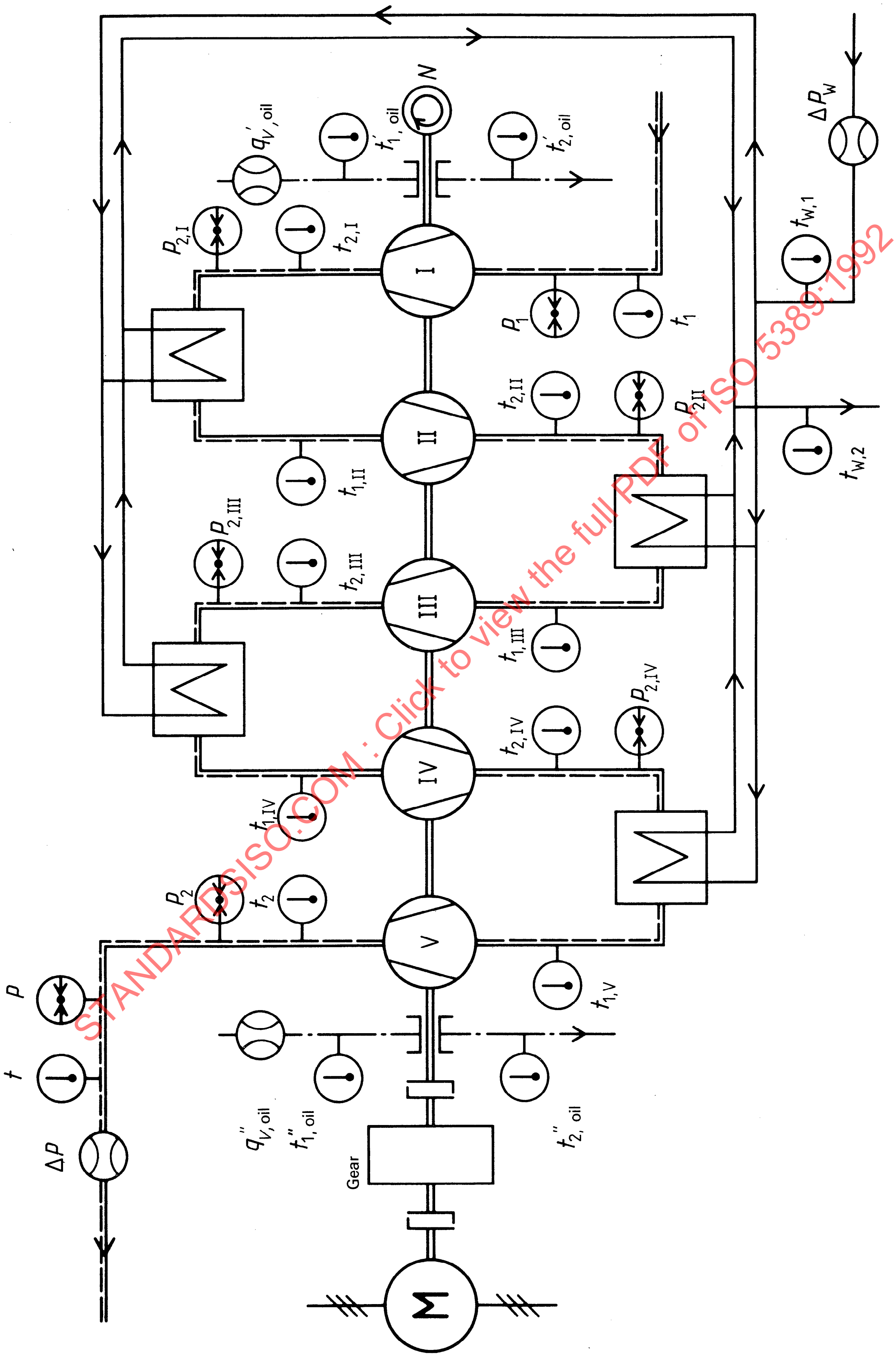


Figure F.10 — Test set-up for example 4

F.4.6 Test results and conversion

See tables F.35 to F.37.

Table F.35 — Test results

Designation	Symbol	Numerical value	Unit
Test number	—	1	—
Date of test	—	1965-04-26	—
Atmospheric pressure	p_a	1,010	bar
Gas constant	R_{Te}	288,3	N·m/(kg·K)
Speed	N_{Te}	4 560	r/min
Mass flow*)	$q_{m,Te}$	33 114	kg/h
Inlet pressure	$p_{t,1,Te}$	0,240 9	bar
Inlet temperature	$t_{t,1,Te}$	15,08	°C
	$T_{t,1,Te}$	288,23	K
Density	$\varrho_{t,1,Te}$	0,289 9	kg/m³
Volumetric flow at inlet (also usable volumetric flow at inlet)	$q_{V,t,1,Te}$	114 226	m³/h
Discharge pressure	$p_{t,2,Te}$	1,605 5	bar

*) Calculated from the mass flow measured in accordance with ISO 5167-1.

Table F.36 — Calculation results

Designation	Symbol	Numerical value	Unit
Pressure ratio	$(p_2/p_1)_{t,Te}$	6,664 6	—
Specific isothermal compression work	$W_{m,T,t,Te}$	157 618	N·m/kg
Power absorbed by the gas (compression sections I-V)			
mass flow	$q_{m,Te}$	33 114	kg/h
discharge temperature	$t_{t,2,Te}$	76,62	°C
inlet temperature	$t_{t,1,Te}$	15,08	°C
temperature difference	$\Delta t_{t,Te}$	61,54	°C
power calculated with c_p = 1,004 8 kJ/(kg·K)	$P_{in,t,1,Te}$	568,8	kW
Power absorbed by cooling water (= power absorbed by the gas in sections II, III, IV)			
flow of cooling water	$q_{V,W,Te}$	186,7	m³/h
outlet temperature of cooling water	$t_{W,2,Te}$	31,04	°C
inlet temperature of cooling water	$t_{W,1,Te}$	24,80	°C
temperature difference	$\Delta t_{W,Te}$	6,24	°C
power calculated with c_{pW} = 4,187 kJ/(kg·K)	$P_{in,Te}$	1 354,7	kW
Power absorbed by the bearing lubricating oil			
— journal bearing: volumetric flow of oil	$q_{V,oil,Te}$	2,75	m³/h
inlet temperature of oil	$t_{1,oil,Te}$	39,61	°C
outlet temperature of oil	$t_{2,oil,Te}$	46,86	°C
temperature difference	$\Delta t_{oil,Te}$	7,25	°C
mean specific heat	$c_{pM,oil,Te}$	1,969 3	kJ/(kg·K)
density	$\varrho_{1,oil,Te}$	$0,866 \times 10^3$	kg/m³
power	$P_{f,Te}$	9,4	kW
— thrust bearing: volumetric flow of oil	$q_{V,oil,Te}$	5,62	m³/h
inlet temperature of oil	$t_{1,oil,Te}$	39,19	°C
outlet temperature of oil	$t_{2,oil,Te}$	50,07	°C
temperature difference	$\Delta t_{oil,Te}$	10,88	°C
mean specific heat	$c_{pM,oil,Te}$	1,978	kJ(/kg·K)
density	$\varrho_{1,oil,Te}$	$0,866 \times 10^3$	kg/m³
power	$P_{f,Te}$	29,0	kW
total mechanical power dissipation	$P_{f,Te}$	38,4	kW
Power requirement			
isothermal compression power	$P_{T,Te}$	1 449,8	kW
internal power	$P_{in,t,Te}$	1 923,5	kW
mechanical compressor power dissipation	$P_{f,Te}$	38,4	kW
power at coupling	$P_{e,t,Te}$	1 961,9	kW
isothermal coupling efficiency	$\eta_{T,t,e,Te}$	73,9	%

Table F.37 — Conversion to guarantee conditions (see 8.2.4.1)

Designation	Symbol	Numerical value	Unit
Test speed	N_{Te}	4 560	r/min
Design speed	N_{Gu}	4 650	r/min
Converted volumetric flow at inlet	$q_{V,t,1,Co}$	116,480	m^3/h
Converted isothermal delivery head	$W_{m,T,t,Co}$	163,901	N·m/kg
Converted pressure ratio	$(p_2/p_1)_{t,Co}$	6,665 3	—
Converted internal power	$P_{in,t,Co}$	8 008,8	kW
Converted mechanical power dissipation	$P_{f,Co}$	39,6	kW
Converted power at coupling	$P_{e,t,Co}$	8 048,4	kW

F.4.7 Test uncertainty and comparison with guarantee

The test uncertainties are calculated in accordance with 9.2.

