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Turbocompressors — Performance test code

Turbocompresseurs — Code d'essais des performances

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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.

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 2.

The main task of technical committees is to prepare International Standards. 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.

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

ISO 5389 was prepared by Technical Committee ISO/TC 118, *Compressors and pneumatic tools, machines and equipment*, Subcommittee SC 1, *Process compressors*.

This second edition cancels and replaces the first edition (ISO 5389:1992), which has been technically revised. In particular, an improved flow Sheet for determination of setting conditions using similarity conditions has been integrated, taking into account the Reynolds number correction method.

Three classes of conversion of test results have been defined, including tests beyond flow similarity conditions.

The subclause on measuring uncertainties has been revised. The tried and proven procedure for determination of measuring uncertainties using the difference method has been added in order to be able to meet all test requirements, including in particular those occurring in the case of multicasing compressors and machine sets consisting of different driving machines and compressors.

The subclause on guarantee comparison has been enlarged, taking into account all possible cases of performance curves and guarantee points.

ISO 5389 was prepared, based on ASME PTC 10 [1] and VDI 2045-1 [2] and VDI 2045-2 [3].

Turbocompressors — Performance test code

1 Scope

This International Standard applies to performance tests on turbocompressors of all types. It does not apply to fans and high-vacuum pumps, or to jet-type compressors with moving drive components

Turbocompressors comprise machines in which inlet, compression and discharge are continuous flow processes. The gas is conveyed and compressed in impellers and decelerated with further increase in pressure in fixed vaned or vaneless stators.

This International Standard is intended to provide standard provisions for the preparation, procedure, evaluation and assessment of performance tests on compressors as specified above. The acceptance test of the performance is based on this performance test code. Acceptance tests are intended to demonstrate fulfilment of the order conditions and guarantees specified in the contract.

2 Normative references

The following referenced documents are indispensable for the application of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ISO 5167-1, *Measurement of fluid flow by means of pressure differential devices inserted in circular cross-section conduits running full — Part 1: General principles and requirements*

3 Symbols and definitions

3.1 Symbols and units

3.1.1 Latin letters

Symbol	Meaning	Unit
A	area	m^2
a	sonic velocity	m/s
B	manufacturing tolerance	%
b	outlet width of 1st impeller	m
c	velocity	m/s
c_p, c_v	specific heat capacity	kJ/(kg·K)
c_i	evaluation coefficients	—
D	outer impeller diameter of the first impeller	m
f	correction factor	—
f_x	mean relative deviation	

Symbol	Meaning	Unit
G	quality grade	%
g	local acceleration due to gravity	m/s^2
h	specific enthalpy	kJ/kg
k	isentropic exponent	—
k_T	isentropic exponent, temperature	—
k_V	isentropic exponent, volume	
l	length of column	mm
Ma	Mach number	—
M_t	torque	Nm
M	molar mass	kg/mol
m	temperature exponent	—
$\dot{m}$	mass flow	kg/s
N	speed of rotation	1/s
n	polytropic exponent	—
P	power	kW
p	pressure	MPa (bar)
$\dot{Q}$	heat flow	kW
R	specific gas constant	J/(kg·K)
Ra	average roughness	µm
R_mol	universal gas constant	J/(kmol·K)
Re	Reynolds number	—
S	digital measuring step	—
s	specific entropy	kJ/(kg·K)
T	thermodynamic temperature	K
t	temperature	°C
u	tip speed, referred to D	m/s
u	specific internal energy	kJ/kg
V	confidence interval or measuring uncertainty	—
v	specific volume	m^3/kg
$\dot{V}$	volume flow	m^3/s
W	result function	—
w	mass fraction	—
X	compressibility function	—
X_N	ratio of reduced speeds of rotation	—
x	vapour content referred to moist mass of vapour of the same gas	kg/kg
$x_\mathrm{(Subscript)}$	vapour content of vapour/gas mixtures referred to dry gas	kg/kg
Y	compressibility function	—
y	function value	—

Symbol	Meaning	Unit
y	specific compression work	kJ/kg
Z	compressibility factor	—
z	number of stage groups	—

3.1.2 Greek letters

Symbol	Meaning	Unit
α	coefficient of heat transfer	W/(m²·K)
β	coefficient of cubic expansion	1/K
γ	weighting factor	—
Δ	difference	—
ε	calculation coefficient	—
η	efficiency	—
η	dynamic viscosity	Ns/m^2
ϑ	ratio of $(RZ_1\ T_1)$ values	—
κ	ratio of specific heat capacities	—
ν	polytropic ratio	—
ν	kinematic viscosity	m^2/s
Π	pressure ratio	—
ρ	density	kg/m^3
τ	relative uncertainty of measurement	—
ϕ	ratio of volume flow ratios	—
φ	flow coefficient	—
$\varphi_{(Subscript)}$	relative humidity	—
ψ	reference process work coefficient	—
ω	angular speed	1/s

3.1.3 Subscripts

Index	Meaning
1	inlet (suction side)
2	outlet (discharge side)
I, II, III, ..., z	stages, numbered in direction of flow
∞	at an infinitely large Reynolds number
A	uncooled section of an intercooled compressor
air	dry air
amb	ambient (air, temperature)
an	assumption, driving machine
av	average
B	cooled section of a multi-stage intercooled compressor
cal	calibration

Index	Meaning
co	converted to guarantee conditions
cog	converted to the pressure ratio and inlet volume flow of the guarantee point
comb	combined sections
cond	condensate
cou	coupling
crit	critical
d	dynamic
dev	deviation
dr	driving machine
dry	dry
eff	effective
Ex	extreme value of ϕ
g	guarantee or reference conditions
gas	gas
i	ith term of a sum (i = 1, 2, 3, ...)
i	internal
in	input
j	number of stage group (j = I, II, III, ..., z)
k	isentropic exponent
L	leakage
lub	lubricant
M	measurement, motor
$\dot{m}$	mass flow
mech	mechanical
n	standard state
N	frequency of rotation
out	output
p	polytropic
P	power
Pr	reference or standard process
pr	precalculated or predicted test results
rad	radiation and convection
ran	relevant measuring range of instrument
Re	referred to Reynolds number
red	reduced speed
ref	reference value
res	result
s	isentropic
sat	saturated steam/vapour

Index	Meaning
seal	sealing liquid
side	sidestream or extractions
st	static
sup	supply
sur	surface
sys	system
T	isothermal
t	temperature
te	test result
term	terminals
tol	permissible deviation
tot	total
u	tip or peripheral
us	usable
V	volume
vap	vapour, steam
wet	moist
wf	working fluid
W	cooling water or coolant
x	between inlet and outlet
y	function value

Where no specific remark is made to the contrary, the thermodynamic variables of state used without indices in this International Standard describe total state.

3.2 Definitions

For the purposes of this document, the following terms and definitions apply. Additional terms and definitions are given in Annex E.

3.2.1
ratio of volume flow ratios

$$\phi = \frac{\left(\dot{V}_1 / \dot{V}_2\right)_{\text{te}}}{\left(\dot{V}_1 / \dot{V}_2\right)_{\text{g}}} \tag{1}$$

3.2.2
ratio of reduced speeds of rotation

$$X_N = \frac{\left(\dfrac{N}{\sqrt{R \cdot Z_1 \cdot T_1}}\right)_{\text{te}}}{\left(\dfrac{N}{\sqrt{R \cdot Z_1 \cdot T_1}}\right)_{\text{g}}} \tag{2}$$

3.2.3
tip Mach number

$$Ma_u = \frac{u}{a_1} \quad (3)$$

3.2.4
tip Reynolds number

$$Re_u = \frac{ub}{\upsilon_1} \quad (4)$$

3.2.5
volume flow coefficient

$$\varphi = \frac{\dot{V}_1}{\frac{\pi}{4} \cdot D^2 \cdot u} \quad (5)$$

3.2.6
reference process work coefficient

$$\psi_{Pr} = \frac{y_{Pr}}{u^2/2} \quad (6)$$

3.2.7
enthalpy coefficient

$$\psi_i = \frac{\Delta h}{u^2/2} \quad (7)$$

3.2.8
RZ_1T_1 **ratio**

$$\vartheta_j = \frac{(R \cdot Z_1 \cdot T_1)_j}{(R \cdot Z_1 \cdot T_1)_I} \qquad \vartheta_{j,B} = \frac{(R \cdot Z_1 \cdot T_1)_j}{(R \cdot Z_1 \cdot T_1)_{I,B}} \quad (8)$$

where I,B is the first stage of cooled section B

3.2.9
section
one to several successive stages of a turbocompressor without intercooling through which the same mass flow flows

4 Guarantees

4.1 General

The customer and the manufacturer shall make a contractual agreement specifying which properties and characteristics of the compressor are to be guaranteed and demonstrated by the acceptance test. Verification of these properties is effected by means of the values measured in the acceptance test and converted to the guarantee conditions.

Fulfilment of the guarantee may be demanded only if all components of the compressor system are in correct condition at the acceptance test (see 6.1.3).

4.2 Preconditions for the guarantee

The conditions that apply as a precondition for the guarantee, modification of which will affect the functioning of the compressor, shall be specified in the contract of supply. These conditions can include the following:

a) inlet pressure (or discharge pressure in the case of suction-type compressors) and inlet temperature;

b) in the case of inward sidestreams, their thermodynamic states and the ratio of the side mass flows to the inlet mass flow, in the case of intermediate extraction the ratio of the extracted mass flows to the inlet mass flow and the extraction pressure;

c) in the case of intercooled compressors, the recooling temperatures and pressure drops between the relevant compressor sections;

d) physical properties of the gas or vapour and its composition in volume or mass fractions;

e) coolant, its mass flow and inlet temperature;

f) operating conditions of the driving machine (e.g. enthalpy differences, inlet and outlet state, heat value of the fuel, type, voltage and frequency of electrical current, speed);

g) inlet and outlet state referred to the inlet and outlet flow area of the compressor;

h) speed (necessary deviations to meet the guarantee points shall be agreed upon between customer and manufacturer).

4.3 Object of the guarantee

The following values can be guaranteed under the preconditions specified in 4.2:

a) actual inlet volume flow as defined in E.4.2;

b) discharge pressure (or inlet pressure in the case of suction-type compressors) and intermediate pressures in the case of inward sidestreams and intermediate extraction;

c) the power for specified inlet volume flows and discharge pressures (or inlet pressures in the case of vacuum-type compressors) in the form of

- compressor power at the compressor coupling, or
- power of the compressor with gearbox at the coupling of the driving machine, or
- electrical power at the terminals of the drive motor, or
- driving machine fuel consumption.

Where the compressor and driving machine have common components (e.g., bearings, oil pumps, etc.), an agreement shall be made specifying the manner in which the losses occurring inside the components are to be apportioned (see 5.9).

The related power or the efficiency related to a suitable reference process (see E.5) may also be guaranteed instead of power.

d) the power of auxiliary machinery (e.g. oil pumps or cooling-water pumps) where such is not included in the guaranteed power;

e) operating range limits, as follows:

- maximum actual inlet volume flow at a specified discharge pressure or maximum pressure at a specified actual inlet volume flow,
- minimum actual inlet volume flow at a specified discharge pressure,
- surge limit.

See E.9.

4.4 Supplementary guarantees

Additional guarantees (for part-load efficiencies, sealants, temperature of the gas compressed, cooling efficiency of coolers and condensers) can be required in cases were they are of significance for operation, or for any other reasons.

4.5 Guarantee comparison

In case of an acceptance test, the test results measured and converted to the guarantee conditions shall be assessed against the values guaranteed (see Clause 8), making allowance for the limits of measuring uncertainties (see 6.4).

Any manufacturing tolerances for the guarantee shall be deemed to constitute a component of the contract of supply and not of this International Standard.

4.6 Guarantees for series production

Where a series of compressors of the same design are manufactured within a short period of time, it is not customary to perform an acceptance test on each individual compressor. Such a test performed on a few compressors selected at random from the series and completed successfully, constituting a type-test, shall be deemed to suffice. The details of this procedure shall be governed by the contract of supply.

5 Measuring methods and measuring equipment

5.1 General

5.1.1 Measuring methods and measuring uncertainties

Following measuring methods and measuring instruments inclusive of the rules necessary for their use shall be used if applicable.

Other measuring methods may be used upon agreement regarding testing and fitting.

5.1.2 Facilities for measurement

The measuring points and equipment for measurement of pressure, temperature, flow, power and speed shall be incorporated into the compressor during design and during its installation into the subsequent system. Above all, it shall be ensured at all points for measurement of flow as specified in ISO 5167-1 that adequate lengths of straight pipe are available and suitable flanged joints for installation of the orifices and nozzles. Figures E.3 and E.4 illustrate a suitable arrangement for two measuring points each for pressure and temperature on the compressor. Guarantees should be referred to the measuring points provided and prepared. Sockets for reference instruments should be provided at the main measuring points.

5.1.3 Measuring instruments

The following measuring instruments shall be used for acceptance tests:

a) measuring instruments which have been calibrated by comparison with measuring instruments as specified in 5.1.3 c),

b) measuring instruments for which a calibration or test certificate issued by an accredited authority is submitted,

c) other tried and proven measuring instruments of a known accuracy, the use of which has been agreed between the parties to the contract.

All measuring instruments (and orifices and nozzles in particular) shall be checked immediately before installation and/or before and after the test for condition and dimensional accuracy. It shall, in addition, be ensured that the installation point, installation itself, and the measuring instrument itself comply with the relevant specifications. The result of this check shall be recorded.

5.1.4 Use of transducers; data acquisition

When electronic measuring instruments are used with transducers of any type and digital evaluation is possible, the transducers shall be calibrated and a record kept of calibration. It shall be possible to check the measuring systems by suitable means. This provision applies analogously to the use of data acquisition systems and electronic data processing.

5.2 Pressures

5.2.1 Static pressure

The static pressure present at a wall should be measured by means of holes drilled in the wall. Such holes shall have neither a burr on the wall surface, nor a flared opening. The diameter of the holes shall be kept as small as possible; the lower limit is that adequate to avoid the danger of blockage.

In long straight pipes, flow parallel to the pipe axis is established. The static pressure may then be assumed to be constant in every flat flow cross-section perpendicular to the axis of the pipe; sampling of pressure by means of a hole drilled in the pipe wall then suffices for the purpose of measurement (see Figures E.3 and E.4 for the pressure-sampling apparatus).

5.2.2 Dynamic pressure and total pressure

Where an average velocity, c, is known from flow measurement and flow area, an average dynamic pressure, p_d, can be calculated from this and with the static pressure, p, an average total pressure, p_tot, can be calculated as follows:

For the average velocity:

$$c = -\frac{c_p \cdot p \cdot A}{\dot{m} \cdot R \cdot Z} + \sqrt{\left(\frac{c_p \cdot p \cdot A}{\dot{m} \cdot R \cdot Z}\right)^2 + 2 \cdot c_p \cdot T_\mathrm{tot}} \qquad (9)$$

For the ratio of total to static pressure:

$$\frac{p_\mathrm{tot}}{p} = \frac{p + p_\mathrm{d}}{p} = \left(\frac{T_\mathrm{tot}}{T}\right)^{\frac{k}{k-1}} \qquad (10)$$

This approximation for the calculation of the dynamic and total pressure with the average velocity, c, is regarded as sufficiently accurate in the scope of the present rules.

5.2.3 Installation of measuring lines

Measuring lines installed between the sampling point and the display instrument shall be installed with great care. Any leaks shall be eliminated. Provisions shall be made to prevent blockage by foreign bodies. Where condensate occurs in the measuring lines, such lines shall be completely filled with condensate or shall be reliably kept free of condensate (e.g. by arranging the measuring instrument at a geodetic higher level than the measuring point).

5.3 Temperatures

The static temperature, T, and total temperature, T_{tot}, cannot be directly measured as variables of state of a gas in flow.

Ratio of total to static temperature:

$$\frac{T_{tot}}{T} = \frac{1}{1 - \frac{c^2}{2 \cdot c_p \cdot T_{tot}}} \quad (11)$$

Temperature sensors of conventional type and size (liquid thermometers, thermocouples, resistance thermometers with or without thermowells for installation) gravitate, even when correctly installed, to their so-called characteristic temperature, which is located between T and T_{tot}, as soon as they are exposed to the flowing gas. There are, however, temperature probes ("total temperature measurement instruments") such as plate-type, hook, and diffusor thermometers, the indication of which approximates extremely closely to the total temperature (temperature at rest) of the gas.

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.

5.4 Gas density

For gases and vapours of known composition, density can be determined from equations of state, state charts, or tables. In the case of gas mixtures of unknown composition, density should be measured directly using an acknowledged method.

5.5 Gas composition

5.5.1 General

Where mixtures of gases or gas/vapour mixtures are being compressed, the composition of the mixture shall, if necessary, be checked at regular intervals using an acknowledged method. The frequency, nature and accuracy of such checks will vary according to fluctuations in gas composition.

5.5.2 Moisture content

5.5.2.1 Air humidity

The relative humidity, expressed in percent, of air at atmospheric pressure (p_{amb}) can be calculated as follows using the temperatures read on the wet (t_{wet}) and dry (t_{dry}) thermometer of a psychrometer (as defined, for instance, by *Assmann*) using Sprung's approximation equation:

$$\varphi_{vap} = \frac{p_{sat} - 0{,}5 \cdot \left(t_{dry} - t_{wet}\right) \cdot \frac{p_{amb}}{755}}{p_{dry}} \cdot 100 \qquad (12)$$

where

p_{sat} is the saturated vapour pressure at t_{wet};

p_{dry} is the saturated vapour pressure at t_{dry};

p_{amb} is the ambient pressure reading.

Relative humidity (φ_{vap}) can be read from an h_{air} - x_{air} chart for any pressure, p, of the air at known values for t_{wet} and t_{dry} and the barometer level p_{amb}.

The relative humidity of compressed air can be determined by diverting a side stream from the centre of the pressure line and depressurizing it to atmospheric pressure. The relative humidity, φ_{vap}, measured at atmospheric pressure, shall then be converted to the state in the line.

Recognized methods other than the psychrometric measuring method are also permissible (e.g. the dewpoint, freezing-out, lithium chloride and absorption methods).

5.5.2.2 Moisture in other gases

The other methods mentioned in 5.5.2.1 are recommended for use with gases other than air [instead of Equation (12)].

5.6 Gas velocity

5.6.1 Quantitative measurement

The numerical value for local velocity can be measured using indicating anemometers or probes (e.g. Prandtl or pitot tube), which are non-direction-dependent within certain limits (see 5.7.3).

5.6.2 Determination of direction

The direction of velocity can be determined using fixed calibrated probes, or by means of the pressure differences measured at adjustable probes.

Determination of direction is not necessary in long straight piping sections.

5.7 Volume flow and mass flow

5.7.1 Flow measurement using orifices and nozzles

ISO 5167-1 is definitive for measurement of flow using orifices and nozzles. Measurement may be effected using non-standardized orifices and nozzles if special agreements to this effect have been made (see e.g. References [4] and [5]).

5.7.2 Measurement using gas meters

Volume flow measurements can be effected using calibrated gas meters.

It shall be ensured that the gas flows through the meter without disruption by pulsating surges. The meter shall also be checked for leaks at the drums or bellows and for precise filling with sealant liquid and for changes in the gas-saturation level of the sealing liquid.

5.7.3 Other measuring methods

If one of the measuring methods mentioned in 7.5.1 and 7.5.2 is not be practicable for technical or economical reasons, other measuring methods may be used upon agreement between the customer and the manufacturer.

In a constant flow, the volume or mass flow can be determined from a calibrated pressure difference or by means of measurement of the velocity profile (e.g. Reference [6]). The mass flow can also be calculated from suitable energy balances, with the inclusion of drive power or of the process.

5.8 Speed of rotation

Where measurement of the speed of rotation is necessary for the performance test, it shall be determined with the accuracy necessary for this purpose using a cyclometer, tachometer, frequency meter, etc.

5.9 Power

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

a) by performing an energy balance on the driver in accordance with the appropriate test codes for the particular type of machine;

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

c) 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.

In case 5.9 a), 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.

In the case 5.9 b) of measuring the torque, 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.

In the case 5.9 c) of establishing a total energy balance of the compressor-heat exchange with the ambient air by means of conduction and radiation shall be taken into account:

$$\dot{Q}_{\text{rad}} = \alpha \cdot A_{\text{rad}} \cdot \left(t_{\text{sur}} - t_{\text{amb}}\right) \tag{13}$$

A coefficient of heat transfer α = 14 [W/(m^2·K)] can be used for estimation of these losses. A_{rad} is the external surface of the compressor between inlet and discharge. t_{sur} is a mean surface temperature of the compressor, either measured or estimated from the gas temperatures in the compressor. If the radiation heat loss, $\dot{Q}_{\text{rad}}$, is already known when evaluating the test values, test power can already be corrected by adding $\dot{Q}_{\text{rad,te}}$ to the gas power, $P_{\text{i},\Delta\text{t,te}}$, evaluated from mass flow and temperature rise.

$$P_{i,te} = P_{i,\Delta t,te} + \dot{Q}_{rad,te} \tag{14}$$

Otherwise, e.g. in case of an online test evaluation, $\dot{Q}_{rad}$ is converted separately (see 7.2.4.5).

6 Performance test

6.1 Preparation for the test

6.1.1 General

It shall be ensured when preparing for the performance test that measuring instruments, the measuring inaccuracies of which ensure the necessary level of accuracy (see 6.4.2), are selected.

6.1.2 Test procedure

The type, scope and chronological sequence of measurements, the location of the measuring points and the measuring methods to be used, should all be specified in a test schedule. The diagrams and drawings required for comprehension should be attached to the test procedure.

In the case of performance tests, this procedure should be agreed between the supplier and the purchaser on the basis of the guarantee conditions.

The operating points at test shall be selected in accordance with 7.2. Bypass lines from the pressure to the suction side of the compressor and from the hot-water to the cold-water side of the coolers, including the flow-restriction elements, etc., can be installed, if necessary, as an aid to adaptation of test conditions to guarantee conditions.

6.1.3 Inspections and preliminary test

It shall be ensured before (and after) the performance test that all lines are free of obstructions and all parts of the system are in correct condition. It shall also be ensured that all supply and return lines not in use during the test are correctly closed, by installing blind discs if necessary. All relevant pipes shall be checked for tightness. Any components in the system exposed to fouling, and surface heat-exchange coolers in particular, shall be cleaned on the water and gas sides before the test is started. If this is not possible, corresponding agreements shall be made on the implications.

All measuring instruments and measuring lines shall be carefully checked for correct adjustment and correct connection (see 5.1).

Also in the case of performance tests to be performed at the installation location, the supplier may first perform his own preliminary test. Such preliminary tests can also be used to familiarize the test staff and to test and check the instruments and equipment used. If this test is successful, it can be accepted as a performance test by the customer.

6.2 Execution of the test

6.2.1 General

Performance tests should, wherever possible, take place under the operating conditions specified. It is recommendable to isolate the compressor system from operational fluctuations.

Where the performance test is performed in the system, the adjustment of operating parameters may be performed only in consultation with the person responsible for the system.

During a performance test on a compressor or a compressor system, no modification that would influence the compressor performance and that could not be retained under normal operating conditions may be implemented.

The performance test shall be carried out with all values in steady-state condition.

The data measured, the time of measurement and unusual occurrences shall be documented during the test.

The most important measured values shall (wherever possible) all be registered simultaneously. After the test, the supplier's and purchaser's representatives and any neutral parties attending shall all be supplied with a copy of the documentation.

The type, number and duration of measurements and their frequency will vary according to the importance of the particular measurements, taking into account the special characteristics of the measuring equipment and of operation. An agreement shall be made on this item.

In the case of cooled compressors, it is also advisable to ascertain in a test the effectiveness of the intercooler under design conditions.

6.2.2 Permissible mean value deviations from the values specified in the guarantee conditions and permissible fluctuations of individual values around the mean values

If the operating conditions deviate from the guarantee conditions, the test shall be valid, provided the mean value deviations from the values in the guarantee preconditions are within certain limits. The limits can be found in Tables 1 and 2 (7.2), in Figure 2 and Annex A.

Still greater deviations can be allowed, provided corresponding agreements have been made between the supplier and the purchaser.

Where individual values fluctuate substantially, it is necessary to make an agreement regarding the permissibility and possible enlargement of the measurement uncertainty range, depending on the particular circumstances (see e.g. Reference [7]).

6.3 Evaluation of test results

6.3.1 Averaging

Readings from the values that influence the calculation linearly, taken at equal time intervals, can be averaged arithmetically.

Readings from values that do not influence the calculation linearly, taken at equal time intervals, shall be averaged in the equivalent form.

6.3.2 Mass flow and inlet volume flow

Effective inlet volume flow, $\dot{V}_{1,\mathrm{us,wet}}$, can be determined from measured mass flow, $\dot{m}_{\mathrm{te}}$, (see E.4.2).

6.3.3 Power (power at coupling), fluid consumption

The power (power at coupling), P_{cou}, of the compressor can be determined in accordance with 5.9.

Where a gearbox is used, the gear losses have to be determined separately (by means, for instance, of measurement of the losses dissipated in the form of heat in the gearbox oil).

Where the compressor is driven by thermal machines, the fluid consumption can be determined from the acceptance measurements in accordance with the rules for acceptance of the respective driving machine (see 5.9).

6.3.4 Power of the reference process

The power of the reference process can be calculated using the measured inlet and outlet state. Selection of the reference process (isentropic, polytropic, isothermal) depends on the type and manner of operation of the compressor (see E.5.1).

6.3.5 Specific working fluid consumption

Where a thermal engine is used as driving machine and the operating conditions of the compressor and the driver are constant, the performance of the compressor may be expressed in terms of the mass flow of the driver's working fluid per unit effective inlet volume flow of the compressor.

Where compressor operating conditions are subject to change, but the operating conditions of the driving machine are constant, working fluid consumption should preferably be referred to the power of the reference process, e.g. $\dot{m}_{\text{wf}} / P_{\text{Pr}}$.

6.4 Measuring uncertainty of test results

6.4.1 Basic principles

Any measurement involves a degree of uncertainty. Uncertainties also arise from conversion (see 7.2.5).

The data contained in 6.4 presuppose that the requirements specified in Clause 5 are fulfilled. If this is not the case, an agreement shall be made regarding an appropriate increase in the measuring uncertainties for the individual measured variables and of the confidence ranges for the gas data. It is further assumed that all registrable systematic errors in the measurement of individual measured quantities and gas data have been eliminated by means of corrections. A further precondition is that the confidence limits of the reading error and integration error have been rendered negligible by means of an adequate number of readings. The (small) non-registrable systematic errors are also covered by the measuring uncertainties. Quality grades and error limits are sometimes used for determination of the measuring uncertainties of individual measured quantities, since the registrable systematic error of the measuring instruments used, with some exceptions, covers only a fraction of the quality grade or error limit.

The data regarding the determination of measuring uncertainties for individual measured quantities (6.4.2), for confidence ranges of gas data (6.4.3) and for variables of state, are approximations. These approximations can be improved only with a corresponding level of complexity and expense.

In accordance with Reference [7], the measuring uncertainties defined in this International Standard should be taken at the 95 % confidence limits.

The instructions regarding determination of overall uncertainties of measuring results (6.4.4) and their application as semi-axes for the measuring uncertainty ellipses (8.2.4) include convenient simplifications, such as ignoring certain relationships; see Reference [8].

6.4.2 Measuring uncertainty of individual measured variables

6.4.2.1 Measuring uncertainty of pressures

6.4.2.1.1 Precision pressure gauges and pressure transducers

The relative measuring uncertainty, expressed in percent, for pressure difference is

$$\tau_{\Delta p} = \frac{V_{\Delta p}}{\Delta p} \cdot 100 = \pm G \frac{\Delta p_{\text{ran}}}{\Delta p_{\text{te}}} \tag{15}$$

Where the measuring instrument has a quality grade of $G < 0{,}2$, the term $G = 0{,}2$ should nonetheless be used in the equation, in order to make allowance for mounting errors.

6.4.2.1.2 Liquid columns

If liquid columns are applied, the measuring uncertainty depends above all on the readability of deflection Δl. If no special aids are used, a measuring uncertainty, $V_{\Delta l}$, of ± 1 mm can be achieved.

Relative measuring uncertainty, expressed in percent, in the 100 mm $\leqslant \Delta l \leqslant$ 1 000 mm range is

$$\tau_{\Delta l} = \frac{V_{\Delta l}}{\Delta l} \cdot 100 = \frac{1}{\Delta l} \cdot 100 \tag{16}$$

For $\Delta l >$ 1 000 mm, relative measuring uncertainty, expressed in percent, is

$$\tau_{\Delta l} = 0{,}1$$

6.4.2.1.3 Absolute pressures

The measuring uncertainty of an absolute pressure, p, depends on the uncertainty of the measured ambient pressure, p_{amb}, and the pressure difference, $p - p_{\text{amb}}$:

$$\tau_p = \sqrt{\left(\frac{p_{\text{amb}}}{p} \cdot \tau_{p,\text{amb}}\right)^2 + \left(\frac{p - p_{\text{amb}}}{p} \cdot \tau_{p-p,\text{amb}}\right)^2} \tag{17}$$

6.4.2.2 Measuring uncertainties of temperatures

6.4.2.2.1 General

National standards contain information on the calculation of errors and error limits, with the inclusion of unavoidable minor boundary influences. The provisions in 6.4.2.2.2 to 6.4.2.2.4 are intended to facilitate selection.

6.4.2.2.2 Liquid-in-glass thermometers

The error limit determined by means of calibration and enlarged by mounting allowances should be used as measuring uncertainty, V_{t}. Normally, V_{t} = 1 K.

6.4.2.2.3 Thermocouples

Where the entire measuring system has been calibrated recently and precision measuring instruments (quality grade 0,1) are being used for measurement, a measuring uncertainty, V_{t}, of ± 1,0 K can be used for temperatures to 300 °C.

Substantially smaller measuring uncertainties can be achieved via the use of special instrument combinations, particularly for small temperature differences.

6.4.2.2.4 Resistance thermometers

Where the entire measuring system has been calibrated recently, a measuring uncertainty, V_{t}, of ± 1,0 K can be used for temperatures to 300 °C. The most accurate system for the particular application of the measuring methods should, however, be used for this purpose.

6.4.2.3 Measuring uncertainties of flow

Tolerance, $\tau_{\dot{m}}$, of the flow measurement using standardized orifices and nozzles shall be calculated in accordance with ISO 5167-1. In cases where it is not possible to completely eliminate pulsation surges, correction factors shall be applied. In addition, the tolerance, $\tau_{\dot{m}}$, shall be enlarged by 20 % of the correction factor.

Where measurement is performed using meters (e.g. for oil volume flow), the measuring uncertainties of the instruments (specified, for instance, by means of a test certificate) shall be used.

6.4.2.4 Measuring uncertainties of speed of rotation

The relative measuring uncertainty, expressed in percent, of the speed of rotation using calibrated analogue measuring instruments is

$$\tau_N = \frac{V_N}{N} \cdot 100 = \pm G \frac{N_{\text{ran}}}{N_{\text{te}}} \tag{18}$$

The relative measuring uncertainty, expressed in percent, of speed of rotation using calibrated digital measuring instruments is

$$\tau_N = \frac{V_N}{N} \cdot 100 = \pm \frac{S}{N_{\text{te}}} \cdot 100 \tag{19}$$

6.4.2.5 Measuring uncertainty of torque

The relative measuring uncertainty, expressed in percent, of torque using calibrated torsion dynamometers is

$$\tau_{M,\text{t}} = \frac{V_{M,\text{t}}}{M_{\text{t}}} \cdot 100 = \pm G \frac{M_{\text{t,ran}}}{N_{\text{te}}} \cdot 100 \tag{20}$$

The measuring uncertainty stated by the manufacturer of the measuring equipment can be used for torque measurement using cradle-type motors.

6.4.2.6 Measuring uncertainty of power at the coupling of the driving machine

The relative measuring uncertainty, expressed in percent, of power at the coupling via the measured electrical power of an electric motor is

$$\begin{aligned} \tau_{P,\text{cou}} &= \frac{V_{P,\text{cou}}}{P_{\text{cou}}} \cdot 100 = \pm \sqrt{\left(\frac{V_{P,\text{el}}}{P_{\text{el}}}\right)^2 + \left(\frac{V_{\eta,\text{M}}}{\eta_{\text{M}}}\right)^2} \cdot 100 \\ &= \pm \sqrt{\tau_{P,\text{el}}^2 + \tau_{\eta,\text{M}}^2} \end{aligned} \tag{21}$$

where

$V_{P,\text{el}}$ is the measuring uncertainty of electrical energy consumed;

$V_{\eta,\text{M}}$ is the uncertainty of motor efficiency. The supplier of the electric motor shall supply, with the motor, curves stating motor efficiency as a function of load and shall state the degree of uncertainty.

For the calculation of power at the coupling from measured electrical power consumption and measured individual losses, the measuring uncertainties of these individual losses shall be taken into account in a manner appropriate to the method of their measurement.

For the measurement of power at the coupling on other driving machines, the relative measuring uncertainty, $\tau_{P,\text{cou}}$, shall be calculated in accordance with the corresponding standard.

6.4.2.7 Measuring uncertainties of power from temperature difference and mass flow

Where power (e.g. gas power, mechanical power losses) is determined from a temperature difference and a mass flow, the measuring uncertainty, expressed in percent, is

$$\tau_P = \frac{V_P}{P} \cdot 100 = \pm \sqrt{\tau_{\dot{m}}^2 + \tau_{\mathrm{cp}}^2 + \frac{V_{t1}^2 + V_{t2}^2}{\left(t_2 - t_1\right)^2} \cdot 100^2} \qquad (22)$$

6.4.2.8 Measuring systems

Since the measured value is generally displayed on measuring instruments at the end of a measuring system, the rules for measuring systems specified in applicable standards, e.g. Reference [7], shall also be observed.

6.4.3 Confidence ranges for gas data

6.4.3.1 General

Where gas composition fluctuates, particular care shall be devoted to suitable and correct sampling. The confidence ranges for gas data shall be increased if these fluctuations exceed the ranges that can be balanced out by means of suitable sampling.

The information in 6.4.3.2 to 6.4.3.4 also presupposes suitable chemical or physical analytical methods for determination of gas composition.

6.4.3.2 Gas constant

6.4.3.2.1 Pure gases

Where the gas constant is taken from recognized equations of state, its confidence range, V_R, can be ignored.

6.4.3.2.2 Gas mixtures

The confidence range, V_R, of the gas constants can be ignored, provided the conditions of 6.4.3.1 are met. If the gas constant is determined by means of measurement of density using precision measuring instruments as specified in 5.4, a relative confidence range V_R/R of ± 0,5 % should be used.

6.4.3.3 Compressibility factor

6.4.3.3.1 Pure gases

The confidence range, V_Z, of the compressibility factor can be found in the relevant literature for the pure gases most commonly compressed; see Reference [9].

Where the compressibility factor is determined using equations of state, confidence range, V_Z, should be estimated.

6.4.3.3.2 Gas mixtures

The greatest accuracy level can be achieved by means of measurement of the compressibility factor of the gas mixture.

For the estimation of the confidence range of a compressibility factor determined from equations of state, it is principally the confidence range, V_Z, of the compressibility factor of the component occupying the greatest proportion by volume and the confidence range, V_Z, of the component whose compressibility factor most greatly deviates from 1 that should be used.

6.4.3.4 Isentropic exponent

6.4.3.4.1 Pure gases

Where the isentropic exponent for approximately perfect gases is taken from recognized tables, the confidence range, V_k, of the isentropic exponent can be ignored.

No accurate data are available on the confidence ranges, V_k, of the isentropic exponents of gases which deviate greatly from perfect behaviour; they can be estimated.

6.4.3.4.2 Gas mixtures

The same remarks as those made in 6.4.3.4.1 apply, provided the conditions of 6.4.3.1 are met.

6.4.4 Uncertainty of measuring results

6.4.4.1 General

The equations for calculation of the relative uncertainties of measured results are compiled in 6.4.4.2. They state the semi-axes for the measuring uncertainty ellipses (see 8.2.4) and shall be expanded by the additional tolerances if necessary (see 7.2.5):

$$\tau_{\text{tot}} = \pm\left(\left|\tau_{\text{res}}\right| + \left|\tau_{\text{dev}}\right|\right) \tag{23}$$

These measuring uncertainty ellipses are plotted around the measured points.

In the event that only one guarantee point and one test point are available, a relative total uncertainty of measuring results can be determined for the power or related power if this is converted to the guaranteed inlet volume flow and pressure ratio, Π_{g}. In this case, the equations for relative measuring uncertainties for related power apply approximately (see 6.4.4.2.4). This should be applied to the guarantee comparison in accordance with 8.2.2.

6.4.4.2 Relative uncertainty of measuring results calculated by differentiation

Formulae derived according to Annex D.

6.4.4.2.1 For the inlet volume flow $\tau_{\text{res},\dot{V}}$

The relative uncertainty of measuring results for inlet volume flow is

$$\tau_{\text{res},\dot{V}} = \pm\sqrt{\tau_{\dot{m}}^2 + \tau_N^2 + \tau_{p1}^2 + \tau_{T1}^2 + \tau_{Z1}^2} \tag{24}$$

6.4.4.2.2 For the pressure ratio, $\tau_{\text{res},\,\Pi}$

The relative uncertainty of measuring results for the pressure ratio is

$$\tau_{\text{res},\Pi} = \pm\frac{1}{{X_N}^2}\sqrt{\left(\ln\Pi\right)_v^2\left(4\cdot\tau_N^2 + \tau_{T1}^2 + \tau_R^2 + \tau_{Z1}^2\right) + \tau_{p1}^2 + \tau_{p2}^2} \tag{25}$$

6.4.4.2.3 For the specific polytropic compression work, $\tau_{res,y,p}$

The relative uncertainty of measuring results for the specific polytropic compression work is

$$\tau_{res,y,po} = \frac{V_{y,p}}{y_p} = \pm\sqrt{\left(\frac{1}{\ln\frac{p_2}{p_1}}\right)^2 \cdot \left(\tau_{p1}^2 - \tau_{p2}^2\right) + \left(\frac{T_2}{T_2-T_1} - \frac{1}{\ln\frac{T_2}{T_1}}\right)^2 \cdot \tau_{T2}^2 + \left(\frac{T_1}{T_2-T_1} - \frac{1}{\ln\frac{T_2}{T_1}}\right)^2 \cdot \tau_{T1}^2 + \tau_R^2 + \tau_{Z,m}^2} \quad (26)$$

6.4.4.2.4 For power, $\tau_{res,P,cou}$, related power, $\tau_{res(P,cou/\dot{V})}$, and efficiency, $\tau_{res,\eta,cou}$

The relative uncertainties of measuring results required for formulation of the equations for relative uncertainty of measuring results for power, related power and efficiency, and the factors by which these relative individual measuring uncertainties are multiplied, are compiled in Table 1.

Table 1 — Factors or relative measuring uncertainties for individual measured variables for determination of uncertainty of measuring results for determination of power, related power and efficiency in accordance with 6.4.4.2.4

Relative measuring uncertainty for individual measured values	Uncooled compressor			Cooled compressor $\vartheta_{j,te} = \vartheta_{j,g}$			Cooled compressor $\vartheta_{j,te} \neq \vartheta_{j,g}$		
	Case 1 [c]	Case 2 [d]	Case 3 [e]	Case 1 [c]	Case 2 [d]	Case 3 [e]	Case 1 [c]	Case 2 [d]	Case 3 [e]
$\tau_{P,cou}$	0	$\frac{1}{1+1/\varepsilon_2}$	0	0	$\frac{1}{1+1/\varepsilon_2}$	0	0	$\frac{1}{1+1/\varepsilon_2}$	0
τ_{Pi}	$\frac{P_{i,co}}{P_{cou,co}}$	0	0	$\frac{P_{i,co}}{P_{cou,co}}$	0	0	$\frac{P_{i,co}}{P_{cou,co}}$	0	0
$\tau_{P,mech}$	$\frac{P_{mech,co}}{P_{cou,co}}$	$\frac{1}{1+\varepsilon_2}$	$\frac{1}{1+\varepsilon_2}$	$\frac{P_{mech,co}}{P_{cou,co}}$	$\frac{1}{1+\varepsilon_2}$	$\frac{1}{1+\varepsilon_2}$	$\frac{P_{mech,co}}{P_{cou,co}}$	$\frac{1}{1+\varepsilon_2}$	$\frac{1}{1+\varepsilon_2}$
τ_{Mt}	0	0	$\frac{1}{1+1/\varepsilon_2}$	0	0	$\frac{1}{1+1/\varepsilon_2}$	0	0	$\frac{1}{1+1/\varepsilon_2}$
τ_N [a]	1	1	0	1	1	0	$1+2\varepsilon_3 \ln \Pi_{A,co}$		$2\varepsilon_3 \ln \Pi_{A,co}$
	0	0	1	0	0	1	$2\varepsilon_3 \ln \Pi_{A,co}$		$1+\ldots$ $\ldots 2\varepsilon_3 \ln \Pi_{A,co}$
$\tau_{\dot{m}}$ [a]	0								
	1								
τ_{p1} [a]	$1-\frac{1}{\ln \Pi_{te}}$								
	$\frac{1}{\ln \Pi_{te}}$								
τ_{T1} [a]	0						$\varepsilon_3 \ln \Pi_{A,co}$		
	1						$1+\varepsilon_3 \ln \Pi_{A,co}$		
τ_{Z1} [a]	0						$\varepsilon_3 \ln \Pi_{A,co}$		
	1						$1+\varepsilon_3 \ln \Pi_{A,co}$		

Table 1 *(continued)*

Relative measuring uncertainty for individual measured values	Uncooled compressor			Cooled compressor $\vartheta_{j,\mathrm{te}} = \vartheta_{j,\mathrm{g}}$			Cooled compressor $\vartheta_{j,\mathrm{te}} \neq \vartheta_{j,\mathrm{g}}$		
	Case 1 [c]	**Case 2** [d]	**Case 3** [e]	**Case 1** [c]	**Case 2** [d]	**Case 3** [e]	**Case 1** [c]	**Case 2** [d]	**Case 3** [e]
τ_R [a]	0						$\varepsilon_3 \ln \Pi_{\mathrm{A,co}}$		
	1						$1+\varepsilon_3 \ln \Pi_{\mathrm{A,co}}$		
τ_{p2}	$1/\ln \Pi_{\mathrm{te}}$								
τ_k	ε_1	ε_1	ε_1	0	0	0	0	0	0
$\tau_{T1,\mathrm{B}}$	0	0	0	0	0	0	ε_3	ε_3	ε_3
$\tau_{Z1,\mathrm{B}}$	0	0	0	0	0	0	ε_3	ε_3	ε_3
$\tau_{T1,j}$ [b]	0	0	0	$\frac{z-1}{z}$	$\frac{z-1}{z}$	$\frac{z-1}{z}$	$\frac{z-2}{z-1}$	$\frac{z-2}{z-1}$	$\frac{z-2}{z-1}$

[a] Here, the factors for determination of uncertainty of measuring results for power at coupling $\tau_{\mathrm{res},P\mathrm{cou}}$, not including τ_{dev}, are shown in the top line, and those for related power $\tau_{\mathrm{res}(P\mathrm{cou}/\dot{V})}$ and for efficiency $\tau_{\mathrm{res},\eta\mathrm{cou}}$, not including τ_{dev}, in the bottom line.