The converted inlet volume is slightly higher than guaranteed (see table F.38), even when taking into account the test uncertainties.

The converted specific power consumption is lower than guaranteed by about 2,96 % (see table F.38).

The guarantee can be considered to be fulfilled.

NOTE — When calculating the power absorbed by the gas in compressor sections I and V, the leakage losses were neglected. As the power consumption is well below the guarantee figure, this simplification is justified.

The test uncertainty for example 4 is shown in table F.39.

Table F.38 — Comparison with the guarantee (see 9.4)

Designation	Symbol	Numerical value	Unit
Guaranteed pressure ratio	$(p_2/p_1)_{t,Gu}$	6,640 8	—
Isothermal delivery head at $(p_2/p_1)_{t,Gu}$	$W_{m,T,Gu}$	163,583	N·m/kg
Conversion factor	$\dfrac{W_{m,T,Gu}}{W_{m,T,Co}}$	0,998 1	—
Internal power absorption related to $(p_2/p_1)_{t,Gu}$	$P_{in,Co}$	7 993,3	kW
Power at coupling	$P_{e,Co}$	8 032,9	kW
Specific power at coupling related to $(p_2/p_1)_{t,Gu}$	$\dfrac{P_{e,Co}}{q_{V,t,1,Co}}$	0,068 96	kWh/m^3
Guaranteed specific power at coupling	$\dfrac{P_{e,Gu}}{q_{V,t,1,Gu}}$	0,071 06	kWh/m^3
Deviation from guarantee value of specific power at coupling		– 2,96	%

Table F.39 — Resulting test uncertainty

Designation	Symbol	Numerical value	Comments
Uncertainty in volumetric flow at inlet	$\tau_{q_m,\mathrm{Te}}$	1,0 %	In accordance with ISO 5167-1
	$\tau_{N,\mathrm{Te}}$	0,22 %	Quality class 0,2; final scale value 5 000 (See 5.9)
	$\tau_{p_1,\mathrm{Te}}$	0,56 %	1 mmHg on absolute pressure (See 5.9)
	$\tau_{T_1,\mathrm{Te}}$	0,35 %	1 ° on absolute temperature (See 5.9)
	$\tau_{\mathrm{res},q_V,1}$	1,22 %	See 9.2.5
Uncertainty in pressure ratio	N_r	1,0	
	$\ln(p_2/p_1)$	1,890	
	$\tau_{p_2,\mathrm{Te}}$	0,64 %	Quality class 0,6; final value 10 atm gauge
	$\tau_{\mathrm{res},p_2/p_1}$	1,555 %	See 9.2.6
Uncertainty in power at coupling	τ_{P_in}	1,5 %	Energy balance estimated
	τ_{P_f}	10 %	Estimated
	$\tau_{T_{1,j}}$	0,33 %	1 ° on absolute temperature (see 5.9)
	$\tau_{\mathrm{res},P_\mathrm{e}}$	1,55 %	See 9.2.7 and table 9

F.4.8 Comparison of different calculation methods

A comparison of different calculation methods is given in table F.40.

Table F.40 — Comparison of different calculation methods

Example 4	Total (p_t, t_t)	Total $\left(W_m = W_{m,\text{static}} + \frac{c_2^2 - c_1^2}{2}\right)$	Suction side: total Discharge side: static	Static
Guarantee conditions				
Inlet pressure	$p_{t,1,Gu} = 0{,}983\,5$ bar	$p_{t,1,Gu} = 0{,}983\,5$ bar	$p_{t,1,Gu} = 0{,}983\,5$ bar	$p_{1,Gu} = 0{,}981$ bar
Inlet temperature	$t_{t,1,Gu} = 25{,}31$ °C	$t_{t,1,Gu} = 25{,}31$ °C	$t_{t,1,Gu} = 25{,}31$ °C	$t_{1,Gu} = 25$ °C
Relative humidity	$\varphi_{Gu} = 70$ %			
Gas constant	$R_{Gu} = 289{,}5$ Nm³/(kg·K)			
Isentropic exponent	$\kappa_{Gu} = 1{,}4$			
Volumetric flow of cooling water	$q_{V,W,Gu} = 800$ m³/h			
Inlet temperature of cooling water	$t_{W,1,Gu} = 27$ °C			
Object of the guarantee				
Volumetric flow at inlet	$q_{V,t,1,Gu} = 114\,826$ m³/h	$q_{V,t,1,Gu} = 114\,826$ m³/h	$q_{V,t,1,Gu} = 114\,826$ m³/h	$q_{V,1,Gu} = 115\,000$ m³/h
Discharge pressure	$p_{t,2,Gu} = 6{,}531\,2$ bar	$p_{2,Gu} = 6{,}472$ bar	$p_{2,Gu} = 6{,}472$ bar	$p_{2,Gu} = 6{,}472$ bar
Specific power requirement at compressor coupling	$\left(\frac{P_e}{q_{V,t,1}}\right)_{Gu} = 0{,}071\,06$ kWh/m³	$\left(\frac{P_e}{q_{V,t,1}}\right)_{Gu} = 0{,}071\,06$ KWh/m³	$\left(\frac{P_e}{q_{V,t,1}}\right)_{Gu} = 0{,}071\,06$ KWh/m³	$\left(\frac{P_e}{q_{V,1}}\right)_{Gu} = 0{,}070\,96$ kWh/m³
Speed (design)	$N_{Gu} = 4\,650$ r/min			
Outside diameter of 1st impeller	$D_I = 1{,}12$ m			
Density at inlet	$\frac{0{,}983\,5 \times 10^5}{289{,}5 \times 298{,}46} = 1{,}138\,3$ kg/m³	$\frac{0{,}983\,5 \times 10^5}{289{,}5 \times 298{,}46} = 1{,}138\,3$ kg/m³	1,138 3 kg/m³	$\varrho_{1,Gu} = \frac{p_{1,Gu}}{Z \times R \times T} = \frac{0{,}981 \times 10^5}{289{,}5 \times 298{,}15} = 1{,}138\,3$ kg/m³
Inlet temperature of air ahead of				
Stage II	$t_{t,1,II,Gu} = 40$ °C			
Stage III	$t_{t,1,III,Gu} = 42$ °C			
Stage IV	$t_{t,1,IV,Gu} = 45$ °C			
Stage V	$t_{t,1,V,Gu} = 48$ °C			

Table F.40 — Comparison of different calculation methods *(continued)*

Example 4	Total (p_t, t_t)	Total $\left(W_m = W_{m,\text{static}} + \frac{c_2^2 - c_1^2}{2}\right)$	Suction side: total Discharge side: static	Static
Test setting	$N_{Te} = N_{Gu}\sqrt{\frac{R_{Te}T_{t,1,Te}}{R_{Gu}T_{t,1,Gu}}}$ $= 4\,650\sqrt{\frac{288{,}3 \times 288{,}23}{289{,}5 \times 298{,}46}}$ $= 4\,560$ r/min	$N_{Te} = 4\,560$ r/min	$N_{Te} = 4\,560$ r/min	$N_{Te} = N_{Gu}\sqrt{\frac{R_{Te}T_{1,Te}}{R_{Gu}T_{1,Gu}}}$ $= 4\,650\sqrt{\frac{288{,}3 \times 287{,}95}{289{,}5 \times 298{,}15}}$ $= 4\,560$ r/min
Test results				
Test number	1			
Date of test	1975-04-26			
Atmospheric pressure	$p_a = 1{,}010$ bar			
Gas constant	$R_{Te} = 288{,}3$ N·m/(kg·K)			
Speed	$N_{Te} = 4\,560$ r/min			
Mass flow	$q_{m,Te} = 33\,114$ kg/h			
Inlet pressure	$p_{t,1,Te} = 0{,}240\,9$ bar	$p_{1,Te} = 0{,}240\,3$ bar	$p_{t,1,Te} = 0{,}240\,9$ bar	$p_{1,Te} = 0{,}240\,3$ bar
Inlet temperature	$t_{t,1,Te} = 15{,}08$ °C	$t_{t,1,Te} = 14{,}8$ °C	$t_{t,1,Te} = 15{,}08$ °C	$t_{1,Te} = 14{,}08$ °C
Density at inlet	$\varrho_{t,1,Te} = 0{,}289\,9$ kg/m³	$\varrho_{t,1,Te} = \frac{p_{t,1,Te}}{ZR_{Te}T_{t,1,Te}}$ $= \frac{0{,}240\,9 \times 10^5}{288{,}3 \times 288{,}23}$ $= 0{,}289\,9$ kg/m³	$\varrho_{t,1,Te} = 0{,}289\,9$ kg/m³	$\varrho_{1,Te} = \frac{p_{1,Te}}{ZR_{Te}T_{1,Te}}$ $= \frac{0{,}240\,3 \times 10^5}{288{,}3 \times 287{,}95}$ $= 0{,}289\,5$ kg/m³
Volumetric flow at inlet	$q_{V,t,1,Te} = 114\,226$ m³/h	$q_{V,t,1,Te} = \frac{q_{m,Te}}{\varrho_{t,1,Te}}$ $= \frac{33{,}114}{0{,}289\,9}$ $= 114\,226$ m³/h	$q_{V,t,1,Te} = 114\,226$ m³/h	$q_{V,1,Te} = \frac{q_{m,Te}}{\varrho_{1,Te}}$ $= \frac{33{,}114}{0{,}289\,5}$ $= 114\,383$ m³/h
Discharge pressure	$p_{t,2,Te} = 1{,}605\,5$ bar	$p_{2,Te} = 1{,}591$ bar	$p_{2,Te} = 1{,}591$ bar	$p_{2,Te} = 1{,}591$ bar