[b] For $\vartheta_{j,\mathrm{te}} = \vartheta_{j,\mathrm{g}}$ $T_{1j\mathrm{av}} = \frac{\sum_{j=\mathrm{II}}^{j=z} T_{1j}}{z-1}$, $\dot{V}_{tj\mathrm{av}} = \frac{\sum_{j=\mathrm{II}}^{j=z} V_t}{z-1}$ and $\tau_{T1j} = \frac{\dot{V}_{tj\mathrm{av}}}{T_{1j\mathrm{av}}}$

For $\vartheta_{j,\mathrm{te}} \neq \vartheta_{j,\mathrm{g}}$ $T_{1j\mathrm{av}} = \frac{\sum_{j=\mathrm{III}}^{j=z} T_{1j}}{z-2}$, $\dot{V}_{tj\mathrm{av}} = \frac{\sum_{j=\mathrm{III}}^{j=z} V_t}{z-2}$ and $\tau_{T1j} = \frac{\dot{V}_{tj\mathrm{av}}}{T_{1j\mathrm{av}}}$

[c] Case 1: Power at the coupling, $P_{\mathrm{cou,te}}$, is determined by means of measurement of gas power $P_{\mathrm{i,te}}$ and mechanical losses $P_{\mathrm{mech,te}}$.

[d] Case 2: Power at the coupling is measured on the driving machine.

[e] Case 3: Power at the coupling is determined by means of measurement of torque $M_{\mathrm{t,te}}$ and rotational speed N_{te}.

Coefficients:

$$\varepsilon_1 = \frac{1}{1-k_{\mathrm{te}}} + \frac{1}{k_{\mathrm{te}}} \left(\frac{\ln \Pi_{\mathrm{te}}}{1-\Pi_{\mathrm{te}}^{\left(\frac{1-k}{k}\right)_{\mathrm{te}}}} \right) \quad (27)$$

$$\varepsilon_2 = \frac{P_{\mathrm{cou,te}}}{P_{\mathrm{mech,co}} \frac{P_{\mathrm{i,te}}}{P_{\mathrm{i,co}}} - P_{\mathrm{mech,te}}} \quad (28)$$

$$\varepsilon_3 = \frac{P_{\mathrm{iB,co}}}{P_{\mathrm{i,co}}} \quad (29)$$

EXAMPLE The equation for relative uncertainty of measured results for power at the coupling of a cooled compressor ($\vartheta_{j,te} = \vartheta_{j,g}$) can be formulated as follows in accordance with Table 2 if power at the coupling in the test $P_{cou,te}$ has been determined by means of measurement of torque and rotational speed (case 3):

$$\tau_{res,(P cou)} = \pm \left\{ \left(\frac{1}{1+\varepsilon_2} \tau_{P mech} \right)^2 + \left(\frac{1}{1+\frac{1}{\varepsilon_2}} \tau_{M t} \right)^2 + \left(2 \cdot \varepsilon_3 \cdot \ln \Pi_{A,co} \cdot \tau_N \right)^2 + \left[\left(1 - \frac{1}{\ln \Pi_{te}} \right) \tau_{p1} \right]^2 + \ldots \right.$$

$$\ldots + \left(\varepsilon_3 \cdot \ln \Pi_{A,co} \cdot \tau_{T1} \right)^2 + \left(\varepsilon_3 \cdot \ln \Pi_{A,co} \cdot \tau_{Z1} \right)^2 + \left(\varepsilon_3 \cdot \ln \Pi_{A,co} \cdot \tau_R \right)^2 + \ldots$$

$$\left. \ldots + \left(\frac{1}{\ln \Pi_{te}} \tau_{p2} \right)^2 + \left(\varepsilon_3 \cdot \tau_{T_{1,B}} \right)^2 + \left(\varepsilon_3 \cdot \tau_{Z_{1,B}} \right)^2 + \left(\frac{z-2}{z-1} \tau_{T_{1,j}} \right)^2 \right\}^{1/2} \tag{30}$$

6.4.4.3 Determination of measuring uncertainty using the difference method

The uncertainty of measuring results $\tau_{res,W}$ of a result function, W, (e.g. steam consumption) can be determined as follows, particularly in the case of complicated functional interactions whose derivation by means of closed mathematical solutions is difficult.

The result function, W, is based on all measured values, converted to the guarantee preconditions and corrected to the guarantee value.

For example, for $W = P_{cou,g}$, with $\tau_{res,W}$ expressed in percent:

$$W = P_{i,co} \cdot \frac{y_{T,g}}{y_{T,co}} \cdot \frac{V_{1,g}}{V_{1,co}} + P_{mech,co,g} \tag{31}$$

$$\tau_{res,W} = \frac{V_W}{W} \cdot 100 = \pm \sqrt{\sum \left(\frac{\partial W}{\partial x_i} \cdot \frac{V_{xi}}{W} \right)^2} \cdot 100$$

$$= \pm \sqrt{\sum f_{xi}^2} \cdot 100 \tag{32}$$

$$f_{xi} = \frac{W(x_i + V_{xi}) - W(x_i - V_{xi})}{2 \cdot W(x_i)} \tag{33}$$

For this purpose, the result function, W, for all measured variables and gas data, x_i, contained therein is calculated using the values increased or decreased by the individual measuring uncertainty V_{xi}, and the mean relative deviation f_{xi} of the result function, W, is calculated from their difference at the position of the measured value x_i.

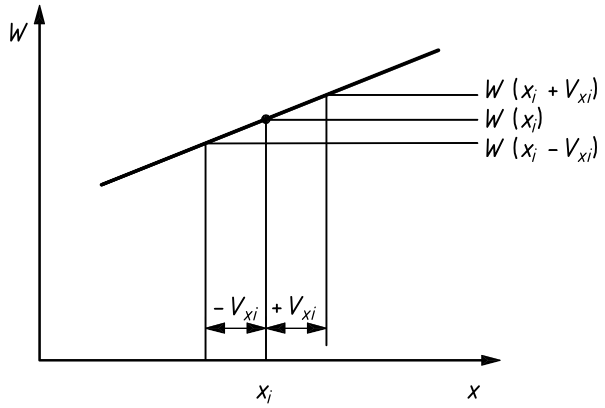


Figure 1 — Variation of a result function versus an individual measuring uncertainty

6.4.4.4 Relative uncertainty of measuring results for single-stage measurements on multi-stage compressors

The following equations apply in the case of the composition of the overall performance curve from the curves of single stages or sections measured separately:

for the relative measuring uncertainty for volume flow:

$$\tau_{\text{comb},\dot{V}} = \left(1 + 0{,}2 \cdot \sqrt{z-1}\right) \frac{\sum \tau_{\dot{V},j}}{z} \tag{34}$$

for the relative measuring uncertainty for the pressure ratio:

$$\tau_{\text{comb},\Pi} = \left(1 + 0{,}2 \cdot \sqrt{z-1}\right) \cdot \frac{\sum\left(\tau_{\Pi,j} \cdot W_{\text{co},j}\right)}{\sum W_{\text{co},j}} \tag{35}$$

for the relative measuring uncertainty for power:

$$\tau_{\text{comb},P} = \left(1 + 0{,}2 \cdot \sqrt{z-1}\right) \cdot \frac{\sum\left(\tau_{P,j} \cdot P_{\text{co},j}\right)}{\sum P_{\text{co},j}} \tag{36}$$

The factors 0,2 and $z - 1$ make allowance for the unavoidable inaccuracies of separate measurement of the individual sections and in compilation of the results.

6.4.4.5 Weighted relative uncertainty of measuring results

If different measuring methods have been used, weighted measuring values and uncertainties can be obtained following Reference [7]:

Weighted measuring result:

$$\bar{W}_{\gamma} = \frac{\sum\left(W_i \cdot \gamma_i\right)}{\sum \gamma_i} \tag{37}$$

with

$$\gamma_i = \left(\frac{1}{V_{Wi}}\right)^2 \quad (38)$$

Weighted uncertainty of measuring results:

$$V_{\bar{W}y} = \frac{1}{\sqrt{\sum \gamma_i}} = \frac{1}{\sqrt{\sum \left(\frac{1}{V_{Wi}}\right)^2}} \quad (39)$$

Weighted relative uncertainty of measuring results:

$$\tau_{\bar{W}y} = \frac{V_{\bar{W}y}}{\bar{W}_\gamma} \quad (40)$$

7 Conversion of test results to guarantee conditions

7.1 General

7.1.1 Purpose of conversion

The test results can be directly compared with the guarantee values only if the compressor is measured precisely under the guarantee operating conditions during the acceptance test.

If the operating conditions during the test deviate from those specified in the guarantee, the test results shall be converted to the guarantee operating conditions. Only such converted test data may be compared with the guarantee values in the guarantee comparison as specified in Clause 8.

7.1.2 Object of conversion

The values converted are essentially the following:

— effective inlet volume flow, $\dot{V}_{1,\text{us}}$;

— pressure ratio, Π, or head;

— and power at the coupling, P_{cou}.

Power at the coupling P_{cou} is composed additively of gas power P_{i} and mechanical loss P_{mech}, which are converted separately. Allowance shall be made, if necessary, for the influence of leakage flows.

7.2 Conversion

7.2.1 Adherence to the requirements deriving from similarity theory

Conversion of test results from test conditions to the guarantee conditions is generally possible if similarity of the flow in the compressor is ensured during conversion of a test point to guarantee conditions, i.e., provided the essential conditions can be maintained for identical reference process work coefficients, see Equation (6), and for identical flow coefficients, see Equation (5).

Where variable geometry systems for control of flow in the compressor (e.g. adjustable inlet guide vanes or diffuser vanes) are installed, the conversion will apply only to one constant setting of such systems. These similarity conditions relate only to the flow in the compressor, and not to mechanical losses. For this reason, these shall be separately measured and converted for the guarantee comparison (see 7.2.4.4).

a) Identical reference process work coefficients and volume flow coefficients

Under identical reference process work coefficients and flow coefficients, ψ and φ, the ratio of a characteristic flow velocity in the compressor to tip speed has an identical value under test and under guarantee conditions. For this reason, it is necessary, but not sufficient, that φ and ψ or $\dot{V}_1/N$ or Y/N^2 be kept constant for conversion of the test point.

b) Identical isentropic exponents

The change of state of the compressed gas can be kept identical under test and guarantee conditions in all stages of the compressor only if the isentropic exponents are identical.

c) Identical Mach numbers

In order for the velocity ratios to be identical in a gas at every flow path location, the condition shall be imposed, in addition to the requirement for identical reference process work and flow coefficients, that the volume flow ratios (volume flow referred to inlet volume flow) remain constant at every flow path location under test and guarantee conditions. The requirement for identical volume flow ratios in all stages of the compressor is met — always assuming identical isentropic exponents — if the tip Mach numbers Ma_u are identical under test and guarantee conditions. Under these preconditions, identical tip Mach number signifies simultaneously identical local Mach number (flow velocity referred to the respective local sonic velocity).

d) Identical ϑ_j ratios in the individual stages

The similarity condition of identical ϑ_j ratios in the individual stages signifies that the values $\vartheta_j = (RZ_1T_1)_j/(RZ_1T_1)_\text{I}$ are constant (j = I, II, ...). In uncooled compressors, this requirement is fulfilled anyway given identical isentropic exponents and identical tip Mach number.

In cooled compressors, the condition ϑ_j = const shall be achieved by means of corresponding adjustment of the intercooler.

The cooler performance cannot be assessed if the test values diverge from the guarantee conditions. If necessary, performance shall be tested separately.

e) Identical Reynolds numbers

In order that the boundary layer of the flow, and thus also the flow pattern influenced by it, remain constant, the Reynolds number, as well as the parameters already mentioned, shall also remain constant in the conversion calculation.

f) Identical heat exchange characteristic

In cases where heat exchange has an influence on the compression process, the corresponding characteristic variables shall also remain identical.

7.2.2 Approximations to the requirements deriving from similarity theory

7.2.2.1 General

Since it is, in general, not possible to fulfil all similarity conditions simultaneously, it will be necessary to neglect individual conditions to a greater or minor extent; see Reference [10].

In compressors operating at flow velocities in the sonic velocity range, it is necessary to check whether the deviations in Mach numbers during the test from those in the guarantee conditions are within the permissible range. In this context, however, it is not the tip Mach numbers Ma_u for which allowance shall be made, but instead the local Mach numbers (ratio of local flow velocity to local sonic velocity).

7.2.2.2 Negligence of influence of certain characteristic variables

If it is not possible to satisfy all the conditions required for the characteristic variables simultaneously, it is necessary to dispense with the equality of those characteristic variables which generally have only secondary influence on efficiency and are of significance only in boundary areas, once their magnitude has been checked under test and guarantee conditions.

As heat exchange generally has only a slight influence on the compression process in uncooled compressor stages, the corresponding characteristic variables for heat transfer play a role only in case of extreme deviations in test conditions. Intercoolers are not considered in this context.

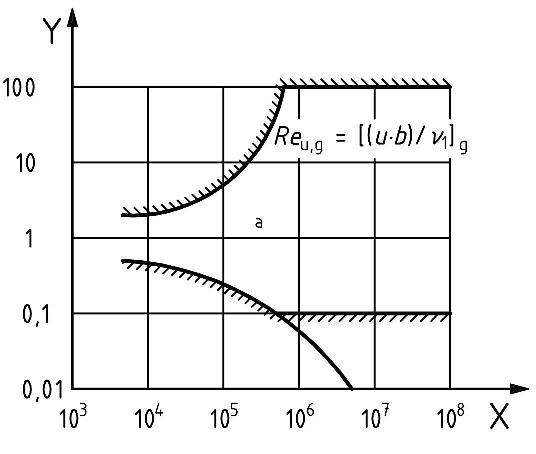
7.2.2.3 Permissible deviations of characteristic variables whose Influence cannot be neglected

7.2.2.3.1 Preconditions

Similarity in the entire compressor is ensured provided ψ_p and φ of each stage, Ma_u, k, ϑ_j and if necessary Re remain identical.

Approximations may be appropriate in cases where the tip speed Mach number or the isentropic exponent k, or both variables, in test conditions cannot be precisely adopted to the guarantee conditions in uncooled compressors; and in cooled compressors if, in addition to the two conditions mentioned above, the temperature ratios cannot be maintained either.

Deviations of the Reynolds number are permissible up to certain limits (see Figure 2). If the Reynolds number in the test is lower than the limits of Figure 2, only a Reynolds number correction, in accordance with Annex C, within the limits of Figure 2 shall be made. The influence of the Reynolds number on efficiency, specific compression work and flow coefficient shall be taken into account when determining the test conditions (see 7.2.2.3.2) and converting test results to guarantee conditions (see 7.2.4.1).



Key

X guarantee Reynolds number, $Re_{u,g}$

Y Reynolds number ratio, $\frac{Re_{u,te}}{Re_{u,g}}$

a Permissible range of application.

Figure 2 — Permissible range of application for conversion

The approximation procedures start from the precondition that at $\varphi_{te} = \varphi_g$ or $\left(\dot{V}_1 / N\right)_{te} = \left(\dot{V}_1 / N\right)_g$ the velocity ratios, and thus the volume flow ratios in the compressor during the test, can deviate up to a certain percentage from the ratios under guarantee conditions without a substantial effect on efficiency and specific work of compression.

7.2.2.3.2 Permissible deviation of volume flow ratio, ϕ

The greatest deviations in volume flow always occur for $n_{te} = n_g$ at the end of the compression process.

For $n_{te} \neq n_g$, the condition of identical volume flow ratios throughout the compression process is achievable only approximately, since the maximum volume flow deviations can occur within the compressor as a result of a divergent pattern in change of state.

The inner tolerance limit for deviation of the ratio of volume flow ratios ϕ is $\Delta\phi_{tol} = \pm$ 0,01. It shall be checked if this limit can be maintained by means of variation of test values N_{te}, R_{te}, $Z_{1,te}$ or $T_{1,te}$. In such cases, the test should be performed without the use of a supplementary tolerance. Otherwise, it has to be checked whether the test can still be performed within the outer tolerance limit $\Delta\phi_{tol}$ = 0,05 (see 7.2.5). In this case, the test can be performed with approximated similarity using a supplementary tolerance (Figures 6 and 7).

The permissible ratio of reduced speeds $X_{N,tol}$, Equation (2), can be calculated using the permissible deviation $\Delta\phi_{tol}$ of the ratio of volume flow ratios, ϕ, Equation (1); see Annex A.

If the outer tolerance limits do not suffice, it shall be checked on a case-to-case basis if tests may still be performed according to the method described in Annex B.

If the check of the test values indicates that the values $\eta_{p,te}$ and $(p_2/p_1)_{te}$ deviate from the values $\eta_{p,pr}$ and $(p_2/p_1)_{pr}$ predicted for the test conditions, the check for similarity conditions should be repeated using these values.

7.2.2.3.3 Permissible deviations of tip Mach Number

The permissible deviations of the volume flow ratio themselves include a limitation on the permissible deviations of tip Mach Number.

$$\frac{Ma_{u,te}}{Ma_{u,g}} = X_N \cdot \sqrt{\frac{k_{1,g}}{k_{1,te}}} \tag{41}$$

Mach number effects shall be considered if the critical relative Mach number Ma_{crit} (localy sonic velocity in the stage) is reached under guarantee or test conditions for the test point and the converted point.

7.2.2.3.4 Permissible deviations of the Reynolds number

It is necessary to check the deviation of the test Reynolds number from that of the guarantee conditions. Allowance is made for the influence of this deviation on operation of compressors by means of suitable correction provisions, which, however, may be used only in specific cases.

The cases for application of the correction equations and the selection of a suitable test Reynolds number are determined by two factors:

— accuracy of the correction equation for various Reynolds numbers;

— accuracy of the test results achieved at reduced inlet pressures or lower speeds.

In the case of centrifugal compressors, the well-proven method for correction of the Reynolds number (see Annex C) shall be used. The limits of application of the equations are shown in Figure 2.

In the case of axial compressors, the correct method for Reynolds number correction depends on the blade characteristics used by the compressor manufacturer. For this reason, the method and the application ranges should be agreed between the manufacturer and the user.

7.2.3 Subdivision of conversion cases

7.2.3.1 Classes of conversion

According to the test conditions, the following classes derive for adjustment, test and conversion:

— Class A: Test, maintaining the inner tolerance limit $\Delta\phi_{tol} = \pm 0{,}01$. If this is not possible:

— Class B: Test, maintaining the outer tolerance limit $\Delta\phi_{tol} = \pm 0{,}05$. If this is not possible:

— Class C: Test beyond the outer tolerance limit.

7.2.3.2 Conversion according to classes A and B

These cases are shown schematically in Tables 2 and 3 and in Annex A.

7.2.3.3 Conversion according to class C

This case is described in Annex B.

Table 2 — Adjustment, test, conversion: uncooled compressor

	$n_{te} = n_g$		$n_{te} \neq n_g$	
Case	**3a**	**3b**	**3c**	**3d**
Example			**Annex F, Example 1**	**Annex F, Example 5**
Ratio of reduced speeds, see 7.2.2.3.2	Set compressor at $X_{N,tol} = \sqrt{\psi_{p,g}/\psi_{p,te}}$. Deviations within tolerance limit $\Delta\phi_{tol} = \pm 0{,}01$ as per Annex A are permissible. No supplementary tolerance for conversion.	If adjacent condition cannot be met, set compressor at $X_{N,tol}$ within outer tolerance limit $\Delta\phi_{tol} = \pm 0{,}05$ as per Annex A. Supplementary tolerance for conversion as per 7.2.5.	Set compressor with X_N to be within inner tolerance limit $\Delta\phi_{tol} = \pm 0{,}01$ as per Annex A. No supplementary tolerance for conversion.	If adjacent condition cannot be met, set compressor with X_N to be within outer tolerance limit $\Delta\phi_{tol} = \pm 0{,}05$ as per Annex A. Supplementary tolerance for conversion as per 7.2.5.
Mach number: see 7.2.2.3.3	If $Ma_{u,te} \neq Ma_{u,g}$, check whether changes caused by Mach number occur in the performance curve range relevant for the guarantee comparison (critical Mach number, choke Mach number).			
Reynolds number: see 7.2.2.3.4	Check whether $Re_{u,te}/Re_{u,g}$ is within the range permissible for conversion of efficiency (in accordance with Figure 2 for centrifugal compressors).			
Performance curve	Operation of one point in the vicinity of the guarantee point or not less than two test points which enclose the guarantee value for specific work of compression or for inlet volume flow (depending on guarantee comparison).			
Conversion	7.2.4.1, Figure 3.			
Converted values	Check whether the similarity conditions have been met in the test.			
Guarantee comparison	Clause 8.			

Table 3 — Adjustment, test, conversion: intercooled compressor

<table>
<tr><td></td><td colspan="2">$n_{te} = n_g$ and $Re_{u,te} \approx Re_{u,g}$</td><td colspan="2">Other application cases, for instance:
a) $n_{te} = n_g$ and $Re_{u,te} \neq Re_{u,g}$
b) $n_{te} \neq n_g$
c) Inward sidestreams or extractions</td></tr>
<tr><th>Case</th><th>4a</th><th>4b</th><th>4c</th><th>4d</th></tr>
<tr><th>Example</th><th>Annex F, Example 4</th><th>Annex F, Example 3</th><th></th><th>Annex F, Example 2</th></tr>
<tr><td>Ratio of reduced speeds: See 7.2.2.3.2</td><td>Set first section at $X_{N,tol} = 1$.
Deviations within inner tolerance limit $\Delta\phi_{tol} = \pm 0{,}01$ as per Annex A are permissible.
No supplementary tolerance for conversion.</td><td>Set the individual uncooled sections with X_N to be within inner tolerance limit $\Delta\phi_{tol} = \pm 0{,}01$ as per Annex A.
No supplementary tolerance for conversion.
If the above condition cannot be met, set first section at X_N within outer tolerance limit $\Delta\phi_{tol} = \pm 0{,}05$ as per Annex A.
Supplementary tolerance for first section for conversion as per 7.2.5.</td><td>Set the individual uncooled sections with X_N to be within inner tolerance limit $\Delta\phi_{tol} = \pm 0{,}01$ as per Annex A.
No supplementary tolerance for conversion for these sections.</td><td>If the adjacent condition cannot be met for all uncooled sections, set the relevant sections at X_N within the outer tolerance limit $\Delta\phi_{tol} \pm 0{,}05$ as per Annex A.
Supplementary tolerance for conversion as per 7.2.5 for the relevant sections.</td></tr>
<tr><td>RZ_1T_1, ratio as defined in 3.2.8</td><td>Set stage inlet temperatures in such a way that
$\frac{\vartheta_{j,te}}{\vartheta_{j,g}} = 1$</td><td>If the adjacent condition cannot be met, subdivide into uncooled first section and intercooled downstream section at
$\frac{\vartheta_{j,B,te}}{\vartheta_{j,B,g}} = 1$</td><td colspan="2">Wherever possible, set the stage inlet temperatures in such a way that the test can be run at an identical speed for all sections.</td></tr>
<tr><td>Mach numbers: see 7.2.2.3.3</td><td colspan="4">If $Ma_{u,te} \neq Ma_{u,g}$, check whether changes due to Mach number occur in the characteristic curve range relevant for the guarantee comparison (critical Mach number, choke Mach number).</td></tr>
<tr><td>Reynolds number: see 7.2.2.3.4</td><td>Overall conversion via specific isothermal work of compression and isothermal total efficiency only possible if $Re_{u,te} \approx Re_{u,g}$.
(No change in polytropic stage efficiency.)</td><td colspan="3">Check that $Re_{u,te}/Re_{u,g}$ is in the permissible range for conversion of efficiency (in accordance with Figure 2 for centrifugal compressors).</td></tr>
</table>

Table 3 *(continued)*

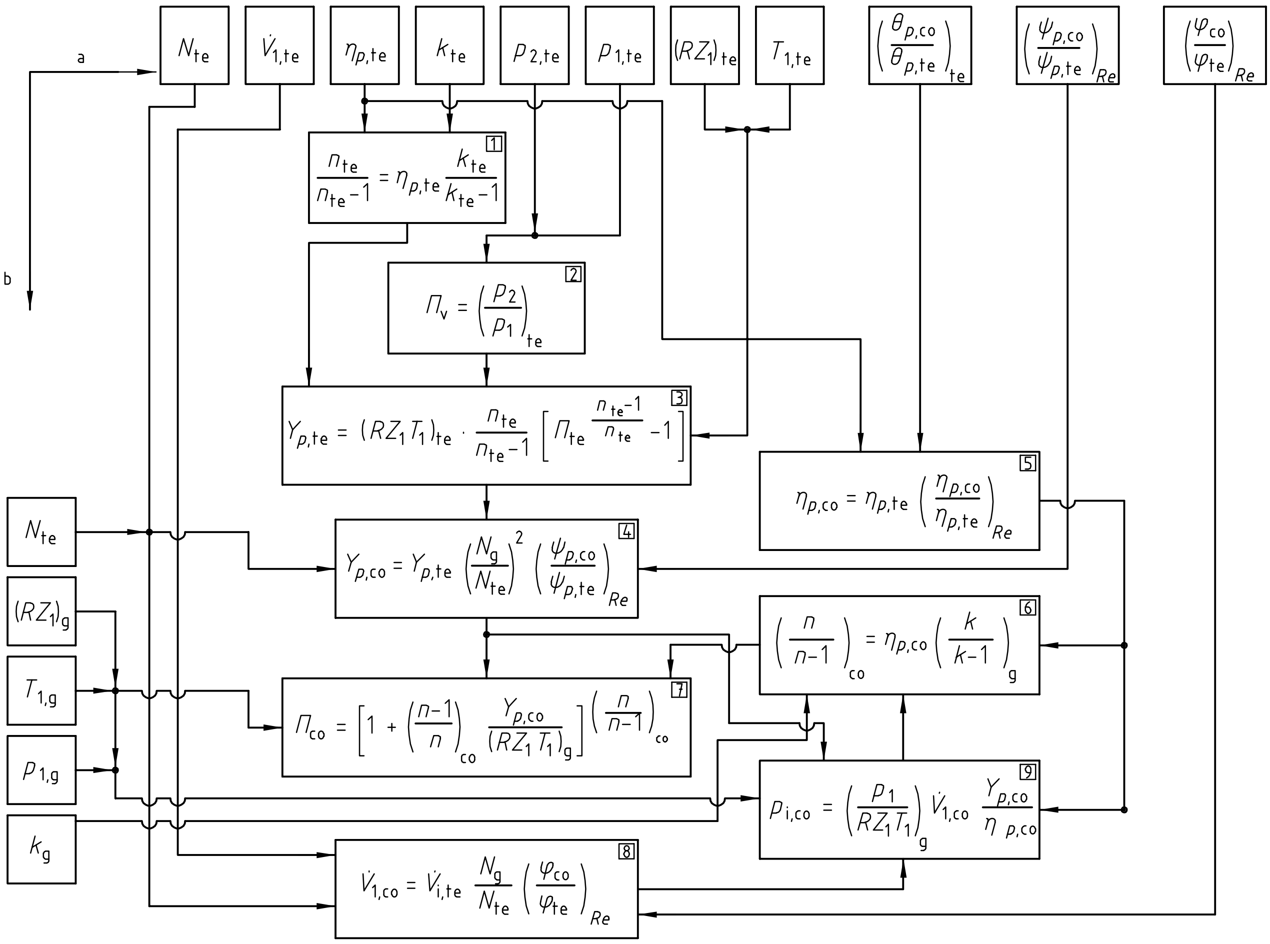
	$n_{te} = n_g$ and $Re_{u,te} \approx Re_{u,g}$		Other application cases, for instance: a) $n_{te} = n_g$ and $Re_{u,te} \neq Re_{u,g}$ b) $n_{te} \neq n_g$ c) Inward sidestreams or extractions	
Case	**4a**	**4b**	**4c**	**4d**
Example	**Annex F, Example 4**	**Annex F, Example 3**		**Annex F, Example 2**
Performance curve	As for uncooled compressors in Table 2.	First uncooled section as the intercooled high-pressure section indicated in Table 2 with adequate number of characteristic curve points for superpositioning.	First uncooled section as in Table 2, all other sections with adequate number of performance curve points for superpositioning.	
Conversion	7.2.4.2.1, Figure 4	7.2.4.2.1, Figure 5.	As per Table 2 for each uncooled section, then superpositioning with recooling temperatures and pressure losses, and mass flow ratios as per guarantee conditions if necessary, Section 7.2.4.2.2.	
Converted values	Check whether similarity conditions were met in test.			
Guarantee comparison	Clause 8.			

7.2.4 Conversion equations

7.2.4.1 Conversion for uncooled compressors or sections

Provided the conditions specified in 7.2.1 and 7.2.2 are met and the gas behaves approximately perfect, the test values can be converted to guarantee conditions using the procedure shown in Figure 3.

For real gas behaviour, the variables [1], [3], [6] and [7] in Figure 3 should be calculated from the temperatures and pressures measured, either using the compressibility functions; see Equations (E.22) and (E.23), or using gas data programs. At small pressure ratios, calculation can also be effected by means of the isentropic change of state; see Equation (E.74).



a Test values.

b Guarantee conditions.

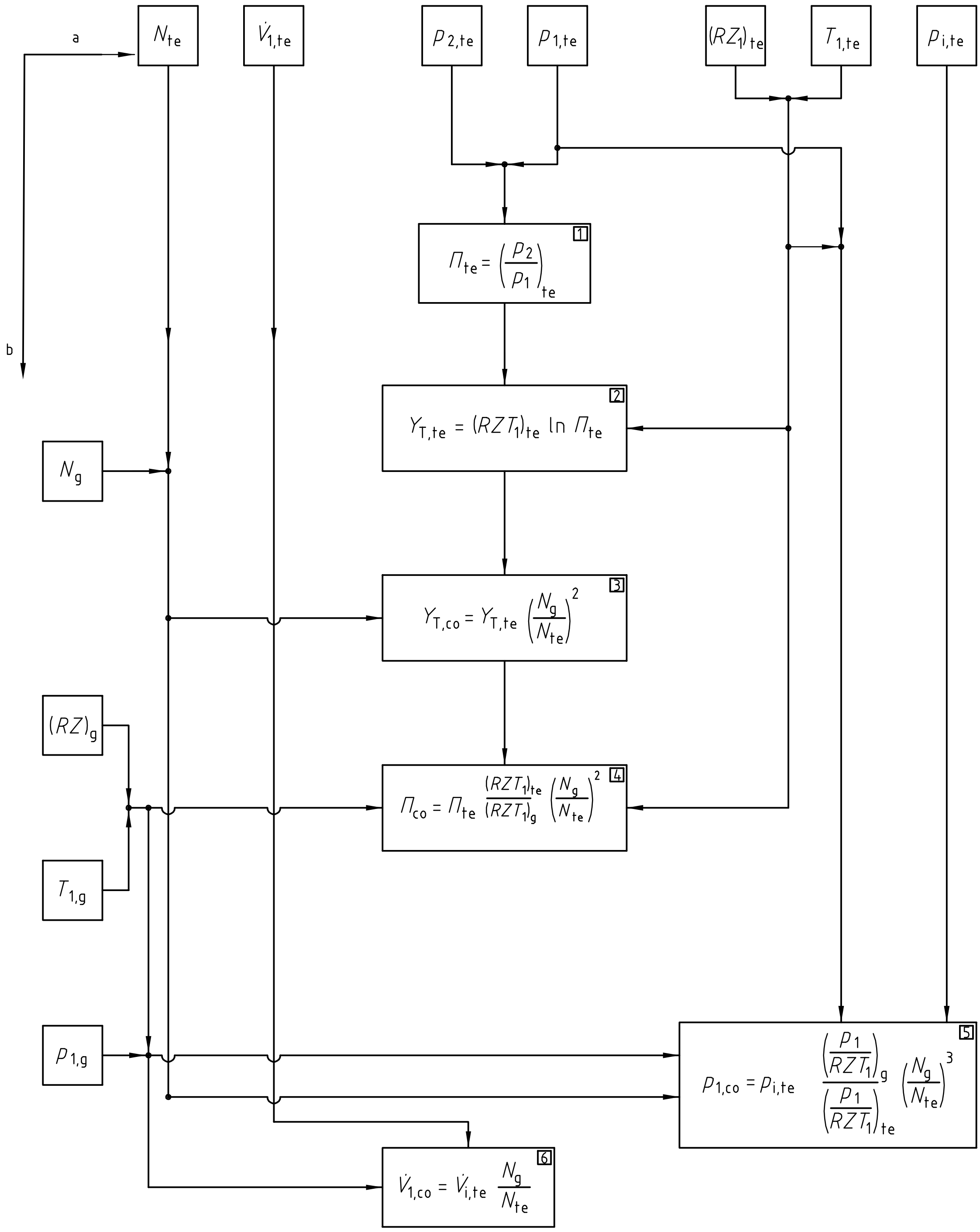
Figure 3 — Calculation for uncooled compressors or sections with approximately perfect gas behaviour

7.2.4.2 Conversion for cooled compressors

7.2.4.2.1 Overall conversion

Provided the conditions specified in 7.2.1 and 7.2.2 are fulfilled and that the test conditions achieved mean that there is no need to make allowance for the influence of the Reynolds number, the test results can be converted to the guarantee conditions using the procedure shown in Figures 4 and 5.

Where $\vartheta_{j,\mathrm{te}} = \vartheta_{j,\mathrm{g}}$, conversion can be carried out as shown in Figure 4.



[a] Test values.

[b] Guarantee conditions.

Figure 4 — Overall conversion for cooled compressors where $\vartheta_{j,te} = \vartheta_{j,g}$, $n_{te} = n_g$ and with approximately perfect gas behaviour

Where $\vartheta_{j,te} \neq \vartheta_{j,g}$, the test results can be converted in accordance with Figure 5 It is presupposed here only that, where multiple intercooling is installed, the products RZ_1T_1 downstream of the intercoolers have the same ratio to one another in the test as in the guarantee conditions.

The measured gas power of the compressor shall be subdivided into a section A for the uncooled section and a section B for the cooled section. It will, in general, be possible to perform this subdivision during the test.

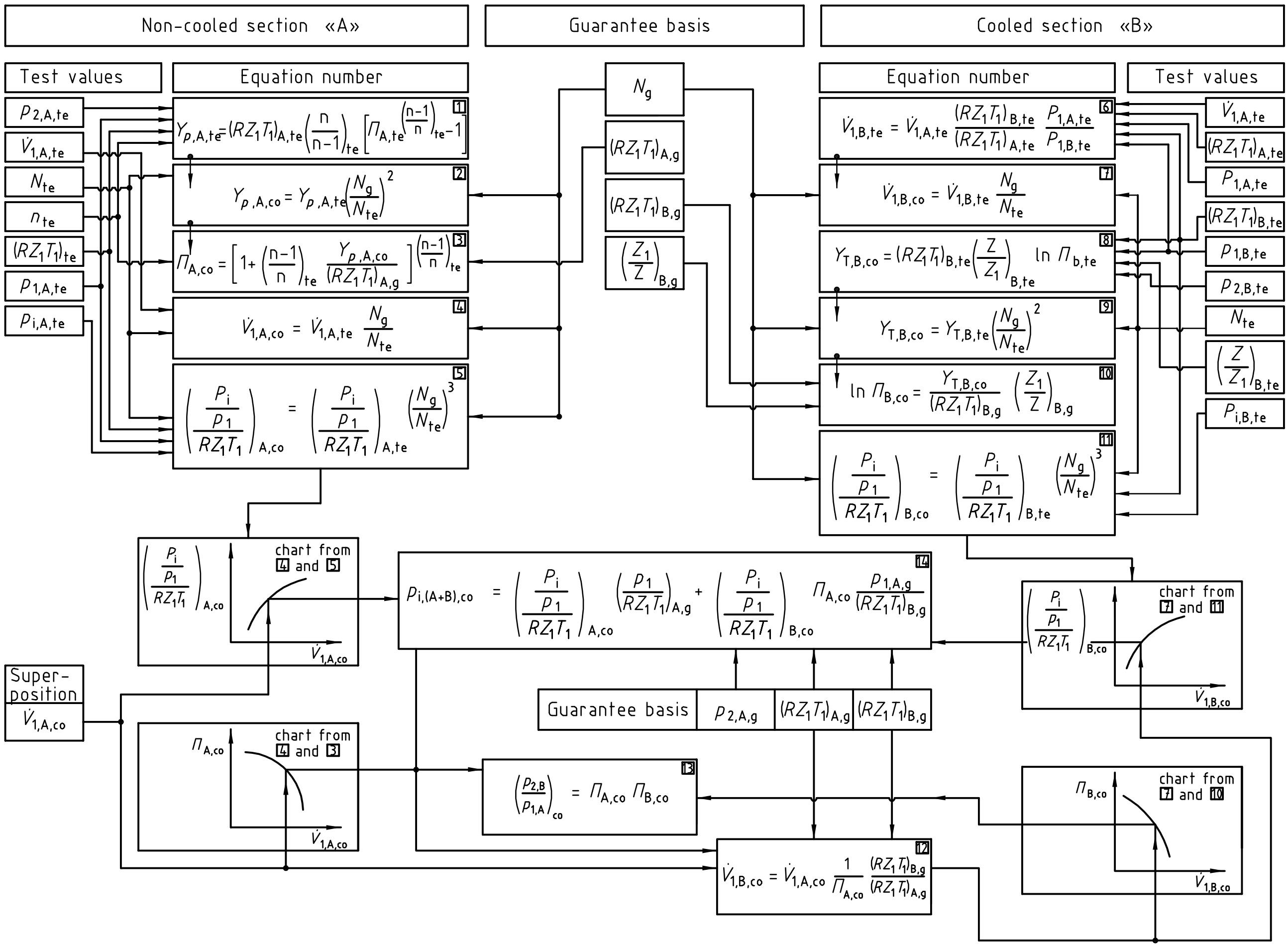


Figure 5 — Conversion for cooled compressors where $\vartheta_{j,te} \neq \vartheta_{j,g}$, but $\vartheta_{jB,te} = \vartheta_{jB,g}$, $n_{te} = n_g$ and approximately with perfect gas behaviour

Where this is not possible, the subdivision can be performed proportional to the specific design work of compression.

These power portions shall then be converted to the guarantee conditions using the procedure shown in Figure 5.

7.2.4.2.2 Conversion by means of uncooled sections

Where the conditions for an overall conversion are not met (e.g. greater Reynolds number deviations, different isentropic exponent, intercooler operating conditions diverging from guarantee conditions), the conversion shall be effected by means of superpositioning of the section performance curves converted in accordance with 7.2.4.1.

7.2.4.3 Provision for leakage flows

Allowance shall be made in the conversion for changes in leakage flows where the test conditions in the test diverge significantly from the guarantee conditions.

7.2.4.4 Conversion of mechanical power loss

Mechanical power loss, P_{mech}, is the sum of all individual mechanical losses. These depend on the respective test and guarantee conditions, and on speed, power consumption, axial thrust, and the viscosity and temperature of the lubricant, in particular.

Individual mechanical losses occur in axial and radial bearings, lubricant pumps, compressor gear boxes, liquid and gas operated shaft seals, mechanical contact seals, etc., in particular.

The losses are normally measured on the basis of oil-temperature rise and/or calculated from design dimensions and the test data.

The sum of the converted individual losses is the mechanical power loss, $P_{\text{mech,co}}$, from which $P_{\text{cou,co}}$ can now be calculated:

$$P_{\text{cou,co}} = P_{\text{i,co}} + P_{\text{mech,co}} \tag{42}$$

The power loss due to bearing friction can be determined for plain bearings.

The influence of speed on the mechanical losses can be estimated by the equation:

$$P_{\text{mech,co}} = P_{\text{mech,te}} \left(\frac{N_{\text{g}}}{N_{\text{te}}} \right)^b \tag{43}$$

where b = 1,5 to 2,0

7.2.4.5 Correction of gas power due to radiation

If not directly added to measured gas power according to 5.9, the heat transferred to the ambient air during the test in case of an energy balance is converted to guarantee preconditions by the equation

$$\dot{Q}_{\text{rad,co}} = \dot{Q}_{\text{rad,te}} \cdot \frac{P_{\text{i},\Delta t,\text{co}}}{P_{\text{i},\Delta t,\text{te}}} \tag{44}$$

and added to the gas power, $P_{\text{i},\Delta h,\text{co}}$, measured over mass flow and enthalpy rise and converted to guarantee preconditions

$$P_{\text{i,co}} = P_{\text{i},\Delta t,\text{co}} + \dot{Q}_{\text{rad,co}} \tag{45}$$

The converted discharge temperature $t_{2,\text{co}}$ then can be corrected by

$$t_{2,\text{co}} = t_{1,\text{g}} + \left(t_{2,\text{co},\Delta t} - t_{1,\text{g}} \right) \cdot \frac{P_{\text{i,co}}}{P_{\text{i},\Delta t,\text{co}}} \tag{46}$$

$\dot{Q}_{\text{rad,co}}$ is not the heat transferred to ambient air by radiation at guarantee preconditions.

7.2.5 Supplementary tolerances

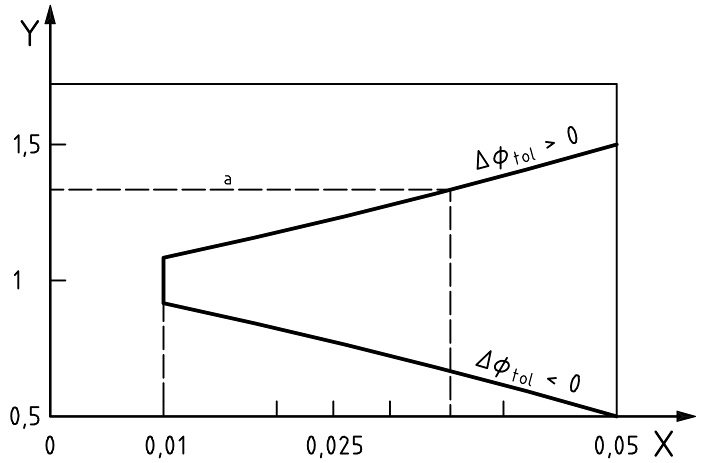
Where it has not been possible to meet the inner tolerance limit during the test, a supplementary tolerance for power and specific compression work shall be determined as follows:

For the pressure ratio, Π, and the polytropic exponents, n_{te}, and n_{g}, the permissible upper and lower limit values of $X_{N,\text{tol}}$ with the parameter $\Delta\phi_{\text{tol}}$ shall be calculated according to the flow diagram in Annex A or

$$X_{N,\text{tol}}\sqrt{\frac{\psi_{\text{p,te}}}{\psi_{\text{p,g}}}} \tag{47}$$

at the inner tolerance limit and at the outer tolerance limits shall be taken from the relevant figures and $X_{N,\text{tol}}$ shall be calculated with this value.

Figure 6 shows an example of such a diagram, which shall be drafted newly in each case.



Key

X deviation, $|\Delta\phi|$, of the ratio of the volume flow ratios

Y X_N

[a] Achievable in the test.

Figure 6 — Procedure to determine the deviation of the ratio, ϕ, of volume flow ratios for determination of supplementary tolerance

This figure supplies the corresponding value for $\Delta\phi$ for the test value X_N.

The supplementary tolerance τ_{dev} can then be calculated:

If	$\lvert\Delta\phi\rvert < 0{,}01$	$\tau_{\text{dev}} = 0$
If	$0{,}01 < \lvert\Delta\phi\rvert < 0{,}05$	$\tau_{\text{dev}} = 25\ (\lvert\Delta\phi\rvert - 0{,}01)$ in %
If	$\lvert\Delta\phi\rvert > 0{,}05$ (Test class C)	$\tau_{\text{dev}} = 1{,}0\ \%$

7.2.6 Special notes

Where a portion of the gas condensates, e.g. in the intercoolers in the test and/or guarantee case, the condensated amount shall be taken into account in power in accordance with the compression work necessary for it, and in the volume flow. It shall be borne in mind that the amount actually condensated up to stage "j" of the compressor is, in general, smaller than its thermodynamically calculated quantity (separation efficiency < 1). Where it is necessary to convert from a test state at a specified moisture content to a guarantee state of a different moisture content, the separation efficiency of the respective coolers shall be assumed constant in the test and guarantee conditions by way of approximation. The measured amounts of condensated water in the test shall be converted at the ratio of the amounts which would result in each case at a separation efficiency of 1. The power for each stage shall be corrected by the amount by which the gas mass flow differs in the test and guarantee conditions due to differing amounts of water condensation. Precise conversion, on the other hand, is possible from a test with moist gas to the guarantee state (dry gas) by adding to the power for each stage the extra energy required for compression of the gas which, in the guarantee case, would remain in place of the fluid condensated out during the test.

Furthermore, during the compression process or in the compression system and its measuring points, chemical reactions which modify gas contents, volumes and temperatures, in particular, can occur.

Where a compressor features inward sidestreams and/or extractions, the volume sidestream flows or extraction flows under test and guarantee conditions shall be harmonized in proportion to the main flow. Here, conversion shall be performed on the basis of the mixture states.

Where the compressor is operated under test and guarantee conditions at differing pressure levels, leakage losses shall be considered.

Where a compressor consists of several casings or where its design makes it possible to remove intercoolers and install measuring lines in their place, the compressor can be subdivided into separate compressor units for test purposes.

The acceptance test does not necessarily furnish proof that the intercoolers fulfil the guarantee conditions with regard to re-cooling temperature, pressure loss, coolant flow, etc.

8 Guarantee comparison

8.1 Object

The guarantee comparison is comprised of the following:

a) verification of the guaranteed absolute and/or related values for power or fluid consumption, and/or for the efficiency of the compressor under guarantee conditions (see 4.3);

b) verification of the guaranteed upper limit of the operating range of the compressor under guarantee conditions and, possibly, also of the lower limit of the operating range and the corresponding efficiencies under guarantee conditions.

8.2 Execution

8.2.1 General

In order to verify fulfilment of the guarantee, the guarantee values are compared with the test results converted to the guarantee conditions. Inclusion of total uncertainty τ_{tot} (6.4.4.1) in the guarantee comparison is dealt with below.

The guarantee comparison is carried out in most cases by means of graphic presentation. The inlet volume flow, $\dot{V}$, is selected as the abscissa X. The equations for the relative uncertainty of measured results shall be derived anew for other plotted features. The variable to be verified, e.g., efficiency η_{cou}, is plotted as the ordinate Y. The methods for the guarantee comparison described below apply, provided no contractual agreements to the contrary have been made.

8.2.2 Comparison of single test points with single guarantee points

Power at the coupling $P_{cou,\,co}$ converted to guarantee conditions in accordance with 6.3 is converted to the values taken as a basis for the guarantee for inlet volume flow, $\dot{V}_{1,g}$, and pressure ratio $\Pi_g\,[y_g = f(\Pi_g)]$; see Figure 7 assuming constant gas efficiency:

$$P_{cou,cog} = P_{i,co} \cdot \frac{\dot{V}_{1,g}}{\dot{V}_{1,co}} \cdot \frac{y_g}{y_{co}} + P_{mech,cog} \qquad (48)$$

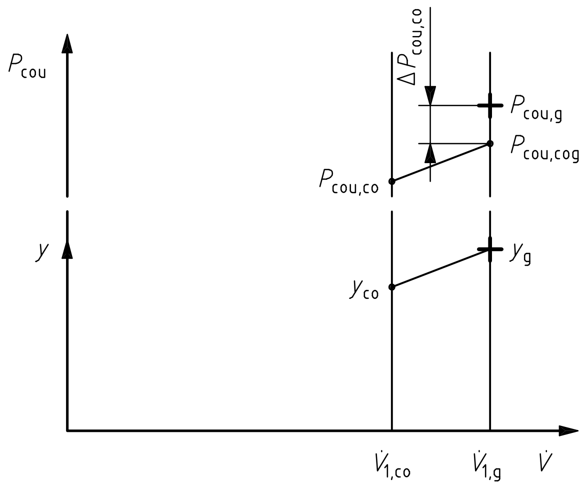


Figure 7 — Guarantee comparison for single test points

The applicability of this procedure is restricted by the permissibility of the assumption of a constant efficiency.

A guarantee comparison with respect to the performance curve shape is not possible. The measuring uncertainty for $P_{cou,co}$ shall, in this case, be calculated using the equation for related power at the coupling in 6.4.4.2.4

8.2.3 Comparison of measured performance curves with guarantee points

8.2.3.1 Compressors with fixed geometry and speed

Where a manufacturing tolerance is agreed for the inlet volume flow and/or discharge pressure (pressure ratio) for a compressor due to the lack of facilities for control to vary the performance curve, the guarantee comparison for power shall be carried out in accordance with Equation (48). The following point of the performance curve is taken as a basis for this, according to the type of manufacturing tolerance agreed:

- manufacturing tolerance on inlet volume flow;
- performance curve point at guaranteed discharge pressure (pressure ratio); see Figure 8;
- manufacturing tolerance on discharge pressure (pressure ratio): performance curve point at guaranteed inlet volume flow; see Figure 9;
- manufacturing tolerance on inlet volume flow and discharge pressure (pressure ratio): intersection of the performance curve with the straight line from the guarantee point to the minimum or maximum point of the tolerance field; see Figure 10.

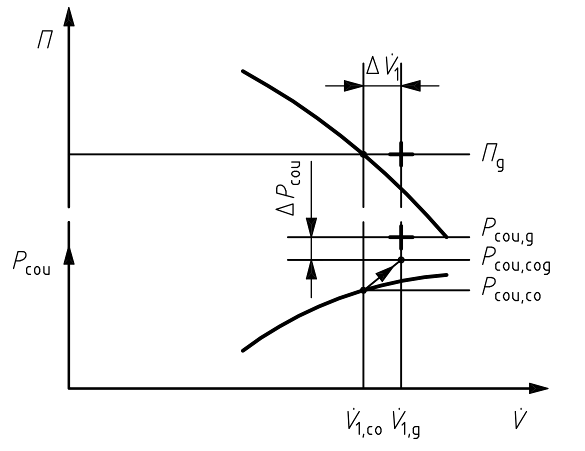


Figure 8 — Guarantee comparison for a compressor at guaranteed pressure ratio

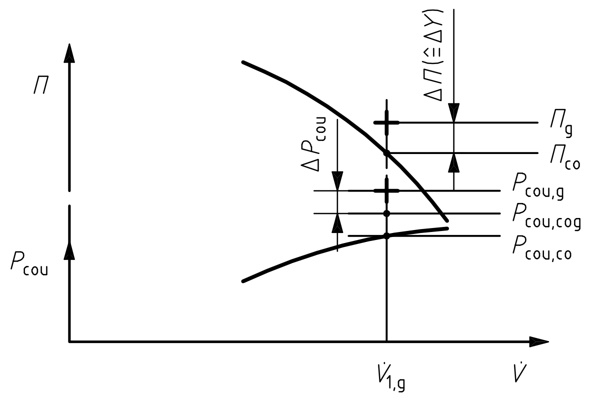


Figure 9 — Guarantee comparison for a compressor with guaranteed inlet volume flow

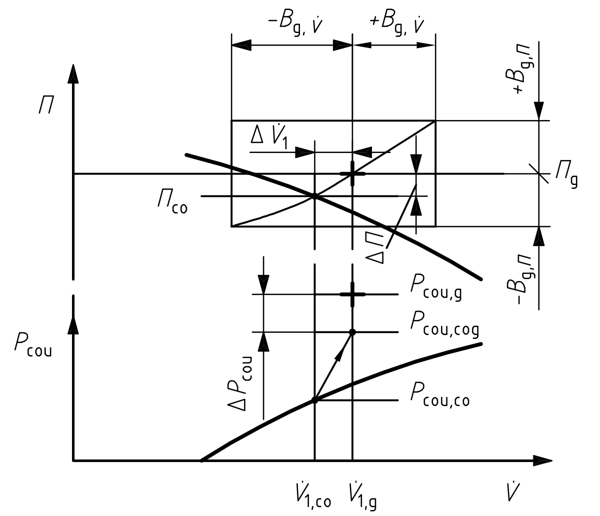


Figure 10 — Guarantee comparison for adjustable pressure ratio and inlet volume flow
(e.g. where manufacturing tolerance applies to pressure ratio and inlet volume flow)

8.2.3.2 Compressors with variable geometry or variable speed

Here, the guarantee comparison can be carried out directly in accordance with Figure 11 using a performance curve through the guarantee point $\left(\dot{V}_{1,g}, \Pi_g\right)$.

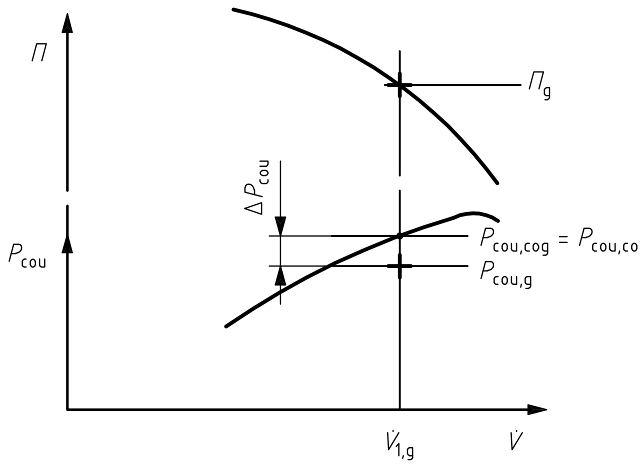


Figure 11 — Guarantee comparison for a compressor with variable geometry or variable speed

This performance curve can be operated directly, or may be produced from adjacent performance curves in conjunction with the similarity conditions or by means of interpolation.

8.2.3.3 Special case: inlet throttle control

Where the converted performance curve for the inlet volume flow of the guarantee point exhibits a greater pressure ratio than the guaranteed pressure ratio and inlet throttling is possible at the place where the compressor is installed, the guarantee comparison may be carried out making allowance for inlet throttle losses. Under perfect gas behaviour and without intercooling (Figure 12):

$$P_{cou,cog} = (P_{cou,co} - P_{mech,co}) \frac{p_{2,g}}{p_{2,co}} + P_{mech,co} \quad (49)$$

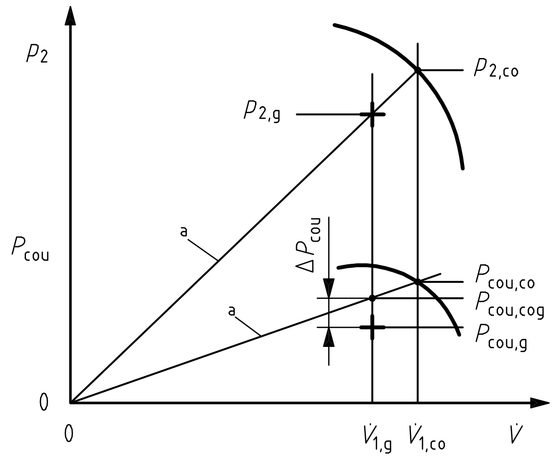
Where p_2/p_1 is a constant

$$\frac{\dot{V}_{1,g}}{\dot{V}_{1,co}} = \frac{p_{2,g}}{p_{2,co}} \quad (50)$$

The throttled inlet pressure is

$$p_{1,cog} = \frac{p_{2,g}}{p_{2,co}/p_{1,g}} \quad (51)$$

This suction pressure shall be in the tolerable operating range.



[a] Straight line.

Figure 12 — Guarantee comparison with suction throttle control

8.2.4 Measuring uncertainties and manufacturing tolerances with regard to the guarantee comparison in 8.3.2

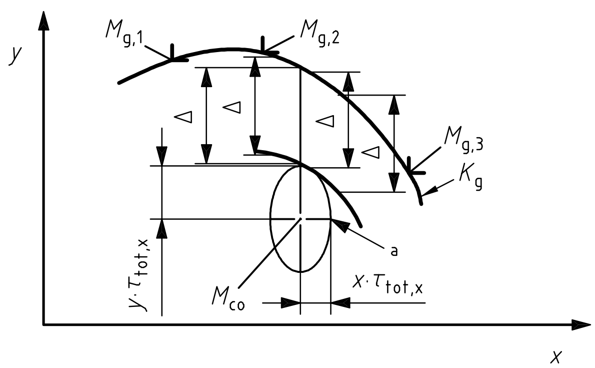
8.2.4.1 General

In the following, total uncertainties, τ_{res}, see 6.4.4.1, of the abscissa and ordinate values are plotted around the converted measured points, M_{co}, in the form of horizontal and vertical ellipse semi-axes. The appurtenant measuring uncertainty ellipses are entered. For example, for guarantee comparison of the efficiency of a compressor, total uncertainty, $\tau_{res,\dot{V}}$, for inlet volume flow $\dot{V}_1$ is entered horizontally, and total uncertainty, $\tau_{res,\eta cou}$, for efficiency, η_{cou}, is entered vertically.

These measuring uncertainty ellipses shall be plotted only if they have a significance for the guarantee comparison. This is, for instance, not the case where the deviation at the guarantee point is smaller than the magnitude of the corresponding semi-axis.

8.2.4.2 Given: One test point and one guarantee curve

Where several guarantee points, M_g, whose connecting line produces a performance (guarantee curve, K_g) and only one test point, M_{co}, are given, the procedure shown in Figure 18 should be used.



[a] Measuring uncertainty ellipse.

Figure 13 — Comparison of one test point with the guarantee curve, where the guarantee is not met

The deviation from the guarantee is provided in the case illustrated by the vertical difference Δ between the guarantee curve and the guarantee curve shifted vertically up to contact with the measuring uncertainty ellipse. In the case illustrated, the guarantee is not met.

The guarantee is regarded as met if the measuring uncertainty ellipse is intersected by the guarantee curve as in Figure 14.

The guarantee is also regarded as met if the measuring uncertainty ellipse touches the guarantee curve, K_g, as in Figure 15.

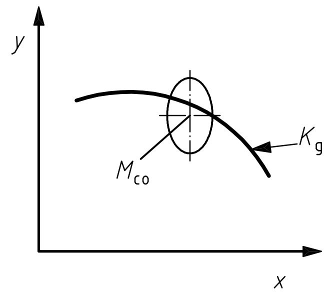


Figure 14 — Illustration of a test point with a given guarantee curve, in which the guarantee is met, partially taking into account the measuring uncertainty

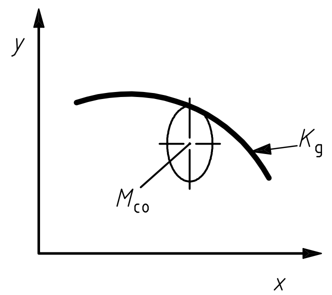
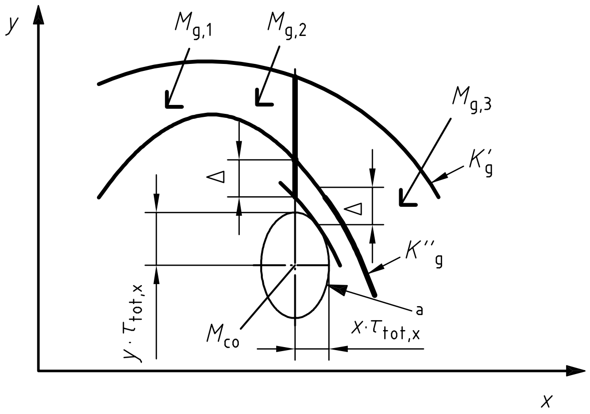


Figure 15 — Comparison of a test point with a guarantee band, in which the guarantee is not met fully taking into account the measuring uncertainty

8.2.4.3 Given: one test point and a guarantee band

Where manufacturing tolerances have been agreed (see 4.5), the guarantee curve is converted to a guarantee band (see for example Reference [7]).

In this cases, a guarantee band, B_g, is given (see, for example, Reference [7]) which, in Figure 16, for instance, is limited by the curves K'_g and K''_g. The remarks made in 8.2.4.1 apply with regard to the construction of measured points, M_{co}, and the measuring uncertainty ellipses.



[a] Measuring uncertainty ellipse.

Figure 16 — Comparison of several test points with one guarantee curve

The deviation from the guarantee in the case illustrated is provided by the vertical difference, Δ, between the lower limit of the guarantee band, K''_g, and the lower limitation curve of guarantee band, K''_g, shifted vertically to contact with the measuring uncertainty ellipse. In the case illustrated, the guarantee is not met.

The guarantee is regarded as met if curve K''_g intersects the measuring uncertainty ellipse, as in Figure 17.

The guarantee is also regarded as met if the measuring uncertainty ellipse touches curve K''_g as in Figure 18.

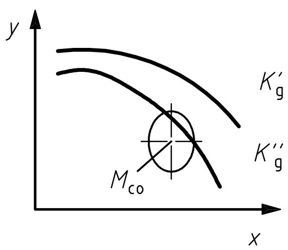


Figure 17 — Illustration of a test point with given guarantee band, in which the guarantee is met, partially taking into account the measuring uncertainty

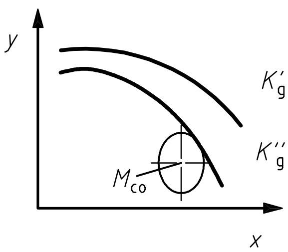


Figure 18 — Illustration of a test point with given guarantee band, in which the guarantee is met, fully taking into account the measuring uncertainty

8.2.4.4 Given: several test points and one guarantee curve

Where several guarantee points, whose connecting line produces a characteristic, are given and where several points were measured, the procedure described in 8.2.4.1 should be used for each valid test point. Figure 19 shows an example of two valid test points. The measured result for test point $M_{co,1}$ diverges from the guarantee by more than the measuring uncertainty, because the measuring uncertainty ellipse fails to contact with guarantee curve, K_g. The guarantee is met at test point $M_{co,2}$.

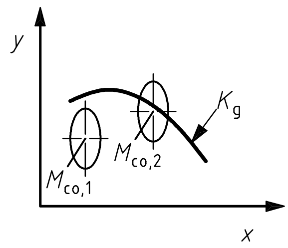
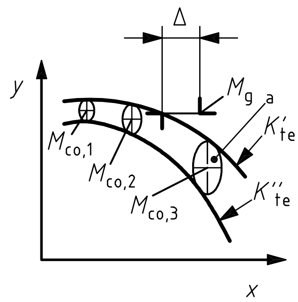


Figure 19 — Comparison of several test points with one guarantee curve

8.2.4.5 Given: single guarantee points and one test result band

Where only one guarantee point or single guarantee points are given and a sufficiently large number of test points converted to guarantee conditions, whose connecting line can be regarded as a characteristic, the procedure shown in Figure 20 should be used. Guarantee point M_g should be plotted. The converted test points, M_{co}, should be plotted and the measuring uncertainty ellipse entered as described in 8.2.4.1. Envelope curves K'_{te} and K''_{te} are then placed onto the measuring uncertainty ellipses. The envelope curves limit the result band. In the case illustrated, the horizontal distance, Δ, between the guarantee point and the upper envelope, K'_{te}, of the result band is, for instance, a measure of the deviation from the guarantee. The guarantee is considered not to have been met in the case illustrated.



[a] Measuring uncertainty ellipse.

Figure 20 — Comparison of a guarantee point with a test result band, where the guarantee is not met

Where the guarantee point is within the test result band, see Figure 21, the guarantee is considered to have been met.

Where the test result band touches the guarantee point, see Figure 22, i.e., the guarantee point is located on curve K''_{te} the guarantee is also considered to have been met.

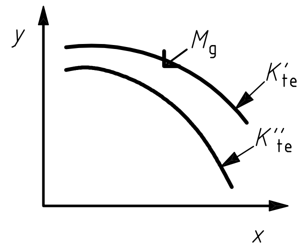


Figure 21 — Illustration of a test result band with given guarantee point, in which the guarantee is met, partially taking into account the measuring uncertainty

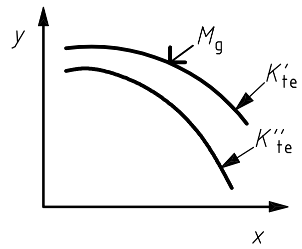


Figure 22 — Illustration of a test result band with given guarantee point, in which the guarantee is met, completely taking into account the measuring uncertainty

8.2.4.6 Given: single guarantee points with manufacturing tolerance and one test result band

The manufacturing tolerance is plotted in the manner specified in the contract of supply around the guarantee point, M_g, extending this to the guarantee range, B_g. Figure 23 shows an example, in which a positive and negative manufacturing tolerance of equal magnitude are plotted parallel to the abscissa.

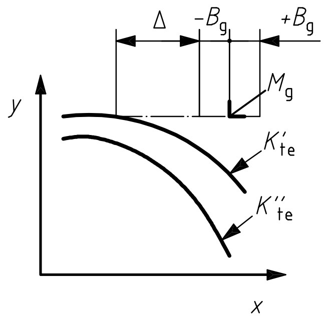


Figure 23 — Comparison of a test result band with a guarantee point and agreed single-axis manufacturing tolerance in the x **direction, in which the guarantee is not met**

In the case illustrated, the smallest distance, Δ, between guarantee range, B_g, and the result band in the direction of the manufacturing tolerance provides a measure of the deviation from the guarantee. The guarantee is considered to have been met if the test result band includes a portion of the guarantee range, as in Figure 24.

Where the guarantee range, B_g, touches the test result band, as in Figure 25, the guarantee is also considered to have been met.

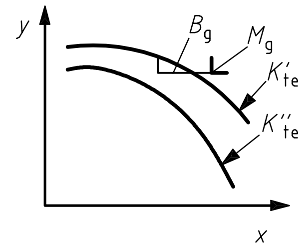


Figure 24 — Illustration of a test result band with given guarantee point with agreed single-axis manufacturing tolerance in the *x* direction, in which the guarantee is met, partially taking into account the measuring uncertainty and manufacturing tolerance

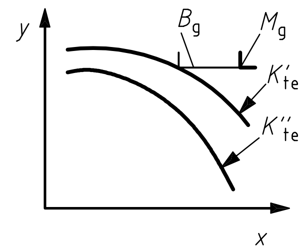


Figure 25 — Illustration of a test result band with given guarantee point with agreed single-axis manufacturing tolerance in the *x* direction, in which the guarantee is met, fully taking into account the measuring uncertainty and manufacturing tolerance

8.3 Special notes

8.3.1 Linking of manufacturing tolerances

Where a manufacturing tolerance for the inlet volume flow or for the pressure ratio has been agreed for a guarantee point, this manufacturing tolerance also applies to all other guarantee points, unless agreements to the contrary have been made in the contract.

8.3.2 Linking of guarantee points

The percentage deviations Δ* from the guarantee values are calculated from the absolute deviations Δ as defined in 8.2.4 and the weighted average deviation formed in accordance with

$$\Delta_m^* = \frac{\Sigma\left(\Delta_i^* c_i\right)}{\Sigma c_i} \qquad (52)$$

The various values of c_i are the evaluation characteristic numbers (weightings) allocated to the guarantee points. They shall be assumed equal to 1 where they have not been agreed in the contract.

Where it is not possible, for reasons beyond the supplier's control, to verify all guarantee points, the guarantees for the non-verifiable points will in all cases be deemed to have been fulfilled.

In cases where the supplier is demonstrably disadvantaged with regard to the overall guarantee comparison, separate agreements shall be made.

8.3.3 Upper limit value of the operating range on compressors with a performance map

The guarantee for the upper limit value of the operating range is deemed to have been met where the value for inlet volume flow or pressure ratio measured and converted at the upper maximum setting of the guide vanes or at maximum permissible speed falls below the relevant guaranteed limit value by less than the total measuring uncertainty.

Where the guarantee is not fulfilled at the maximum permissible speed (to be specified in the contract of supply), the speed necessary for fulfilment of the guarantee shall be calculated. The increase in speed shall not endanger the system or parts thereof (due, for instance, to mechanical loads, vibration, generation of heat). The supplier shall expressly declare the excess speed to be permissible for continuous duty. Where the driving machine is not manufactured by the compressor supplier, the compressor supplier shall be deemed obliged to obtain the agreement of the driving machine supplier to the necessary increase in speed.

8.3.4 Lower limit value of the operating range

The lower limit of the operating range is defined by the anti-surge control line plus the tolerance as agreed in the contract. If a guarantee comparison in respect of the power for a guarantee point below the actually measured anti-surge line has to be made, then the power of the measuring point at the just still stably operable inlet volume flow shall be referred to the inlet volume flow for the guarantee point in respect of the related power for this guarantee point.

9 Test report

A test report shall be written on the result of the acceptance test. It should include the following:

a) technical data on the compressor: customer/operator, installation location, application, type, machine number, year of manufacture, short technical description, power rating, rated speed, other relevant special features of the machine/system;

b) technical data for the driving machine: data as far as necessary for the guarantee comparison;

c) guarantee conditions (see 4.2) and the object of the guarantees (see 4.3);

d) date, location, persons in charge and witnessing parties;

e) test procedure (see 6.1.2) and flow scheme, with measuring points;

f) report on the execution of the acceptance test, with the attachment of a table of mean values of individual read-offs of significance for evaluation with statement of times; records made shall be attached, as shall documentation of analyses of the compressed gas, data on the instruments used for measurement, calibration certificates if required, etc.;

g) documentation for conversion to guarantee conditions: tables and diagrams used shall be stated; deviations from the standard specifications for measuring and conversion procedures shall be stated;

h) calibration method for gas properties, if relevant, shall be stated as agreed upon;

i) documentation for the guarantee proof: this should prove unequivocally whether, and the manner in which, the guarantees have been fulfilled. Allowance shall be made for total uncertainties in accordance with 6.4 and, possibly, in accordance with 8.2.4.

Annex A
(normative)

Flow diagram and figures for volume flow ratio

Flow diagram within flow similarity conditions — Calculation of test setting conditions for an uncooled section.

	Start	Result	Next step
	Guarantee conditions: Gas data: Inlet → discharge $\Psi_{p,g}$, $\eta_{p,g}$, N_g Geometry 1st stage Test conditions: Gas data: Inlet $\Psi_{p,te,an} = \Psi_{p,g}$ $\eta_{p,te,an} = \eta_{p,g}$ $n_{te,an} = \frac{1}{1 - \frac{k_{1,te} - 1}{k_{1,te} \cdot \eta_{p,g}}}$ $\lvert \Delta\phi_{tol} \rvert = 0 \div 0{,}05$ [a]		
A	$\left\lvert \frac{n_{te}}{n_g} - 1 \right\rvert \leqslant 0{,}001$	yes no	cont. B
	$X_{N,tol} = \sqrt{\frac{\Psi_{p,g}}{\Psi_{p,te}} \cdot \frac{\left(1 + \Delta\phi_{tol}\right)^{n_{te}-1} \cdot \left(\frac{p_2}{p_1}\right)_g^{\frac{n_g - 1}{n_g}} - 1}{\left(\frac{p_2}{p_1}\right)_g^{\frac{n_g - 1}{n_g}} - 1}}$		C
C	$N_{te} = X_{N,tol} \sqrt{\frac{R_{te} Z_{1te} T_{1te}}{R_g Z_{1g} T_{1g}}} \cdot N_g$ Correction of the Reynolds number [b] $Re_{u,te} = f\left(N_{te}, \ldots\right)$ $\eta_{p,te} = f\left(Re_{u,te}, \ldots\right)$ $\Psi_{p,te} = f\left(\eta_{p,te}, \ldots\right)$ $\Delta h_{te} = \frac{\Psi_{p,te}}{\eta_{p,te}} \cdot \frac{u_{2,te}^2}{2}$ Real gas calculation: $n_{te} = f\left(p_{1,te}, T_{1,te}, \Delta h_{te}, \eta_{p,te}\right)$		

	Start	Result	Next step
	$\left\lvert \frac{\eta_{p,te}}{\eta_{p,te,an}} - 1 \right\rvert < 0{,}001$ and $\left\lvert \frac{\eta_{te}}{\eta_{te,an}} - 1 \right\rvert < 0{,}001$	yes no	end cont.
	$n_{te,an} = n_{te}$ $\eta_{p,te,an} = \eta_{p,te}$ $\Psi_{p,te,an} = \Psi_{p,te}$		A
B	$\phi_2 = 1 + \Delta\phi_{tol}$		D
D	$F = \left\{ \frac{n_{te} - 1}{n_{te} - n_g} \left[1 - \frac{\left(\frac{p_2}{p_1}\right)_g^{\frac{n_g - 1}{n_g}} - 1}{\phi_2^{n_{te} - 1} \left(\frac{p_2}{p_1}\right)_g^{\frac{n_{te} - 1}{n_g}} - 1} \right] \right\}$		
	$F < 0$	yes no	E F
E	$X_{N,tol} = \sqrt{\frac{\Psi_{p,g}}{\Psi_{p,te}} \cdot \frac{\frac{n_{te}}{n_{te} - 1} \cdot \left[\phi_2^{n_{te} - 1} \left(\frac{p_2}{p_1}\right)_g^{\frac{n_{te} - 1}{n_g}} - 1 \right]}{\frac{n_g}{n_g - 1} \left[\left(\frac{p_2}{p_1}\right)_g^{\frac{n_g - 1}{n_g}} - 1 \right]}}$		C
F	$\left(\frac{p_x}{p_1}\right)_{Ex} = F^{\frac{n_g}{n_g - 1}}$		
	$1 < \left(\frac{p_x}{p_1}\right)_{Ex} < \left(\frac{p_2}{p_1}\right)_g$	yes no	cont. E
	$\phi_{Ex} = \frac{1}{\left(\frac{p_x}{p_1}\right)_{Ex}^{1/n_g}} \left\{ 1 + \frac{\left(\frac{p_x}{p_1}\right)_{Ex}^{\frac{n_g - 1}{n_g}} - 1}{\left(\frac{p_2}{p_1}\right)_g^{\frac{n_g - 1}{n_g}} - 1} \left[\phi_2^{n_{te} - 1} \left(\frac{p_2}{p_1}\right)_g^{\frac{n_{te} - 1}{n_g}} - 1 \right] \right\}^{\frac{1}{n_{te} - 1}}$		
	$\phi_{Ex} > 1 + \lvert \Delta\phi_{tol} \rvert$	yes no	cont. G
	$\phi_2 = \phi_2 - 0{,}001$		

	Start	Result	Next step
	$\phi_2 < 1 - \lvert\Delta\phi_{tol}\rvert$	yes no	H D
G	$\phi_2 < 1 - \lvert\Delta\phi_{tol}\rvert$	yes no	H cont.
	$\phi_{Ex} < 1 - \lvert\Delta\phi_{tol}\rvert$	yes no	cont. I
	$\phi_2 = \phi_2 + 0{,}001$		
	$\phi_2 > 1 + \lvert\Delta\phi_{tol}\rvert$	yes no	H D
I	$\phi_2 > 1 + \lvert\Delta\phi_{tol}\rvert$	yes no	H E
H	Can another test gas be used or the section be further subdivided for measuring purposes ?	yes no	J cont.
	No test possible within $\lvert\Delta\phi_{tol}\rvert$		end
J	Proceed using another test gas or further sub-divisions of the section		A

[a] Rotational speed of test, calculated with
$\Delta\phi_{tol} = \pm\, 0{,}01$ inner tolerance limit;
$\Delta\phi_{tol} = \pm\, 0{,}05$ outer tolerance limit.

[b] For Reynolds number correction, see Annex C.

Annex B
(normative)

Tests for volume flow ratio beyond flow similarity

B.1 General

In the event that no setting conditions for a test can be found where a flow similarity in accordance with the limitations of 7.2 and Annex A can be fulfilled, the following method can be used.

For setting conditions as near as practicable to flow similarity, the manufacturer shall predict a test performance curve.

Each point of the guarantee performance curve has a reference point on the predicted test performance curve.

Due to deviations of the volume ratios under guarantee and test conditions at the same flow coefficient of the first stage, the flow coefficients of the following stages are different under guarantee and test conditions.

In case of a 3-stage compressor as in example (see Figure B.1), the last stage has the largest deviation $\Delta\varphi$ from the guarantee conditions (see Figure B.2). The smallest deviations for all stages during test from guarantee conditions has reference point R on the predicted test curve "pr" with the same average flow coefficient in accordance with Figure B.3.

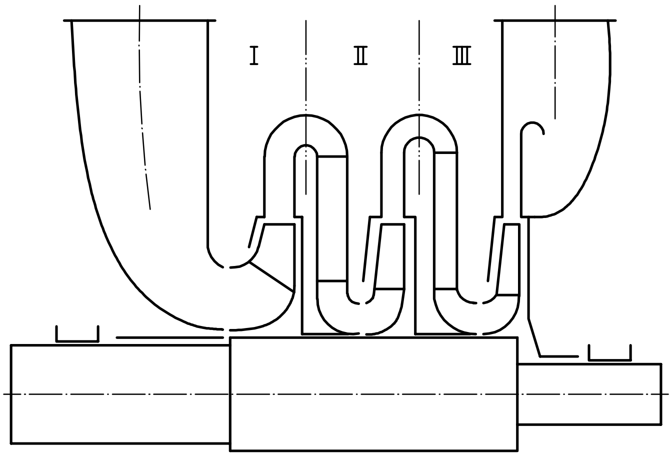


Figure B.1 — 3-stage compressor

Average flow coefficient

$$\varphi_{\text{av}} = \varphi_{\text{I}} \cdot \frac{v_{\text{av}}}{v_{1,\text{I}}} \tag{B.1}$$

with inlet flow coefficient of the first stage

$$\varphi_{\text{I}} = \frac{\dot{V}_{1,\text{I}}}{D_{2,\text{I}}^2 \cdot \frac{\pi}{4} \cdot u_{2,\text{I}}} \tag{B.2}$$

and ratio of average and inlet specific volume with

$$v_{\text{av}} = \frac{\int v \cdot \mathrm{d}p}{\Delta p} = \frac{y_{\text{p}}}{\Delta p} \tag{B.3}$$

$$\frac{v_{\text{av}}}{v_{1,\text{I}}} = \frac{y_{\text{p}}}{R \cdot Z_{1,\text{I}} \cdot T_{1,\text{I}}} \cdot \frac{1}{\frac{p_{2,\text{z}}}{p_{1,\text{I}}} - 1} = \frac{y_{\text{p}}}{p_{2,\text{z}} - p_{1,\text{I}}} \cdot \frac{\dot{m}}{\dot{V}_{1,\text{I}}} \tag{B.4}$$

where

$D_{2,\text{I}}$	outer diameter of the impeller of stage I	m
$u_{2,\text{I}}$	circumferential speed of the impeller of stage I	m/s
$\dot{V}_{1,\text{I}}$	inlet volume flow of the first stage of the stage group	m^3/s
$v_{1,\text{I}}$	inlet specific volume	m^3/kg
$p_{1,\text{I}}$	inlet pressure	MPa (bar)
$Z_{1,\text{I}}$	inlet compressibility factor	—
$T_{1,\text{I}}$	inlet temperature	K
v_{av}	average specific volume	m^3/kg
$p_{2,\text{z}}$	discharge pressure of the last stage group	MPa (bar)
$v_{2,\text{z}}$	discharge specific volume	m^3/kg
R	gas constant	J/kg/K
y_{p}	polytropic specific compression work	J/kg
$\dot{m}$	mass flow	kg/s

Each test point is converted to guarantee conditions at constant average flow coefficient $\varphi_{\text{av,T}} = \varphi_{\text{av,R}} = \varphi_{\text{av,G}}$ (see Figure B.3).

For test conditions with the same volume ratio

$$\left[\frac{v_{2,\text{z}}}{v_{1,\text{I}}}\right]_{\text{pr}} = \left[\frac{v_{2,\text{z}}}{v_{1,\text{I}}}\right]_{\text{g}} \tag{B.5}$$

curves “pr” and “g” would be identical (full flow similarity) with

$$\varphi_{\text{I,R}} = \varphi_{\text{I,G}} \text{ at } \varphi_{\text{av,R}} = \varphi_{\text{av,G}} \tag{B.6}$$

B.2 Conversion of test points (see Figure B.3)

Deviations of each point T on the measured test curve from the predicted test point R will then be transferred to point G on the guarantee curve with the same percentage resulting in corrected test point C according to the following relation:

Inlet flow coefficient

$$\varphi_{\mathrm{I,C}} = \frac{\varphi_{\mathrm{I,T}}}{\varphi_{\mathrm{I,R}}} \cdot \varphi_{\mathrm{I,G}} \quad \text{(B.7)}$$

Head coefficient

$$\psi_{\mathrm{p,C}} = \frac{\psi_{\mathrm{p,T}}}{\psi_{\mathrm{p,R}}} \cdot \psi_{\mathrm{p,G}} \quad \text{(B.8)}$$

The same relation is valid for other characteristic numbers like enthalpy coefficient and efficiency of the stage group.

B.3 Correction of surge volume flow and choke volume flow (see Figure B.3)

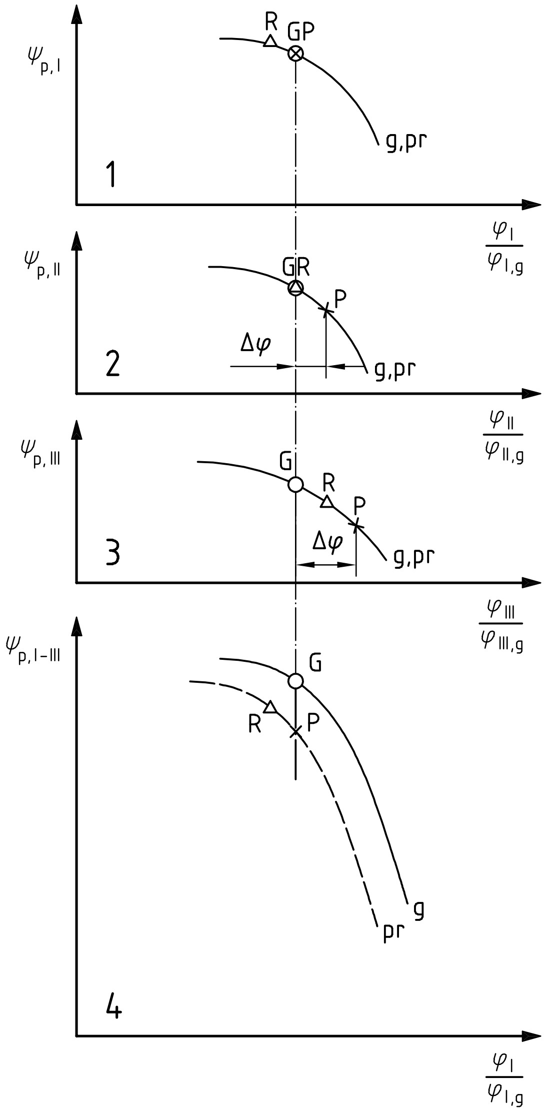
Surge volume flow and choke volume flow can be compared directly between measured test curve “te” and predicted test curve “pr” and transferred to the converted test curve co and the guarantee curve g, because they are not influenced by the average of all stages but normally by only one stage.