Table F.40 — Comparison of different calculation methods *(continued)*

Example 4	Total (p_t, t_t)	Total $\left(W_m = W_{m,\text{static}} + \frac{c_2^2 - c_1^2}{2}\right)$	Suction side : total Discharge side : static	Static
Test calculation				
Pressure ratio	$\frac{p_{t,2,Te}}{p_{t,1,Te}} = \frac{1{,}605\ 5}{0{,}240\ 9}$ $= 6{,}664\ 6$	$\frac{p_{2,Te}}{p_{1,Te}} = \frac{1{,}591}{0{,}240\ 3}$ $= 6{,}620\ 9$	$\frac{p_{2,Te}}{p_{t,1,Te}} = \frac{1{,}591}{0{,}240\ 9}$ $= 6{,}604\ 4$	$\frac{p_{2,Te}}{p_{1,Te}} = \frac{1{,}591}{0{,}240\ 3}$ $= 6{,}620\ 9$
Specific isothermal compression work	$W_{m,T,Te} = R_{Te} T_{t,1,Te} \ln\left(\frac{p_{t,2,Te}}{p_{t,1,Te}}\right)$ $= 288{,}3 \times 288{,}23 \times \ln 6{,}664\ 6$ $= 157\ 618$ N·m/kg	$W_{m,T,Te} =$ $= R_{Te} T_{1,Te} \ln\left(\frac{p_{2,Te}}{p_{1,Te}}\right) + \frac{c_2^2 - c_1^2}{2}$ $= 288{,}3 \times 287{,}95 \times 95 \times \ln 6{,}620\ 9 +$ $+ \frac{44{,}77^2 - 21{,}19^2}{2}$ $= 157\ 697$ N·m/kg	$W_{m,T,Te} = R_{Te} T_{t,1,Te} \ln\left(\frac{p_{2,Te}}{p_{1,t,Te}}\right)$ $= 288{,}3 \times 288{,}23 \times \ln 6{,}604\ 4$ $= 156\ 865$ N·m/kg	$W_{m,T,Te} = R_{Te} T_{1,Te} \ln\left(\frac{p_{2,Te}}{p_{1,Te}}\right)$ $= 288{,}3 \times 287{,}95 \times 95 \times \ln 6{,}620\ 9$ $= 156\ 919$ N·m/kg
a) Power absorbed by gas				
Discharge temperature total	$t_{t,2,Te} = 76{,}62$	$t_{t,2,Te} = 76{,}62$	$t_{t,2,Te} = 75{,}2$	$t_{2,Te} = 75{,}2$
Temperature difference	$\Delta t = 61{,}54$ K	$\Delta t = 61{,}54$ K	$\Delta t = 61{,}54$ K	$\Delta t = 61{,}54$ K
Power calculated with $c_p = 1{,}004\ 8$ kJ/(kg·K)	$P_{in,Te} = 568{,}8$ kW	$P_{in,Te} = 568{,}8$ kW	$P_{in,Te} = 568{,}8$ kW	$P_{in,Te} = 568{,}8$ kW
b) Power absorbed by cooling water	$P_{in,Te} = 1\ 354{,}7$ kW	$P_{in,Te} = 1\ 354{,}7$ kW	$P_{in,Te} = 1\ 354{,}7$ kW	$P_{in,Te} = 1\ 354{,}7$ kW
c) Power absorbed by the bearing oil	$P_{f,Te} = 38{,}4$ kW	$P_{f,Te} = 38{,}4$ kW	$P_{f,Te} = 38{,}4$ kW	$P_{f,Te} = 38{,}4$ kW
Internal power	$P_{in,Te} = 1\ 923{,}5$ kW	$P_{in,Te} = 1\ 923{,}5$ kW	$P_{in,Te} = 1\ 923{,}5$ kW	$P_{in,Te} = 1\ 923{,}5$ kW
Power at coupling	$P_{e,Te} = 1\ 961{,}9$ kW	$P_{e,Te} = 1\ 961{,}9$ kW	$P_{e,Te} = 1\ 961{,}9$ kW	$P_{e,Te} = 1\ 961{,}9$ kW
Isothermal compression power	$P_{T,Te} = 1\ 449{,}8$ kW	$P_{T,Te} = 1\ 450{,}5$ kW	$P_{T,Te} = 1\ 442{,}9$ kW	$P_{T,Te} = 1\ 443{,}4$ kW
Isothermal coupling efficiency	$\eta_{T,e,Te} = 73{,}90$ %	$\eta_{T,e,Te} = 73{,}93$ %	$\eta_{T,e,Te} = 73{,}55$ %	$\eta_{T,e,Te} = 73{,}57$ %

Table F.40 — Comparison of different calculation methods *(continued)*

Example 4	Total (p_t, t_t)	Total $\left(W_m = W_{m,\text{static}} + \frac{c_2^2 - c_1^2}{2}\right)$	Suction side : total Discharge side : static	Static
Conversion to guarantee conditions				
Test speed	$N_{Te} = 4\ 560$ r/min	$N_{Te} = 4\ 560$ r/min	$N_{Te} = 4\ 560$ r/min	$N_{Te} = 4\ 560$ r/min
Design speed	$N_{Gu} = 4\ 650$ r/min	$N_{Gu} = 4\ 650$ r/min	$N_{Gu} = 4\ 650$ r/min	$N_{Gu} = 4\ 650$ r/min
Converted volumetric flow at inlet	$q_{V,t,1,Co} = q_{V,t,1,Te} \frac{N_{Gu}}{N_{Te}}$ $= 114\ 226 \times \frac{4\ 650}{4\ 560}$ $= 116\ 480$ m³/h	$q_{V,t,1,Co} = q_{V,t,1,Te} \frac{N_{Gu}}{N_{Te}}$ $= 114\ 226 \times \frac{4\ 650}{4\ 560}$ $= 116\ 480$ m³/h	$q_{V,t,1,Co} = q_{V,t,1,Te} \frac{N_{Gu}}{N_{Te}}$ $= 116\ 480$ m³/h	$q_{V,1,Co} = q_{V,1,Te} \frac{N_{Gu}}{N_{Te}}$ $= 114\ 383 \times \frac{4\ 650}{4\ 560}$ $= 116\ 640$ m³/h
Converted isothermal specific compression work	$W_{m,T,t,Co} = W_{m,Te} \left(\frac{N_{Gu}}{N_{Te}}\right)^2$ $= 157\ 618 \times \frac{4\ 650}{4\ 560}$ $= 163\ 901$ N·m/kg	$W_{m,T,t,Co} = W_{m,Te} \left(\frac{N_{Gu}}{N_{Te}}\right)^2$ $= 157\ 618 \times \frac{4\ 650}{4\ 560}$ $= 163\ 983$ N·m/kg	$W_{m,T,t,Co} = W_{m,Te} \left(\frac{N_{Gu}}{N_{Te}}\right)^2$ $= 156\ 865 \times \frac{4\ 650}{4\ 560}$ $= 163\ 118$ N·m/kg	$W_{m,T,Co} = 156\ 919 \times \frac{4\ 650}{4\ 560}$ $= 163\ 174$ N·m/kg
Converted pressure ratio	$\left(\frac{p_2}{p_1}\right)_{Co} = 6{,}665\ 3$	$\left(\frac{p_2}{p_1}\right)_{Co} = 6{,}684\ 8$	$\left(\frac{p_2}{p_1}\right)_{Co} = 6{,}605\ 1$	$\left(\frac{p_2}{p_1}\right)_{Co} = 6{,}622\ 4$
Converted internal power	$P_{in,Co} = P_{in,Te} \frac{\varrho_{1,Gu}}{\varrho_{1,Te}} \left(\frac{N_{Gu}}{T_{Te}}\right)^3$ $= 1\ 923{,}5 \times \frac{1{,}138\ 3}{0{,}289\ 9} \times \left(\frac{4\ 650}{4\ 560}\right)^3$ $= 8\ 008{,}8$ kW	$P_{in,Co} = P_{in,Te} \frac{\varrho_{1,Gu}}{\varrho_{1,Te}} \left(\frac{N_{Gu}}{T_{Te}}\right)^3$ $= 1\ 923{,}5 \times \frac{1{,}138\ 3}{0{,}289\ 9} \times \left(\frac{4\ 650}{4\ 560}\right)^3$ $= 8\ 008{,}8$ kW	$P_{in,Co} = P_{in,Te} \frac{\varrho_{1,Gu}}{\varrho_{1,Te}} \left(\frac{N_{Gu}}{T_{Te}}\right)^3$ $= 1\ 923{,}5 \times \frac{1{,}138\ 3}{0{,}289\ 9} \times \left(\frac{4\ 650}{4\ 560}\right)^3$ $= 8\ 008{,}7$ kW	$P_{in,Co} = P_{in,Te} \frac{\varrho_{1,Gu}}{\varrho_{1,Te}} \left(\frac{N_{Gu}}{T_{Te}}\right)^3$ $= 1\ 923{,}5 \times \frac{1{,}136\ 5}{0{,}289\ 5} \times \left(\frac{4\ 650}{4\ 560}\right)^3$ $= 8\ 007$ kW
Converted mechanical power	$P_{f,Co} = 39{,}6$ kW	$P_{f,Co} = 39{,}6$ kW	$P_{f,Co} = 39{,}6$ kW	$P_{f,Co} = 39{,}6$ kW
Converted power at coupling	$P_{e,Co} = 8\ 048{,}4$ kW	$P_{e,Co} = 8\ 048{,}4$ kW	$P_{e,Co} = 8\ 048{,}3$ kW	$P_{e,Co} = 8\ 046{,}6$ kW
Comparison with the guarantee				
Guarantee pressure ratio	$\left(\frac{p_2}{p_1}\right)_{Gu} = 6{,}640\ 8$	$\left(\frac{p_2}{p_1}\right)_{Gu} = 6{,}640\ 8$	$\left(\frac{p_2}{p_1}\right)_{Gu} = 6{,}580\ 6$	$\left(\frac{p_2}{p_1}\right)_{Gu} = 6{,}597\ 3$