Correction of surge flow coefficient, $\varphi_{\mathrm{I,S,co}}$:

$$\varphi_{\mathrm{I,S,co}} = \varphi_{\mathrm{I,S,g}} + \left(\varphi_{\mathrm{I,S,te}} - \varphi_{\mathrm{I,S,pr}}\right) \quad \text{(B.9)}$$

Correction of choke flow coefficient, $\varphi_{\mathrm{I,CH,co}}$:

$$\varphi_{\mathrm{I,CH,co}} = \frac{\varphi_{\mathrm{I,CH,te}}}{\varphi_{\mathrm{I,CH,pr}}} \cdot \varphi_{\mathrm{I,CH,g}} \quad \text{(B.10)}$$

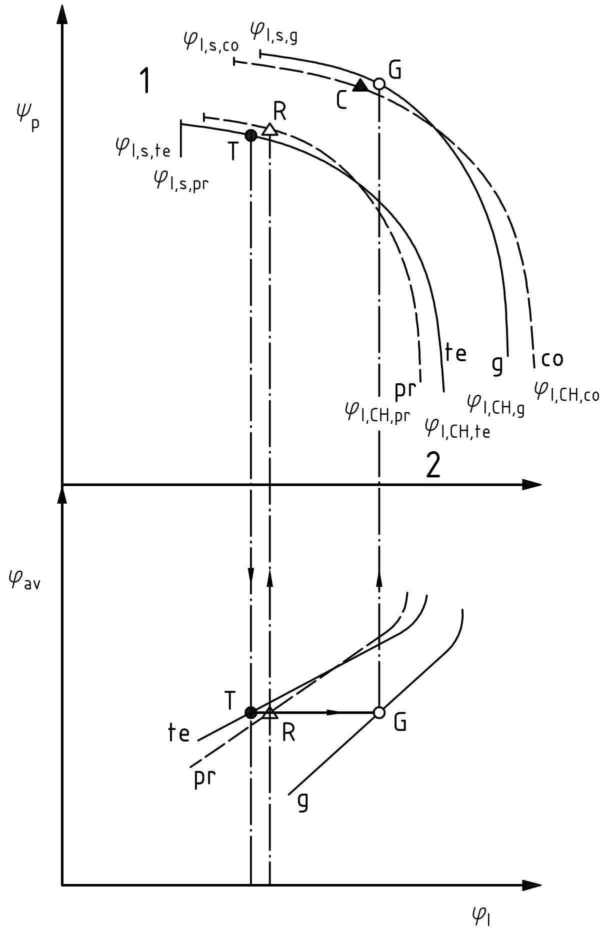


Key

1 stage I
2 stage II
3 stage III
4 stage I + II + III

g guarantee curve
pr predicted curve
G guarantee conditions
P predicted test conditions assuming $\varphi_{I,G} = \varphi_{I,R}$
R predicted test conditions $\varphi_{av,G} = \varphi_{av,R}$

Figure B.2 — Polytropic head coefficient of single stages and complete compressor



Key

1 surge flow

2 choke flow

T point on measured test curve “te”

R reference point to T on predicted test curve “pr”

G reference point to T on guarantee curve “g”

C corrected point for the converted curve “co”

Figure B.3 — Determination of the reference points

Annex C
(normative)

Correction method for the influence of Reynolds Number on the performance of centrifugal compressors

C.1 Summary

This method (Reference [11]) provides equations that are needed in case of differing Reynolds numbers in test and guarantee conditions, for corrections to efficiency, specific compression work and volume flow. The total losses are subdivided into losses irrespective of Reynolds number, covered by a coefficient of 0,3, and losses due to friction, which are regarded as being dependent on a representative value of the coefficient of friction, λ. The representative value of λ is dependent on a reference Reynolds number and a reference value for the relative average roughness of the compressor. These corrections can be applied in the permissible range as shown in Figure C.2.

C.2 Definitions

The representative value of the Reynolds number for the first stage of a section is stated by

$$Re_{\text{u}} = \frac{u \cdot b}{\nu_1} \qquad \text{(C.1)}$$

where

u is the tip speed of the first impeller of the section, expressed in metres per second;

b is the outlet width of first impeller of the section, expressed in metres;

ν_1 is the kinematic viscosity for total inlet state, expressed in square metres per second.

Average roughness, Ra, signifies average roughness starting from the centre line of the roughness peaks of the impeller and its diffusor. Ra can either be measured, or readoff from the manufacturer's drawing (assuming agreement on this procedure between the manufacturer and the purchaser). The representative relative roughness of the stage is stated by Ra/b.

In multi-stage compressors, the reference values for the Reynolds number and the relative roughness of the first stage of each section are taken.

The correction method relates only to internal flow losses. Separate allowance shall therefore be made for leakage mass flows and mechanical losses.

C.3 Correction equation for efficiency

The equation for correction of efficiency in the optimum efficiency range is stated by the equation:

$$\frac{1-\eta_{\text{p,co}}}{1-\eta_{\text{p,te}}} = \frac{0{,}3 + 0{,}7 \cdot \dfrac{\lambda_{\text{g}}}{\lambda_{\infty}}}{0{,}3 + 0{,}7 \cdot \dfrac{\lambda_{\text{te}}}{\lambda_{\infty}}} \qquad \text{(C.2)}$$

where

η_p is the polytropic efficiency of section;

λ is the coefficient of pipe friction.

with the subscripts

co converted from test to guarantee conditions or the guarantee value during determination of test setting conditions;

g guarantee conditions;

te test conditions;

∞ at an infinitely large Reynolds number.

Equations for calculations of λ values:

(von Karman):

$$\frac{1}{\sqrt{\lambda_\infty}} = 1{,}74 - 2 \cdot \log_{10}\left(2 \cdot \frac{Ra}{b}\right) \tag{C.3}$$

(Colebrook)

$$\frac{1}{\sqrt{\lambda}} = 1{,}74 - 2 \cdot \log_{10}\left(2 \cdot \frac{Ra}{b} + \frac{18{,}7}{Re_u \cdot \sqrt{\lambda}}\right) \tag{C.4}$$

Values for λ and λ_∞ can be taken from Figure C.3.

C.4 Correction equations for specific compression work, enthalpy difference and volume flow

Correction of specific work of compression:

$$\frac{\psi_{p,co}}{\psi_{p,te}} = 0{,}5 + 0{,}5 \cdot \frac{\eta_{p,co}}{\eta_{p,te}} \tag{C.5}$$

Correction of enthalpy difference:

$$\frac{\psi_{i,co}}{\psi_{i,te}} = 0{,}5 + 0{,}5 \cdot \frac{\eta_{p,te}}{\eta_{p,co}} \tag{C.6}$$

Correction of volume flow

$$\frac{\varphi_{co}}{\varphi_{te}} = \sqrt{\frac{\psi_{p,co}}{\psi_{p,te}}} \tag{C.7}$$

where

ψ_p is the polytropic reference process
$= 2\, y_p/u^2$;

ψ_i is the enthalpy coefficient of section
$= 2\, \Delta h/u^2$;

φ is the volume flow coefficient of the section
$= 4\dot{V}_1/\left(D^2 \pi u\right)$;

y_p is the specific polytropic compression work of section, expressed in joules per kilogram;

Δh is the enthalpy difference of section, expressed in joules per kilogram;

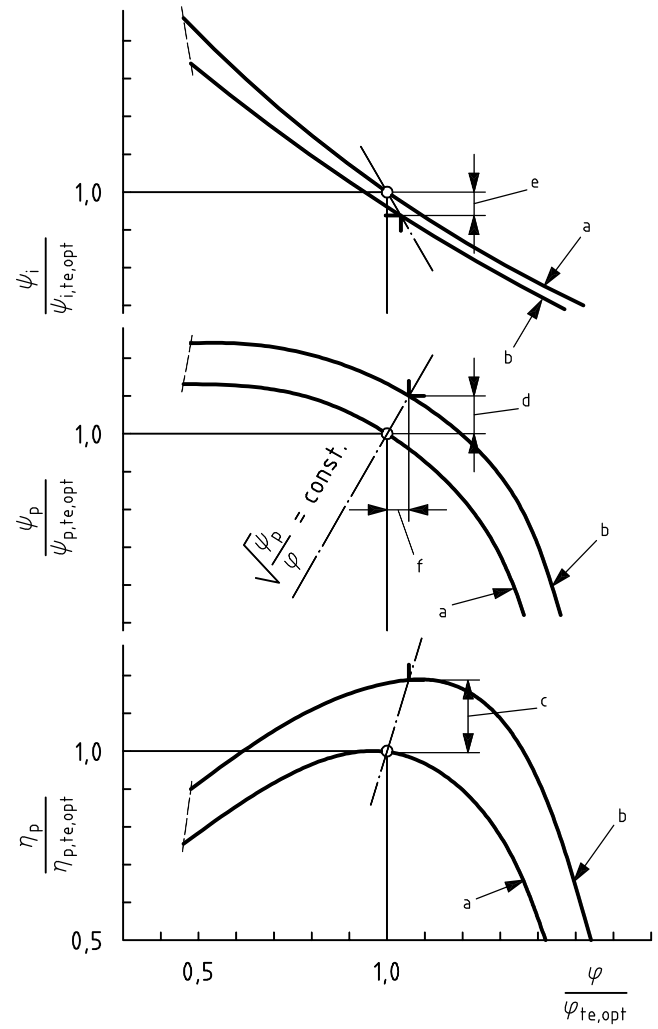
$\dot{V}_1$ is the inlet volume flow of section under total inlet state, expressed in cubic metres per second;

D is the outer diameter of first impeller of section, expressed in metres.

The above equations define the change in the performance curve point at optimum efficiency, (see Figure C.1). The other performance field points are converted at the same ratio as the optimum point, with the result that the performance curve shape remains unchanged.

C.5 Permissible range of application

The permissible range of application for the correction equations is shown in Figure C.2 (see 7.2.2.3.1).



[a] Test.

[b] Converted.

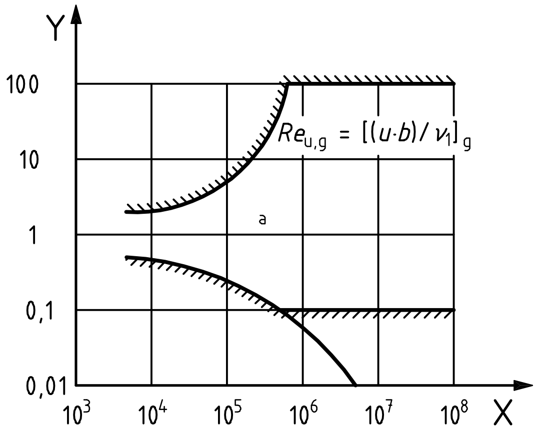
[c] Equation (C.2).

[d] Equation (C.5).

[e] Equation (C.6).

[f] Equation (C.7).

Figure C.1 — Correction of Reynolds number, diagrams for conversion



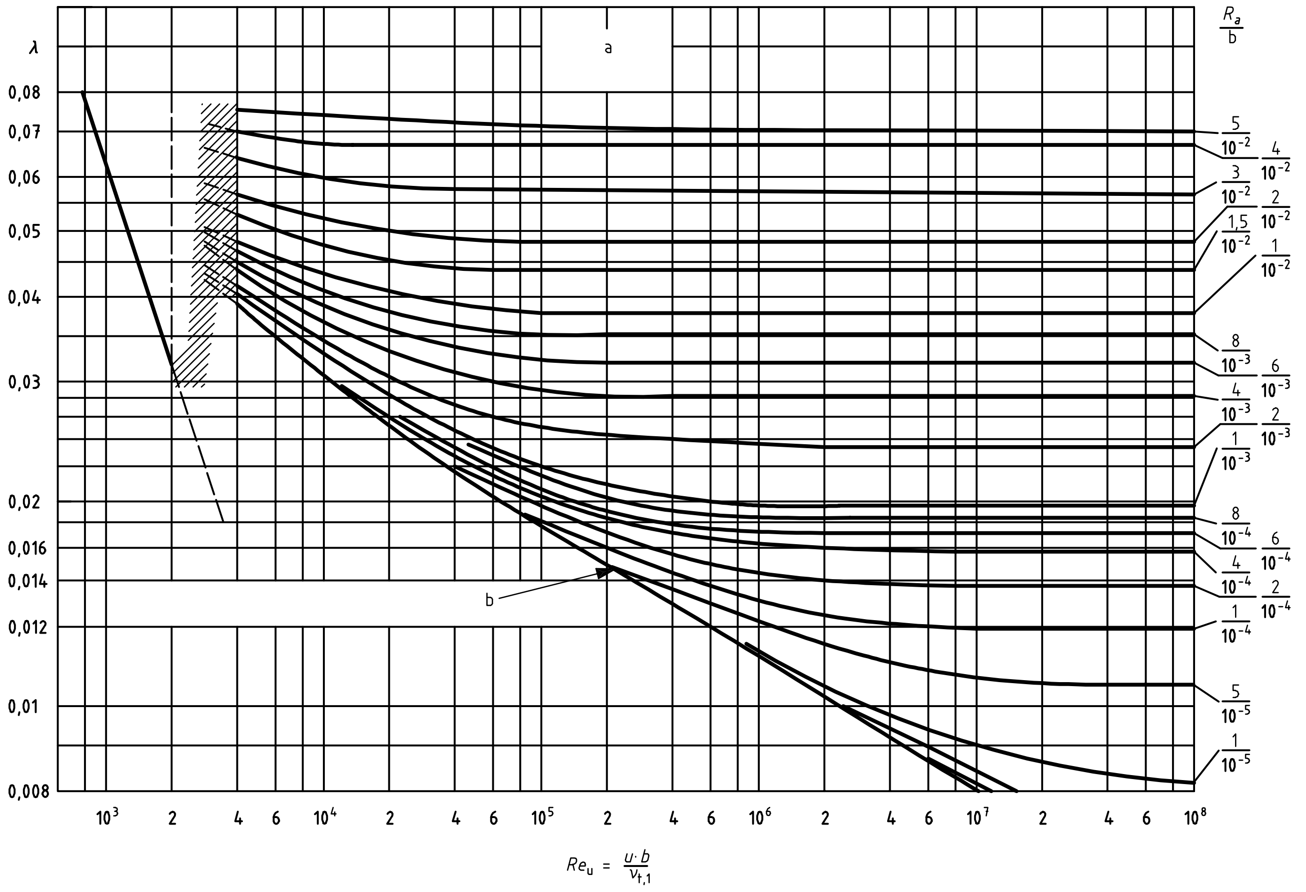
Key

X Guarantee Reynolds number, $Re_{u,g}$

Y Reynolds number ratio $\frac{Re_{u,te}}{Re_{u,g}}$

[a] Permissible range of application.

Figure C.2 — Permissible range of application for conversion



[a] Hydraulically rough pipes.

[b] Hydraulically smooth pipes.

Figure C.3 — Friction factor for turbulent flow in rough pipes

Annex D
(informative)

Derivation of equations for calculating the uncertainty of measuring results

The uncertainty of measuring results, $\tau_{\mathrm{res},W}$, expressed in percent, of a result function W (e.g. specific polytropic compression work) can be determined as follows, particularly in the case of simple functional interactions, whose derivation by means of closed mathematical solutions is possible:

$$\tau_{\mathrm{res},W} = \frac{V_W}{W} \cdot 100 = \pm \sqrt{\sum \left(\frac{\partial W}{\partial x_i} \cdot \frac{V_{xi}}{W} \right)^2} \cdot 100 \tag{D.1}$$

EXAMPLE

Basic equation for specific polytropic compression work:

$$y_{\mathrm{p}} = \Delta h \cdot \eta_{\mathrm{p}} = \frac{k}{k-1} \cdot R \cdot Z_{\mathrm{m}} \cdot \left(T_2 - T_1 \right) \cdot \frac{k-1}{k} \cdot \frac{\ln \frac{p_2}{p_1}}{\ln \frac{T_2}{T_1}} \tag{D.2}$$

$$y_{\mathrm{p}} = R \cdot Z_{\mathrm{m}} \cdot \left(T_2 - T_1 \right) \cdot \frac{\ln \frac{p_2}{p_1}}{\ln \frac{T_2}{T_1}} \tag{D.3}$$

Relative uncertainty of specific polytropic compression work:

$$\tau_{\mathrm{res},y,\mathrm{p}} = \frac{V_{y,\mathrm{p}}}{y_{\mathrm{p}}} \cdot 100 = \pm \sqrt{\sum \left(\frac{\partial y_p}{\partial x_i} \cdot \frac{V_{xi}}{y_{\mathrm{p}}} \right)^2} \cdot 100 \tag{D.4}$$

Derived from Equation (D.2):

$$\frac{\partial y_{\mathrm{p}}}{\partial p_2} = \frac{R \cdot Z_{\mathrm{m}} \cdot \left(T_2 - T_1 \right)}{\ln \frac{T_2}{T_1}} \cdot \frac{1}{\frac{p_2}{p_1}} \cdot \frac{1}{p_1} = \frac{R \cdot Z_{\mathrm{m}} \cdot \left(T_2 - T_1 \right)}{\ln \frac{T_2}{T_1}} \cdot \frac{1}{p_2} \tag{D.5}$$

$$\frac{\partial y_{\mathrm{p}}}{\partial p_2} \cdot \frac{V_{p2}}{y_{\mathrm{p}}} = \frac{1}{\ln \frac{p_2}{p_1}} \cdot \frac{V_{p2}}{p_2} \tag{D.6}$$

$$\frac{\partial y_{\mathrm{p}}}{\partial p_1} = \frac{R \cdot Z_{\mathrm{m}} \cdot \left(T_2 - T_1 \right)}{\ln \frac{T_2}{T_1}} \cdot \frac{1}{\frac{p_2}{p_1}} \cdot -\frac{p_2}{p_1^2} = -\frac{R \cdot Z_{\mathrm{m}} \cdot \left(T_2 - T_1 \right)}{\ln \frac{T_2}{T_1}} \cdot \frac{1}{p_1} \tag{D.7}$$

$$\frac{\partial y_{\mathrm{p}}}{\partial p_1} \cdot \frac{V_{p1}}{y_{\mathrm{p}}} = -\frac{1}{\ln \frac{p_2}{p_1}} \cdot \frac{V_{p1}}{p_1} \tag{D.8}$$

$$\frac{\partial y_\mathrm{p}}{\partial T_2} = R \cdot Z_\mathrm{m} \cdot \ln\frac{p_2}{p_1} \cdot \frac{\ln\frac{T_2}{T_1} - (T_2 - T_1) \cdot \frac{1}{\frac{T_2}{T_1}} \cdot \frac{1}{T_1}}{\left(\ln\frac{T_2}{T_1}\right)^2} = R \cdot Z_\mathrm{m} \cdot \ln\frac{p_2}{p_1} \cdot \frac{\ln\frac{T_2}{T_1} - (T_2 - T_1) \cdot \frac{1}{T_2}}{\left(\ln\frac{T_2}{T_1}\right)^2} \quad \text{(D.9)}$$

$$\frac{\partial y_\mathrm{p}}{\partial T_2} \cdot \frac{V_{T2}}{y_\mathrm{p}} = \left(\frac{T_2}{T_2 - T_1} - \frac{1}{\ln\frac{T_2}{T_1}}\right) \cdot \frac{V_{T2}}{T_2} \quad \text{(D.10)}$$

$$\frac{\partial y_\mathrm{p}}{\partial T_1} = R \cdot Z_\mathrm{m} \cdot \ln\frac{p_2}{p_1} \cdot \frac{-\ln\frac{T_2}{T_1} + (T_2 - T_1) \cdot \frac{1}{\frac{T_2}{T_1}} \cdot \frac{T_2}{T_1^2}}{\left(\ln\frac{T_2}{T_1}\right)^2} = R \cdot Z_\mathrm{m} \cdot \ln\frac{p_2}{p_1} \cdot \frac{-\ln\frac{T_2}{T_1} + (T_2 - T_1) \cdot \frac{1}{T_1}}{\left(\ln\frac{T_2}{T_1}\right)^2} \quad \text{(D.11)}$$

$$\frac{\partial y_\mathrm{p}}{\partial T_1} \cdot \frac{V_{T1}}{y_\mathrm{p}} = -\left(\frac{T_1}{T_2 - T_1} - \frac{1}{\ln\frac{T_2}{T_1}}\right) \cdot \frac{V_{T1}}{T_1} \quad \text{(D.12)}$$

$$\frac{\partial y_\mathrm{p}}{\partial R} = Z_\mathrm{m} \cdot (T_2 - T_1) \cdot \frac{\ln\frac{p_2}{p_1}}{\ln\frac{T_2}{T_1}} \quad \text{(D.13)}$$

$$\frac{\partial y_\mathrm{p}}{\partial R} \cdot \frac{V_R}{y_\mathrm{p}} = \frac{V_R}{R} \quad \text{(D.14)}$$

$$\frac{\partial y_\mathrm{p}}{\partial Z_\mathrm{m}} = \frac{V_{Z\mathrm{m}}}{Z_\mathrm{m}} \quad \text{(D.15)}$$

$$\tau_{\mathrm{res},y,\mathrm{p}} = \frac{V_{y\mathrm{p}}}{y_\mathrm{p}} = \pm\sqrt{\left(\frac{1}{\ln\frac{p_2}{p_1}}\right)^2 \cdot \left(\tau_{p1}^2 - \tau_{p2}^2\right) + \left(\frac{T_2}{T_2 - T_1} - \frac{1}{\ln\frac{T_2}{T_1}}\right)^2 \cdot \tau_{T2}^2 + \left(\frac{T_1}{T_2 - T_1} - \frac{1}{\ln\frac{T_2}{T_1}}\right)^2 \cdot \tau_{T1}^2 + \tau_R^2 + \tau_{Z\mathrm{m}}^2} \quad \text{(D.16)}$$

Further derived equations are given in 6.4.4.2. For complicated functional interactions, see 6.4.4.3.

Annex E
(informative)

Special terms for compressors

The basic function of a compressor is to transfer energy to a mass flow of a gas, gas mixture or vapour, in order to increase its pressure; generally such fluids undergo a change in their other thermodynamic variables of state, too.

E.1 Thermodynamic state

The thermodynamic state of a simple system in which the constant chemical composition of the fluid, the absence of magnetic, electrical and capillary effects and the negligibility of the field of gravity (generally fulfilled in present-day compressors) are assumed is determined by two independent thermal or caloric variables of state. This thermodynamic state of a system is referred to as the state of equilibrium when it no longer changes even following isolation from the effects of its environment.

E.1.1 Thermal variables of state

The state of a simple homogeneous system is defined by two of the three thermal variables of state:

— p absolute pressure

— v specific volume or reciprocal $\rho = 1/v$ (density)

— T thermodynamic (absolute) temperature

E.1.1.1
pressure
quotient of the perpendicular force, F_n, that is exerted on an area, A, divided by this area

NOTE The term “pressure” and the units of measurement to be used in conjunction with it are defined in ISO 31-3 [14].

E.1.1.1.1
absolute pressure
p
pressure composed of reference pressure, p_o, and the pressure difference Δp (positive or negative) determined relative to this reference pressure using a pressure measuring instrument

$$p = p_o + \Delta p \qquad \text{(E.1)}$$

where p_o is the atmospheric pressure in most cases.

E.1.1.1.2
static pressure
⟨flowing fluid⟩ pressure that would be indicated by a pressure measuring instrument located in the flow of fluid and moving with it at the same speed

NOTE In the case of fluids flowing in a straight line, this is also the pressure that the fluid exerts on a wall located parallel to the direction of flow. It is stated in the form of absolute static pressure, p.

E.1.1.1.3
dynamic pressure
stagnation pressure
Δp_d
amount by which the static pressure of the fluid flowing at velocity c would rise if its kinetic energy were converted to the work of compression without losses (reversibly) and without heat exchange with the adjacent gas particles, i.e., isentropically and adiabatically

NOTE The following applies with an error of less than 0,5 % for $\Delta p_d / p \leqslant 0{,}01$, which is generally fulfilled at the compressor's inlet and outlet nozzles:

$$\Delta p_d \approx \frac{c^2}{2} \cdot \rho \tag{E.2}$$

Where $\Delta p_d / p > 0{,}01$, the precise relationship between Δp_d and c should be determined, in order to prevent the error from becoming too large:

$$\frac{c^2}{2} = \frac{k}{k-1} \cdot R \cdot Z \cdot T \cdot \left[\left(1 + \frac{\Delta p_d}{p} \right)^{\frac{k-1}{k}} - 1 \right] \tag{E.3}$$

E.1.1.1.4
total pressure
p_{tot}
absolute total pressure is the sum of absolute static and dynamic pressure, which equals:

$$p_{tot} = p + \Delta p_d \tag{E.4}$$

E.1.1.2
temperature

NOTE The unit of thermodynamic temperature is the kelvin (unit symbol: K). Where Celsius temperatures are stated, the unit name “degrees Celsius” (unit symbol: °C) is used. This is T = 273,15 + t, with T in K and t in °C.

E.1.1.2.1
static temperature
⟨complete temperature equilibrium⟩ static temperature prevails in a motionless fluid

NOTE A static temperature also exists in every flowing fluid. It is the temperature which a thermometer located in the flow of fluid and moving with it at the same velocity would indicate.

E.1.1.2.2
dynamic temperature
stagnation temperature
⟨ideal gases⟩ temperature increase (dynamic or stagnation temperature) that results from an isentropic adiabatic conversion of total kinetic energy to work of compression as follows:

$$\Delta t_d = \frac{c^2}{2 \cdot c_p} \tag{E.5}$$

where c_p is the mean isobaric specific heat capacity between t and $t + \Delta t_d$.

NOTE Given ideal gas behaviour, the following numerical value equation applies approximately to air:

$$\Delta t_d \approx \frac{c^2}{2\,000} \tag{E.6}$$

where c is expressed in metres per second.

E.1.1.2.3
total temperature

$$t_{\text{tot}} = t + \Delta t_{\text{d}}$$

or

total thermodynamic temperature

$$T_{\text{tot}} = T + \Delta t_{\text{d}} \tag{E.7}$$

NOTE Together with the total absolute pressure, p_{tot}, as defined in E.1.1.1.4, it constitutes the "stagnation state" of the gas prior to isentropic acceleration or following isentropic retardation. Given ideal gas behaviour, the following relationship exists between the total and static variables of state:

$$\frac{T_{\text{tot}}}{T} = \left(\frac{p_{\text{tot}}}{p}\right)^{\frac{k-1}{k}} \tag{E.8}$$

E.1.1.3
specific volume
v
volume of a unit of mass

NOTE 1 The common unit is cubic metres per kilogram.

NOTE 2 The reciprocal, $\rho = 1/v$, is referred to as "density", i.e., the mass of the unit of volume, expressed in kilograms per cubic metre.

E.1.2 Caloric variables of state

E.1.2.1 Enthalpy

Total enthalpy, given by the equation:

$$h_{\text{tot}} = h + \frac{c^2}{2} + gz \tag{E.9}$$

is the sum of static enthalpy, h, the kinetic energy, $c^2/2$ and the potential energy, gz, which derives from the acceleration due to gravity and geodesic elevation, z. The term gz can be ignored in the case of the enthalpy differences under observation here.

Given ideal gas behaviour, enthalpy, h, depends only on T. Under real gas conditions, enthalpy is additionally dependent on pressure, a circumstance for which allowance is made, for instance, in the h,s diagrams for the various gases.

Since, in the case of gases, pressure and temperature are related to one another on the saturation curve, one of these two variables of state suffices here for determination. In the two-phase range, two independent variables of state shall be supplied for pure gases. Vapour content, x, can be used as one of these.

E.1.2.2 Entropy

Each system has a variable of state, s, referred to as specific entropy, the differential of which is defined as

$$\mathrm{d}s = \frac{\mathrm{d}h - v\mathrm{d}p}{T} \tag{E.10}$$

The entropy of an adiabatic system can never decrease. In all natural (non-reversible) processes, the entropy of the adiabatic system increases; in all reversible processes, constituting a boundary case of the non-reversible processes, it remains constant.

E.1.3 Thermal and caloric equation of state

The thermal variables of state have a fixed relationship to the caloric variables of state. A thermodynamic state can therefore also be described using suitable combinations of thermal and caloric variables of state.

Molar mass, M, (unit symbol: kg/kmol) has the following relationship to the specific gas constant, R:

$$R = \frac{R_{\text{mol}}}{M} \quad \text{(E.11)}$$

R_{mol} = 8 314,4 J/(kmol K) is the universal gas constant.

At constant pressure, specific heat capacity is

$$c_p = \left(\frac{\partial h}{\partial T}\right)_p \quad \text{(E.12)}$$

and, at constant volume

$$c_v = \left(\frac{\partial u}{\partial T}\right)_v \quad \text{(E.13)}$$

where u is the specific internal energy.

E.1.3.1 Ideal gas behaviour

At low pressures ($p \rightarrow 0$), all gases exhibit exceptionally simple behaviour: the thermal and caloric equations of state conform to simple boundary laws. This state range is referred to as the ideal gas behaviour range. In thermodynamic terms, ideal gas behaviour is defined by the thermal equation of state

$$p \cdot v = R \cdot T \quad \text{(E.14)}$$

and the caloric equation of state

$$u = u(T) \quad \text{(E.15)}$$

Given ideal gas behaviour, the isentropic exponent

$$k = -\frac{v}{p} \cdot \left(\frac{\partial p}{\partial v}\right)_{\text{s}} \quad \text{(E.16)}$$

is equal to the ratio of specific heat capacities c_p/c_v:

$$k = \kappa = \frac{c_p}{c_v} \quad \text{(E.17)}$$

Given ideal gas behaviour, k, is constant, or dependent only on temperature.

E.1.3.2 Real gas behaviour

At high pressures and low temperatures, deviations from ideal gas behaviour occur. The thermal equation of state is then:

$$p \cdot v = R \cdot Z \cdot T \quad \text{(E.18)}$$

and the caloric equation of state with specific internal energy

$$u = u(v, T) \quad \text{(E.19)}$$

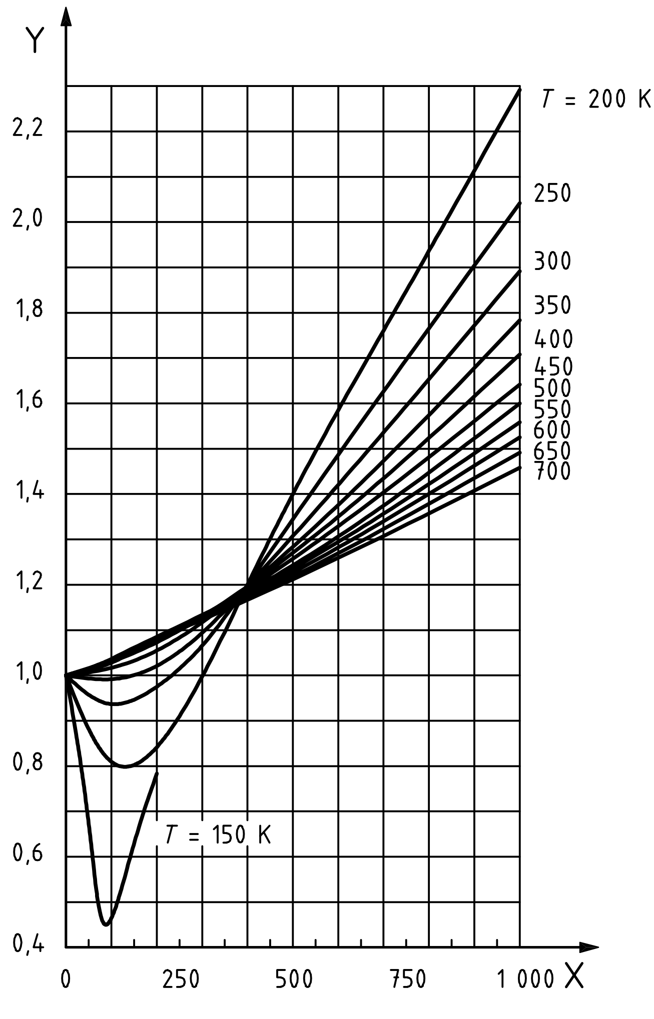
The compressibility factor

$$Z = \frac{p \cdot v}{R \cdot T} \quad \text{(E.20)}$$

is then used to make allowance for the deviations from the ideal gas law. The factor Z depends on p and T.

Gases whose critical temperatures are well below 0 °C, such as oxygen, nitrogen, hydrogen, air, etc., have compressibility factors which differ only slightly from 1 in the temperature range from 0 °C to 200 °C and pressure range up to approximately 2 MPa (20 bar) which are of significance for compressors. The rate of change in Z increases as the critical point (p_{crit}, t_{crit}) is approached.

The compressibility factors and variables of state are determined from gas data equations. Figure E.1 shows the plot of compressibility factor, Z, as a function of the two variables of state, p and T, using the example of dry air. Major differences for Z can arise depending on the gas data equation used, in particular in the case of pronounced real behaviour, i.e. in the vicinity of the critical point. It is recommended, particularly in such cases, to make agreements in advance on the gas data equations to be used in the test evaluation.



Key

X pressure, p, bars

Y compressibility factor, Z

Figure E.1 — Compressibility factor of dry air

Real gas behaviour is characterized by the following compressibility functions:

$$X = \frac{T}{v} \cdot \left(\frac{\partial v}{\partial T} \right)_p - 1 = \frac{T}{Z} \cdot \left(\frac{\partial Z}{\partial T} \right)_p \tag{E.21}$$

$$Y = \frac{p}{v} \cdot \left(\frac{\partial v}{\partial p} \right)_T = 1 - \frac{p}{Z} \cdot \left(\frac{\partial Z}{\partial p} \right)_T \tag{E22}$$

Given real gas behaviour, the following equations apply to isentropic changes of state:

$$\frac{v_1}{v_{2,s}} = \left(\frac{p_2}{p_1} \right)^{\frac{1}{k_v}} \tag{E.23}$$

and

$$\frac{T_{2,s}}{T_1} = \left(\frac{p_2}{p_1} \right)^{\frac{k_T - 1}{k_T}} = \left(\frac{p_2}{p_1} \right)^{m_s} = \frac{Z_1}{Z_{2,s}} \cdot \left(\frac{p_2}{p_1} \right)^{\frac{k_v - 1}{k_v}} \tag{E.24}$$

where

$$k_v = -\frac{v}{p} \cdot \left(\frac{\partial p}{\partial \text{v}} \right)_s = \frac{\kappa}{Y} \tag{E.25}$$

and

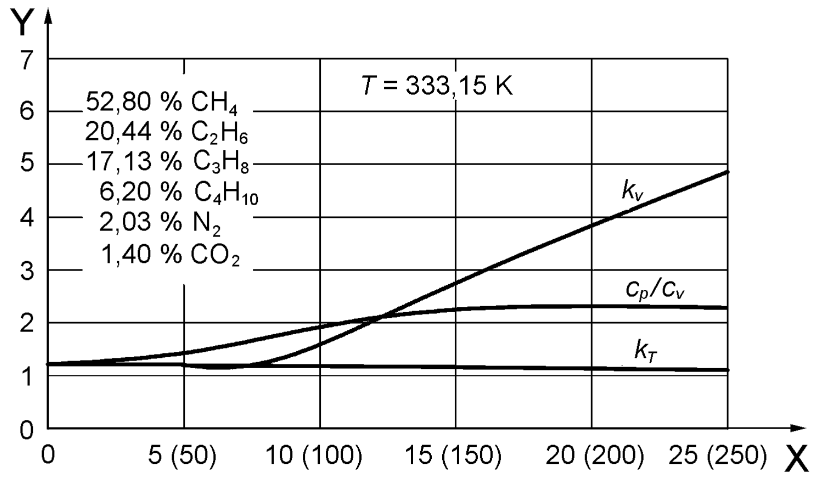
$$m_s = \frac{k_T - 1}{k_T} = \frac{\kappa - 1}{\kappa} \cdot \frac{Y}{1 + X} \tag{E.26}$$

and

$$k_T = \frac{1}{1 - \frac{p}{T} \cdot \left(\frac{\partial T}{\partial p} \right)_s} = \frac{1}{1 - \frac{\kappa - 1}{\kappa} \cdot \frac{Y}{1 + X}} \tag{E.27}$$

In the case of air, k = 1,4 ± 0,03 in the ranges p = 0 MPa to 1 MPa (0 bar to 10 bar) and t = 0 °C to 200 °C, signifying that $k = \kappa = c_p/c_v = k_v = k_T$ = constant can be used for calculation in this range with sufficient accuracy for the gas specified. Figure E.2 shows an example of greatly differing values for c_p/c_v, k_v and k_T, for a natural gas mixture.

Calculation of the compressibility function, Y, can be performed using the relationship derived for the ideal gas and modified by the inclusion of compressibility factor, Z, and a mean isentropic exponent, k, with adequate accuracy within the limits specified in Table E.1 for the compressibility functions at each point of the change of state and for the ratio of minimum and maximum isentropic exponents, $k_{\text{max}}/k_{\text{min}}$ occurring during the change of state. Outside these limits, test evaluation and/or conversion to guarantee conditions shall be performed with more precise allowance made for real gas behaviour.



Key

X pressure, expressed in MPa (bars)

Y ratio of specific heat capacities c_p/c_v and the isentropic exponents k_v and k_T

Figure E.2 — Ratio of specific heat capacities and isentropic exponents for a natural gas as a function of pressure

Table E.1 — Allowable departure for simplified calculation of changes of state [9]

Pressure ratio	Maximum ratio	Maximum compressibility function		Minimum compressibility function	
	$\frac{k_{max}}{k_{min}}$	X	Y	X	Y
1,4	1,12	0,279	1,071	–0,344	0,925
2	1,10	0,167	1,034	–0,175	0,964
4	1,09	0,071	1,017	–0,073	0,982
8	1,08	0,050	1,011	–0,041	0,988
16	1,07	0,033	1,008	–0,031	0,991
32	1,06	0,028	1,006	–0,025	0,993

The basis for this are equations of state, state diagrams and gas data tables. Where no gas data documentation is available, calculation may be performed by way of approximation using the compressibility functions obtained with the aid of the reduced pressures and temperatures.

E.2 Determination of gas properties in the case of mixtures

E.2.1 Mixtures of gases

A mixture of n gases can be described in three ways, by stating for each component gas, i (i = 1, 2, ..., n) either

— the molar fraction (equal to the volume fraction given ideal gas behaviour), r_i,

— or mass fraction, w_i,

— or partial pressure, p_i.

Dalton's Law states that every gas behaves in a mixture of ideal gases as if it were present alone at its partial pressure. In the gas/vapour mixtures, the vapour, too, may also be regarded with sufficient accuracy as an ideal gas, since the vapour fraction is generally relatively small. This is particularly true in the case of vapour in air in the range of 0 °C to 50 °C under atmospheric conditions.

E.2.1.1 Conversion

With the aid of the molar masses M_i of all individual gases, molar proportion, r_i, can be converted to mass proportion w_i and vice versa by means of Table E.2 and, where pressure, p, of the mixture is known, also to partial pressures, p_i, and vice versa.

The equivalent molar mass for a gas mixture is

$$M = \sum\left(r_i M_i\right) = \frac{1}{\sum\left(\frac{w_i}{M_i}\right)} = \sum\left(\frac{p_i}{p} M_i\right) \qquad \text{(E.28)}$$

and the gas constant, R, of the mixture in accordance with Equation (E.11) or

$$R = \sum\left(w_i R_i\right) \qquad \text{(E.29)}$$

In dry atmospheric air, the volume and mass proportion of oxygen and nitrogen approximate to the following:

	Oxygen	**Nitrogen**
r_i	0,210	0,790
w_i	0,233	0,767
M_i	32	28,2

Table E.2 — Computational relationship between mass proportion, partial pressure and mole proportion

Unknown: / Given:	**Mass proportion**	**Partial pressure**	**Mole proportion** (for ideal gases identical with volume proportion)
Mass proportion w_i; Σw_i = 1	—	$p_i = \frac{w_i / M_i}{\Sigma\left(w_i / M_i\right)} p$	$r_i = \frac{w_i / M_i}{\Sigma\left(w_i / M_i\right)} = w_i \frac{R_i}{R}$
Partial pressure p_i; $\Sigma p_i = p$	$w_i = \frac{p_i M_i}{\Sigma\left(p_i M_i\right)}$	—	$r_i = \frac{p_i}{p}$
Mole proportion (for ideal gases identical with volume proportion) r_i; Σr_i = 1	$w_i = \frac{r_i M_i}{\Sigma\left(r_i M_i\right)} = \frac{r_i R}{R_i}$	$p_i = r_i p$	—

The number of moles of the components per kilogram mixture (mol/kg) is

$$m_i = \frac{w_i}{M_i} \qquad \text{(E.30)}$$

The number of moles per kilogram mixture is then

$$m = \sum m_i \tag{E.31}$$

With M, m and R:

$$r_i = \frac{m_i}{m} \tag{E.32}$$

$$w_i = \frac{p_i}{p} \cdot \frac{M_i}{M} = \frac{p_i}{p} \cdot \frac{R}{R_i} \tag{E.33}$$

isobaric specific heat capacity:

$$c_p = \frac{\sum \left(r_i \cdot M_i \cdot c_{p,i} \right)}{\sum \left(r_i \cdot M_i \right)} = \sum \left(w_i \cdot c_{p,i} \right) \tag{E.34}$$

$$= \frac{\sum \left(\frac{p_i}{p} \cdot M_i \cdot c_{p,i} \right)}{\sum \left(\frac{p_i}{p} \cdot M_i \right)}$$

Only under the precondition that all individual gases and vapours conform to the ideal equation of state does the isentropic exponent equal the ratio of the specific heat capacities of the mixture:

a) from the specific heat capacities, $c_{p,i}$, at constant pressure of the individual gases:

$$k = \kappa = \frac{1}{1 - \frac{R_{\text{mol}}}{\sum \left(r_i \cdot M_i \cdot c_{p,i} \right)}} \tag{E.35}$$

b) from the ratios, κ_{i}, of the specific heat capacities of the individual gases:

$$k = \kappa = \frac{1}{1 - \frac{1}{\sum \left(r_i \cdot \frac{\kappa_i}{\kappa_i - 1} \right)}} \tag{E.36}$$

In general, it is not possible to calculate the compressibility factor of a gas mixture using simple mixture rules from the proportions of the individual gases and their compressibility factors; instead, experimentally obtained influencing factors incorporated in gas data equations shall be used. Where suitable gas data programs are not available, calculation can be performed using the "mixture rule" as an orientation point for the deviation from the gas law:

$$Z = \frac{\sum \left(w_i \cdot R_i \cdot Z_i \right)}{\sum \left(w_i \cdot R_i \right)} = \sum \left(r_i \cdot Z_i \right) = \frac{\sum \left(m_i \cdot Z_i \right)}{m} \tag{E.37}$$

Here, w_i, and r_i represent the mass and molar fractions, respectively, m_i is the number of moles of the ideal individual gas in the mixture and R_i is the gas constant. Z_i represents the compressibility factors of the individual gases at mixture temperature, T, and mixture pressure, p.

E.2.2 Mixtures consisting of gases and vapours

E.2.2.1 General relationships

The vapour fraction should be regarded as adequately small, with the result that the vapour component alone conforms satisfactorily to the ideal gas law and the compressibility factor of the vapour can, therefore, be equated to 1. Z is introduced, however, for the dry gas. In numerical calculations, Z should, then, be used corresponding to the mixture temperature and the partial pressure of the dry gas.

Mixing gases with vapours is possible without the formation of condensate only as long as the partial pressure of the vapour, p_{vap} remains below the saturation pressure, $p_{vap,sat}$, relating to the mixture temperature. The relative humidity of the mixture is described as

$$\varphi = \frac{p_{vap}}{p_{vap,sat}} \tag{E.38}$$

The value $\varphi = 1$ identifies the maximum possible vapour pressure and, simultaneously, the maximum vapour fraction in a mixture volume.

As soon as the value $p_{vap}/p_{vap,sat}$ becomes greater than 1 — irrespective of whether this occurs due to cooling at the same pressure or due to a reduction in volume or increase in pressure at the same temperature — a portion of the vapour will condense (dew-line). The “dewpoint” is the temperature at which the dew-line is reached due to cooling while pressure and vapour fraction remain the same.

In a similar way in the case of gas mixtures, the following equations derive when p_{gas} is the partial pressure of the gas and p_{vap} is the partial pressure of the vapour.

For the pressure of the mixture:

$$p = p_{vap} + p_{gas} = \varphi \cdot p_{vap,sat} + p_{gas} \tag{E.39}$$

For the molar volume vapour fraction in the mixture:

$$r_{vap} = \frac{p_{vap}}{p} = \frac{\varphi \cdot p_{vap,sat}}{p} \tag{E.40}$$

For the mass vapour fraction in the mixture:

$$w_{vap} = \frac{p_{vap}}{p} \cdot \frac{R}{R_{vap}} = \varphi \cdot \frac{p_{vap,sat}}{p} \cdot \frac{R}{R_{vap}} \tag{E.41}$$

Where the behaviour of the gas (without vapour) and/or of the vapour deviates significantly from that of a mixture of two ideal gases with gas constants R_{gas} and R_{vap} in a practical case, precise precalculation is no longer possible without knowledge of the compressibility factor, Z, applicable to the mixture (from gas data programs or special measurements, for instance). Each individual gas may be dealt with separately using the relevant compressibility factor, Z_{gas}, in each case for partial pressure p_{gas} and Z_{vap} for partial pressure p_{vap} at mixture temperature T only as long as the molecules of the individual gases in the mixture do not exert too pronounced an interaction on each other (sufficient remoteness from a tendency to liquefy).

The following applies where the mass of the vapour is related not to the mass of the mixture but instead to that of the dry gas and is designated vapour fraction, x:

$$x = \frac{w_{vap}}{1 - w_{vap}} = \frac{\varphi \cdot p_{vap,sat}}{p - \varphi \cdot p_{vap,sat}} \cdot \frac{R_{gas}}{R_{vap}} = \frac{r_{vap}}{1 - r_{vap}} \cdot \frac{R_{gas}}{R_{vap}} \tag{E.42}$$

It follows therefrom:

$$w_{\text{vap}} = \frac{x}{x+1} \tag{E.43}$$

$$r_{\text{vap}} = \frac{\varphi \cdot p_{\text{vap,sat}}}{p} = \frac{x}{x + \dfrac{R_{\text{gas}}}{R_{\text{vap}}}} \tag{E.44}$$

The gas constant of the mixture, expressed by means of relative vapour saturation, φ, is

$$R = R_{\text{gas}} \cdot \frac{1}{1 - \dfrac{\varphi \cdot p_{\text{vap,sat}}}{p} \cdot \left(1 - \dfrac{R_{\text{gas}}}{R_{\text{vap}}}\right)} \tag{E.45}$$

or, expressed by means of the mole or volume fraction, r_v, of the vapour,

$$R = R_{\text{gas}} \cdot \left[\frac{1}{1 - r_{\text{vap}} \cdot \left(1 - \dfrac{R_{\text{gas}}}{R_{\text{vap}}}\right)} \right] \tag{E.46}$$

or, expressed by means of the fraction, x, of the vapour component,

$$R = R_{\text{gas}} \cdot \left[1 + \frac{x}{x+1} \cdot \left(\frac{R_{\text{vap}}}{R_{\text{gas}}} - 1 \right) \right] \tag{E.47}$$

For identical values of x, the gas constant of the mixture, R, is, according to Equation (E.47), not affected by pressure and temperature provided the dewpoint is not reached.

E.2.2.2 Mixture of air and water vapour (Humid air)

Equations (E.42) to (E.47) continue to supply the following numerical relationship at

R_{vap} = 461,52 J/(kg·K) for water vapour;

R_{air} = 287,1 J/(kg·K) for dry air.

The water vapour content (absolute humidity) related to the dry mass of air is

$$x_{\text{air}} = 0{,}622 \cdot \frac{\varphi \cdot p_{\text{vap,sat}}}{p - \varphi \cdot p_{\text{vap,sat}}} \tag{E.48}$$

The gas constant of humid air is

$$R_{\text{wet}} = R_{\text{air}} \cdot \frac{1}{1 - \dfrac{\varphi \cdot p_{\text{vap,sat}}}{p} \cdot 0{,}378} \tag{E.49}$$

or

$$R_{\text{wet}} = R_{\text{air}} \cdot \left(1 + \frac{x_{\text{air}}}{x_{\text{air}} + 1} \cdot 0{,}608 \right) \tag{E.50}$$

The relative humidity of air is

$$\varphi = \frac{p}{p_{\text{vap,sat}}} \cdot \frac{x_{\text{air}}}{x_{\text{air}} + 0{,}622} \tag{E.51}$$

The dew line is reached at $\varphi = 1$.

The isentropic exponent (see E.1.3) for humid air is

$$k_{\text{wet}} \approx k_{\text{dry}} \left(1 - 0{,}11 \cdot x_{\text{air}}\right) \tag{E.52}$$

The influence of x_{air} on k_{wet} can generally be ignored. However it is somewhat more significant for the exponent relevant for the conversion of a measured ratio of absolute temperatures to the relevant pressure ratio:

$$\frac{k_{\text{wet}}}{k_{\text{wet}} - 1} \approx \frac{k_{\text{dry}}}{k_{\text{dry}} - \left(1 + 0{,}11 \cdot x_{\text{air}}\right)} \tag{E.53}$$

E.3 Reference boundaries of the compressor

E.3.1 Definition

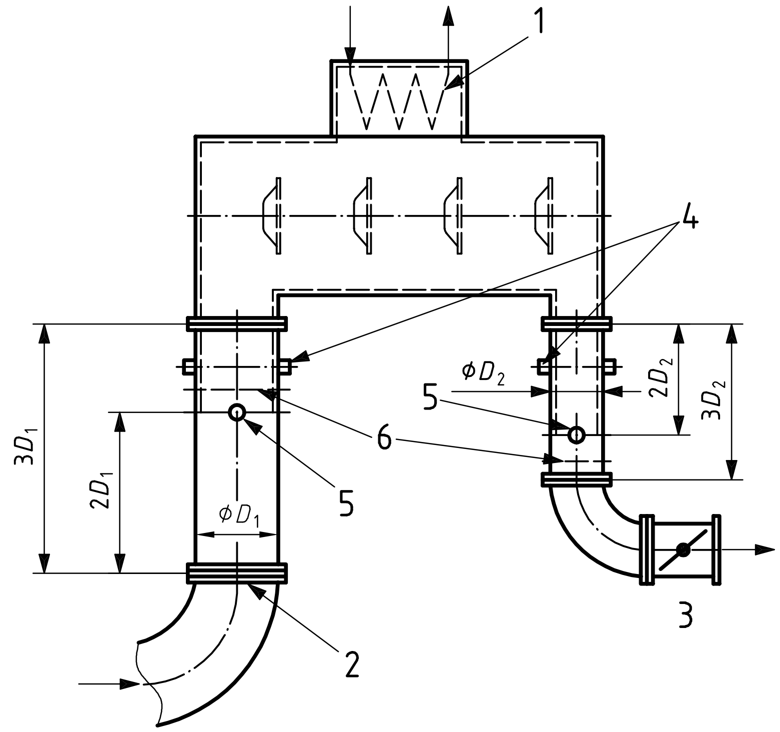
The compressor's thermodynamic reference boundary is formed by the inner surface area of the compressor casing in contact with the compressed fluid and, if applicable, by the surface area of the intercooler wetted by the compressed fluid and by the planes of the inlet and outlet areas of the suction and discharge pipes. The inner surface area of the casing of these boundaries can generally be regarded with good approximation as heat-impermeable (adiabatic). Where necessary, allowance can be made for heat losses on the casing exterior surfaces as detailed in 5.9.

The cooler surface area in contact with the compressed fluid shall be regarded as heat-permeable (diabatic).

In addition to the surface areas mentioned, the measuring planes for inlet and outlet state shall also be defined; these should, wherever possible, coincide with the inlet and outlet surface areas. Figures E.3 and E.4 show the thermodynamic reference boundaries for various types of compressors and a useful arrangement of the measuring points. The numerical values stated for the location of the measuring planes and the length of the measuring sockets should be regarded as guide figures.

Intercoolers are identified by the number of the preceding and succeeding stage (Roman numerals).

Temperatures can also be measured in the suction-side and discharge-side plenum. Pressure is measured via borings in the wall, i.e., it is the static pressure which is measured. The number of measuring stations per measuring plane depends on the diameter D of the suction and pressure lines. Where $D \leqslant 150$ mm, one pressure- and one temperature-measuring stations are required; where $D > 150$ mm, two pressure and two temperature stations are necessary. Where $D > 150$ mm, the pressure measuring stations shall be installed at intervals of 90°, and the temperature measuring station at intervals of 180°.



Key

1 intercooler integrated into casing

2 perforated plate, if necessary

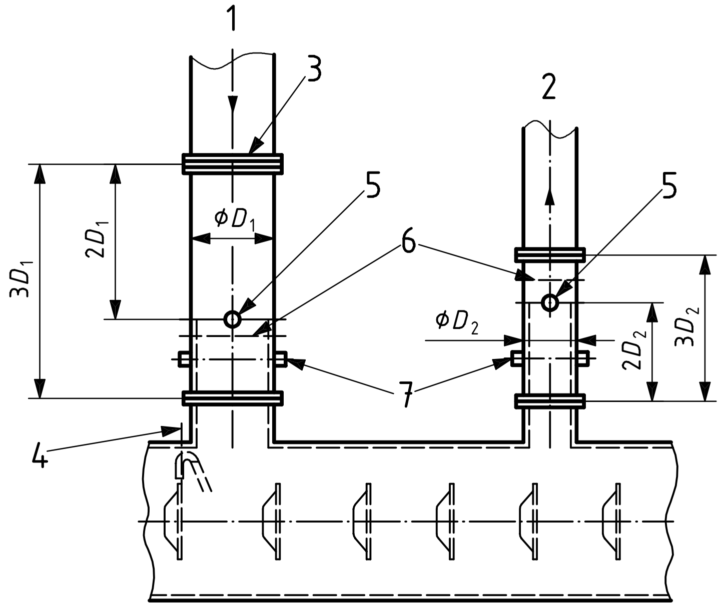
3 flow control by means of throttle

4 measuring stations for temperature, at least two spaced at an angle of 180°

5 measuring stations for static pressure, at least two spaced at an angle of 180°

6 measuring stations for dynamic or total pressure, at least two spaced at an angle of 90°, if relevant

Figure E.3 — Thermodynamic reference boundaries for a compressor, with useful arrangement of the measuring stations in the inlet and outlet nozzles — Thermodynamic reference boundary



Key

1 feed
2 extraction
3 perforated plate, if necessary
4 measuring stations for temperature, at least two on the circumference of the casting
5 measuring stations for static pressure, at least two spaced at an angle of 180°
6 measuring stations for dynamic or total pressure, at least two spaced at an angle of 90°, if relevant
7 measuring stations for temperature, at least two spaced at an angle of 180°

Figure E.4 — Additional measuring stations for side streams (e.g. admission, extraction) — Thermodynamic reference boundary

E.3.2 Inlet

The unobstructed cross-section in the plane of the suction flange is referred to as the inlet of the compressor.

Where a suction throttle control as part of the compressor is installed directly upstream of the suction flange, the unobstructed cross-section upstream of this throttle applies as the inlet plane.

All variables relating to this inlet plane are identified using the subscript 1.

Where the compressor is subdivided for the purpose of conversion into several sections or individual stages, the inlet area for each of these sections shall be analogously specified according to the measurement options.

The inlet concerned shall then be identified using the subscript 1 and, additionally, using the consecutive Roman numeral applicable to the section concerned.

E.3.3 Outlet

Analogously to the situation at the inlet, the unobstructed cross-section in the plane of the discharge flange of the outlet nozzle applies as the outlet.

All variables relating to this outlet plane shall be identified using the subscript 2. Where the compressor is subdivided into several sections or individual stages for the purpose of conversion, the outlet area for each of these sections shall be analogously specified according to the measurement options. In this case, the outlet concerned shall be identified using the subscript 2 and, additionally, using the consecutive Roman numeral for the section concerned.

E.4 Fluid flows

Fluid flows can be stated in the form of mass or volume flows.

E.4.1 Mass flow

The mass flow at the outlet from the compressor is

$$\dot{m}_2 = \dot{m}_1 + \sum \dot{m}_{\mathrm{L,in}} - \sum \dot{m}_{\mathrm{L,out}} + \sum \dot{m}_{\mathrm{cond,in}} - \sum \dot{m}_{\mathrm{cond,out}} + \sum \dot{m}_{\mathrm{side,in}} - \sum \dot{m}_{\mathrm{side,out}} \tag{E.54}$$

where

$\dot{m}_1$ is the mass flow at the inlet (suction pipe) to the compressor;

$\dot{m}_2$ is the mass flow at the outlet (discharge pipe) from the compressor;

$\sum \dot{m}_{\mathrm{L}}$ is the mass leakage flow;

$\sum \dot{m}_{\mathrm{cond}}$ is the mass flow of condensate precipitated or of liquid injected;

$\sum \dot{m}_{\mathrm{side}}$ are the subsidiary flows (sidestreams and extractions).

Process mass flow, $\dot{m}_{\mathrm{us}}$, shall be used for assessment of the compressor. The definitive side will be the inlet (in the case of suction compressors, for instance) or the outlet, depending on the function of the compressor. In cases of doubt, it should be specified in the contract.

The mass flow handled by the compressor shall, wherever possible, be measured on the process side. To determine the process mass flow, $\dot{m}_{\mathrm{us}}$, external leakages $\sum \dot{m}_{\mathrm{L}}$ through e.g. shaft seals have to be accounted for theoretically or by separate measurement. The same applies to liquid entering or leaving (e.g. condensate precipitated in intercoolers).

E.4.2 Volume flow

Volume flow is related to mass flow via the thermal equation of state:

$$\dot{V} = \dot{m} \frac{R \cdot Z \cdot T}{p} \tag{E.55}$$

The inlet volume flow $\dot{V}_1$ is the volume flow of the gas at inlet state. The process inlet volume flow $\dot{V}_{1,\mathrm{us}}$ is the inlet volume flow corrected by the external leakage losses and the amounts of condensate. Where no external leakage exists and no condensate is precipitated $\dot{V}_{1,\mathrm{us}} = \dot{V}_1$.

Vapour content, x, is included in the calculation for the volume flow of gas/water vapour mixtures.

The process moist inlet volume flow $\dot{V}_{1,\mathrm{us,wet}}$ is definitive for assessment of the compressor.

Where the discharge side is the process side and the inlet mass flow $\dot{m}_1$ is measured,

$$\dot{V}_{1,\mathrm{us,wet}} = \frac{(R \cdot Z)_{\mathrm{wet},1} \cdot T_1}{p_1} \cdot \left(\dot{m}_1 - \sum \dot{m}_{\mathrm{L,out}} + \sum \dot{m}_{\mathrm{L,in}} - \sum \dot{m}_{\mathrm{side,out}} + \sum \dot{m}_{\mathrm{side,in}} \right) \tag{E.56}$$

If, on the other hand, mass flow $\dot{m}_2$ is measured on the outlet side,

$$\dot{V}_{1,\mathrm{us,wet}} = \frac{(1 + x_1) \cdot (R \cdot Z)_{\mathrm{wet},1} \cdot T_1}{(1 + x_2) \cdot p_1} \cdot \dot{m}_2 \tag{E.57}$$

Where the suction side is the process side, and $\dot{m}_1$ is measured,

$$\dot{V}_{1,\text{us,wet}} = \frac{(R \cdot Z)_{\text{wet},1} \cdot T_1}{p_1} \cdot \dot{m}_1 \quad \text{(E.58)}$$

If, on the other hand, $\dot{m}_2$ is measured

$$\dot{V}_{1,\text{us,wet}} = \frac{(R \cdot Z)_{\text{wet},1} \cdot T_1}{p_1} \cdot \left(\dot{m}_2 + \sum \dot{m}_{\text{L,out}} - \sum \dot{m}_{\text{L,in}} - \sum \dot{m}_{\text{cond,in}} + \sum \dot{m}_{\text{cond,out}} + \sum \dot{m}_{\text{side,out}} - \sum \dot{m}_{\text{side,in}}\right) \quad \text{(E.59)}$$

Standard volume flow
The following relationship exists between mass flow and standard volume flow, $\dot{V}_n$, provided no constituents have precipitated out:

$$\dot{m} = \frac{p_n}{R \cdot Z_n \cdot T_n} \cdot \dot{V}_n \quad \text{(E.60)}$$

Standard state (subscript n) is constituted by pressure p_n = 0,101 325 MPa (1,013 25 bar) and temperature T_n = 273,15 K (t = 0 °C).

Preference should be given to statement in the form of mass flow.

The statement of a dry standard volume flow, in which the condensable constituent is not contained in Z_n and R in Equation (60), is also customary where condensable constituents (e.g. water in air) are present. The condensable constituents at the reference boundaries (see E.2.2) shall be added to the dry mass flow calculated in this way.

E.5 Change in thermodynamic state and specific compression work

E.5.1 General

The specific compression work and efficiency normally are calculated by computer programs. These programs, based on recognized equations of state, cover the gas behaviour along the compression path suitable from ideal to strong real gas behaviour.

In the range of Table E.1, the equations for approximately ideal gas behaviour can be used (see E.1.3.1).

Up to a certain extent outside the limits of Table E.1, these modified equations (by Schultz [12]) can be used.

E.5.2 Selection of the reference process

During its passage through the compressor, the fluid changes its state in accordance with the energy added, the flow losses, the process of conversion between static and kinetic energy and heat exchange with the environment. Since the actual change of state which occurs can be determined only with great difficulty, a thermodynamic reference process is taken as a basis. The reference process (generally subscripted "Pr") for compression is selected in such a way that it either approximates as closely as possible to the actual change of state (polytropic) or constitutes for reference purposes an idealized compression process (isothermal, isentropic).

Here, the same inlet state, p_1, T_1, as for the actual compressor, and the same discharge pressure, p_2, and, in the case of a polytropic reference process, the same discharge temperature, T_2, too, shall be used. Since the inlet and outlet velocities of the compressors are generally low and, additionally, are of approximately the same magnitude, calculation of change of state can be effected for the sake of simplicity using total variables of state.

It is rational, for the purpose of comparison of acceptance test results with guarantee data, to relate variables determined by the test to corresponding reference process variables. The reference process also serves as the basis for comparison of compressors of differing manufacture, type and size working under similar operating conditions.

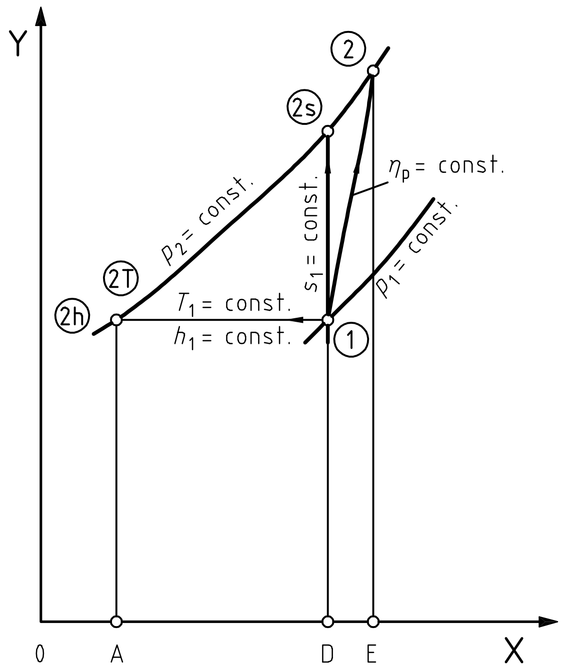
It is not possible to specify rigid rules for the selection of the reference process. The following reference processes, however, are generally applied to compressors for gases and gas/vapour mixtures:

a) isothermal (T = constant, subscript T) for single-stage cooled compressors and for multi-stage cooled compressors;

b) isentropic (s = constant, subscript s) for uncooled single-stage and multistage compressors, particularly such of moderate pressure ratio;

c) polytropic (polytropic ratio $\nu = 1/\eta_p$ = constant, subscript p)for uncooled compressors, in particular such with a high pressure ratio and in the case of real gas behaviour.

The changes of state are indicated in the T, s diagram which a gas undergoes given perfect (Figure E.5) and real (Figure E.6) gas behaviour when compressed using these reference processes. The state 1 at entry to the compressor is defined by two variables of state, p_1, T_1.



Key

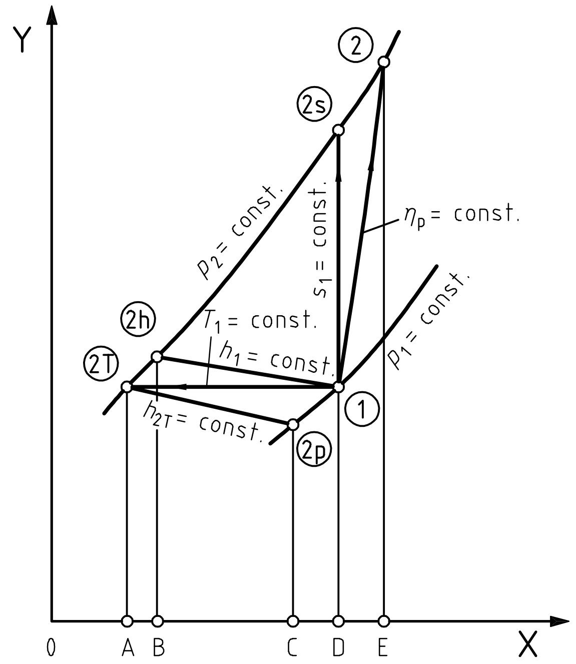
X entropy, s

Y temperature, t

Figure E.5 — Reference processes in the T, s diagram for ideal gas behaviour

Lines of constant temperature coincide with lines of constant enthalpy, since $c_p = f(T)$

Compression process	Process description	Change of state	Specific compression work = area in the T, s diagram
Isentropic	Reversible adiabatic	1 → 2s	y_s = A – D – 2s – 2T – A
Reference process polytropic	Irreversible adiabatic	1 → 2	y_p = A – D – 1 – 2 – 2T – A
Isothermal	Reversible diabatic	1 → 2T = 2h	y_T = A – D – 1 – 2T – A
Actual	Irreversible	1 → 2	Δh = A – E – 2 – 2T – A



Key

X entropy, s

Y temperature, t

Figure E.6 — Reference processes in the T, s diagram for real gas behaviour

Lines of constant enthalpy are inclined with respect to lines of constant temperature, since $c_p = f(T,p)$

Compression process	Process description	Change of state	Specific compression work = area in the T, s diagram
Isentropic	Reversible adiabatic	$1 \rightarrow 2s$	y_s = B – D – 2s – 2h – B
Reference process polytropic	Irreversible adiabatic	$1 \rightarrow 2$	y_p = B – D – 1 – 2 – 2h – B
Isothermal	Reversible diabatic	$1 \rightarrow 2T$	y_T = A – C – 2p – 1 – 2T – A
Actual	Irreversible	$1 \rightarrow 2$	Δh = B – E – 2 – 2h – B

The isobaric curve, p_2, is reached, according to the reference processes selected (at point 2 in the case of polytropic compression, at point 2s in the case of isentropic compression, and at point 2T in the case of isothermal compression). The relevant specific works of compression can be plotted as areas in the T, s diagram.

Generally, in every reference process the specific reversible work of compression is:

$$y_{\mathrm{Pr}} = \int_{p_1}^{p_2} v_{\mathrm{Pr}} \mathrm{d}p \qquad \text{(E.61)}$$

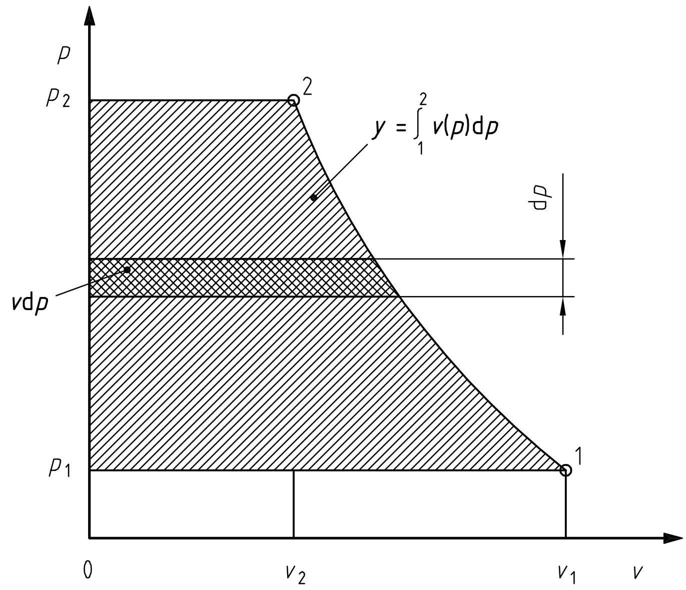


Figure E.7 — Specific compression work, area in the p, v diagram

y_{Pr} is dependent on the compression path $v_{Pr} = f(p)$, i.e. on the path and, therefore, on the reference process selected. The specific compression work y_T, y_s and y_p described in the following sections thus results.

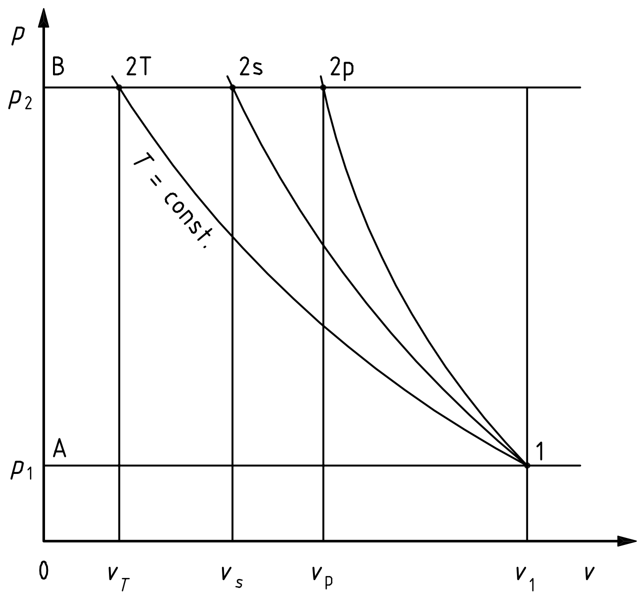


Figure E.8 — Reference processes in the p, v diagram valid for ideal and real gas behaviour

Compression process	Process description	Change of state	Specific work of compression = area in the p,v diagram, Figure E.7	Path of specific volume
Reference process:				
Isentropic	Reversible adiabatic	$1 \rightarrow 2s$	$y_s = A-1-2s-B-A$	$v(p)_s = v_1 \left[\frac{p_1}{p}\right]^{\left[\frac{1}{k_u}\right]}$
Polytropic	Irreversible adiabatic	$1 \rightarrow 2p$	$y_p = A-1-2p-B-A$	$v(p)_p = v_1 \left[\frac{p_1}{p}\right]^{\left[\frac{1}{n}\right]}$
Isothermal	Irreversible diabatic	$1 \rightarrow 2T$	$y_T = A-1-2T-B-A$	$v(p)_T = \frac{Z(p)}{Z_1} \frac{p_1}{p}$

E.5.3 Isothermal compression

Isothermal compression, given ideal gas behaviour, follows the law:

$$p \cdot v = p_1 \cdot v_1 = R \cdot T_1 = \text{constant} \tag{E.62}$$

Under real gas behaviour, specific isothermal work of compression is

$$y_T = R \cdot Z \cdot T_1 \cdot \ln \frac{p_2}{p_1} \tag{E.63}$$

Suitable mean values should be used throughout the reference process for compressibility factor, Z, and gas constant, R, e.g.

$$Z = \frac{Z_1 + Z_2}{2} \tag{E.64}$$

Isothermal compression is particularly suitable for comparison of cooled compressors of any type incorporating intercooling or jacket cooling.

To incorporate approximately the influence of the recooling temperatures on a result function (see 6.4.4.3.), an isothermal stage compression work can be defined, which takes into account the number of uncooled and cooled stages:

$$y_{T,Z} = \left(\frac{1}{z} + \frac{z-1}{z} \cdot \frac{T_{1,j,\text{av}}}{T_{1,\text{I}}} \right) \cdot R \cdot Z \cdot T_{1,\text{I}} \cdot \ln \frac{p_{2,Z}}{p_{1,\text{I}}} \tag{E.65}$$

E.5.4 Isentropic compression

The isentropic compression, given ideal gas behaviour, follows the law:

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_{2s}} \right)^k = \left(\frac{T_{2s}}{T_1} \right)^{\frac{k}{k-1}} \tag{E.66}$$

where k is the isentropic exponent.

A suitable mean value should be used in the range of the change of state, e.g.

$$k = \frac{k_1 + k_{2s}}{2} \tag{E.67}$$

For gas behaviour within the limits of Table E.1, the specific isentropic compression work is

$$y_s = \frac{k}{k-1} \cdot p_1 \cdot v_1 \cdot \left[\left(\frac{p_2}{p_1} \right)^{\frac{k-1}{k}} - 1 \right] \tag{E.68}$$

or

$$y_s = \frac{k}{k-1} \cdot R \cdot Z_1 \cdot T_1 \cdot \left[\left(\frac{p_2}{p_1} \right)^{\frac{k-1}{k}} - 1 \right] \tag{E.69}$$

The specific isentropic compression work, y_s, corresponds to the isentropic enthalpy difference:

$$y_s \equiv \Delta h_s = h_{2s} - h_1 \tag{E.70}$$

which is generally valid for ideal and real gas behaviour.

The enthalpy difference at constant entropy (Figure E.9) shall be determined from known gas data sources.

The change of isentropic enthalpy takes course on the line of constant entropy, s_1, from point 1 of compressor inlet state $h_1, s_1(p_1, T_1)$ to point $2s$ of compressor discharge state $h_{2s}, s_1(p_2, T_{2s})$.

T_{2s} is determined in the crossing point of the line of constant entropy s_1 and the isobaric curve p_2.

The actual, irreversible process runs from point 1 to point 2 of real compressor discharge state h_2, $s_2(p_2, T_2)$. These data are completely known by measurement procedure.

The enthalpy change, Δh, of the actual, irreversible process derives from:

$$y_s = h_{2,s} - h_1 \tag{E.71}$$

$$\Delta h = h_2 - h_1 \tag{E.72}$$

Each non-intercooled downstream stage draws in a greater volume flow at a higher temperature than in the case with isentropic compression in the upstream stages and has higher power. These differences in discharge temperatures become the greater, the higher the compression ratio p_2/p_1 and the lower the efficiency of the previous stage.

Assuming a constant isentropic stage efficiency, the isentropic efficiency of the compressor becomes the poorer, the greater p_2/p_1.

In turbo-machines, the extra energy consumption resulting from flow losses is converted directly into heat. Thus, for stages with no external heat transfer, the discharge temperature exceeds the isentropic temperature, $T_{2,s}$, at the same pressure.

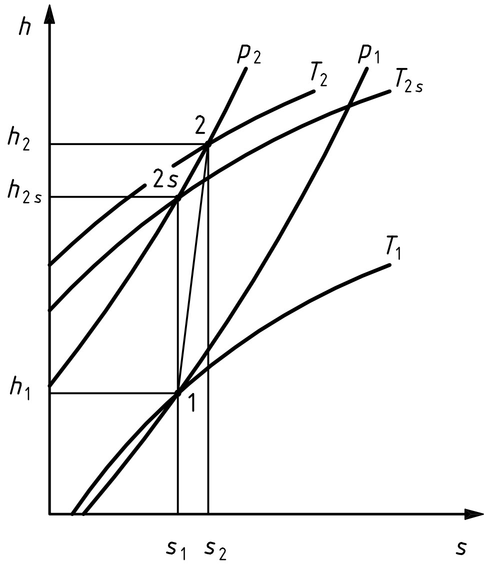


Figure E.9 — The isentropic reference process and actual process in the *h*, *s* diagram

E.5.5 Polytropic compression

The polytropic compression is the more suitable reference process for conversion of test results to guarantee conditions for uncooled sections in compressors with high pressure ratios and, in particular, in the case of real gas behaviour. For these, the ratio of differential enthalpy (originating generally from internal friction) to differential work of compression throughout the entire compression process is constant. The polytropic ratio

$$\nu = \frac{\mathrm{d}h}{v\mathrm{d}p} \tag{E.73}$$

is thus the same for all fractions of the compression ratio. For this reason, the entire enthalpy change and the entire specific work of compression are at the same ratio to one another. The inlet and outlet states of this reference process accord with the actual states. In the case of adiabatic compression (with no external heat transfer), the polytropic ratio is

$$\nu = \frac{1}{\eta_\mathrm{p}} \tag{E.74}$$

The specific polytropic work of compression

$$y_\mathrm{p} = \int_{p_1}^{p_2} v\mathrm{d}p \tag{E.75}$$

produces, with

$$p \cdot v^n = \text{constant} \tag{E.76}$$

for gas behaviour within the limits of Table E.1 and at constant polytropic exponent, n

$$y_p = p_1 \cdot v_1 \cdot \frac{n}{n-1} \cdot \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] \tag{E.77}$$

$$y_p = R \cdot Z_1 \cdot T_1 \cdot \frac{n}{n-1} \cdot \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] \tag{E.78}$$

$$y_p = R \cdot Z_m \cdot \left(T_2 - T_1 \right) \cdot \frac{\ln \frac{p_2}{p_1}}{\ln \frac{T_2}{T_1}} = \left(h_2 - h_1 \right) \cdot \eta_p \tag{E.79}$$

The polytropic discharge temperature, T_{2p}, is identical with the discharge temperature, T_2, actually measured for the compression process

$$\frac{T_2}{T_1} = \frac{T_{2p}}{T_1} = \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} \tag{E.80}$$

The polytropic exponent, n, derives in accordance with the ideal gas law from pressures p_1 and p_2 and temperatures T_1 and T_2:

$$n = \frac{\ln(\frac{p_2}{p_1})}{\ln(\frac{p_2}{p_1}) - \ln(\frac{T_2}{T_1})} \tag{E.81}$$

or, via polytropic efficiency η_p

$$\frac{n-1}{n} = \frac{\frac{k-1}{k}}{\eta_p} \tag{E.82}$$

or

$$n = \frac{1}{1 - \frac{k-1}{k \cdot \eta_p}} \tag{E.83}$$

Equations (E.81) and (E.83) can, given ideal gas behaviour, be applied both to uncooled compressors and to any stage of cooled compressors to determine the polytropic exponent, n.

The relationship between polytropic and isentropic efficiency is

$$\frac{\eta_\mathrm{p}}{\eta_s} = \frac{k-1}{k \cdot \eta_s} \cdot \frac{\ln \frac{p_2}{p_1}}{\ln \left\{ \frac{1}{\eta_s} \left[\left(\frac{p_2}{p_1} \right)^{\frac{k-1}{k}} - 1 \right] + 1 \right\}} \tag{E.84}$$

Given real gas behaviour, the determination of the polytropic exponent shall account for the change of the compressibility factor during the compression process.

$$n = \frac{\ln \frac{p_2}{p_1}}{\ln \frac{v_1}{v_2}} = \frac{\ln \frac{p_2}{p_1}}{\ln \frac{p_2}{p_1} - \ln \left[\frac{Z_2}{Z_1} \cdot \frac{T_2}{T_1} \right]} \tag{E.85}$$

The temperature ratio then derives from

$$\frac{T_2}{T_1} = \frac{Z_1}{Z_2} \cdot \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} \tag{E.86}$$

With the compressibility functions, in accordance with Reference [9]

$$n = \frac{1 + X}{\frac{1}{k_V} \cdot \left(\frac{1}{\eta_\mathrm{p}} + X \right) - Y \cdot \left(\frac{1}{\eta_\mathrm{p}} - 1 \right)} \tag{E.87}$$

with

$$\eta_\mathrm{p} = \frac{1}{\frac{k_V \cdot (1 + X)^2}{k_V \cdot Y - 1} \cdot \frac{\ln \frac{T_2}{T_1}}{\ln \frac{p_2}{p_1}} - X} \tag{E.88}$$

and

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1} \right)^m \tag{E.89}$$

with

$$m = \frac{\left(\frac{k_V \cdot Y - 1}{k_V} \right) \cdot \left(\frac{1}{\eta_\mathrm{p}} + X \right)}{(1 + X)^2} \tag{E.90}$$

Suitable mean values throughout the compression process from p_1 to p_2 should be used for the compressibility functions X and Y in Equation (E.87), e.g. $X = (X_1 + X_2)/2$ and $Y = (Y_1 + Y_2)/2$.

With the mean polytropic exponent, the polytropic work is

$$y_\mathrm{p} = f \cdot R \cdot Z_1 \cdot T_1 \cdot \frac{n}{n-1} \cdot \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] \tag{E.91}$$

with correction factor of Schultz [12]:

$$f = \frac{h_{2s} - h_1}{\frac{k_V}{k_V - 1} \cdot \left(p_2 \cdot v_{2s} - p_1 \cdot v_1 \right)} \tag{E.92}$$

and

$$k_V = \frac{\ln\left(\frac{p_2}{p_1} \right)}{\ln\left(\frac{v_1}{v_{2s}} \right)} \tag{E.93}$$

Under real gas behaviour the specific compression work $y_s = h_{2s} - h_1$ shall be known from gas data sources like tables or charts.

Under pronounced real gas behaviour (f diverging more greatly from 1,0), significant deviations from the polytropic compression at v = constant, which shall, for similarity reasons, be met (presupposing identical flow losses in each step for both the test and the conversion), may still occur when this calculation method is used.

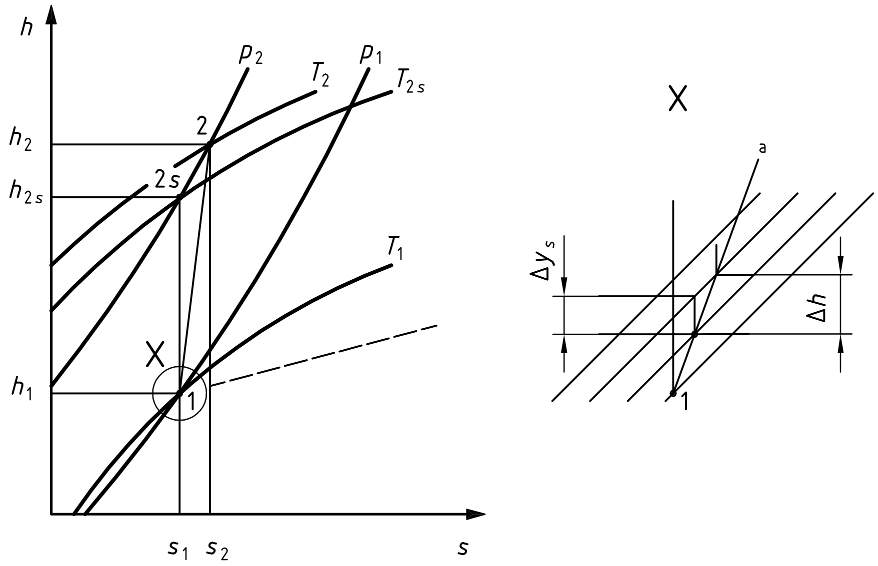
This polytropic compression can be approximated by means of a step-by-step isentropic compression with a large number of steps which are so small that

$$y_\mathrm{p} = \sum_1^2 \Delta y_s = \eta_\mathrm{p} \sum \Delta h \tag{E.94}$$

with $\Delta y_s \rightarrow \mathrm{d}y_s$, $\Delta h \rightarrow \mathrm{d}h$.

R.A. Huntington's [13] approximation method supplies a good approximation to this "reference polytropic compression" (see Figure E.10) even under extremely pronounced real gas behaviour.

Polytropic calculation methods which supply the same polytropic efficiencies and the same specific polytropic work of compression for the same compression process shall be used for evaluation of the test results and their conversion.



a Polytropic path.

Figure E.10 — Approximation for the reference polytropic compression

E.6 Power and efficiencies

E.6.1 Power terms

The following power terms are defined:

a) For the compressor:

1) compression power of the reference process, P_{pr}

2) gas power, P_i

3) mechanical power loss, P_{mech}

4) power at coupling, P_{cou}

b) For the driving machine:

1) driving machine power at coupling, $P_{cou,dr} = P_{cou}$

2) power input, $P_{in,dr}$

3) power losses, $P_{loss,dr}$

E.6.2 Compression power of the reference processes

Compression power

$$P_{Pr} = \dot{m}_{us} \cdot y_{Pr} \qquad (E.95)$$

Here, the total conditions at the inlet and outlet are used (see E.5.2).

E.6.3 Gas power

On the basis of the definitions in E.4, the gas power of a compressor section is derived as follows:

$$P_i = \dot{m}_2 \cdot (h_2 - h_1) + \sum(\dot{m}_L \Delta h_L) + \sum(\dot{m}_{cond} \Delta h_{cond}) + \dot{Q}_{amb} \quad (E.96)$$

where

Δh_L and Δh_{cond} are the enthalpy changes for the corresponding mass flow components;

$\dot{Q}_{amb}$ is the power component resulting from conduction, $\dot{Q}_W$, convection and radiation, $\dot{Q}_{rad}$, transmitted over the boundary defined in E.3.1: positive in the case of output, negative in the case of input.

E.6.4 Mechanical power losses

Mechanical losses occur in the bearings, in shaft seals and in gearboxes, lubricant pumps, etc., appurtenant to the compressor. They have no effect on the compression process.

E.6.5 Power at coupling

Power at coupling, P_{cou}, is the power input to the compressor measured at the compressor coupling.

Where a gearbox exists between the driving machine and the compressor, either the connection between the gearbox and the compressor or that between the gearbox and the driving machine may be specified as the compressor coupling, according to the agreement made in the contract.

Power at coupling is the sum of the compressor gas power, P_i, and mechanical power losses, P_{mech}.