Table F.40 — Comparison of different calculation methods *(concluded)*

Example 4	Total (p_t, t_t)	Total $\left(W_m = W_{m,\text{static}} + \frac{c_2^2 - c_1^2}{2}\right)$	Suction side: total Discharge side: static	Static
Comparison with the guarantee *(continued)* Isothermal specific compression work at $(p_2/p_1)_{Gu}$	$W_{m,T,t,Gu} = R_{Gu} T_{t,1,Gu} \ln\left(\frac{p_{t,2}}{p_{t,1}}\right)_{Gu}$	$W_{m,T,t,Gu} = R_{Gu} T_{1,Gu} \ln\left(\frac{p_2}{p_1}\right)_{Gu} + \frac{c_2^2 - c_1^2}{2}$	$W_{m,T,t,Gu} = R_{Gu} T_{t,1,Gu} \ln\left(\frac{p_2}{p_{t,1}}\right)_{Gu}$	$W_{m,T,Gu} = R_{Gu} T_{1,Gu} \ln\left(\frac{p_2}{p_1}\right)_{Gu}$
	$= 289{,}5 \times 298{,}46 \times \ln 6{,}640\,8$	$= 289{,}5 \times 298{,}15 \times \ln 6{,}597\,3 + \frac{45{,}65^2 - 21{,}61^2}{2}$	$= 289{,}5 \times 298{,}46 \times \ln 6{,}580\,6$	$= 289{,}5 \times 298{,}15 \times \ln 6{,}597\,3$
	$= 163\,583$ N·m/kg	$= 163\,654$ N·m/kg	$= 162\,796$ N·m/kg	$= 162\,846$ N·m/kg
Internal power related to $(p_2/p_1)_{Gu}$	$P_{in,Co} = 7\,993{,}3$ kW	$P_{in,Co} = 7992{,}8$ kW	$P_{in,Co} = 7\,992{,}9$ kW	$P_{in,Co} = 7990$ kW
Power at coupling	$P_{e,Co} = 8\,032{,}9$ kW	$P_{e,Co} = 8\,032{,}4$ kW	$P_{e,Co} = 8\,032{,}5$ kW	$P_{e,Co} = 8\,029{,}6$ kW
Specific power at coupling related to $(p_2/p_1)_{Gu}$	$\frac{P_{e,Co}}{q_{V,t,1,Co}} = \frac{8\,032{,}9}{116\,480}$	$\frac{P_{e,Co}}{q_{V,t,1,Co}} = \frac{8\,032{,}9}{116\,480}$	$\frac{P_{e,Co}}{q_{V,t,1,Co}} = \frac{8\,032{,}5}{116\,480}$	$\frac{P_{e,Co}}{q_{V,1,Co}} = \frac{8\,032{,}9}{116\,640}$
	$= 0{,}068\,96$ kWh/m³	$= 0{,}068\,96$ kWh/m³	$= 0{,}068\,96$ kWh/m³	$= 0{,}068\,84$ kWh/m³
Deviation from guarantee	− 2,96 %	− 2,96 %	− 2,96 %	− 2,99 %

Example 4

$$A_1 = 1{,}5 \times 1 = 1{,}5 \text{ m}^2$$

$$A_2 = \frac{0{,}406\,4 \times \pi}{4} = 0{,}129\,7 \text{ m}^2$$

$$q_{V,1,\mathrm{Te}} = 31{,}78 \text{ m}^3/\text{s}$$

$$C_{1,\mathrm{Te}} = \frac{q_{V,1,\mathrm{Te}}}{A_1} = \frac{31{,}78}{1{,}5} = 21{,}19 \text{ m/s}$$

$$C_{1,\mathrm{Gu}} = 21{,}19 \frac{N_{\mathrm{Gu}}}{N_{\mathrm{Te}}} = 21{,}19 \times \frac{4\,650}{4\,560} = 21{,}61 \text{ m/s}$$

$$q_{V,2,\mathrm{Te}} = 5{,}806\,2 \text{ m}^3/\text{s}$$

$$C_{2,\mathrm{Te}} = \frac{q_{V,2,\mathrm{Te}}}{A_2} = \frac{5{,}806\,2}{0{,}129\,7} = 44{,}77 \text{ m/s}$$

$$C_{2,\mathrm{Gu}} = 44{,}77 \frac{N_{\mathrm{Gu}}}{N_{\mathrm{Te}}} = 44{,}77 \times \frac{4\,650}{4\,560} = 45{,}65 \text{ m/s}$$

$$Ma_{\mathrm{t},1,\mathrm{Gu}} = \frac{q_{m,\mathrm{Gu}}}{A_1 p_{1,\mathrm{Gu}}} \sqrt{\frac{ZRT_{\mathrm{t},1,\mathrm{Gu}}}{\kappa}} = \frac{35{,}76}{10^5 \times 1{,}5 \times 0{,}981} \times \sqrt{\frac{1{,}0 \times 289{,}5 \times 298{,}15}{1{,}4}} = 0{,}060\,3$$

$$Ma_{\mathrm{t},1,\mathrm{Te}} = \frac{9{,}198\,3}{10^5 \times 1{,}5 \times 0{,}240\,3} \times \sqrt{\frac{1{,}0 \times 288{,}3 \times 288{,}15}{1{,}4}} = 0{,}057\,9$$

$$Ma_{\mathrm{t},2,\mathrm{Gu}} = \frac{q_{m,\mathrm{Gu}}}{A_1 p_{2,\mathrm{Gu}}} \sqrt{\frac{ZRT_{\mathrm{t},2,\mathrm{Gu}}}{\kappa}} = \frac{35{,}76}{6{,}472 \times 10^5 \times 0{,}129\,7} \times \sqrt{\frac{289{,}5 \times 348{,}35}{1{,}4}} = 0{,}114\,34$$

$$Ma_{\mathrm{t},2,\mathrm{Te}} = \frac{q_{m,\mathrm{Te}}}{A_2 p_{1,\mathrm{Te}}} \sqrt{\frac{ZRT_{\mathrm{t},2,\mathrm{Te}}}{\kappa}} = \frac{9{,}198\,3}{10^5 \times 0{,}129\,7 \times 1{,}591} \times \sqrt{\frac{288{,}3 \times 348{,}35}{1{,}4}} = 0{,}119\,4$$