It can be measured as follows:

a) directly, using torque and angular velocity at the coupling:

$$P_{cou} = M_t \omega \quad (E.97)$$

b) using the power absorbed by the driving machine, making allowance for the losses of the driving machine and of any intermediate gearbox:

$$P_{cou} = P_{in,dr} - P_{loss,dr} \quad (E.98)$$

c) using an energy balance:

$$P_{cou} = P_i + P_{mech} \quad (E.99)$$

E.7 Efficiencies of compressors

The efficiency, η_{Pr}, of a compressor is the ratio of the power, P_{Pr}, calculated for the reference process selected to the actual power. The process mentioned in E.5.2 can be used as reference process.

The power of the reference process selected in the numerator of the quotient is generally related to gas power, P_i, or power at coupling P_{cou} mentioned in E.6.3 Each of the powers calculated from the above-mentioned reference processes can be related to the gas power or power at coupling.

Gas efficiency:

$$\eta_{\mathrm{Pr,i}} = \frac{P_{\mathrm{Pr}}}{P_{\mathrm{i}}} \tag{E.100}$$

For isentropic process:

$$\eta_{s,\mathrm{i}} = \frac{P_s}{P_{\mathrm{i}}} = \frac{y_s}{h_2 - h_1} = \frac{h_{2s} - h_1}{h_2 - h_1} \tag{E.101}$$

For polytropic process:

$$\eta_{\mathrm{p,i}} = \frac{P_{\mathrm{p}}}{P_{\mathrm{i}}} = \frac{y_{\mathrm{p}}}{h_2 - h_1} \tag{E.102}$$

For isothermal process:

$$\eta_{\mathrm{T,i}} = \frac{P_{\mathrm{T}}}{P_{\mathrm{i}}} = \frac{y_{\mathrm{T}}}{h_2 - h_1 + q_{\mathrm{out}}} = \frac{\ln \dfrac{p_{2,Z}}{p_{1,I}}}{\displaystyle\sum_{j=1}^{Z} \frac{\left(R \cdot Z_1 \cdot T_1\right)_j}{\left(R \cdot Z_1 \cdot T_1\right)_{\mathrm{I}}} \cdot \frac{k}{k-1} \cdot \left[\left(\frac{p_2}{p_1}\right)_j^{\frac{k-1}{k}} - 1\right] \cdot \frac{1}{\eta_{s,j}}} \tag{E.103}$$

Efficiency at coupling:

$$\eta_{\mathrm{Pr,cou}} = \frac{P_{\mathrm{Pr}}}{P_{\mathrm{cou}}} = \frac{P_{\mathrm{Pr}}}{P_{\mathrm{i}}} \cdot \frac{P_{\mathrm{i}}}{P_{\mathrm{cou}}} = \eta_{\mathrm{Pr,i}} \cdot \eta_{\mathrm{mech}} \tag{E.104}$$

The ratio resulting from gas power and power at coupling is referred to as mechanical efficiency:

$$\eta_{\mathrm{mech}} = \frac{P_{\mathrm{i}}}{P_{\mathrm{i}} + P_{\mathrm{mech}}} = \frac{P_{\mathrm{i}}}{P_{\mathrm{cou}}} \tag{E.105}$$

E.8 Characteristic numbers

E.8.1 Significance of the characteristic numbers

Dimensionless characteristic numbers are used as criteria for similarity. These contain the decisive influencing factors, which are rendered dimensionless by means of suitable reference values. Here, the axial projection of the impeller area of the outer diameter, D, is defined as the impeller cross-section area and the velocity at the outer impeller diameter, D, as tip speed, u. The following characteristic numbers are of particular significance in compressor engineering.

E.8.2 Characteristic numbers

E.8.2.1 Flow coefficient, φ

The flow coefficient is a flow velocity formed from the inlet volume flow and an impeller cross-section area and rendered dimensionless by the tip speed of the impeller.

Therefore

$$\varphi = \frac{\dot{V}_1}{\frac{\pi}{4} \cdot D^2 \cdot u} \tag{E.106}$$

E.8.2.2 Head coefficient, ψ_{Pr}

The head coefficient is the specific compression work, y_{Pr}, of the reference process rendered dimensionless by the kinetic energy of tip speed, u. Therefore

$$\psi_{\text{Pr}} = \frac{y_{\text{Pr}}}{\frac{u^2}{2}} \tag{E.107}$$

In multi-stage machines and in sections, the specific compression work of the reference process for the entire multi-stage machine or section can be related to the kinetic energy of the tip speed of the first stage, i.e.,

$$\psi_{\text{Pr,I}-j} = \frac{y_{\text{Pr,I}-j}}{\frac{u_{\text{I}}^2}{2}} \tag{E.108}$$

E.8.2.3 Enthalpy coefficient, ψ_{i}

The enthalpy coefficient is the enthalpy rise rendered dimensionless by the kinetic energy of tip speed u.

$$\psi_{\text{i}} = \frac{\Delta h}{\frac{u^2}{2}} \tag{E.109}$$

E.8.2.4 Tip speed Mach number, Ma_{u}

Analogously to the Mach number as a ratio of a flow velocity to the sonic velocity of the gas in question referred to a specified state, the tip speed Mach number is formed by the ratio of a metal velocity, in this case, tip speed u, to the speed of sound of the fluid inlet state, i.e.,

$$Ma_{\text{u}} = \frac{u}{a_1} = \frac{u}{\sqrt{k_1 \cdot R \cdot Z_1 \cdot T_1}} \tag{E.110}$$

If the ratio of tip speed Mach numbers is formed for test (subscript "te") and guarantee (subscript "g"), for instance, u can be replaced by speed of rotation, N, due to the unchanged geometry of the same compressor.

$$\frac{Ma_{\text{u,te}}}{Ma_{\text{u,g}}} = \frac{\left(\frac{N}{\sqrt{k_1 \cdot R \cdot Z_1 \cdot T_1}}\right)_{\text{te}}}{\left(\frac{N}{\sqrt{k_1 \cdot R \cdot Z_1 \cdot T_1}}\right)_{\text{g}}} \tag{E.111}$$

Given real gas behaviour, $k_{V,1}$ should be used in place of k_1.

E.8.2.5 Reduced speed ratio, X_N

It has proved useful (see Reference [10]) to define a "reduced speed ratio" for conversion of test results to guarantee conditions as follows:

$$X_N = \frac{N_{\text{red,te}}}{N_{\text{red,g}}} = \frac{\left(\dfrac{N}{\sqrt{R \cdot Z_1 \cdot T_1}}\right)_{\text{te}}}{\left(\dfrac{N}{\sqrt{R \cdot Z_1 \cdot T_1}}\right)_{\text{g}}} \quad \text{(E.112)}$$

Where the isentropic exponent $k_V = k_{\text{g}}$, the following is precisely true:

$$X_N = \frac{N_{\text{red,te}}}{N_{\text{red,g}}} = \frac{Ma_{\text{u,te}}}{Ma_{\text{u,g}}} \quad \text{(E.113)}$$

E.8.2.6 Tip Reynolds number, Re_{u}

The Reynolds number is the ratio of the forces of inertia to the viscous forces in a flow, a characteristic flow velocity and a characteristic geometrical dimension of the body in contact with the flow appearing in the numerator. Analogously to the Reynolds number, the tip speed u is used in place of flow velocity in the formation of tip Reynolds number, and impeller outlet width b for the geometrical dimensions in a centrifugal compressor.

Kinematic viscosity ν in the denominator is referred to the inlet state of the stage, i.e., the tip Reynolds number is:

$$Re_{\text{u}} = \frac{u \cdot b}{\nu_1} \quad \text{(E.114)}$$

Kinematic viscosity ν can be calculated from dynamic viscosity η and density ρ:

$$\nu = \frac{\eta}{\rho} \quad \text{(E.115)}$$

The influence of the tip Reynolds number for test and guarantee for a given compressor is taken into account to determine the adjustment conditions and the conversion of test results to guarantee conditions (see 7.2.2.3, 7.2.4.1 and Annex F).

E.9 Performance curves and performance maps

The behaviour of a compressor referred to constant operating conditions and constant compressor geometry can conveniently be presented in the form of a performance curve.

Diagrams featuring absolute values are generally preferred for the guarantee comparison. The effective inlet volume flow $\dot{V}_{1,\text{us}}$ is used as the abscissa.

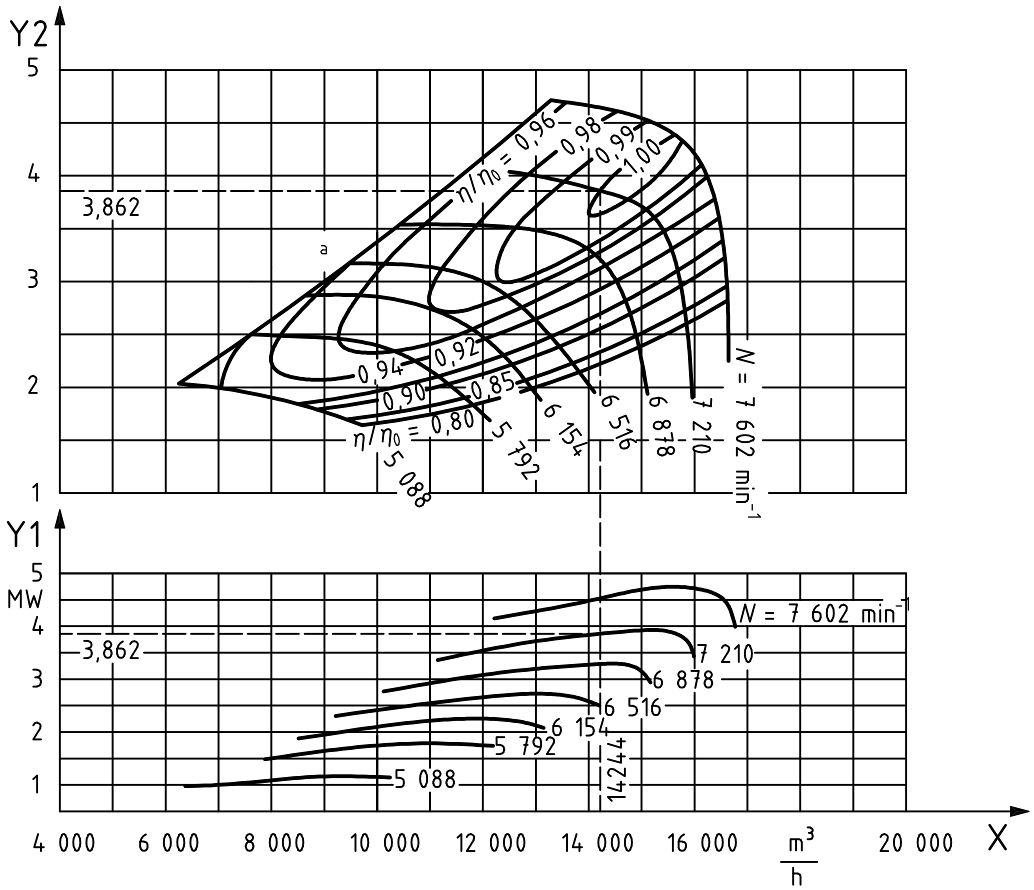
The dependent variables, e.g. pressure ratio $\Pi = p_2/p_1$, specific compression work y_{Pr}, efficiency η_{Pr}, power P_{cou}, etc., are plotted as ordinates. Allowance is generally made for the influence of a third variable as a parameter, resulting in the generation of a performance map. The operating conditions necessary for clear interpretation should always be stated as numerical values for all performance curves, either on the diagram itself or in the relevant key.

The following, primarily, can be selected as parameters for these performance curves:

a) speed of rotation, N;

b) position, δ, of the guide vanes, on the impeller inlet side (adjustable inlet guide vanes) or discharge side (adjustable diffusers) for centrifugal compressors, for instance.

The upper section of Figure E.11 shows the performance map of a compressor with speed of rotation, N, as parameter; the pressure ratio is plotted versus effective inlet volume flow. Lines of constant efficiency are also plotted. Power is plotted versus volume flow at identical speeds in the lower section of Figure E.11.



Key

X inlet volume flow, $\dot{V}_{1,us}$

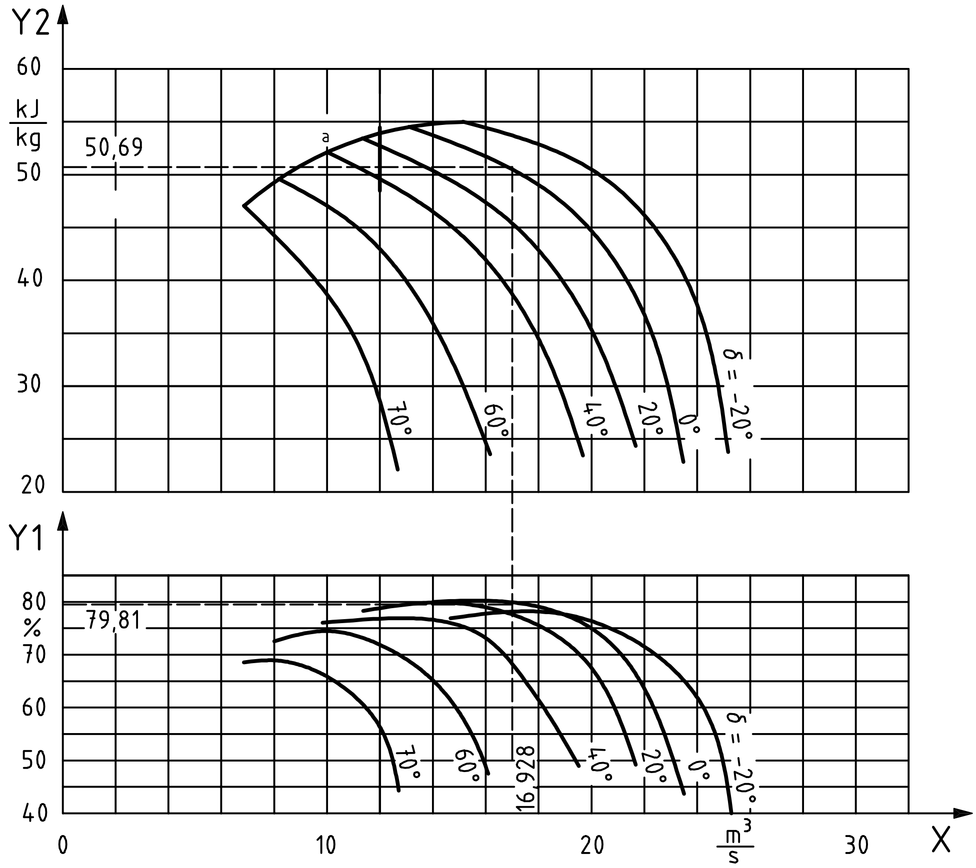
Y1 power at coupling, P_{cou}

Y2 pressure ratio $\frac{P_2}{P_1}$

a Surge limit.

Figure E.11 — Performance map of a compressor with variable speed

The upper section of Figure E.12 shows the performance map of a compressor using the inlet guide-vane angle as the parameter; specific polytropic compression work of the reference process, y_p, is plotted versus inlet volume flow. The relevant efficiencies are plotted in the lower section of Figure E.12.



Key

X inlet volume flow, $\dot{V}_{1,us}$

Y1 gas polytropic efficiency, η_p

Y2 specific polytropic compression work, y_p

[a] Surge limit.

Figure E.12 — Performance map of the section of a compressor with adjustable inlet guide vane

In compressors, pressure ratio, Π, rises from the design point as volume flow, $\dot{V}_{1,us}$, declines. The pressure ratio, Π, starts to decline again, however, when volume flow falls below a certain minimum. In conjunction with the storage capacity of the pipes, this results in unstable flow, characterized by periodic fluctuations in pressure and flow. The compressor is, then, operating in the unstable range that is separated on the performance curve from the stable range by the so-called surge point.

The surge limit is defined by the line connecting the surge points for various performance curves. The performance map is limited on its low inlet volume flow side by the surge line. The compressor's stable working range is limited by a line located to the right of the surge limit, defined by the surge control value opening.

Stable operation can be achieved by means of blowing-off or by-passing of a corresponding part flow even in cases where the required volume flow is in the unstable range below the surge limit.

Other limitations of the performance map depend, under given operating conditions (R, T_1, t_W, $\dot{m}_W$), on for example

a) the maximum drive power,

b) permissible maximum speed,

c) choke line, where appropriate,

d) permissible thrust-bearing load,

e) permissible maximum temperature,

f) safety-valve setting pressure.

Annex F
(informative)

Examples of acceptance test reports

F.1 General

The following examples illustrate the way in which the variables obtained in an acceptance test should be compared in a guarantee comparison with the guarantees contractually warranted by the supplier. In order to conduct a correct guarantee comparison, the schedule for the acceptance tests, the variables to be measured and the measuring methods to be used, and, possibly the gas data equations and evalutation systems and procedures should be agreed upon between the purchaser and the supplier and/or any third party also involved at a sufficiently early stage (if possible, during the actual contract negotiations) on the basis of the applicable standards and guidelines (see also 5.1).

F.2 Test examples

Overview

Test Example Number	Gas type	Speed adjustable	Number of sections	Cooling	Polytropic exponent	Effective inlet volume flow m³/h	Absolute pressure inlet MPa (bar)	Absolute pressure discharge MPa (bar)
1	Gas mixture	yes	1	—	$n_{te} \neq n_g$	4 002	15,75 (157,5)	18,7 (187)
2	Propane	yes	2	Feed	$n_{te} \neq n_g$	15 862	0,137 (1,373)	1,51 (15,1)
3	Air	no	4	Water	$n_{te} = n_g$	25 949	0,098 (0,98)	0,686 (6,86)
4	Air	no	3	Water	$n_{te} = n_g$	24 490	0,099 4 (0,994)	0,65 (6,5)
5	Natural gas	yes	1	—	$n_{te} \neq n_g$	4 930	4,9 (49)	7,5 (75)

F.2.1 Test example 1

Uncooled compressor, polytropic exponent $n_{te} \neq n_g$, speed adjustable (see 7.2.3, Table 2, case 3c).

F.2.1.1 Purpose of tests

Verification of guaranteed power for one guarantee point.

F.2.1.2 System configuration

Four-stage compressor for recycle gas, driven by steam turbine.

F.2.1.3 Guarantee conditions

	Symbol	Numerical value	Unit	Remarks
Inlet pressure	$p_{1,g}$	15,75 (157,5)	MPa (bar)	—
Inlet temperature	$t_{1,g}$	40	°C	—
Gas composition:				
Hydrogen	H_2	92,911 2	mol %	—
Water vapour	H_2O	0,04	mol %	—
Hydrogen sulfide	H_2S	0,880 1	mol %	—
Nitrogen	N_2	1,926 2	mol %	—
Methane	CH_4	2,630 3	mol %	—
Ethane	C_2H_6	0,220 0	mol %	—
Propane	C_3H_8	0,544 1	mol %	—
Iso-butane	C_4H_{10}	0,454 1	mol %	—
n-butane	C_4H_{10}	0,184 0	mol %	—
n-hexane	C_6H_{14}	0,210 0	mol %	—
Molar mass	M_g	4,000	kg/mol	—
Gas constant	R_g	2,078 8	kJ/(kg·K)	—

F.2.1.4 Object of guarantee

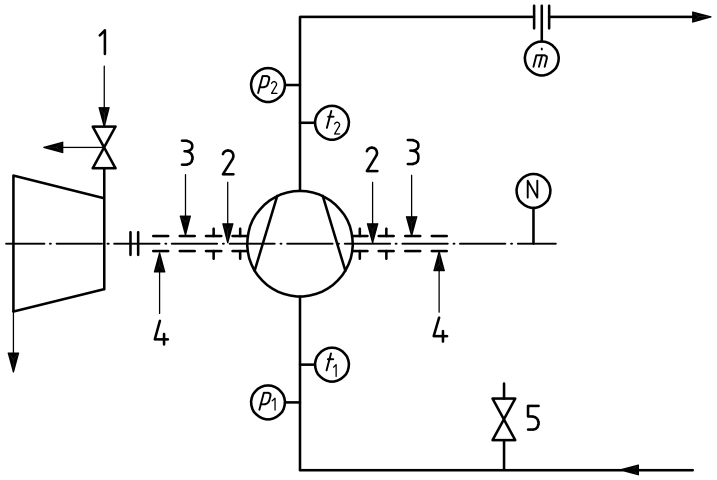
	Symbol	Numerical value	Unit	Remarks
Inlet volume flow	$\dot{V}_{1,us,g}$	1,111 8 4 002	m^3/s m^3/h	—
Discharge pressure	$p_{2,g}$	18,7 (187)	MPa (bar)	—
Power at coupling	$P_{cou,g}$	3 930	kW	—

F.2.1.5 Other design data

	Symbol	Numerical value	Unit	Remarks
Impeller outer diameter of first stage	D	336	mm	—
Outlet width of the first impeller	b	16,1	mm	—
Speed of rotation	N_{g}	13 850	1/min	—
Specific polytropic compression work	$y_{\text{p,g}}$	127,299	kJ/kg	Equation (E.92)
Discharge temperature	$t_{2,\text{g}}$	58,7	°C	—
Density	$\rho_{1,\text{g}}$	22,039 5	kg/m^3	—
Density	$\rho_{2,\text{g}}$	24,387 9	kg/m^3	—
Isentropic exponent, volume	$k_{V,\text{g}}$	1,541 9	—	Equation (E.94)
Specific isentropic enthalpy	$h_{2s,\text{g}}$	2 200,07	kJ/kg	as per RKS equation of state
Polytropic exponent	n_{g}	1,695 7	—	Equation (E.86)
Polytropic efficiency	$\eta_{\text{p,g}}$	0,821 5	—	as per Equation (E.104) $\eta_{\text{p}} = y_{\text{p}}/(h_2 - h_1)$

F.2.1.6 Test arrangement

Since it is not possible to perform the shop test using the gas of the guarantee conditions and with full gas power, a closed loop test with nitrogen is performed at reduced pressure and thereby reduced gas power. The adjustment conditions are calculated as shown in Annex A.



Key

1 test bench turbine
2 sealing gas
3 sealing oil
4 bearing oil
5 gas sample

Figure F.1 — Test arrangement and measuring stations

F.2.1.7 Setting conditions

	Symbol	Numerical value	Unit	Remarks
Inlet pressure	$p_{1,\mathrm{pr}}$	1,3 (13,0)	MPa (bar)	Specific test rig data, Equations 4 and 7 in Figure 3, applied analogously as per Equation (E.87); $p_{2,\mathrm{pr}}$; $t_{2,\mathrm{pr}}$ determined iteratively
Inlet temperature	$t_{1,\mathrm{pr}}$	25	°C	
Discharge pressure	$p_{2,\mathrm{pr}}$	1,545 (15,45)	MPa (bar)	
Discharge temperature	$t_{2,\mathrm{pr}}$	43,7	°C	
Molar mass	M_{pr}	28,016	kg/mol	Gas data calculated using the equation of state for real gases supplied by Lee-Kesler-Plöcker (LKP) This equation is suitable for N_2
Compressibility factor	$Z_{1,\mathrm{pr}}$	0,997 4	—	
Compressibility factor	$Z_{2,\mathrm{pr}}$	0,999 0	—	
Isentropic exponent, volume	$k_{V,1,\mathrm{pr}}$	1,417 9	—	
Isentropic exponent, volume	$k_{V,2,\mathrm{pr}}$	1,420 8	—	
Compressibility function	$X_{1,\mathrm{pr}}$	0,031 137	—	as per Equation (E.22)
Compressibility function	$X_{2,\mathrm{pr}}$	0,030 010	—	as per Equation (E.22)
Compressibility function	$Y_{1,\mathrm{pr}}$	1,002 186	—	as per Equation (E.23)
Compressibility function	$Y_{2,\mathrm{pr}}$	1,000 422	—	as per Equation (E.23)
Polytropic efficiency (Re-corrected)	$\eta_{\mathrm{p,pr}}$	0,813 3	—	as per Equation (C.2)
Polytropic exponent	n_{pr}	1,566 3	—	as per Equation (E.88)
Tolerance of the volume ratio	$\Delta\phi_{\mathrm{tol}}$	0,008 8	—	given: $>$ 0, in order that N is as high as possible: $<$ 0,01 in order to meet internal tolerance limit in test
Ratio of reduced speeds of rotation	$X_{N,\mathrm{tol}}$	1,002 1	—	from Annex A
Reynolds number influence				
on volume flow	$\varphi_{\mathrm{g}}/\varphi_{\mathrm{pr}}$	1,002 5	—	as per Equation (C.7)
on specific polytropic work	$\psi_{\mathrm{p,g}}/\psi_{\mathrm{p,pr}}$	1,005 0	—	as per Equation (C.5)
Matching speed	N_{pr}	4 878	1/min	from Annex A

F.2.1.8 Test conditions

	Symbol	Numerical value	Unit	Remarks
Speed	N_{te}	4 872,1	1/min	—
Inlet pressure	$p_{1,\mathrm{te}}$	1,325 (13,25)	MPa (bar)	—
Inlet temperature	$t_{1,\mathrm{te}}$	24,6	°C	—
Molar mass	M_{te}	28,016	kg/mol	—
Compressibility factor	$Z_{1,\mathrm{te}}$	0,997 3	—	—
Compressibility factor	$Z_{2,\mathrm{te}}$	0,998 9	—	—
Isentropic exponent, volume	$k_{V,1,\mathrm{te}}$	1,418 3	—	Gas data as per LKP equation
Isentropic exponent, volume	$k_{V,2,\mathrm{te}}$	1,421 4	—	

F.2.1.9 Testing of volume flow ratio during test

	Symbol	Numerical value	Unit	Remarks
Polytropic efficiency	$\eta_{\mathrm{p,te}}$	0,839 4	—	—
Reynolds-number-corrected efficiency	$\eta_{\mathrm{p,co}}$	0,846 3	—	as per Equation (C.2)
Polytropic exponent	n_{g}	1,695 7	—	as per Equation (E.86)
Polytropic exponent	n_{te}	1,540 2	—	as per Equation (E.86)
Reynolds number correction on specific polytropic compression work	$\psi_{\mathrm{p,g}}/\psi_{\mathrm{p,te}}$	1,004 1	—	as per Equation (C.5)
Ratio of reduced speeds of rotation	X_N	1,001 7	—	Equation (2)
Deviation of the volume ratio	$\Delta\phi$	0,010 8	—	as per Figure 6

F.2.1.10 Test results

	Symbol	Numerical value	Unit	Remarks
Gas constant	R_{te}	296,77	J/(kg·K)	—
Speed of rotation	N_{te}	4 872	1/min	—
Mass flow discharge side	$\dot{m}_{2,\mathrm{te}}$	6,006	kg/s	Measured as per ISO 5167; effective mass flow
Mass flow, leaks	$\sum \dot{m}_{\mathrm{L,te}}$	0,126	kg/s	Balance piston and gas seals

F.2.1.11 Inlet and discharge state

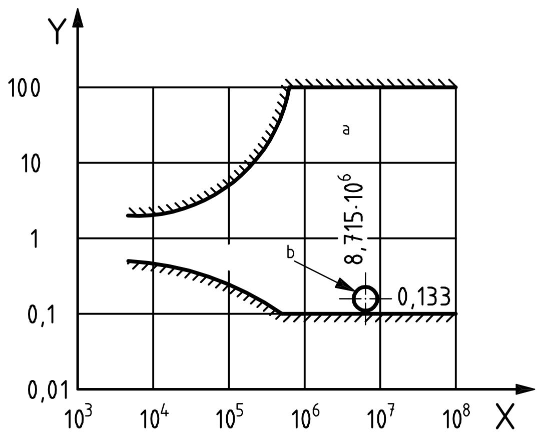
	Symbol	Numerical value	Unit	Remarks
Inlet pressure	$p_{1,\text{te}}$	1,325 (13,25)	MPa (bar)	—
Inlet temperature	$t_{1,\text{te}}$	24,6	°C	—
Inlet density	$\rho_{1,\text{te}}$	15,035	kg/m^3	—
Inlet spec. enthalpy	$h_{1,\text{te}}$	306,209	kJ/kg	from LKP-equation
Discharge pressure	$p_{2,\text{te}}$	1,575 (15,75)	MPa (bar)	—
Discharge temperature	$t_{2,\text{te}}$	42,7	°C	—
Discharge density	$\rho_{2,\text{te}}$	16,821	kg/m^3	—
Discharge spec. enthalpy	$h_{2,\text{te}}$	324,915	kJ/kg	from LKP-equation

F.2.1.12 Calculation results

	Symbol	Numerical value	Unit	Remarks
Inlet volume flow	$\dot{V}_{1,\text{us,te}}$	0,399 5	m^3/s	—
Pressure ratio	Π_{te}	1,188 7	—	—
Polytropic efficiency	$\eta_{\text{p,te}}$	0,839 4	—	as per Equation (E.104) $\eta_\text{p} = y_\text{p} / (h_2 - h_1)$
Specific polytropic compression work	$y_{\text{p,te}}$	15,702 6	kJ/kg	Equation (E.92)
Correction factor	f_{te}	0,999 9	—	Equation (E.93)
Gas power	$P_{\text{i,te}}$	114,71	kW	as per Equation (E.97) $P_\text{i} = \left(\dot{m}_2 + \sum \dot{m}_\text{L}\right) y_\text{p} / \eta_\text{p}$
Radiation losses	$\dot{Q}_{\text{rad,te}}$	0,71	kW	Equation (13)
Bearing losses	$P_{\text{bearing,te}}$	0,66	kW	from measurement of oil
Oil seal losses	$P_{\text{seal,te}}$	7,74	kW	mass flow and Δt_oil
Power at coupling	$P_{\text{cou,te}}$	129,82	kW	—
Isentropic exponent, volume	$k_{V,\text{te}}$	1,420 8	—	Equation (E.94)

F.2.1.13 Calculation of the influence of Reynolds number (Figure F.2)

	Symbol	Numerical value	Unit	Remarks
Speed of rotation	N_{g}	13 850	1/min	—
Speed of rotation	N_{te}	4 872	1/min	—
Impeller diameter of 1st stage	D	336	mm	—
Impeller outlet width of 1st stage	b	16,1	mm	—
Average roughness	Ra	2,5	µm	—
Kinematic viscosity	ν_{g}	$4{,}5 \cdot 10^{-7}$	m^2/s	from gas data calculation
Kinematic viscosity	ν_{te}	$1{,}195 \cdot 10^{-6}$	m^2/s	—
Reynolds number	$Re_{\mathrm{u,g}}$	$8{,}715 \cdot 10^{6}$	—	—
Reynolds number	$Re_{\mathrm{u,te}}$	$1{,}155 \cdot 10^{6}$	—	—
Reynolds number ratio	$Re_{\mathrm{u,te}} / Re_{\mathrm{u,g}}$	0,133	—	—
Reynolds number influence on polytropic efficiency	$\eta_{\mathrm{p,co}} / \eta_{\mathrm{p,te}}$	1,008 2	—	Equation (C.2)
Polytropoic efficiency	$\eta_{\mathrm{p,co}}$	0,846 3	—	—
Polytropic efficiency	$\eta_{\mathrm{p,te}}$	0,839 4	—	—
Reynolds number influence on polytropic specific work	$\psi_{\mathrm{p,co}} / \psi_{\mathrm{p,te}}$	1,004 1	—	Equation (C.5)
Reynolds number influence on volume flow	$\varphi_{\mathrm{co}} / \varphi_{\mathrm{te}}$	1,002 1	—	Equation (C.7)



Key

X Reynolds number under guarantee conditions, $Re_{\mathrm{u,g}}$

Y Reynolds number ratio, $\dfrac{Re_{\mathrm{u,te}}}{Re_{\mathrm{u,g}}}$

a Permissible range.

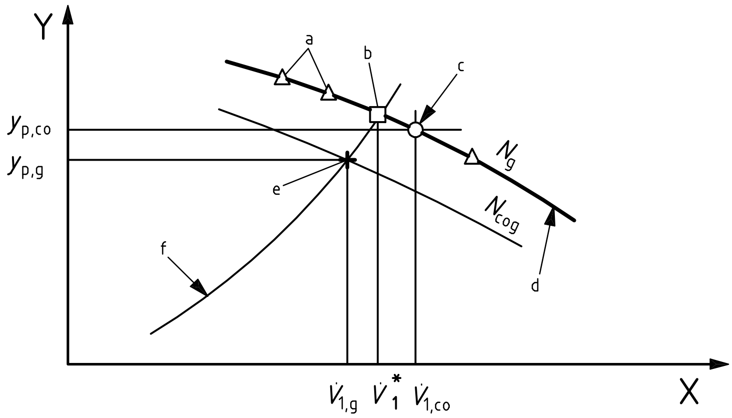
b Measuring point.

Figure F.2 — Checking of the permissibility of Reynolds number correction

F.2.1.14 Conversion to guarantee conditions

Conversion to guarantee conditions at an efficiency, $\eta_{p,co}$, maintained constant is effected here as an iterative procedure. $p_{2,co}$ and $t_{2,co}$ are firstly assumed and the gas data calculated from the corresponding equation of state (in this case, RKS). Improved values for $p_{2,co}$ and $t_{2,co}$ are obtained via $y_{p,co}$. The calculation procedure is repeated until a sufficiently accurate level of accordance is achieved. Each calculation operation includes renewed determination of gas data:

	Symbol	Numerical value	Unit	Remarks
Speed of rotation	N_g	13 850	1/min	—
Volume flow	$\dot{V}_{1,us,co}$	1,138 0	m^3/s	Figure 3
Discharge mass flow	$\dot{m}_{2,co}$	25,081	kg/s	—
Mass flow, leakage	$\sum \dot{m}_{L,co}$	0,460	kg/s	Conversion using labyrinth flow equation at constant flow coefficient
Polytropic efficiency	$\eta_{p,co}$	0,846 3	—	Equation (C.2)
Specific polytropic compression work	$y_{p,co}$	127,42	kJ/kg	Figure 3
Pressure ratio	Π_{co}	1,187 7	—	Figure 3
Discharge pressure	$p_{2,co}$	18,705 (187,05)	MPa (bar)	—
Discharge temperature from temperature measurement	$t_{2,\Delta t,co}$	58,07	°C	Equation (E.90)
Polytropic exponent	n_{co}	1,668 4	—	Equation (E.86)
Temperature exponent	m_{co}	0,326 8	—	Equation (E.91)
Isentropic exponent, volume	$k_{V,co}$	1,542 8	—	Equation (E.94)
Compressibility factor	$Z_{2,co}$	1,111 9	—	from RKS equation
Gas power from temperature measurement	$P_{i,\Delta t,co}$	3 845,5	kW	Equation (E.97)
Radiation losses	$\dot{Q}_{rad,co}$	23,8	kW	Equation (45)
Corrected gas power	$P_{i,co}$	3 869,3	kW	Equation (46)
Corrected discharge temperature	$t_{2,co}$	58,18	°C	Equation (47)
Bearing losses	$P_{bearing,co}$	34,9	kW	empirical
Oil seal losses	$P_{seal,co}$	22,0	kW	—
Power at coupling	$P_{cou,co}$	3 926,2	kW	Equation (E.100)



Key

X suction volume flow, $\dot{V}_1$

Y specific polytropic work of compression, y_p

[a] Further performance curve point.

[b] Auxiliary point.

[c] Characteristic curve point used for guarantee comparison.

[d] Characteristic curve converted to accord with guarantee conditions.

[e] Guarantee point.

[f] Parabola $y_p = c\dot{V}_1^2$.

Figure F.3 — Guarantee comparison

$\dot{V}_1^*$ (the auxiliary point) is determined by graphic means from the point of intersection of the converted performance curve with the parabola which passes through the guarantee point. The speed N_{cog} of the performance curve passing through the guarantee point results to

$$N_{cog} = N_g \cdot \dot{V}_{1g} / \dot{V}_1^* \quad \text{(F.1)}$$

F.2.1.15 Guarantee comparison (Figure F.3)

	Symbol	Numerical value	Unit	Remarks
Specific polytropic compression work	$y_{p,g}$	127,299	kJ/kg	
Inlet volume flow	$\dot{V}_{1,us,g}$	1,111 8 4 002	m^3/s m^3/h	
Power at coupling	$P_{cou,g}$	3 930	kW	
Power at coupling, necessary	$P_{cou,cog}$	3 832	kW	Equation (34)
Power at coupling, deviation	ΔP_{cou}	− 2,5	%	
Speed of rotation, necessary	N_{cog}	1 3774	1/min	
Speed of rotation, deviation	ΔN_g	− 0,5	%	

F.2.2 Text example 2

F.2.2.1 Uncooled compressor with sidestream admission, polytropic exponent $n_{te} \neq n_g$, speed adjustable (see 7.2.3, Table 3, case 4d)

The test conditions deviate from those of the guarantee. Changing the speed makes it possible to achieve volume flow ratios identical to those of the guarantee condition. It is to be established whether the test can be performed at the same test speed for both sections.

F.2.2.2 Purpose of the tests

Verification of guaranteed power at coupling at the guarantee point and achievement of specified intermediate pressure within a tolerance band of 0 % to 4 %.

F.2.2.3 System configuration

Four-stage compressor with sidestream downstream of the second stage for propane, driven by a steam turbine.

F.2.2.4 Guarantee conditions

	Symbol	Numerical value	Unit
Inlet pressure	$p_{1,g}$	0,137 3 (1,373)	MPa (bar)
Inlet temperature	$t_{1,g}$	– 32,3	°C
Gas type	C_3H_8	—	—
Gas constant	R_g	188,6	J/(kg·K)
Sidestream temperature	$t_{side,g}$	– 3	°C
Sidestream mass flow	$\dot{m}_{side,g}$	10,131	kg/s

In this example, inlet temperature, $t_{1,g}$, is not the temperature in the compressor inlet nozzle; rather, it is the temperature decisive for conversion, i.e., at the inlet to the first impeller. The difference derives from the temperature increase caused by leakage mass flow, which is returned to the compressor suction side via the balance piston.

The volume flow of section I and section II also includes the mass flow leakage recirculating via the balance line.

F.2.2.5 Object of the guarantee

	Symbol	Numerical value	Unit
Inlet volume flow	$\dot{V}_{1,g}$	4,406	m^3/s
Sidestream pressure	$p_{side,g}$	0,426 7 (4,267)	MPa (bar)
Discharge pressure	$p_{2,g}$	1,51 (15,1)	MPa (bar)
Power at the compressor coupling	P_{cou}	2 909	kW

F.2.2.6 Other design data

	Symbol	Numerical value	Unit
Speed of rotation of the compressor [a]	N_g	8 261	1/min
Impeller diameter of the 1st impeller	D_I D_{II}	500 575	mm mm
Outlet width of the 1st impeller	b_I b_{II}	34 14	mm mm
Average roughness	Ra_I Ra_{II}	2,8 3,0	µm µm
Temperature at the inlet nozzle [b]	$t_{nozzle,g}$	– 34,7	°C
Circulating mass flow via the balance line [b]	$\dot{m}_{L,g}$	0,431	kg/s
Discharge temperature [b]	$t_{2,g}$	68,4	°C

[a] Where the driving machine has an adjustable speed, speed is not an object of the guarantee; design speed is indicated using the subscript "g".

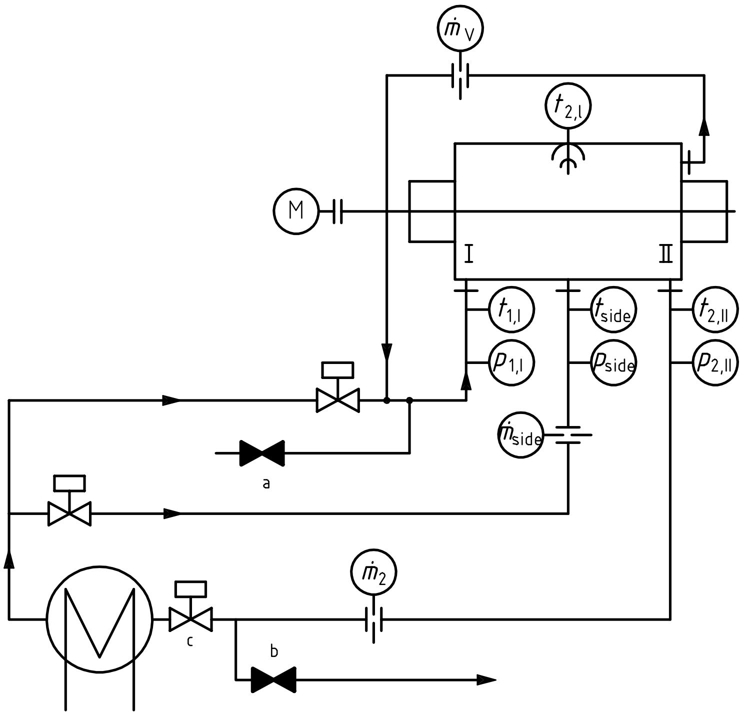
[b] These data are necessary for conversion of the test results to accord with the guarantee conditions.

F.2.2.7 Test arrangement

Since it is not possible to examine the compressor on the supplier's test rig using the original gas, the tests are performed using a substitute gas in a closed loop.

IMPORTANT — This example was calculated with the R12 test gas used in the past. For environmental reasons, however, this gas cannot be used any more. The example is intended to show the procedure for testing a sidestream compressor.

The test arrangement and arrangement of the measuring points can be seen in Figure F.4.



a Gas supply.

b Gas analysis.

c Throttle valve.

Figure F.4 — Test arrangement

F.2.2.8 Setting conditions

	Symbol	Numerical value	Unit
Gas type	CF_2Cl_2 (R12)	—	—
Gas constant	R_{pr}	68,8	J/(kg·K)
Section I:			
Inlet pressure	$p_{1,I,pr}$	0,08 (0,8)	MPa (bar)
Inlet temperature	$t_{1,I,pr}$	40,0	°C
Discharge temperature	$t_{2,I,pr}$	91,2	°C
Section II [a]:			
Inlet pressure [a]	$p_{1,II,pr}$	0,25 (2,5)	MPa (bar)
Inlet temperature [a]	$t_{1,II,pr}$	68,5	°C
Discharge pressure	$p_{2,II,pr}$	0,915 5 (9,155)	MPa (bar)
Discharge temperature	$t_{2,II,pr}$	133,5	°C
Speed of rotation	N_{pr}	5 795	1/min

a Determination of the test conditions for section II and the test speed is made iteratively, since the discharge temperature from section I, in particular, affects the inlet temperature of section II.