$$p_{\mathrm{t},1,\mathrm{Gu}} = p_{1,\mathrm{Gu}} \left(1 + \frac{\kappa}{2} Ma^2_{\mathrm{t},1,\mathrm{Gu}}\right) = 0{,}981 \left(1 + \frac{1{,}4}{2} \times 0{,}060\,3^2\right) = 0{,}983\,5 \text{ bar}$$

$$p_{\mathrm{t},1,\mathrm{Te}} = p_{1,\mathrm{Te}} \left(1 + \frac{\kappa}{2} Ma^2_{\mathrm{t},1,\mathrm{Te}}\right) = 0{,}240\,3 \left(1 + \frac{1{,}4}{2} \times 0{,}057\,9^2\right) = 0{,}240\,9 \text{ bar}$$

$$p_{\mathrm{t},2,\mathrm{Gu}} = p_{2,\mathrm{Gu}} \left(1 + \frac{\kappa}{2} Ma^2_{\mathrm{t},2,\mathrm{Gu}}\right) = 6{,}472 \left(1 + \frac{1{,}4}{2} \times 0{,}114\,34^2\right) = 6{,}531\,2 \text{ bar}$$

$$p_{\mathrm{t},2,\mathrm{Te}} = p_{2,\mathrm{Te}} \left(1 + \frac{\kappa}{2} Ma^2_{\mathrm{t},2,\mathrm{Te}}\right) = 1{,}591 \left(1 + \frac{1{,}4}{2} \times 0{,}114\,1^2\right) = 1{,}605\,5 \text{ bar}$$

$$T_{\mathrm{t},1,\mathrm{Gu}} = T_{1,\mathrm{Gu}} \left(\frac{1}{1 - \frac{\kappa - 1}{\kappa} Ma^2_{\mathrm{t},1,\mathrm{Gu}}}\right) = 298{,}15 \times \left(\frac{1}{1 - \frac{1{,}4 - 1}{1{,}4} \times 0{,}060\,3^2}\right) = 298{,}46 \text{ K} = 25{,}31\ ^\circ\text{C}$$

$$T_{\mathrm{t},1,\mathrm{Te}} = T_{1,\mathrm{Te}} \left(\frac{1}{1 - \frac{\kappa - 1}{\kappa} Ma^2_{\mathrm{t},1,\mathrm{Te}}}\right) = 287{,}95 \times \left(\frac{1}{1 - \frac{1{,}4 - 1}{1{,}4} \times 0{,}057\,9^2}\right) = 288{,}23 \text{ K} = 15{,}08\ ^\circ\text{C}$$

$$T_{\mathrm{t},2,\mathrm{Te}} = T_{2,\mathrm{Te}} \left(\frac{1}{1 - \frac{\kappa - 1}{\kappa} Ma^2_{\mathrm{t},2,\mathrm{Te}}}\right) = 348{,}35 \times \left(\frac{1}{1 - \frac{1{,}4 - 1}{1{,}4} \times 0{,}119\,4^2}\right) = 349{,}77 \text{ K} = 76{,}62\ ^\circ\text{C}$$

F.5 Example 5 — Cooled turbocompressor, isentropic exponent $\kappa_{Te} = \kappa_{Gu}$, speed not variable, temperature ratio of cooled section variable

F.5.1 General

The speed cannot be varied and the temperatures in the cooled section are set by means of the flow of cooling water.

The conversion of uncooled and cooled sections of the compressor is carried out separately.

The guarantee conditions, guaranteed performance and other design values are given in tables F.41 to F.43.

F.5.2 Purpose of tests

The purpose of the tests is to prove the guaranteed power requirement for three guarantee points at constant discharge pressure.

F.5.3 Design of installation

The installation comprises a four-stage turbocompressor for air with three intercoolers, inlet guide vanes to Stage I, driven by electric motor and gear unit.

Table F.41 — Guarantee conditions

Designation	Symbol	Numerical value	Unit
Inlet pressure	$p_{t,1,Gu}$	0,980 7	bar
Inlet temperature	$t_{t,1,Gu}$	20	°C
Relative humidity	$\varphi_{1,Gu}$	70	%
Gas constant	R_{Gu}	288,90	N·m/(kg·K)
Isentropic exponent	κ_{Gu}	1,4	
Overall flow of cooling water	$q_{V,W,Gu}$	0,056 944	m³/s
Inlet temperature of cooling water	$t_{W,1,Gu}$	27	°C
Motor speed	$N_{Pr,Gu}$	1 490	r/min

Table F.42 — Guaranteed performance

Designation	Symbol	Numerical value — Guarantee point (a)	Numerical value — Guarantee point (b)	Numerical value — Guarantee point (c)	Unit
Volumetric flow at inlet	$q_{V,t,1,Gu}$	7,208 33	5,763 89	4,680 56	m³/s
Discharge pressure	$p_{t,2,Gu}$	6,864 9	6,864 9	6,864 9	bar
Power requirement at coupling	$P_{e,Gu}$	1 970	1 610	1 390	kW
Discharge pressure, Stage I	$p_{t,2,I,Gu}$	1,700	1,535	1,425	bar
Discharge temperature, Stage I	$t_{t,2,I,Gu}$	356	351,2	350,5	K

Table F.43 — Other design values

Designation	Symbol	Numerical value	Unit
Inlet temperature			
Stage II	$T_{1,II,Gu}$	310,2	K
Stage III	$T_{1,III,Gu}$	310,2	K
Stage IV	$T_{1,IV,Gu}$	310,2	K

F.5.4 Test set-up

The test is carried out at the compressor site under atmospheric conditions. In this test, it is not possible to judge the intercoolers since a flow of cooling water differing from that of the guarantee is admitted because of the deviation in cooling water inlet temperatures.

The test set-up is shown in figure F.11, and the test conditions are given in table F.44.

F.5.5 Setting conditions

Since the speed is not variable, the compressor is run so that the air temperatures in the cooled section IIc are set to correspond to the design value.

In determining the setting conditions in accordance with the flow chart in figure D.6 the following calculation sequence is used (the numbers given are those designating individual boxes in the flow chart):

1 — 2 — 3 — 4 — 5 — 39 — 40 — 41 — 42 — (43 — 44, but see the note to F.0.1) — 45

Since it is not possible to set the speed or the inlet temperature for Stage I, the compressor has to be divided into two units: 48 — 49.

Unit I: Stage I (without intercooling)

Unit IIc: Stages II to IV (with intercooling)

The additional calculation sequence for Unit I is as follows:

3 — 4 — 5 — 6 — 7 — 8 — 9 — 10 — 11 — 12 — 13 — 21 — 22 — 23 — 24 — 25 — 26 — (43 — 44, but see the note to F.0.1) — 45 — 46

The values $\Delta V_{r,tol} = +0{,}05$ or $\Delta V_{r,tol} = -0{,}05$ are inserted in 23.

The setting conditions for Unit I were maintained in the test and the calculation for Unit I ends with 46.

The additional calculation sequence for Unit IIc is as follows:

3 — 4 — 5 — 39 — 40 — 41 — 42 — (43 — 44, but see the note to F.0.1) — 45

Since the inlet temperature for the stages in Unit IIc can be varied through the intercoolers, it is possible to maintain the setting conditions for Unit IIc. The calculation ends with 46 — 47.

Table F.44 — Test conditions

Designation	Symbol	Numerical value	Unit
Isentropic exponent	κ_{Te}	1,4	
Inlet temperature	$t_{t,1,Te}$	13	°C
Mean inlet pressure	$p_{t,1,Te}$	0,980 7	bar
Cooling water temperature	$t_{W,1,Te}$	19	°C
Gas constant	R_{Te}	287,83	N·m/(kg·K)

F.5.6 Test results and conversion

See tables F.45 to F.47.

Table F.45 — Test results

Designation	Symbol	Numerical value Test No. 1	Test No. 2	Test No. 3	Unit
Date of test	—	1964-03-11	1964-03-11	1964-03-11	—
Atmospheric pressure	p_a	1,003	1,003	1,003	bar
Gas constant	R_{Te}	287,8	287,8	287,8	N·m/(kg·K)
Motor speed	$N_{Pr,Te}$	1 488	1 490	1 492	r/min
Position of inlet guide vanes	δ	+ 10	+ 54	+ 64	degree
Mass flow (also usable mass flow since measured on discharge side)	$q_{m,Te}$	8,608 33	6,755 56	5,669 44	kg/s
Inlet pressure	$p_{t,1,Te}$	0,968 9	0,982 6	0,988 5	bar
Inlet temperature	$t_{t,1,Te}$	12,1	12,9	13,0	°C
	$T_{t,1,Te}$	285,3	286,1	286,2	K
Density	$\varrho_{t,1,Te}$	1,180	1,193	1,200	kg/m³
Volumetric flow at inlet (also usable volumetric flow at inlet)	$q_{V,1,Te}$	7,295 28	5,662 78	4,724 44	m³/s
Discharge pressure	$p_{t,2,Te}$	7,531 5	7,511 9	7,119 6	bar

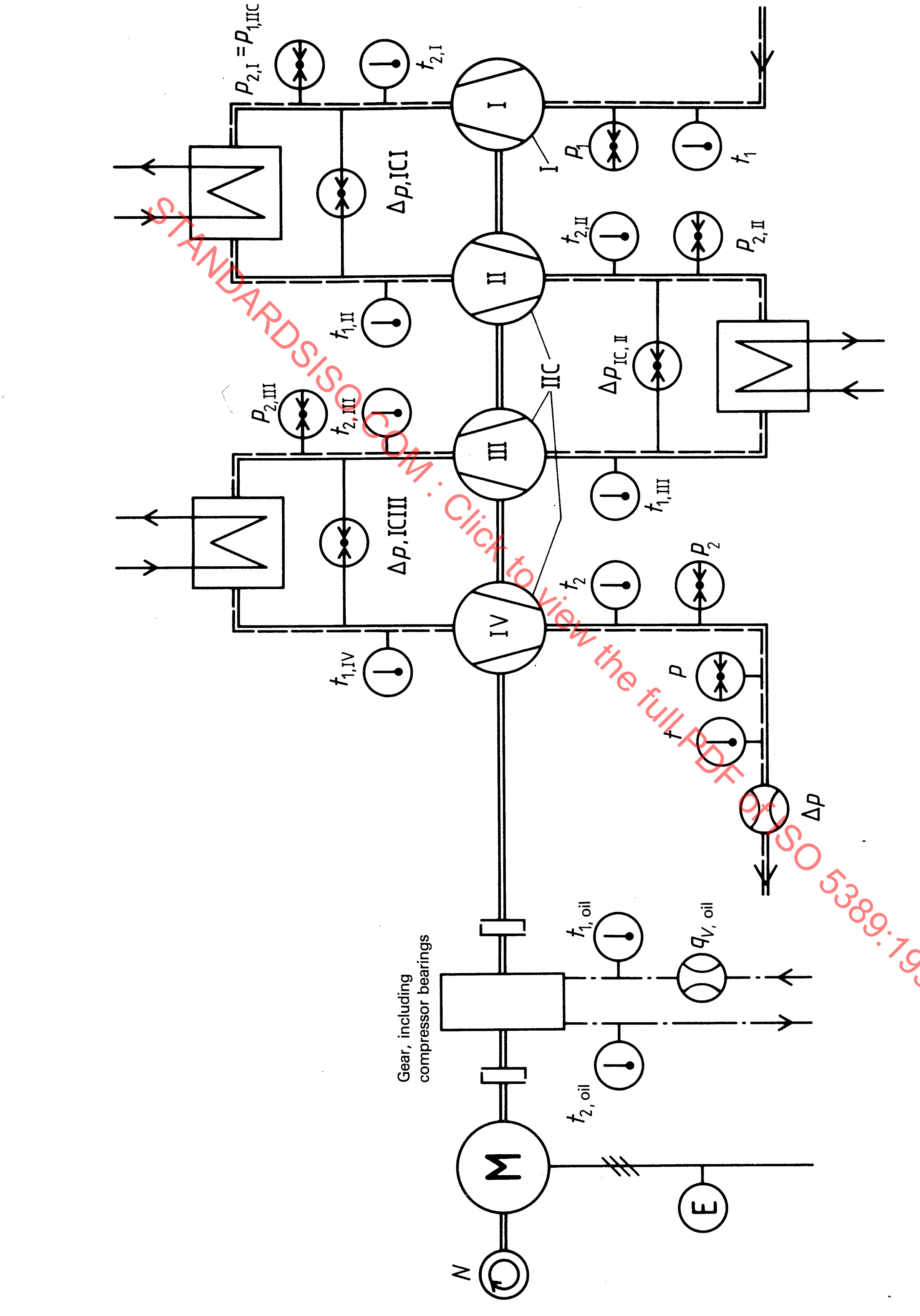


Figure F.11 — Test set-up for example 5

Table F.46 — Calculation results

Designation	Symbol	Numerical value Test No. 1	Test No. 2	Test No. 3	Unit
Pressure ratio	$(p_2/p_1)_t$	7,773	7,645	7,202	—
Specific isothermal compression work	$W_{m,\mathrm{T,Te}}$	168 341	167 409	162 614	N.m/kg
Terminal power of motor	$P_{\mathrm{un,Te}}$	2 168	1 795	1 549	kW
Motor efficiency	$\eta_{\mathrm{Pr,Te}}$	95,4	95,3	95,0	%
Power at coupling	$P_{\mathrm{e,Te}}$	2 068	1 711	1 472	kW
Isothermal compression power	$P_{\mathrm{T,Te}}$	1 449	1 131	922	kW
Isothermal coupling efficiency	$\eta_{\mathrm{T,e,Te}}$	70,1	66,1	62,6	%
Mechanical losses	$P_{\mathrm{f,Te}}$	70	70	70	kW
Internal compression power	$P_{\mathrm{in,Te}}$	1 998	1 641	1 402	kW

Table F.47 — Conversion to guarantee conditions (see figure D.11)

Designation	Symbol	Numerical value Test No. 1	Test No. 2	Test No. 3	Unit
a) Unit I (uncooled Stage I)					
Volumetric flow at inlet	$q_{V,\mathrm{t,1,I,Te}}$	7,295 28	5,662 78	4,724 44	m³/s
Discharge pressure	$p_{\mathrm{t,2,I,Te}}$	1,695 6	1,554 4	1,446 5	bar
Inlet pressure	$p_{\mathrm{t,1,I,Te}}$	0,968 9	0,982 6	0,988 5	bar
Pressure ratio	$(p_2/p_1)_{\mathrm{t,I,Te}}$	1,750	1,582	1,463	—
Specific isentropic compression work	$W_{m,\mathrm{s,I,Te}}$	49 837	40 354	33 097	N.m/kg
Converted volumetric flow at inlet	$q_{V,\mathrm{t,1,I,Co}}$	7,305 00	5,662 78	4,718 06	m³/s
Converted specific compression work	$W_{m,\mathrm{s,I,Co}}$	49 975	40 354	32 950	N.m/kg
Converted pressure ratio	$(p_2/p_1)_{\mathrm{t,I,Co}}$	1,725	1,563	1,446	—
Discharge temperature	$t_{\mathrm{t,2,I,Te}}$	74,6	71,8	70,1	°C
Inlet temperature	$t_{\mathrm{t,1,I,Te}}$	12,1	12,9	13,0	°C
Internal compression power	$P_{\mathrm{in,I,Te}}$	542	401	326	kW
Converted internal compression power	$P_{\mathrm{in,I,Co}}$	534	389	313	kW
b) Unit IIc (cooled Stages II to IV)					
Inlet temperature	$t_{\mathrm{t,1,IIc,Te}}$	36,8	36,6	36,6	°C
Inlet pressure	$p_{\mathrm{t,1,IIc,Te}}$	1,695 6	1,554 4	1,446 5	bar
Density	$\varrho_{\mathrm{t,1,IIc,Te}}$	1,900	1,743	1,622	kg/m³
Volumetric flow at inlet	$q_{V,\mathrm{t,1,IIc,Te}}$	4,530 83	3,875 83	3,495 28	m³/s
Converted volumetric flow at inlet	$q_{V,\mathrm{t,1,IIc,Co}}$	4,536 94	3,875 83	3,490 56	m³/s
Discharge pressure of entire compressor	$p_{\mathrm{t,2,Te}}$	7,531 5	7,511 9	7,119 6	bar
Pressure ratio	$(p_2/p_1)_{\mathrm{t,IIc,Te}}$	4,442	4,833	4,922	—
Specific isothermal compression work	$W_{m,\mathrm{T,IIc,Te}}$	133 076	140 441	142 138	N.m/kg
Converted specific isothermal compression work	$W_{m,\mathrm{T,IIc,Co}}$	133 439	140 441	141 657	N.m/kg
Converted pressure ratio[1]	$(p_2/p_1)_{\mathrm{t,IIc,Co}}$	4,431	4,795	4,862	—
Internal compression power[2]	$P_{\mathrm{in,IIc,Te}}$	1 456	1 240	1 076	kW
Converted power related to density[1]	$\left(\dfrac{P_{\mathrm{in}}}{p_{\mathrm{t,1}}/RZT_{\mathrm{t,1}}}\right)_{\mathrm{IIc,Co}}$	769	711	661	kW·m³/kg