F.2.2.9 Setting conditions

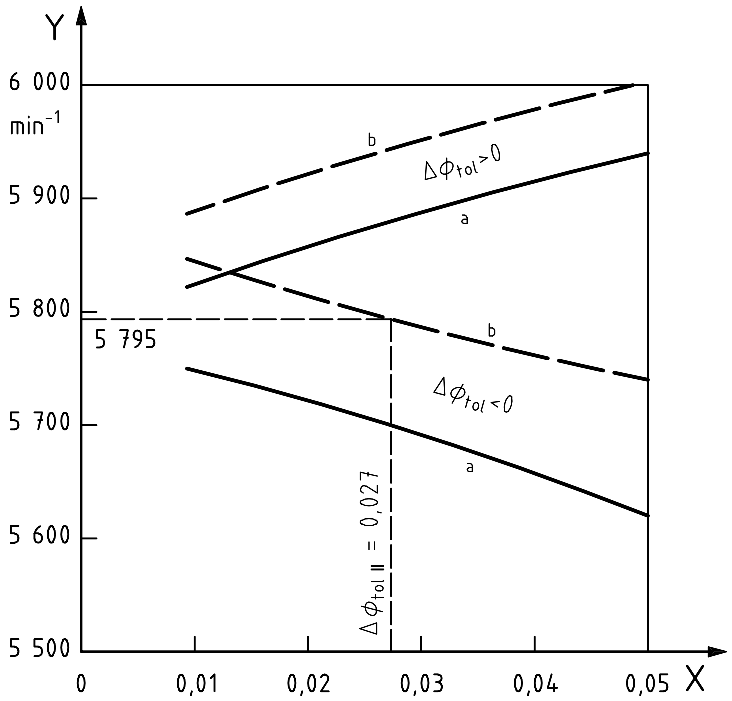
Calculation of the setting conditions was done using Annex A. The relevant setting speeds for both sections were determined for the values $|\Delta\phi|_{tol}$ = 0,01, 0,025 and 0,05.

Calculations variables, which in some cases derived only as a result of the iteration, were as follows:

	Sections	
	I	**II**
$\left(\frac{p_2}{p_1}\right)_g$	3,107 8	3,539 3
n_g	1,146 2	1,074 6
n_{pr}	1,140 3	1,116 5
$\psi_{p,te}/\psi_{p,g}$	0,996 32	0,998 75
$\sqrt{\frac{(RZ_1T_1)_{pr}}{(RZ_1T_1)_g}}$	0,701 6	0,688 1

Figure F.5 shows that adherence to the inner tolerance limit for both sections is not possible. N_{te} = 5 795 rev/min was selected as test speed. This was intended to ensure that the test can be performed within the inner tolerance limit, at least for the first section, even given slight deviations in test conditions.

The deviation $\Delta\phi_{tol}$ for the section II is then – 2,7 %.



Key

X ratio of volume flow ratios, $|\Delta\phi_{tol}|$

Y speed, N_{te}

[a] Section I.

[b] Section II.

Figure F.5 — Determination of test speed

An approximation of the ranges in which the internal tolerance limit is adhered to for each section could be achieved by increasing the inlet temperature of the first section and/or lowering the inlet temperature of the second section. The test apparatus described above, however, does not permit this option, since the inlet temperature cannot be adjusted separately.

F.2.2.10 Calculations

Calculation of the gas properties has been accomplished using the BWRS equation.

Original gas, section I $p_{1,g}$ = 0,137 3 MPa (1,373 bar) $p_{2,g}$ = 0,426 7 MPa (4,267 bar)

$t_{1,g}$ = – 32,3 °C $t_{2,g}$ = 15,8 °C

Determination of the Reynolds numbers for section I:

$$Re_u = \frac{u \cdot b}{\nu_1}$$

Original gas: u_g = 216 m/s; $\nu_{1,g} = 2{,}113 \cdot 10^{-6}$ m²/s, $Re_{u,g} = 3{,}476 \cdot 10^6$

Test gas: u_{pr} = 151,7 m/s; $\nu_{1,pr} = 3{,}213 \cdot 10^{-6}$ m²/s, $Re_{u,pr} = 1{,}605 \cdot 10^6$

Further calculation of Reynolds number correction was effected as detailed in Annex C:

$$\frac{1-\eta_{p,g}}{1-\eta_{p,pr}} = \frac{0{,}3 \cdot \lambda_\infty + 0{,}7 \cdot \lambda_g}{0{,}3 \cdot \lambda_\infty + 0{,}7 \cdot \lambda_{pr}} \quad \text{(F.2)}$$

with

$$\frac{1}{\sqrt{\lambda_\infty}} = 1{,}74 - 2 \cdot \log_{10}\left[2 \cdot \frac{Ra}{b}\right] \quad \text{(F.3)}$$

$$\lambda_\infty = 1{,}155 \cdot 10^{-2}$$

$$\frac{1}{\sqrt{\lambda_g}} = 1{,}74 - 2 \cdot \log_{10}\left[2 \cdot \frac{Ra}{b} + \frac{18{,}7}{Re_{u,g} \cdot \sqrt{\lambda_g}}\right] \quad \text{(F.4)}$$

$$\lambda_g = 1{,}212 \cdot 10^{-2}$$

$$\frac{1}{\sqrt{\lambda_{pr}}} = 1{,}74 - 2 \cdot \log_{10}\left[2 \cdot \frac{Ra}{b} + \frac{18{,}7}{Re_{u,pr} \cdot \sqrt{\lambda_{pr}}}\right] \quad \text{(F.5)}$$

$$\lambda_{pr} = 1{,}268 \cdot 10^{-2}$$

$$\frac{1-\eta_{p,g}}{1-\eta_{p,pr}} = 0{,}968\ 2 \quad \text{(F.6)}$$

With $\eta_{p,g}$ = 0,817 0, the result is $\frac{\eta_{p,pr}}{\eta_{p,g}} = 0{,}99264$

$$\frac{\psi_{p,pr}}{\psi_{p,g}} = 0{,}5 + 0{,}5 \cdot \frac{\eta_{p,pr}}{\eta_{p,g}} = 0{,}996\ 32 \quad \text{(F.7)}$$

Section II:

$Re_{u,g} = 3{,}915 \cdot 10^6$

$Re_{u,pr} = 2{,}178 \cdot 10^6$

$\lambda_\infty = 1{,}374 \cdot 10^{-2}$

$\lambda_g = 1{,}401 \cdot 10^{-2}$

$\lambda_{pr} = 1{,}421 \cdot 10^{-2}$

$$\frac{1-\eta_{p,g}}{1-\eta_{p,pr}} = 0{,}990\ 2 \text{ with } \eta_{p,g} = 0{,}792 \quad \text{(F.8)}$$

$$\frac{\eta_{p,pr}}{\eta_{p,g}} = 0{,}997\ 5 \quad \text{(F.9)}$$

$$\frac{\psi_{p,pr}}{\psi_{p,g}} = 0{,}998\ 75 \quad \text{(F.10)}$$

F.2.2.11 Test results

	Symbol	Numerical value			Unit
Test number		1	2	3	
Test period	—	*xx.xx.xx x.xx*	*xx.xx.xx x.xx*	*xx.xx.xx x.xx*	—
Speed of rotation	N_{te}	5 795	5 795	5 795	1/min
Pressures:					
Inlet pressure, section I	$p_{1,I,te}$	0,080 66 (0,806 6)	0,080 69 (0,806 9)	0,079 54 (0,795 4)	MPa (bar)
Inlet pressure, section II	$p_{side} = p_{2,I,te} = p_{1,II,te}$	0,225 99 (2,259 9)	0,248 91 (2,489 1)	0,259 37 (2,593 7)	MPa (bar)
Discharge pressure, section II	$p_{2,II,te}$	0,710 08 (7,100 8)	0,876 58 (8,765 8)	0,924 14 (9,241 4)	MPa (bar)
Temperatures:					
Inlet temperature, section I	$t_{1,I,te}$	38,9	39,1	39,4	°C
Discharge temp., section I	$t_{2,I,te}$	86,9	89,6	91,6	°C
Sidestream temp., section II	$t_{side,te}$	43,0	43,3	43,9	°C
Discharge temp., section II	$t_{2,II,te}$	130,2	134,2	133,9	°C
Mass flows:					
Outlet, section II	$\dot{m}_{2,te}$	19,838	20,440	21,165	kg/s
Sidestream	$\dot{m}_{side,te}$	7,261	8,456	9,916	kg/s
Balance line	$\dot{m}_{L,te}$	0,161	0,204	0,217	kg/s

F.2.2.12 Calculation results

	Symbol	Numerical value			Unit	Remarks
Test number		1	2	3		
Mass flow at the inlet of the 1st impeller, section I	$\dot{m}_{1,\text{I,te}}$	12,738	12,188	11,466	kg/s	Equation (F.2)
Volume flow	$\dot{V}_{1,\text{I,te}}$	3,363	3,220	3,076	m³/s	Equation (E.55) [a]
Specific polytropic compression work	$y_{\text{p,I,te}}$	23,50	25,79	27,15	kJ/kg	Equation (E.91) [a]
Polytropic efficiency	$\eta_{\text{p,I,te}}$	0,786 4	0,823 5	0,839 0	—	Equation (E.100)
Mixing temperature at the inlet to section II	$t_{1,\text{II,te}}$	71,0	70,6	69,5	°C	Equation (F.12) [a]
Mass flow at the inlet to the 1st impeller, section II	$\dot{m}_{1,\text{II,te}}$	19,999	20,644	21,382	kg/s	Equation (F.13)
Volume flow	$\dot{V}_{1,\text{II,te}}$	2,048	1,912	1,890	m³/s	Equation (E.55) [a]
Specific polytropic compression work	$y_{\text{p,II,te}}$	28,42	31,19	31,33	kJ/kg	Equation (E.91) [a]
Polytropic efficiency	$\eta_{\text{p,II,te}}$	0,767 4	0,795 8	0,792 6	—	Equation (E.100)

[a] Determination of compressibility factor, Z, and correction factor, f, and the enthalpy values using BWRS method.

$$\dot{m}_{1,\text{I,te}} = \dot{m}_{2,\text{te}} - \dot{m}_{\text{side,te}} + \dot{m}_{\text{L,te}} \tag{F.11}$$

$$h\left(t_{1,\text{II,te}}, p_{1,\text{II,te}}\right) = \frac{\dot{m}_{1,\text{I,te}} h\left(t_{2,\text{I,te}}, p_{2,\text{I,te}}\right) + \dot{m}_{\text{side}} h\left(t_{\text{side}}, p_{1,\text{II,te}}\right)}{\dot{m}_{1,\text{I,te}} + \dot{m}_{\text{side,te}}} \tag{F.12}$$

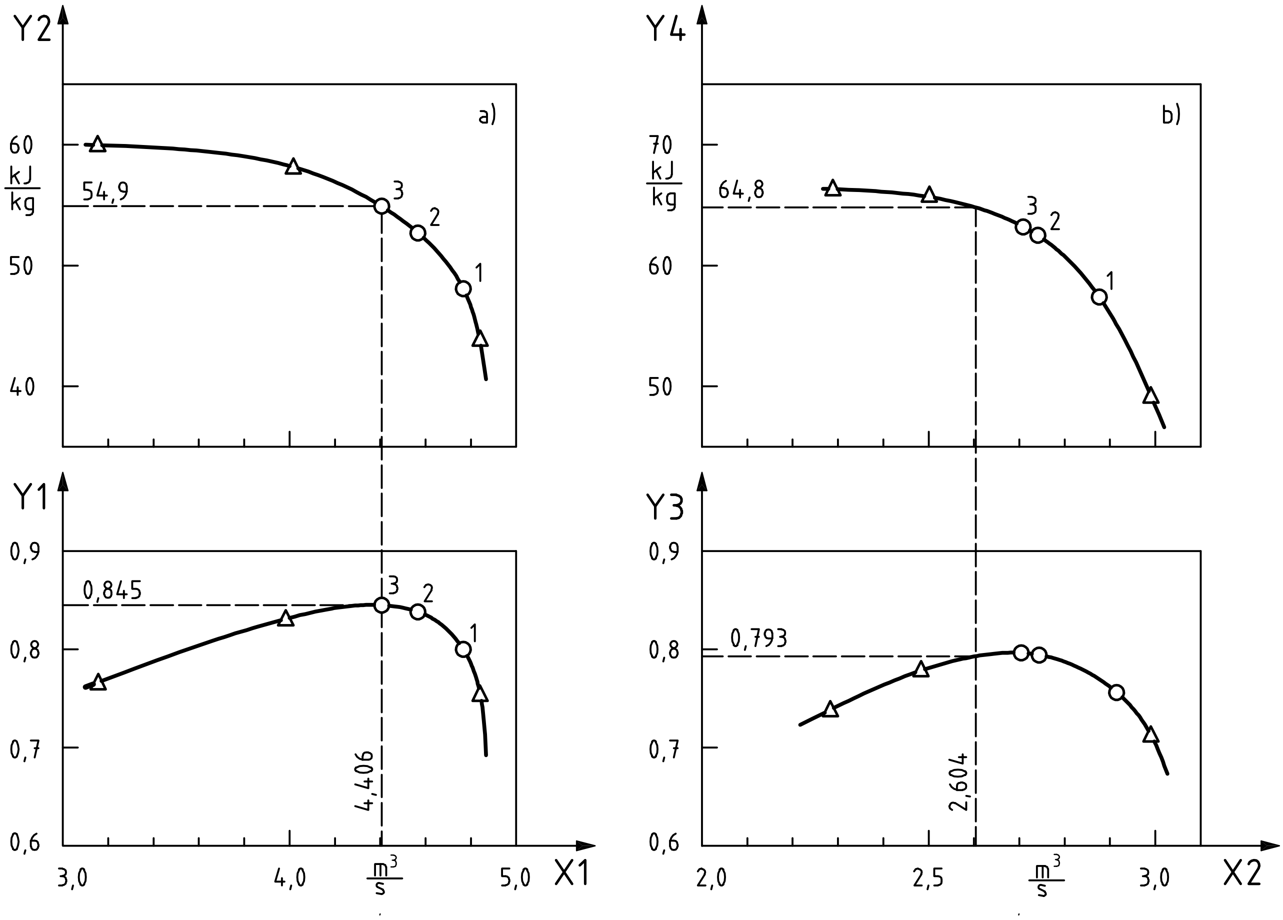
from which the temperature, $t_{1,\text{II,te}}$, can be determined with BWRS

$$\dot{m}_{1,\text{II,te}} = \dot{m}_{1,\text{I,te}} + \dot{m}_{\text{side,te}} \tag{F.13}$$

F.2.2.13 Conversion to guarantee conditions in accordance with 7.2.4.1

	Symbol	Numerical value			Unit
Test number		1	2	3	
Inlet volume flow, section I	$\dot{V}_{1,\text{I,co}}$	4,802	4,598	4,392	m³/s
Specific polytropic compression work	$y_{\text{p,I,co}}$	47,92	52,59	55,36	kJ/kg
Polytropic efficiency	$\eta_{\text{p,I,co}}$	0,791 8	0,829 2	0,844 8	—
Inlet volume flow, section II	$\dot{V}_{1,\text{II,co}}$	2,921	2,727	2,696	m³/s
Specific polytropic compression work	$y_{\text{p,II,co}}$	57,83	63,46	63,75	kJ/kg
Polytropic efficiency	$\eta_{\text{p,II,co}}$	0,769 3	0,797 8	0,794 6	—

The converted sections performance curves can be plotted using this data, as shown in Figure F.6:



Key

X1 suction volume flow, $\dot{V}_{1,\mathrm{I,co}}$

X2 suction volume flow, $\dot{V}_{1,\mathrm{II,co}}$

Y1 gas polytropic efficiency, $\eta_{\mathrm{p,I,co}}$

Y2 specific polytropic compression work, $y_{\mathrm{p,I,co}}$

Y3 gas polytropic efficiency, $\eta_{\mathrm{p,II,co}}$

Y4 specific polytropic compression work, $y_{\mathrm{p,II,co}}$

Figure F.6 — Additional test points, Δ, for performance curves for sections I and II

For test operation, the recirculating leakage mass flow, $\dot{m}_{\mathrm{L,te}}$, resulting when allowance is made for the modified pressures, temperatures, gas constant and gaps, was theoretically determined. This value was confirmed with an adequate level of accuracy by the measurements. For this reason, the calculated changes in mass flow and temperatures increases which have been taken as a basis for design were used for the conversion of the measured performance curves, which related to the impeller inlet, to the specified performance curve which related to the stages.

Demonstration of the achievement of the guarantee values was possible only once the performance curves of the two sections have been superimposed.

At a given speed, complete calculation of the machine for the specified mass flow is done using the following procedure:

$$\dot{m}_{us} = 13{,}585\ \frac{kg}{s} \tag{F.14}$$

$$\dot{m}_{1,I,g} = \dot{m}_{us} + \dot{m}_{L,g} = (13{,}585 + 0{,}431)\frac{kg}{s} = 14{,}016\frac{kg}{s} \tag{F.15}$$

$$\dot{V}_{1,I,g} = \frac{\dot{m}_{1,I,g}}{\rho_{1,I,g}} = 4{,}406\frac{m^3}{s} \tag{F.16}$$

$y_{p,I,co} = 54{,}90\frac{kJ}{kg}$ from the performance curve, Figure F.6 a) (F.17)

$\eta_{p,I,co} = 0{,}845$ from the performance curve, Figure F.6 a)

$$P_{i,I,co} = \dot{m}_{1,I,g}\frac{y_{p,I,co}}{\eta_{p,I,co}} = 911\ kW$$

$p_{side,co} = 0{,}445\ MPa\ (4{,}454\ bar)$ from equation of state

$t_{2,I,co} = 16{,}35\ °C$ from equation of state

$t_{side} = 1{,}8\ °C = t_{sat}$ at $p_{side} = 0{,}445\ MPa\ (4{,}454\ bar)$

$$\dot{m}_{side} = 10{,}131\ \frac{kg}{s}$$

$$\dot{m}_{1,II,g} = \dot{m}_{1,I,g} + \dot{m}_{side} = (14{,}016 + 10{,}131)\ \frac{kg}{s} = 24{,}147\ \frac{kg}{s}$$

$t_{1,II,co} = 8{,}9\ °C$ from mixture calculation for real gases as per Equation (F.12)

$$\dot{V}_{1,II,co} = \frac{\dot{m}_{1,II,g}}{\rho_{1,II,co}} = 2{,}604\ \frac{m^3}{s}$$

$p_{2,co} = 1{,}600\ MPa\ (16{,}00\ bar)$ from equation of state

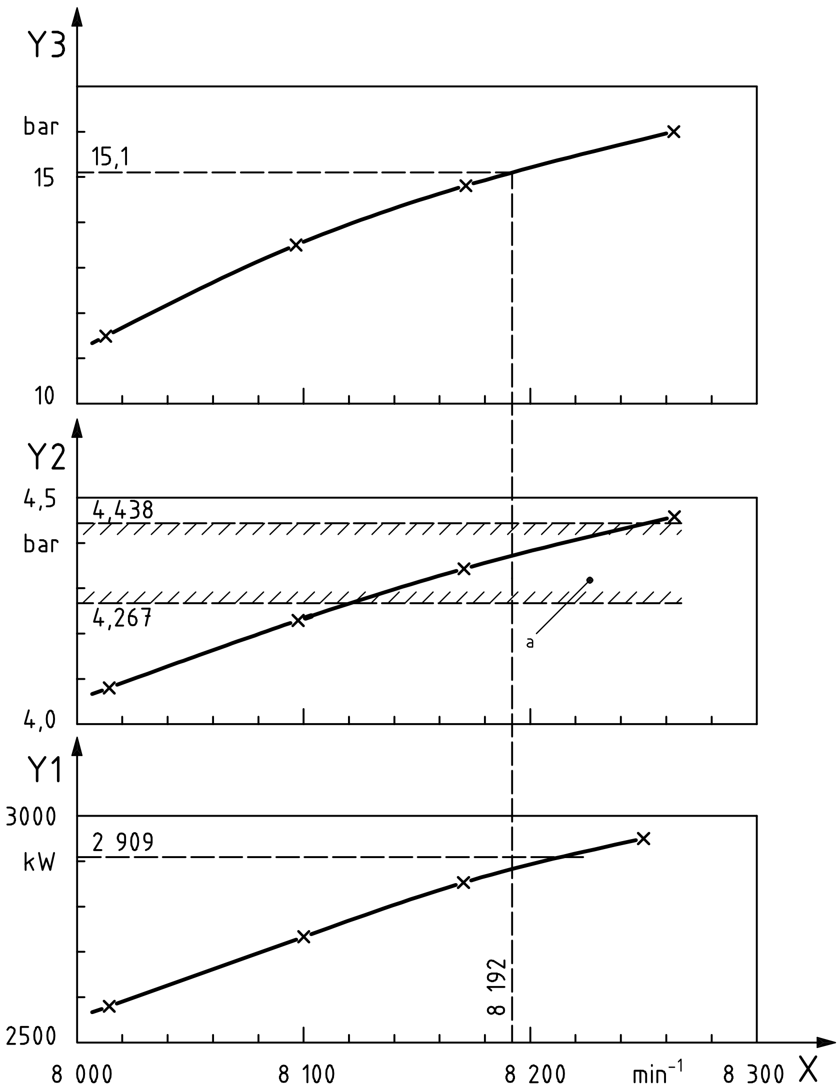
$t_{2,co} = 70{,}5\ °C$ from equation of state

Calculated mechanical losses $P_{mech,g} = 64\ kW$ were added to the converted gas powers $P_{i,co}$ for determination of power at coupling.

The resulting compressor discharge pressure does not conform to the specified discharge pressure at design speed. This can be adjusted in accordance with 8.2.3.2, by adjusting the speed. Starting from test speed, small changes in speed are assumed, the allocation of flow coefficient and head coefficient remaining constant for each performance curve point. The following relationships can be plotted once the above computation has been performed for several speeds:

$$\dot{m}_{\mathrm{us,g}} = 13{,}585\ \frac{\mathrm{kg}}{\mathrm{s}} = \mathrm{constant} \qquad \dot{m}_{\mathrm{side,g}} = 10{,}131\ \frac{\mathrm{kg}}{\mathrm{s}} = \mathrm{constant}$$

× computed values for superpositioning of section performance curves.



Key

X speed, N_g

Y1 power at coupling, $P_{\mathrm{cou,co}}$

Y2 sidestream pressure, $P_{\mathrm{side,co}}$

Y3 discharge pressure, $P_{2,\mathrm{co}}$

[a] Tolerance band for guaranteed intermediate pressure.

Figure F.7 — Guarantee comparison with computational speed variation; boundary conditions under guarantee conditions

F.2.2.14 Guarantee comparison

	Symbol	Numerical value	Unit
Inlet volume flow	$\dot{V}_{1,g}$	4,406	m^3/s
Discharge pressure	$p_{2,g}$	1,51 (15,1)	MPa (bar)
Guaranteed power at the compressor coupling	$P_{cou,g}$	2 909	kW
Converted power at the compressor coupling	$P_{cou,co}$	2 860	kW
Deviation	—	− 1,7	%
Guaranteed sidestream pressure	$p_{side,g}$	0,426 7 (4,267)	MPa (bar)
Converted sidestream pressure at guaranteed discharge pressure	$p_{side,co}$	0,433 6 (4,336)	MPa (bar)
Deviation	—	+ 1,6	%
Agreed tolerance	—	$^{+4}_{\ 0}$	%

The guarantee comparison demonstrates that the values guaranteed were achieved within the agreed tolerances. For this reason, calculation of measuring uncertainty was omitted.

F.2.3 Test example 3

Cooled compressor, polytropic exponent $n_{te} = n_g$, speed not adjustable, $R \cdot Z_1 \cdot T_1$ ratio of cooled section adjustable (see 7.2.3, Table 3, case 4b)

The speed cannot be adjusted, but temperatures in the cooled section can be adjusted by manipulating the flow of cooling water.

Conversion is effected separately for the uncooled and cooled sections of the compressor.

F.2.3.1 Purpose of the tests

Verification of guaranteed performance at three guarantee points at a constant discharge pressure.

F.2.3.2 System configuration

Four-stage compressor for air, incorporating three intercoolers and adjustable inlet guide vane control for the first stage, driven by means of electric motor and with intermediate gear.

F.2.3.3 Guarantee preconditions

	Symbol	Numerical value	Unit
Inlet pressure	$p_{1,g}$	0,098 (0,98)	MPa (bar)
Inlet temperature	$t_{1,g}$	20	°C
Relative air humidity	φ_g	70	%
Gas constant	R_g	288,9	J/(kg·K)
Isentropic exponent	k_g	1,4	—
Cooling water flow, total	$\dot{V}_{W,g}$	0,056 9	m^3/s
Cooling water inlet temperature	$t_{W,1,g}$	27	°C
Motor speed	$N_{M,g}$	1 490	1/min

F.2.3.4 Object of the guarantee

	Symbol	Numerical value			Unit
Guarantee point		a	b	c	
Inlet volume flow	$\dot{V}_{1,g}$	7,208	5,763	4,680	m^3/s
Discharge pressure	$p_{2,g}$	0,686 (6,86)	0,686 (6,86)	0,686 (6,86)	MPa (bar)
Power at the coupling	$P_{cou,g}$	1 960	1 610	1 392	kW

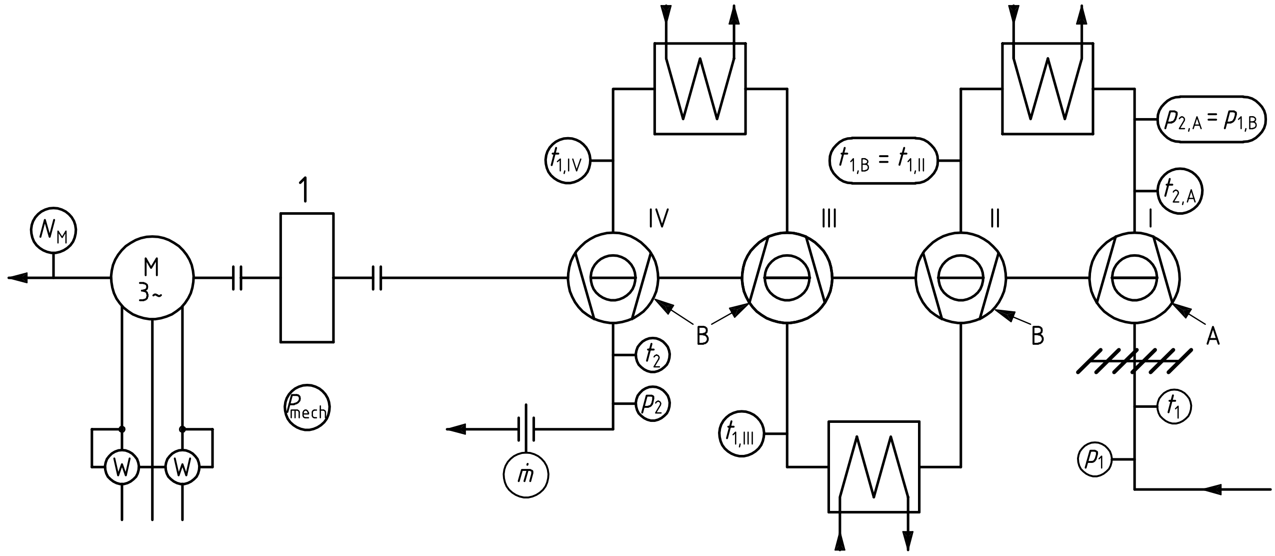
F.2.3.5 Other design figures

	Symbol	Numerical value	Unit
Inlet temperature:			
Stage II	$T_{1,II,g}$	310,2	K
Stage III	$T_{1,III,g}$	312,2	K
Stage IV	$T_{1,IV,g}$	315,2	K

F.2.3.6 Test arrangement

The test is to be performed on site under atmospheric conditions. It will not be possible to assess the intercoolers during this test, since they will be fed with a flow of cooling water differing from that specified in the guarantee, due to differing cooling water inlet temperatures.

The test arrangement can be seen in Figure F.8, which also illustrates the type of measuring instruments used, by means of symbols.



Key

1 gearbox with compressor bearing

Figure F.8 — Test arrangement and measuring stations

F.2.3.7 Test conditions

	Symbol	Numerical value	Unit
Motor speed	$N_{M,te}$	1 490	1/min
Isentropic exponent	k_{te}	1,4	—
Average inlet temperature	$t_{1,te}$	12,7	°C
Average inlet pressure	$p_{1,te}$	0,098 (0,98)	MPa (bar)
Cooling water temperature	$t_{W,1,te}$	19	°C
Gas constant	R_{te}	287,8	J/(kg·K)
Ratio of reduced speeds of rotation of the uncooled section A; Equation (2)	X_N	1,014 6	—

F.2.3.8 Setting conditions

Since the test conditions differ from those of the guarantee, the machine is divided into an uncooled section (A), which is dealt with an accordance with 7.2.3, Table 3, case 3a and a cooled section (B).

The check of the setting conditions for the uncooled section A according Annex A shows that $\Delta\phi$ is near to the inner tolerance limit $\Delta\phi_{tol}$ = 0,001. Therefore no supplementary tolerance according to 7.2.5 is taken into account.

Since the gas constant under test conditions, R_{te}, differs from that in the guarantee conditions, R_g, the recooling temperatures, $T_{1,II,te}$, $T_{1,III,te}$ and $T_{1,IV,te}$, are selected in such a way that the condition $(RZT_1)_{i,te} = (RZT_1)_{i,g}$ is fulfilled. Due to their only slight influence, no allowance is made for the differences in the amounts of condensate due to the differing water contents (x_g = 0,010 6, x_{te} = 0,004). Conversion to guarantee conditions via individual stages may be necessary.

No Reynolds number correction is implemented, since the test conditions deviate only slightly from the guarantee conditions.

The test was performed under the above test conditions.

F.2.3.9 Test results

	Symbol	Numerical value			Unit
Test number		1	2	3	
Day of test		*xx.xx.xxxx*	*xx.xx.xxxx*	*xx.xx.xxxx*	
Gas constant	R_{te}	287,8	287,8	287,8	J/(kg·K)
Motor speed	$N_{M,te}$	1 488	1 490	1492	1/min
Position of adjustable inlet guide vanes	—	+10°	+54°	+64°	—
Mass flow [a]	$\dot{m}$	8,586	6,738	5,650	kg/s
Inlet pressure	$p_{1,te}$	0,096 6 (0,966)	0,098 0 (0,980)	0,098 5 (0,985)	MPa (bar)
Inlet temperature	$t_{1,te}$	12,1	12,9	13,0	°C
	$T_{1,te}$	285,25	286,05	286,15	K
Density	$\rho_{1,te}$	1,177	1,190	1,196	kg/m^3
Discharge pressure	$p_{2,te}$	0,7451 (7,451)	0,743 7 (7,437)	0,703 0 (7,030)	MPa (bar)
Power at motor terminals	$P_{term,te}$	2 127	1 792	1 536	kW
Motor efficiency	$\eta_{M,te}$	95	95	95	%
Power at coupling	$P_{cou,te}$	2 062	1 702	1 459	kW
Mechanical losses	$P_{mech,te}$	70	70	70	kW
Gas power	$P_{i,te}$	1 992	1 632	1 389	KW

[a] Identical with usable mass flow, as per E4.2. since measured on the discharge side, according to ISO 5167.

F.2.3.10 Conversion to guarantee conditions

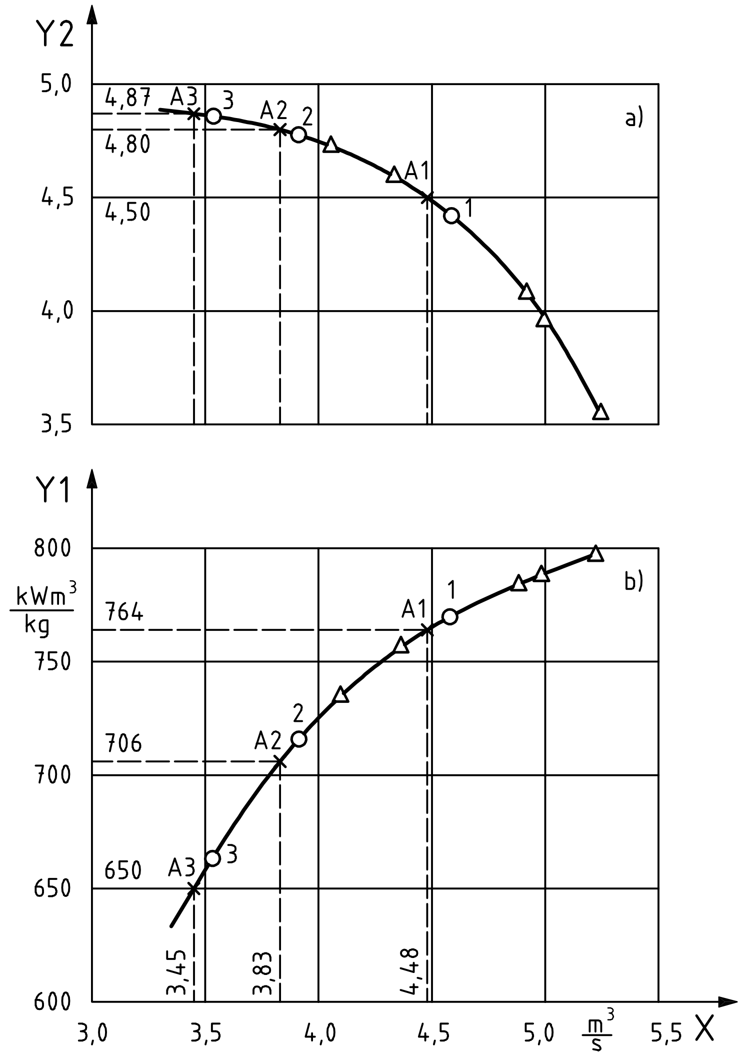
In accordance with 7.2.4.2.2, Figure 5, where $\vartheta_{j,te} \neq \vartheta_{j,g}$ but $\vartheta_{j,B,te} = \vartheta_{j,B,g}$

a) Section A: Uncooled stage I

	Symbol	Numerical value			Unit
Test number		1	2	3	
Inlet volume flow (Identical with usable inlet volume flow)	$\dot{V}_{1,\mathrm{A,te}}$	7,295	5,662	4,724	m^3/s
Inlet pressure	$p_{1,\mathrm{A,te}}$	0,096 6 (0,966)	0,098 0 (0,980)	0,098 5 (0,985)	MPa (bar)
Inlet temperature	$t_{1,\mathrm{A,te}}$	12,1	12,9	13,0	°C
Discharge pressure	$p_{2,\mathrm{A,te}}$	0,169 (1,69)	0,155 (1,55)	0,144 (1,44)	MPa (bar)
Discharge temperature	$t_{2,\mathrm{A,te}}$	74,6	71,8	70,1	°C
Pressure ratio	$\Pi_{\mathrm{A,te}}$	1,75	1,582	1,462	—
Polytropic exponent	$n_{\mathrm{A,te}}$	1,548	1,690	1,920	—
Polytropic efficiency	$\eta_{\mathrm{A,te}}$	0,807	0,700	0,569	—
Specific polytropic compression work	$y_{\mathrm{p,A,te}}$	50,809	41,528	34,305	kJ/kW
Gas power	$P_{\mathrm{i,A,te}}$	542	401	326	kW
Converted inlet volume flow	$\dot{V}_{1,\mathrm{A,co}}$	7,305	5,662	4,718	m^3/s
Converted spec. polytropic compression work	$y_{\mathrm{p,A,co}}$	50,945	41,528	34,213	kJ/kg
Converted pressure ratio	$\Pi_{\mathrm{A,co}}$	1,725	1,563	1,447	—
Converted gas power	$P_{\mathrm{i,A,co}}$	534	389	313	kW

b) Section B: Cooled stages II to IV

	Symbol	Numerical value			Unit	Remarks
Test number		1	2	3		
Inlet temperature	$t_{1,\mathrm{B,te}}$	38,5	38,0	38,1	°C	—
Inlet pressure	$p_{1,\mathrm{B,te}}$	0,169 (1,69)	0,155 (1,55)	0,144 (1,44)	MPa (bar)	—
Density	$\rho_{1,\mathrm{B,te}}$	1,885	1,731	1,608	kg/m^3	—
Inlet volume flow	$\dot{V}_{1,\mathrm{B,te}}$	4,554	3,893	3,514	m^3/s	—
Converted inlet volume flow	$\dot{V}_{1,\mathrm{B,co}}$	4,560	3,893	3,509	m^3/s	as per Figure 5, [7]; see Figure F.9
Discharge pressure overall compressor	$p_{2,\mathrm{te}}$	0,745 1 (7,451)	0,743 7 (7,437)	0,703 0 (7,030)	MPa (bar)	—
Pressure ratio	$\Pi_{\mathrm{B,te}}$	4,409	4,798	4,882	—	—
Specific isothermal compression work	$y_{T,\mathrm{B,te}}$	133,076	140,441	142,038	kJ/kg	—
Converted spec. isothermal compression work	$y_{T,\mathrm{B,co}}$	133,439	140,441	141,657	kJ/kg	—
Converted pressure ratio	$\Pi_{\mathrm{B,co}}$	4,43	4,79	4,86	—	as per Figure 5, [11]; see Figure F.9
Gas power	$P_{\mathrm{i,B,te}}$	1 450	1 231	1 063	kW	$P_{\mathrm{i,B,te}} = P_{\mathrm{i,te}} - P_{1,\mathrm{A,te}}$
Converted power related to density	$P_{\mathrm{i,B},\rho}$	769	711	661	kWm^3/kg	$P_{\mathrm{i,B},\rho} = \left(\frac{P_\mathrm{i}}{\rho_1}\right)_{\mathrm{B,co}}$



Key

X suction volume flow, $\dot{V}_{1,\mathrm{B,co}}$, of cooled section

Y1 density-related power, $P_{\mathrm{i,B},\rho}$

Y2 pressure ratio, $\Pi_{\mathrm{B,co}}$

O test point (calculated in this example)

Δ additional test points

x points at inlet volume flow, $\dot{V}_{1,\mathrm{A,co}}$, of the uncooled section

Figure F.9 — Converted values for pressure ratio and density related power of cooled compressor section B

The converted pressure ratio

$\Pi_{\mathrm{B,co}}$ (see Figure F.9)

and

$$P_{\mathrm{i,B},\rho,\mathrm{co}} = \left(\frac{P_{\mathrm{i}}}{\rho_1}\right)_{\mathrm{B,co}}$$ [see Figure F.9 b)]

are plotted versus the converted inlet volume flow, $\dot{V}_{1,\mathrm{B,co}}$, of Section B.

c) Values for overall compressor (sections A + B)

The following result for the individual test points of the inlet volume flows for cooled Section B (see Figure F.10) allocated when allowance is made for the converted pressure ratio for uncooled Section A:

	Symbol	Numerical value			Unit	Remarks
Test number		1	2	3		
Inlet volume flow	$\dot{V}_{1,B,co}$	4,48	3,83	3,45	m^3/s	as per Figure 5, [12]
Density at inlet state of section B	$\rho_{1,B,co}$	1,886	1,709	1,581	kg/m^3	—
Pressure ratio section B	$\Pi_{B,co}$	4,50	4,80	4,87	—	As per Figure 5
Density related power section B	$P_{i,B,\rho,co}$	764	706	650	kWm^3/kg	—
Converted gas power section B	$P_{i,B,co}$	1 441	1 206	1 028	kW	—

	Symbol	Numerical value			Unit	Remarks
Test number		1	2	3		
Inlet volume flow	$\dot{V}_{1,co}$	7,305	5,66	4,718	m^3/s	see Figure F.10 a) and b)
Converted pressure ratio of section A	$\Pi_{A,co}$	1,725	1,563	1,446	—	—
Converted pressure ratio of section B	$\Pi_{B,co}$	4,50	4,80	4,87	—	—
Converted total pressure ratio	Π_{co}	7,763	7,502	7,042	—	see Figure F.10 b); as per Figure 5
Converted gas power, section A	$P_{i,A,co}$	534	389	313	kW	—
Converted gas power, section B	$P_{i,B,co}$	1 441	1 206	1 028	kW	—
Mechanical losses at test	$P_{mech,te}$	70	70	70	kW	—
Converted mechanical losses	$P_{mech,co}$	70	70	70	kW	—
Converted power at coupling	$P_{cou,co}$	2 045	1 665	1 411	kW	see Figure F.10 a)
Converted spec. isothermal compression work	$y_{T,co}$	173,571	170,680	165,310	kJ/kg	—
Converted isothermal compressor power	$P_{T,co}$	1 467	1 118	902,5	kW	—
Converted isothermal efficiency at coupling	$\eta_{T,cou,co}$	71,7	67,2	63,6	%	—

F.2.3.11 Measuring uncertainty

Measuring uncertainty of results for test point I.