Table F.47 — Conversion to guarantee conditions (see figure D.11) (*concluded*)

Designation	Symbol	Numerical value Test No. 1	Test No. 2	Test No. 3	Unit
c) Unit IIc[3] (see figure D.11)					
Volumetric flow at inlet	$q_{V,\mathrm{t},1,\mathrm{IIc,Co}}$	4,480 28	3,833 06	3,451 94	m³/s
Appropriate pressure ratio	$(p_2/p_1)_{\mathrm{t,IIc,Co}}$	4,46	4,80	4,87	—
Converted internal power	$P_{\mathrm{in,IIc,Co}}$	1 444	1 206	1 037	kW
d) Total compressor					
Volumetric flow at inlet	$q_{V,\mathrm{t},1,\mathrm{Co}}$	7,305 00	5,662 78	4,718 06	m³/s
Converted pressure ratio of Unit I	$(p_2/p_1)_{\mathrm{t,I,Co}}$	1,725	1,563	1,446	—
Converted pressure ratio of Unit IIc	$(p_2/p_1)_{\mathrm{t,IIc,Co}}$	4,460	4,820	4,895	—
Converted total pressure ratio	$(p_2/p_1)_{\mathrm{t,Co}}$	7,694	7,5	7,0	—
Converted internal power Unit I	$P_{\mathrm{in,I,Co}}$	534	389	313	kW
Converted internal power Unit IIc	$P_{\mathrm{in,IIc,Co}}$	1 444	1 206	1 037	kW
Mechanical losses in test	$P_{\mathrm{f,Te}}$	70	70	70	kW
Converted mechanical losses	$P_{\mathrm{f,Co}}$	70	70	70	kW
Converted power at coupling (see figure F.13)	$P_{\mathrm{e,Co}}$	2 048	1 665	1 420	kW
Calculated specific isothermal compression work	$W_{m,\mathrm{T,Co}}$	172 837	171 056	165 828	N·m/kg
Converted isothermal compression power	$P_{\mathrm{T,Co}}$	1 463	1 120	905,8	kW
Converted isothermal coupling efficiency	$\eta_{\mathrm{T,e,Co}}$	71,4	67,5	63,8	%

1) The converted pressure ratio and converted power related to density are shown over the converted volumetric flow at inlet Unit IIc, $q_{V,\mathrm{t},1,\mathrm{IIc,Co}}$.

2) The internal compression power is given by

$$P_{\mathrm{in,IIc,Te}} = P_{\mathrm{in,Te}} - P_{\mathrm{in,I,Te}}$$

3) Taking the converted pressure ratio of the uncooled Unit I into account, the appropriate volumetric flows at inlet for the cooled Unit IIc are obtained for the individual test points.

F.5.7 Test uncertainty and comparison with guarantee

The test uncertainties are calculated in accordance with clause 9.

The comparison with the guarantee is carried out as shown in table F.48 (see figure F.14).

Apart from the test values listed here and designated by the symbol "O" in figures F.12 and F.13, additional test points were run at each of the inlet guide vane positions, and these have been evaluated and converted in the same manner.

All those points have been plotted on figures F.12 and F.13. The power consumption at the guaranteed pressure ratio 7,0 can be obtained from figure F.14.

The guarantees are met for two points.The third point shows a slight excess, even after taking into account the measuring error.

The test uncertainty for example 5 is shown in table F.49.

Table F.48 — Comparison with the guarantee

Designation	Symbol	Numerical value			Unit
		Test No. 1	Test No. 2	Test No. 3	
Volumetric flow at inlet	$q_{V,\mathrm{t},1,\mathrm{Gu}}$	7,208 3	5,763 9	4,680 6	m³/s
Guaranteed power at coupling	$P_{\mathrm{e,Gu}}$	1 970	1 610	1 390	kW
Converted power at coupling	$P_{\mathrm{e,Co}}$	1 960	1 635	1 405	kw

Table F.49 — Resulting test uncertainty

Designation	Symbol	Numerical value	Comments
Uncertainty in volumetric flow at inlet	$\tau_{q_m,\mathrm{Te}}$	1,1 %	In accordance with ISO 5167-1
	$\tau_{N,\mathrm{Te}}$	0,07 %	Digital measuring instrument, 1 revolution on indicator
	$\tau_{p_1,\mathrm{Te}}$	0,14 %	1 mmHg on absolute pressure — See 5.9
	$\tau_{T_1,\mathrm{Te}}$	0,35 %	1 ° on absolute temperature
	$\tau_{\mathrm{res},q_V,1}$	1,165 %	See 9.2.5
Uncertainty in pressure ratio	N_r	1,014	
	$\ln(p_2/p_1)$	2,051	
	$\tau_{p_2,\mathrm{Te}}$	0,9 %	Quality grade 0,6; final value 10 atm gauge (see 5.9)
	$\tau_{\mathrm{res},p_2/p_1}$	1,184 %	See 9.2.6
Uncertainty in power at coupling	ζ_4	2 954	
	τ_{P_e}	0,87 %	Current transformer 0,5; voltage transformer 0,5; wattmeter 0,5
	τ_{P_f}	2,86 %	See 5.9
	$\tau_{T_1,\mathrm{llc},\mathrm{Te}}$	0,323 %	1 ° on absolute temperature
	$\tau_{T_1,j,\mathrm{Te}}$	0,32 %	1 ° on absolute temperature
	$\tau_{\mathrm{res},P_\mathrm{e}}$	1,042 %	See table 9

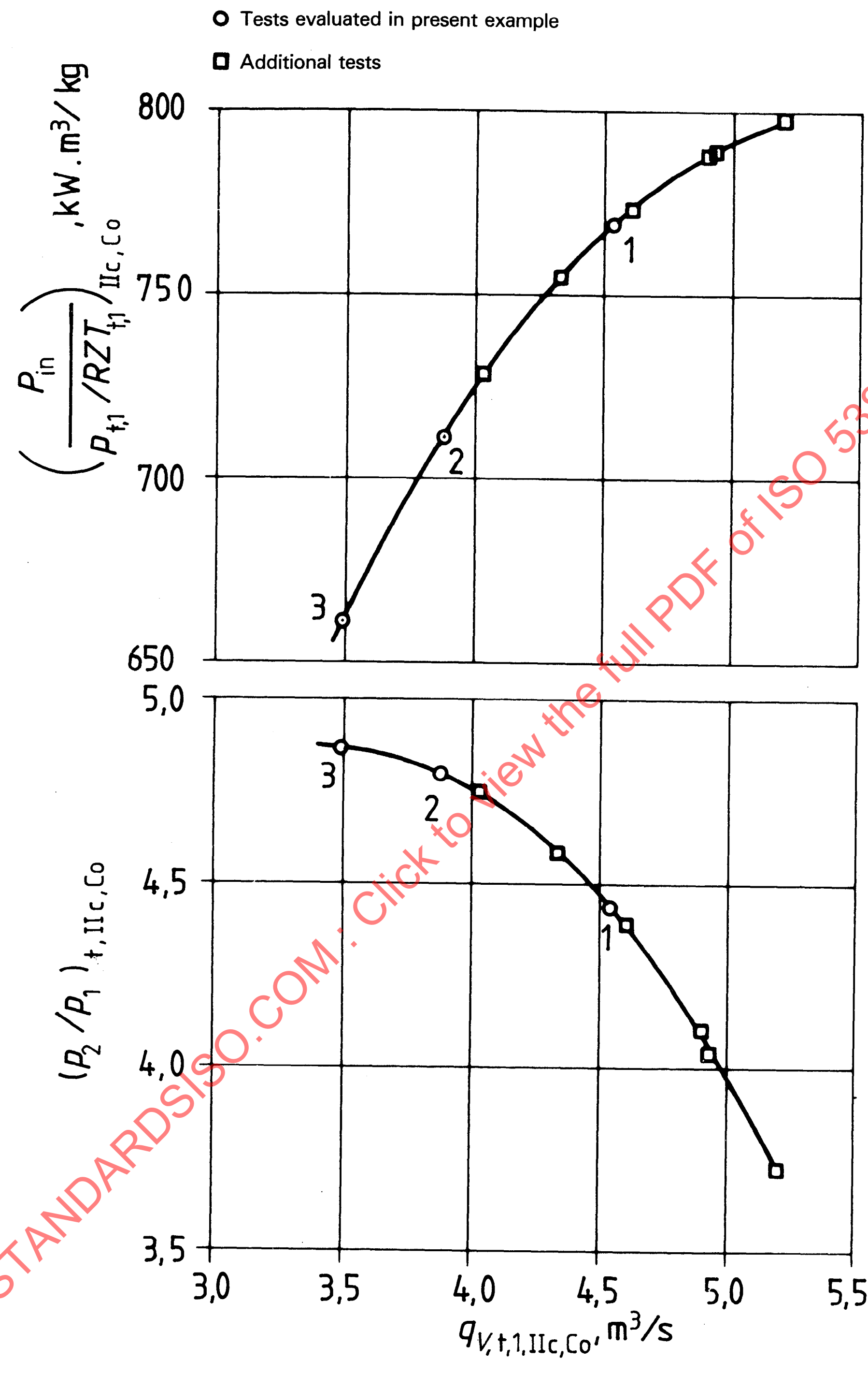


Figure F.12 — Graphs for comparison with guarantee

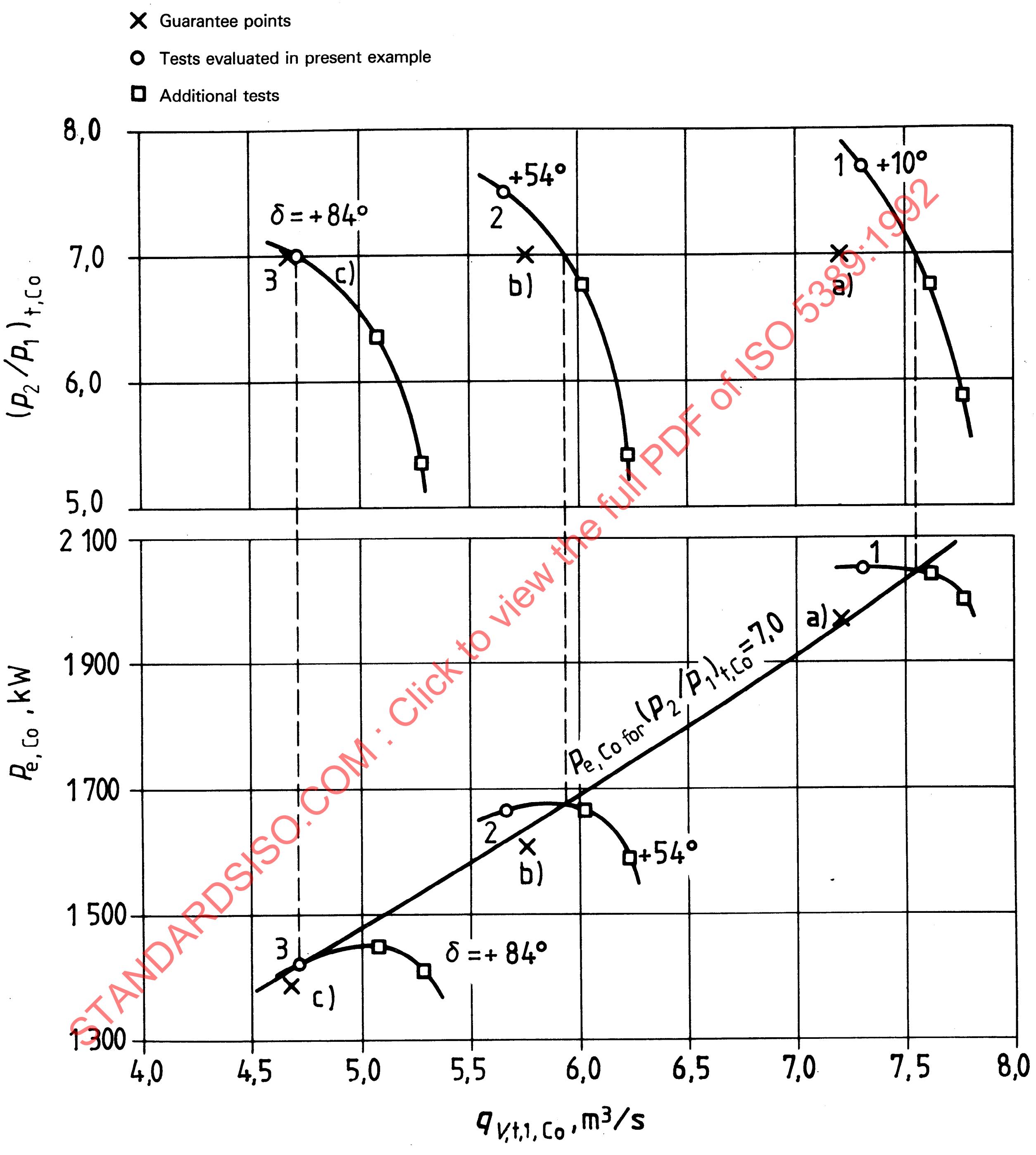


Figure F.13 — Graphs for comparison with guarantee

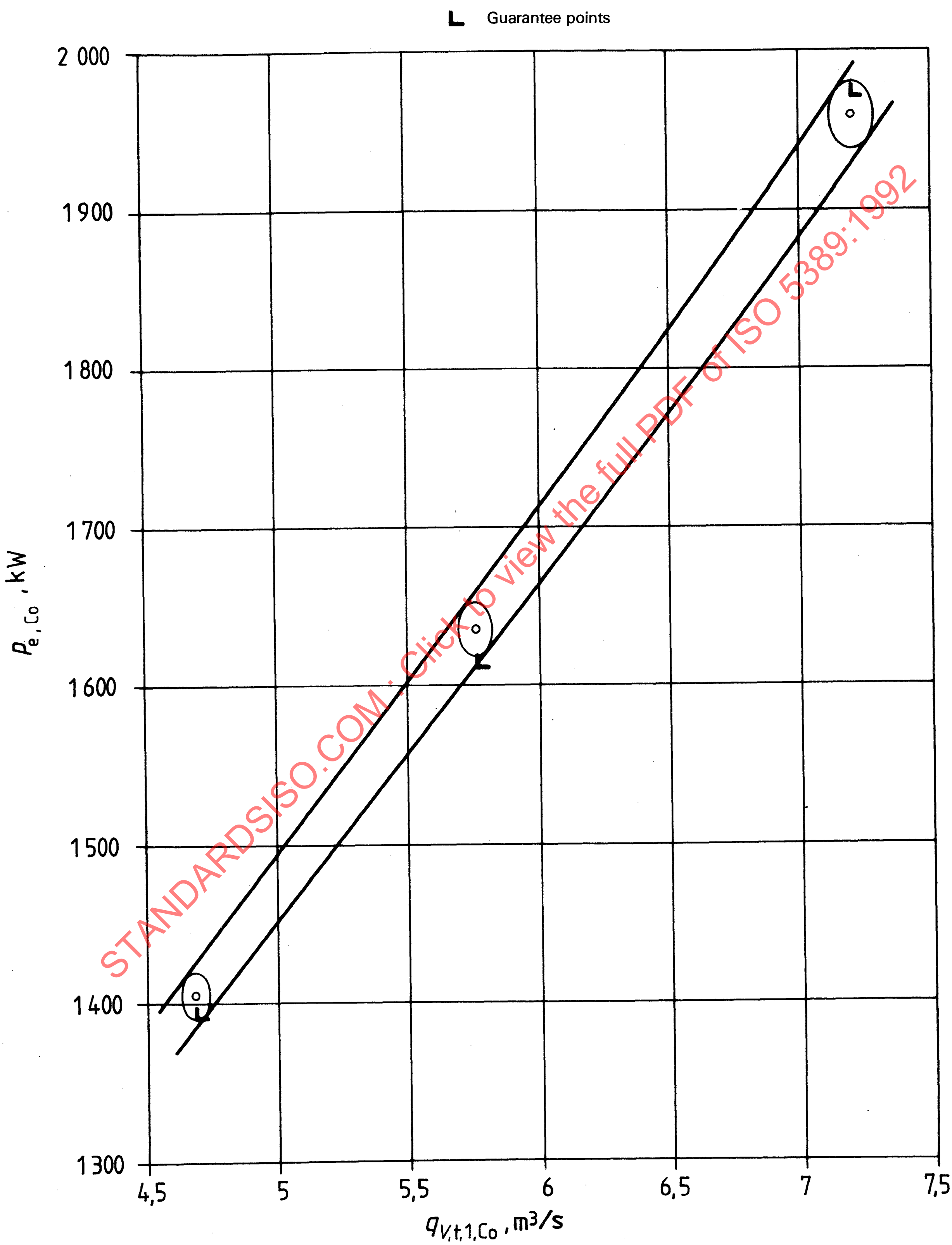


Figure F.14 — Graph for comparison with guarantee