The measuring uncertainties of results were calculated in accordance with 6.4:

For inlet volume flow

	Value %	Remark	Reference
$\tau_{\dot{m}}$	1,1	as per ISO 5167-1	—
τ_N	0,07	digital measuring instrument, 1 Rotation referred to final value	6.4.2.4, Equation (18)
τ_{p1}	0,14	133 Pa (1,33 mbar) to absolute pressure	6.4.2.1.2, Equation (17)
τ_{T1}	0,35	1 K to absolute temperature	6.4.2.2.2 and Table 1
$\tau_{\text{res},\dot{V}}$	1,165	—	6.4.4.2.1, Equation (24)

for pressure ratio

	Value %	Remark	Reference to Section
τ_{p2}	0,9	Quality grade 0,6, final value 0,1 MPa (10 bar)	6.4.2.1.1, Equation (15)
$\tau_{\text{res},\Pi}$	1,160	with X = 1,014 and ln Π = 2,051	6.4.4.2.2, Equation (25) 4.2, Equation (2)

for power at coupling

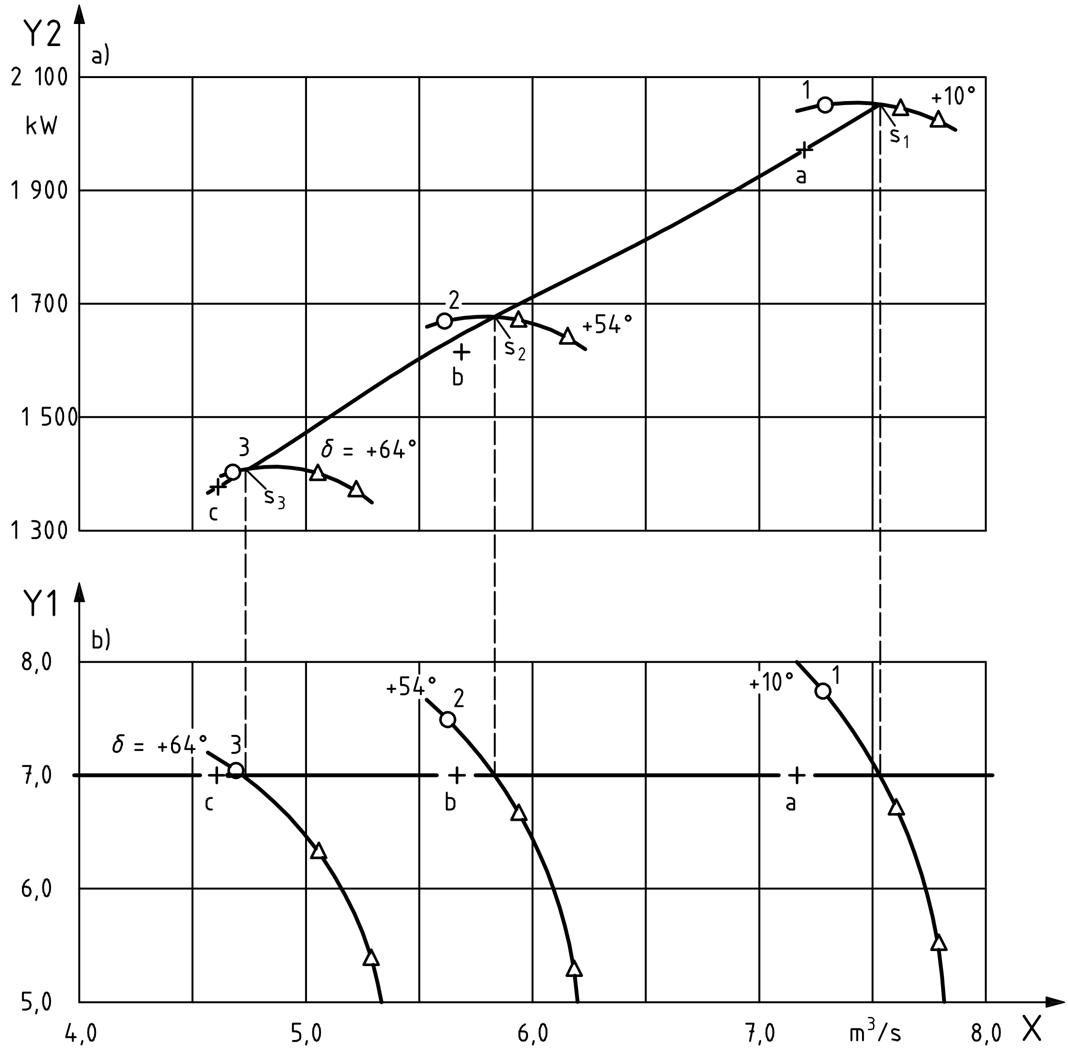
	Value %	Remark	Reference to Section
		with ε_2 = 2 954	6.4.4.2.4, Equation (28)
$\tau_{P,\text{cou}}$	0,87	quality categories: Current transformer, 0,5; voltage transformer, 0,5; wattmeter, 0,5	6.4.2.6, Equation (21)
$\tau_{P,\text{mech}}$	2,86	—	—
$\tau_{T,1,\text{B}}$	0,323	1 K to absolute temperature	6.4.2.2.2
$\tau_{T,1,j}$	0,32	1 K to absolute temperature	—
$\tau_{\text{res},P,\text{cou}}$	1,045	with ε_3 = 0,73 and ln $\Pi_{\text{A,co}}$ = 0,545	6.4.4.2.4, Table 1 $\vartheta_{j,\text{te}} \neq \vartheta_{j,\text{g}}$, case 2

The measuring uncertainty of results should be calculated analogously for test points 2 and 3, in order to be able to perform a guarantee comparison for guarantee points (b) and (c) (see Figure F.11).

It is assumed that the measuring uncertainty of results for points 1, 2 and 3 can be transferred for points s_1, s_2 and s_3.

Comparison with the guarantee

In addition to the test values compiled here [marked with “O” in Figure F.10 a) and b)], further test points were operated [in Figure F.10 a) and b)] at each of the inlet guide vane positions; these were evaluated and converted in the same manner (indicated by Δ in Figure F.10).



Key

X suction volume flow, $\dot{V}_{1,co}$

Y1 pressure ratio, Π_{co}

Y2 power at coupling, $P_{cou,co}$

\+ guarantee points

O test point (calculated in this example)

Δ additional test points

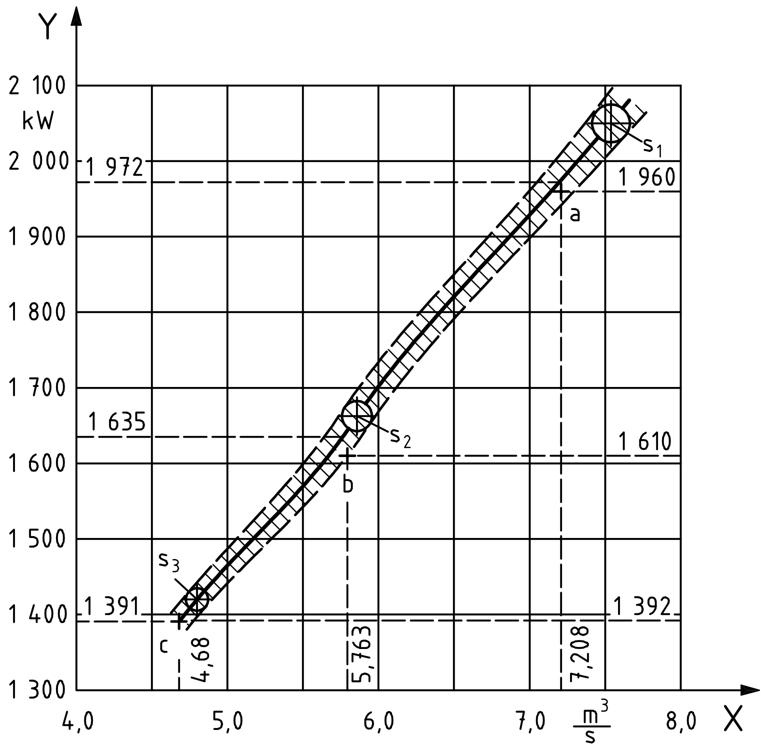
Figure F.10 — Converted values for power at coupling and total pressure ratio

All points are compiled in Figure F.10 a) and b); the powers at the guarantee pressure ratio Π_g = 7,0 have been taken from the plot in Figure F.10 a). Where the guarantee points form a different performance curve, interpolation along the intersection points of this curve with the measured compressor characteristic curves can be performed.

The following guarantee comparison, which is shown in graphic form in Figure F.11, results:

	Symbol	Numerical value			Unit
Guarantee point at $\Pi_g = 7{,}0$		a	b	c	
Inlet volume flow	$\dot{V}_{1,g}$	7,208	5,763	4,680	m³/s
Guaranteed power at coupling	$P_{cou,g}$	1 960	1 610	1 392	kW
Converted power at coupling	$P_{cou,co}$	1 972	1 635	1 391	kW
Deviation	—	+ 0,6	+ 1,55	– 0,1	%
Excess power applying measuring uncertainty	—	—	+ 0,3	—	%

This indicates that the guarantee is fulfilled for two guarantee points while, in the case of the middle point, a slight overshoot still remains even after application of measuring uncertainty.



Key

X suction volume flow, $\dot{V}_{1,co}$

Y power at coupling, $P_{cou,co}$

+ guarantee points

The hatched area applies as the measuring uncertainty band for $P_{cou,co}$ at $\Pi_g = 7{,}0$.

Figure F.11 — Guarantee comparison for $P_{cou,co}$

F.2.4 Test example 4

Cooled compressor, polytropic exponent $n_{te} = n_g$, speed not adjustable (see 7.2.3, Table 3, case 4a)

Speed not adjustable to guarantee conditions, inlet temperature and recooling temperature adjustable, with the result that $\vartheta_{j,te} = \vartheta_{j,g}$ and volume flow ratios can be met (conversion in accordance with Table 3).

F.2.4.1 Purpose of tests

Verification of guaranteed related power at four guarantee points at two different pressure ratios.

F.2.4.2 System configuration

Three-stage centrifugal compressor, for air, with intercooling following each stage; driven by electric motor.

	Symbol	Numerical value	Unit
Inlet pressure	$p_{1,g}$	0,099 4 (0,994)	MPa (bar)
Inlet temperature	$t_{1,g}$	20	°C
Relative air humidity	φ_g	70	%
Gas constant	R_g	288,887	J/(kgK)
Isentropic exponent	k_g	1,4	—
Cooling water volume flow	$\dot{V}_{W,g}$	60	m^3/h
Cooling water inlet temperature	$t_{W,g}$	23	°C

F.2.4.3 Object of the guarantee

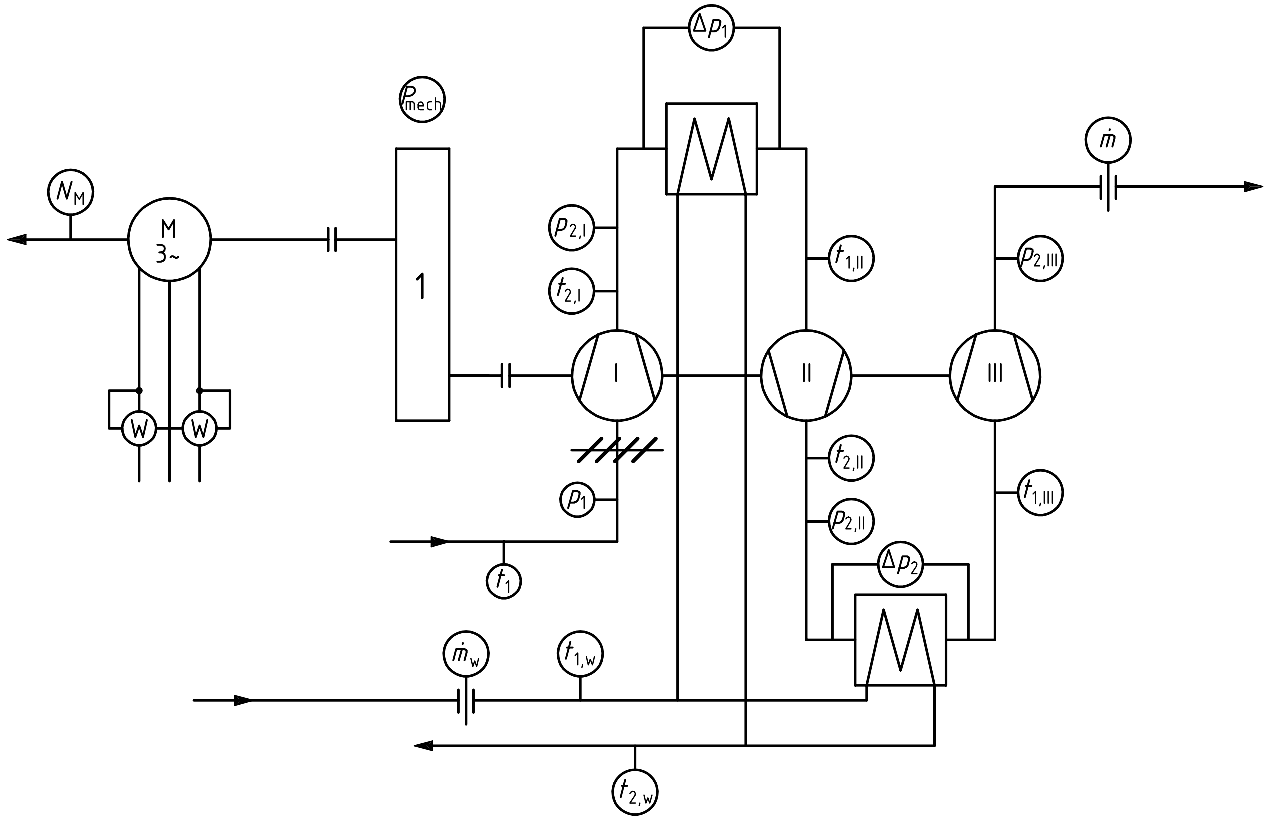
	Symbol	Numerical values				Unit
Guarantee point		a	b	c	d	
Inlet volume flow	$\dot{V}_{1,g}$	24 490	24 490	17 140	14 910	m^3/h
Discharge pressure	$p_{2,g}$	0,65 (6,5)	0,55 (5,5)	0,65 (6,5)	0,55 (5,5)	MPa (bar)
Power at coupling	P_{cou}	1 830	1 730	1 420	1 200	kWh
Related power at coupling	$\left(\frac{P_{cou}}{\dot{V}_1}\right)_g$	0,074 72	0,070 64	0,082 85	0,080 48	$\frac{kWh}{m^3}$

F.2.4.4 Other design data

	Symbol	Numerical value	Unit
Speed of rotation, drive motor	$N_{M,g}$	1 480	1/min
Air inlet temperature:			
Stage II	$t_{1,II,g}$	29	°C
Stage III	$t_{1,III,g}$	29	°C

F.2.4.5 Test arrangement

The acceptance tests are performed on the manufacturer's test bench. The power at the terminals of the electric motor is measured using the two-watt-meter method. Power at coupling is calculated from the power at the terminals using the individual loss method.



Key

1 gearbox

Figure F.12 — Test arrangement and measuring stations

The recooling temperatures are adjusted by setting the cooling water flow and temperature, and the quality of the coolers is assessed simultaneously.

The test apparatus can be seen in Figure F.12. The compressor is equipped with a guide-vane adjustment system installed upstream of the first stage.

Only electrical data transducers in conjunction with a data acquisition system and a controller were used for measurement purposes. The measuring instruments were calibrated before the tests in a certified calibration service laboratory.

Test conditions

For verification of related power at the four guarantee points, a total of five performance curve segments for differing guide-vane positions within the compressor's working range were operated in each case in such a way that the two plant characteristics, Π_{g1} = constant and Π_{g2} = constant, were intersected by the performance curve segments of the compressor.

Only the test points of the performance curve with negative prerotation δ_1 = const. are shown within this example.

The guarantee comparison performed using graphic methods and calculation of measuring uncertainties for the guarantee comparison are performed for all guarantee points.

F.2.4.6 Setting conditions

The recooling temperatures upstream of the second and third stage can be adjusted by changing the flow of cooling water in such a way that the temperature ratios ϑ_j accord with those of the design.

It proved possible to adhere to similarity conditions in all the tests, without taking into account a supplementary tolerance (the setting conditions of the stages are within the inner tolerance $\Delta\phi_{tol} \pm 0{,}01$). Due to the performance of the test using the original gas and the only slight deviations in test conditions from the guarantee conditions, conversion of efficiency as a result of divergent Reynolds numbers is not necessary.

	Symbol	Numerical value			Unit
Test number		1	2	3	
Motor speed	$N_{M,te}$	1 487	1 488	1 487	1/min
Mass flow [a]	$\dot{m}_{te}$	23 389	30 157	30 695	kg/h
Inlet pressure	$p_{1,te}$	0,096 1 (0,961)	0,095 9 (0,959)	0,095 6 (0,956)	MPa (bar)
Inlet temperature	$t_{1,te}$	21,75	22,03	22,99	°C
Air humidity	φ_{te}	50	45	41	%
Gas constant	R_{te}	288,567	288,444	288,402	J/(kgK)
Cooling water inlet temp.	$t_{W,1,te}$	23,3	23,1	23,7	°C
Temperature					
discharge stage I	$t_{2,I,te}$	115,1	115,1	116,2	°C
inlet stage II	$t_{1,II,te}$	30,5	31,0	32,0	°C
discharge stage II	$t_{2,II,te}$	95,9	92,9	91,6	°C
inlet stage III	$t_{1,III,te}$	30,6	31,0	32,1	°C
discharge stage III	$t_{2,III,te}$	91,1	87,5	86,5	°C
Pressure					
discharge stage I	$p_{2,I,te}$	0,210 0 (2,100)	0,202 4 (2,024)	0,197 2 (1,972)	MPa (bar)
inlet stage II	$p_{1,II,te}$	0,204 4 (2,044)	0,195 7 (1,957)	0,190 0 (1,900)	MPa (bar)
discharge stage II	$p_{2,II,te}$	0,374 5 (3,745)	0,323 4 (3,234)	0,333 1 (3,331)	MPa (bar)
inlet stage III	$p_{1,III,te}$	0,371 1 (3,711)	0,316 1 (3,161)	0,328 1 (3,281)	MPa (bar)
discharge stage III	$p_{2,III,te}$	0,649 6 (6,496)	0,534 3 (5,343)	0,520 9 (5,209)	MPa (bar)

	Symbol	Numerical value			Unit
Power at coupling from energy balance:					
Specific enthalpy difference					
stage I	$\Delta h_{\mathrm{I,te}}$	94,47	93,84	94,00	kJ/kg
stage II	$\Delta h_{\mathrm{II,te}}$	65,83	62,33	59,99	kJ/kg
stage III	$\Delta h_{\mathrm{III,te}}$	60,93	59,89	54,76	kJ/kg
Stage gas power					
stage I	$P_{\mathrm{i,I,te}}$	744,99	786,12	801,47	kW
stage II	$P_{\mathrm{i,II,te}}$	519,15	522,12	511,50	kW
stage III	$P_{\mathrm{i,III,te}}$	480,48	476,56	466,93	kW
Total:	$P_{\mathrm{i,te}}$	1 744,60	1 784,80	1 779,90	kW
Mechanical power losses	$P_{\mathrm{mech,te}}$	42	44	44	kW
Heat conduction and radiation losses [b]	$P_{\mathrm{rad,te}}$	5	5	5	kW
Power at coupling	$P_{\mathrm{cou,te}}$	1 791,6	1 833,8	1 828,9	KW

[a] Measured as per ISO 5167-1 on the discharge side; identical with usable mass flow, since no condensation of water in the coolers.

[b] Losses estimated.

	Symbol	Numerical value			Unit	Remarks
Test number		1	2	3		
Pressure ratio	Π_{te}	6,758	6,166	5,447	—	—
Specific isothermal compression work	$y_{T,\mathrm{te}}$	162,51	154,81	144,67	kJ/kg	Equation (E.63)
Inlet density	$\rho_{1,\mathrm{te}}$	1,130 0	1,126 1	1,120 2	$\mathrm{kg/m^3}$	—
Inlet volume flow	$\dot{V}_{1,\mathrm{te}}$	25 124	26 780	27 400	$\mathrm{m^3/h}$	—

F.2.4.7 Power at coupling from measured electrical power

	Symbol	Numerical value			Unit
Test number		1	2	3	
Power at terminals	$P_{\mathrm{term,te}}$	1 890	1 942	1 927	kW
Speed of rotation, electric motor	$N_{\mathrm{M,te}}$	1 487	1 488	1 487	1/min
Power losses, electric motor	$P_{V,\mathrm{te}}$	75,6	77,7	77,1	kW
Power at coupling	$P_{\mathrm{cou,te}}$	1 814,4	1 864,3	1 849,9	kW

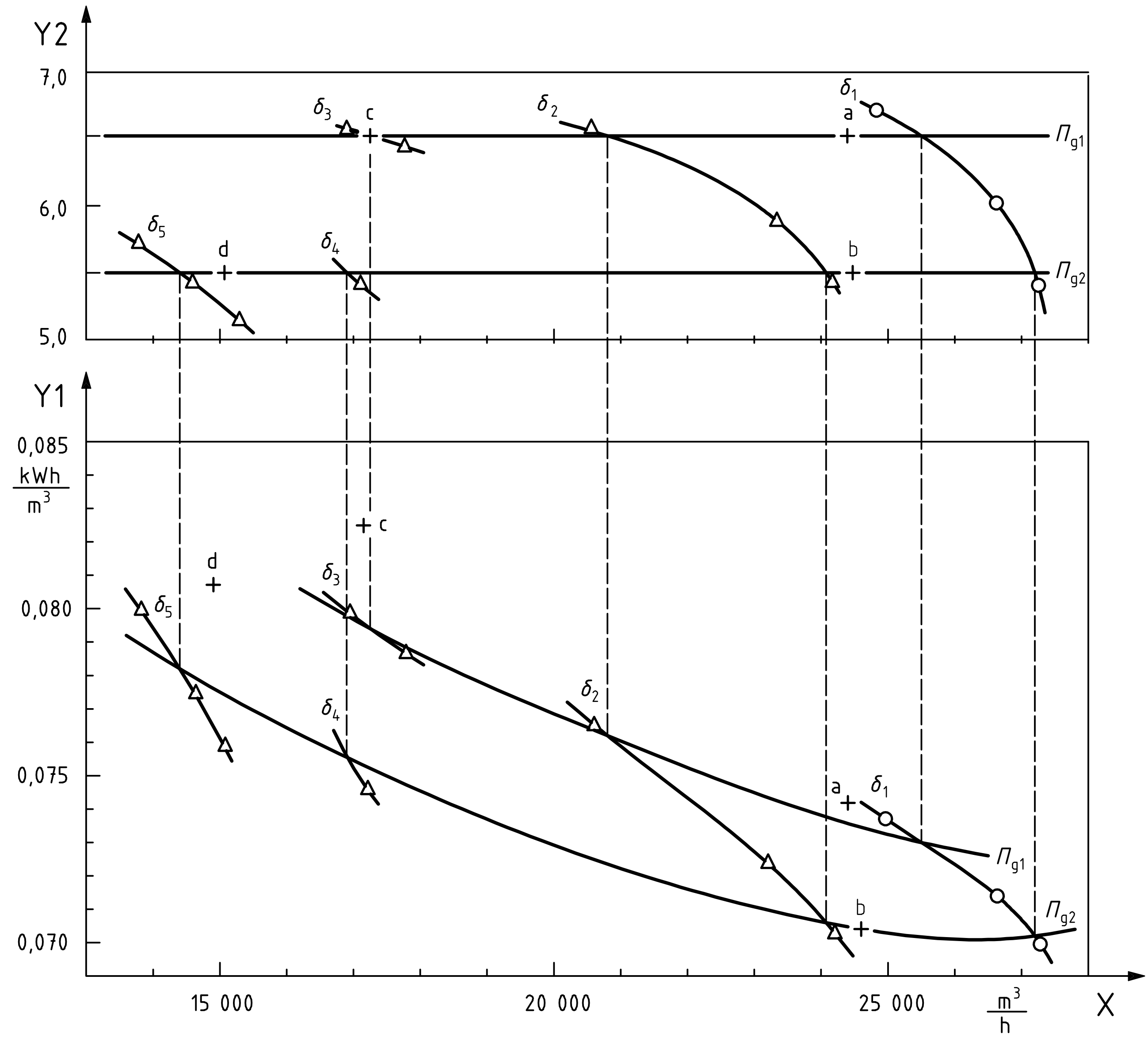
As agreed upon, the measured electrical power is used for further evaluation of the tests.

F.2.4.8 Related power

	Symbol	Numerical value			Unit
Related power at coupling	$\left(\frac{P_{cou}}{\dot{V}_1}\right)_{te}$	0,072 22	0,069 62	0,067 51	$\frac{kWh}{m^3}$
Isothermal compression power	P_T	1 281,6	1 296,9	1 233,5	kW
Isothermal efficiency at coupling	$\eta_{T,cou,te}$	0,706	0,696	0,667	—

F.2.4.9 Conversions to guarantee conditions

	Symbol	Numerical value			Unit	References
Test number		1	2	3		
Test speed	$N_{M,te}$	1 487	1 488	1 487	1/min	—
Design speed	$N_{M,g}$	1 480	1 480	1 480	1/min	—
Converted inlet volume flow	$\dot{V}_{1,co}$	25 006	26 629	27 264	m^3/h	Figure 4
Converted isothermal compression work	$y_{T,co}$	161,59	153,15	143,31	kJ/kg	Figure 4
Converted pressure ratio	Π_{co}	6,766 6	6,097 7	5,429 0	—	Figure 4
Converted power at coupling	$P_{cou,co}$	1 868,1	1 910,6	1 909,6	kW	Figure 4
Converted related power	$\left(\frac{P_{cou}}{\dot{V}_1}\right)_{co}$	0,074 71	0,071 75	0,070 04	$\frac{kWh}{m^3}$	—



Key

X suction volume flow, $\dot{V}_{1,co}$

Y1 related power at coupling, $\left(\frac{P_{cou}}{\dot{V}_1}\right)_{co}$

Y2 compression ratio, Π_{co}

\+ guarantee points

O test points (calculated in this example)

Δ additional test points at various guide vane settings θ

Figure F.13 — Related power at coupling and pressure ratio as a function of inlet volume flow

Measuring uncertainties

The relative measuring uncertainties were calculated for measuring point 1 using the difference method, from section 6.4.4.3. for demonstration. The method from 6.4.4.2.4. could have been applied as well in this case.

The starting equation for relative measuring uncertainty for related power at coupling is

$$W\left(x_i\right)=\frac{P_{\text{cou,cog}}}{\dot{V}_{1,\text{g}}}=\frac{P_{\text{i,cog}}}{\dot{V}_{1,\text{g}}}+\frac{P_{\text{mech,cog}}}{\dot{V}_{1,\text{g}}} \tag{F.18}$$

where

$$P_{\text{i,cog}}=P_{\text{i,co}}\cdot\frac{y_{T,\text{g}}}{y_{T,\text{co}}}\cdot\frac{\dot{V}_{1,\text{g}}}{\dot{V}_{1,\text{co}}} \tag{F.19}$$

where the stage isothermal compression work for approximate incorporation of the influence of recooling temperature is

$$y_{T,z}=\left(\frac{1}{z}+\frac{z-1}{z}\cdot\frac{T_{1,j,\text{av}}}{T_{1,\text{I}}}\right)\cdot R\cdot Z\cdot T_{1,\text{I}}\cdot\ln\frac{p_{2,z}}{p_{1,\text{I}}} \tag{F.20}$$

$$W\left(x_i\right)=K_{\text{c}}\cdot\frac{P_{\text{cou,te}}-P_{\text{mech,te}}}{\dot{m}_{\text{te}}\cdot\left(\frac{1}{z}+\frac{z-1}{z}\cdot\frac{T_{1,\text{II,te}}}{T_{1,\text{I,te}}}\right)\cdot\left(R\cdot Z\cdot T_{1,\text{I}}\cdot\ln\frac{p_{2,\text{III}}}{p_{1,\text{I}}}\right)_{\text{te}}}+\frac{P_{\text{mech,cog}}}{\dot{V}_{1,\text{g}}} \tag{F.21}$$

where

$$K_{\text{c}}=10^{5}\cdot p_{1,\text{I,g}}\left(\frac{1}{z}+\frac{z-1}{z}\cdot\frac{T_{1,\text{II,g}}}{T_{1,\text{I,g}}}\right)\cdot\ln\left(\frac{p_{2,\text{III}}}{p_{1,\text{I}}}\right)_{\text{g}} \tag{F.22}$$

where

$p_{1,\text{I,g}}$ = 99,4 kPa (0,994 bar)

$p_{2,\text{III,g}}$ = 650 kPa (6,5 bar)

$T_{1,\text{I,g}}$ = 293,15 K

$T_{1,\text{II,g}}$ = 302,15 K

$\dot{V}_{1,\text{g}}$ = 24 490 m^3/h

$P_{\text{mech,co}}$ = $P_{\text{mech,te}}$

z = 3

resulting in $W(x_i)$ = 0,073 423 3 kWh/m^3

As in the derivation of the equations in 6.4.4.2, p_1, p_2, and the absolute pressure contained in the mass flow equation are regarded here by way of simplification as measured variables independent of one another, even though these contain the common measuring error of ambient pressure p_{amb}, via Equation (17).

Measured variable	Symbol	Unit	x_i	τ_{xi} %	V_{xi}	$W(x_i + V_{xi})$ kW/m^3	$W(x_i - V_{xi})$ kWh/m^3	f_{xi}	f_{xi}^2
Power at coupling	$P_{\text{cou,te}}$	kW	1 814,4	0,5	9,207	0,073 795	0,073 050	$5{,}073\,3 \cdot 10^{-3}$	$2{,}573\,8 \cdot 10^{-5}$
Mech. power losses	$P_{\text{mech,te}}$	kW	42	10	4,2	0,073 424	0,073 421	$2{,}197\,4 \cdot 10^{-5}$	$4{,}828\,7 \cdot 10^{-10}$
Mass flow	$\dot{m}_{te}$	kg/h	28 389	1,2	340,668	0,072 573	0,074 294	$1{,}172\,1 \cdot 10^{-2}$	$1{,}373\,9 \cdot 10^{-4}$
Inlet temperature, stage I	$T_{1,\text{I,te}}$	K	294,9	0,3	1	0,073 343	0,073 502	$1{,}082\,5 \cdot 10^{-3}$	$1{,}171\,8 \cdot 10^{-6}$
Inlet temperature, stage II	$T_{1,\text{II,te}}$	K	303,65	0,3	1	0,073 264	0,073 582	$2{,}165\,0 \cdot 10^{-3}$	$4{,}687\,3 \cdot 10^{-6}$
Gas constant	R_{te}	J/(kgK)	288,567	0	0	0,073 423	0,073 423	0	0
Compres-sibility factor	Z_{te}	—	1	0	0	0,073 423	0,073 423	0	0
Inlet pressure	$p_{1,\text{I,te}}$	kPa (bar)	96,12 (0,961 2)	10,7 (0,107)	0,102 8 (0,001028)	7,346 3 (0,073 463)	0,073 383	$5{,}464\,6 \cdot 10^{-4}$	$2{,}986\,2 \cdot 10^{-7}$
Discharge pressuret	$p_{2,\text{III,te}}$	kPa (bar)	649,6 (6,496)	25,4 (0,254)	1,65 (0,016 5)	7,332 8 (0,073 328)	0,073 518	$1{,}297\,9 \cdot 10^{-3}$	$1{,}684\,5 \cdot 10^{-6}$
								$\Sigma f_{xi}^2 = 1{,}709\,7 \cdot 10^{-4}$	

$$\tau_{\text{res}} = \pm\sqrt{\Sigma f_x^2}$$

Relative measuring uncertainty of results for the converted related power:

$$\tau_{\text{res}} = 1{,}31\ \%.$$

The measuring uncertainty of results for the other test points should be calculated analogously.

Guarantee comparison

All the test points are plotted in Figure F.13. The guarantee comparison was performed using the graphic method. For this purpose, the intersections of the compressor curves with the plant characteristics (Π_{g1} = constant; Π_{g2} = const.) are projected vertically to the performance curves for reduced power. The lines connecting the points obtained in this way for related power show at the inlet volume flows of the guarantee points the deviations from the guaranteed related power.

Since the converted related powers determined in this way are below the guarantee values at all guarantee points, the plotting of the measuring uncertainty band is omitted.

	Symbol	Numerical values				Unit
Guarantee point		a	b	c	d	
Inlet volume flow	$\dot{V}_{1,g}$	24 490	24 490	17 140	14 910	$\frac{kWh}{m^3}$
Related power as per guarantee	$\left(\frac{P_{cou}}{\dot{V}_1}\right)_g$	0,074 72	0,070 64	0,082 85	0,080 48	$\frac{kWh}{m^3}$
Related power at test converted to guarantee conditions	$\left(\frac{P_{cou}}{\dot{V}_1}\right)_{co}$	0,074 10	0,070 40	0,079 80	0,077 50	$\frac{kWh}{m^3}$
Deviation	$\Delta\left(\frac{P_{cou}}{\dot{V}_1}\right)$	– 0,000 62	– 0,000 24	– 0,003 05	– 0,002 98	$\frac{kWh}{m^3}$
Deviation	$\Delta\left(\frac{P_{cou}}{\dot{V}_1}\right)$	– 0,83	– 0,34	– 3,68	– 3,70	%
Average value (weighing coefficient c_i = 1)	$\Delta\left(\frac{P_{cou}}{\dot{V}_1}\right)$	– 2,14				%

F.2.5 Test example 5

Uncooled pipeline compressor, polytropic exponent $n_{te} \neq n_g$, speed adjustable by gas turbine drive (see 7.2.3, Table 2, case 3d)

The test conditions deviate from those of the guarantee. Changing the speed makes it possible to run the performance tests within allowable deviation of the ratio, ϕ, of volume flow ratios. Example takes into account a calculation of measurement uncertainty.

F.2.5.1 Purpose of the tests

Verification of guaranteed power at coupling at one guarantee point and verification of the specific heat consumption related to isentropic power of the compressor carried out on site. In this example, only the guarantee comparison with respect to the coupling power of the compressor is dealt with.

F.2.5.2 System configuration

One-stage centrifugal compressor for transmission of natural gas in a pipeline. The compressor is driven by a two-shaft gas turbine.

F.2.5.3 Guarantee preconditions

	Symbol	Numerical value	Unit
Inlet pressure	$p_{1,g}$	4,9 (49)	MPa (bar)
Inlet temperature	$t_{1,g}$	10	°C
Inlet density	$\rho_{1,g}$	38,219	kg/m^3
Gas type		natural gas	
Molar mass	M_g	16,460	kg/mol
Gas constant	R_g	0,505 1	kJ/(kg·K)
Speed of rotation	N_g	15 930	1/min

Gas-composition	Volume %
Carbon dioxide	0,17
Nitrogen	0,92
Methane	97,68
Methane	0,84
Propane	0,26
Butane	0,09
Pentane	0,03
Hexane	0,01
Heptane	0,01
Octane	0,01
Benzol	0,01

F.2.5.4 Object of the guarantee

	Symbol	Numerical value	Unit
Inlet volume flow	$\dot{V}_{1,g}$	1,369 4	m^3/s
Discharge pressure	$p_{2,g}$	0,75 (75)	MPa (bar)
Power at coupling	$P_{cou,g}$	3 850	kW

F.2.5.5 Other design figures

	Symbol	Numerical value	Unit
Specific polytropic compression work	$y_{p,g}$	58,4	kJ/kg
Specific isentropic compression work	$y_{s,g}$	57,7	kJ/kg
Polytropic exponent	n_g	1,470 4	—
Polytropic efficiency	$\eta_{p,g}$	81,34	%
Isentropic exponent	$k_{V,g}$	1,358 3	—
Isentropic efficiency	$\eta_{s,g}$	80,36	%

F.2.5.6 Test arrangement

The guarantee tests have been carried out on-site with nearly the original gas. The single test points have been adjusted by means of the available throttle valves at the inlet and outlet of the compressor and the adjustment of the power turbine speed.

Four test points on a constant speed line near the predicted speed to meet the coordinates of the guarantee point (inlet volume flow and polytropic compression work) have been performed.

The test apparatus can be seen in Figure F.14. The coupling power has been measured directly by means of a calibrated torquemeter (to measure torque and speed) and additionally calculated from the gaspower and the mechanical losses taken from the shop test results.

Only electrical data transducers in conjunction with a data acquisition system were used for measurement purposes. The measuring instruments have been calibrated before the tests in a certified service laboratory.

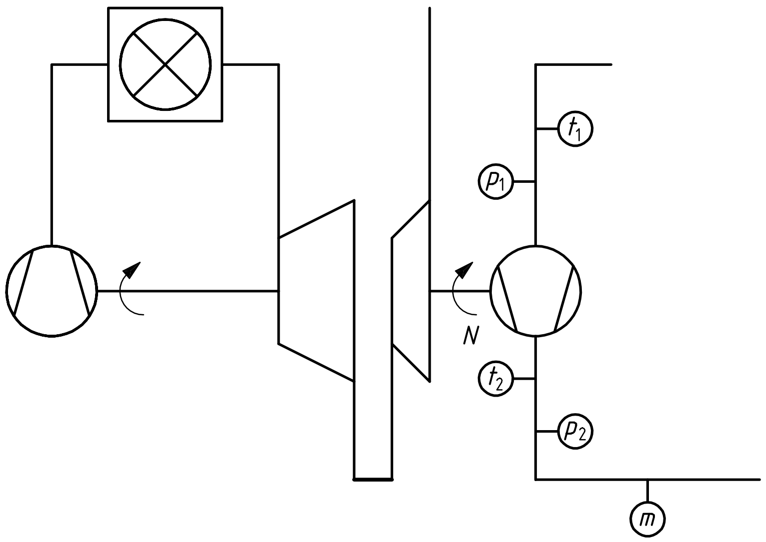


Figure F.14 — Test arrangement and measuring stations

The setting conditions are equal to the guarantee conditions.

F.2.5.7 Test conditions

Gas composition	Volume %
Carbon dioxide	0,167 5
Nitrogen	0,833 9
Methane	97,989 7
Ethane	0,665 9
Propane	0,227 4
Butane	0,083 5
Pentane	0,019 6
Hexane and higher	0,012 5

	Symbol	Numerical value				Unit
Test number		1	2	3	4	
Speed	N_{te}	16 002	15 987	16 004	16 002	min^{-1}
Inlet pressure	$p_{1,te}$	4,913 (49,13)	4,925 (49,25)	4,907 3 (49,073)	4,830 1 (48,301)	MPa (bar)
Inlet temperature	$t_{1,te}$	8,24	8,4	8,4	8,11	°C
Molar mass	M_{te}	16,164	16,164	16,164	16,164	kg/mol
Compressibility factor inlet	$Z_{1,te}$	0,895 5	0,895 5	0,895 9	0,897 0	—

F.2.5.8 Test results

	Symbol	Numerical value				Unit	Remarks
Test number		1	2	3	4		
Mass flow	$\dot{m}_{te}$	60,47	56,11	52,39	47,21	kg/s	—
Inlet density	$\rho_{1,te}$	37,826	37,906	37,766	37,171	kg/m^3	—
Inlet volume flow	$\dot{V}_{1,te}$	1,598 6	1,480 0	1,387 2	1,270 0	m^3/s	—
Discharge pressure	$p_{2,te}$	7,066 (70,66)	7,353 (73,53)	7,523 (75,23)	7,62 (76,2)	MPa (bar)	—
Discharge temperature	$t_{2,te}$	41,66	48,16	48,0	47,24	°C	—
Compressibility factor discharge	$Z_{2,te}$	0,907 1	0,907 5	0,908 4	0,909 9	—	—
Isentropic exponent	$k_{V,te}$	1,349 8	1,353 7	1,355 7	1,356 5	—	Equation (E.67)
Polytropic exponent	n_{te}	1,524 6	1,501 6	1,492 7	1,483 9	—	Equation (E.81)
Difference of specific isentropic enthalpy	$\Delta h_{s,te}$	49,432	54,845	58,697	62,869	kJ/kg	BWR
Specific isentropic compression work	$y_{s,te}$	49,496	54,911	58,762	62,932	kJ/kg	Equation (E.68)
Specific polytropic compression work	$y_{p,te}$	50,108	55,565	59,476	63,719	kJ/kg	Equation (E.77)
Correction factor	f	0,998 7	0,998 8	0,998 9	0,999 0		—
Polytropic efficiency	$\eta_{p,te}$	75,81	78,35	78,33	80,27	%	—
Isentropic efficiency	$\eta_{s,te}$	74,79	77,33	79,37	79,20	%	—
Gas power	$P_{i,te}$	3 997	3 979	3 926	3 748	kW	—
Mechanical losses	$P_{mech,te}$	30	30	30	30	kW	—
Radiation losses	$Q_{rad,te}$	0	0	0	0	kW	—

	Symbol	Numerical value				Unit	Remarks
Power at coupling based on temperature measurement	$P_{\mathrm{cou,te,1}}$	4 027	4 009	3 956	3 778	kW	—
Power at coupling based on torquemeter	$P_{\mathrm{cou,te,2}}$	4 031	4 015	3 972	3 790	kW	—
Weighted power at coupling	$\overline{P}_{\mathrm{cou,te}}$	4 030	4 014	3 969	3 788	kW	—
Ratio of reduced speeds	X_N	0,993 6	0,992 6	0,993 6	0,993 4	—	Equation (E.112)

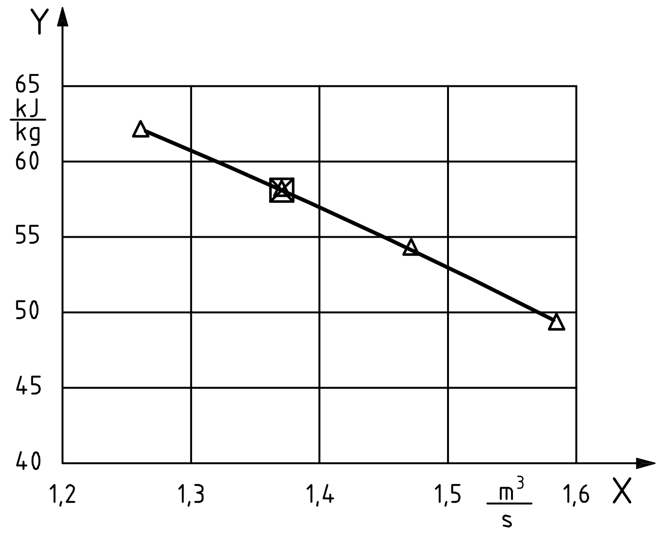
F.2.5.9 Conversion to guarantee conditions

In a first step, the conversion to the guarantee conditions has been done with respect to the inlet density and to a reference speed of the four test runs.

In a second step, the conversion has been performed to the guarantee figures of specific polytropic compression work and inlet volume flow on basis of the above converted values of inlet volume flow, polytropic compression work, gas power and mechanical losses by means of multiplying these values with the linear, square and cube ratio of the speed that meets the guarantee figures and the reference test speed.

Before this was done, the gas power had been corrected due to the weighted average value of the coupling power measured by different and independent methods.

	Symbol	Numerical value				Unit	Remarks
Test number		1	2	3	4		
Reference speed	$N_{\mathrm{te,ref}}$	16 000	16 000	16 000	16 000	min^{-1}	—
Converted inlet volume flow	$\dot{V}_{\mathrm{1,co}}$	1,598 4	1,478 8	1,386 9	1,269 8	m^3/s	—
Converted specific polytropic compression work	$y_{\mathrm{p,co}}$	50,095	55,655	59,446	63,703	kJ/kg	—
Weighted converted coupling power	$P_{\mathrm{cou,co}}$	4 070,5	4 056,9	4 013,7	3 892,6	kW	see 6.4.4.5 Equation (38)
Speed to meet the guarantee figures	$N_{\mathrm{co,g}}$	15 840	15 840	15 840	15 840	min^{-1}	—
Inlet flow converted, g	$\dot{V}_{\mathrm{1,co,g}}$	1,582 4	1,464 0	1,373 0	1,257 1	m^3/s	—
Specific polytropic compression work converted, g	$y_{\mathrm{p,co,g}}$	49,10	54,55	58,26	62,43	kJ/kg	—
Coupling power converted, g	$P_{\mathrm{cou,co,g}}$	3 950	3 936	3 894	3 777	kW	see 8.2.2 Equation (48)



Key

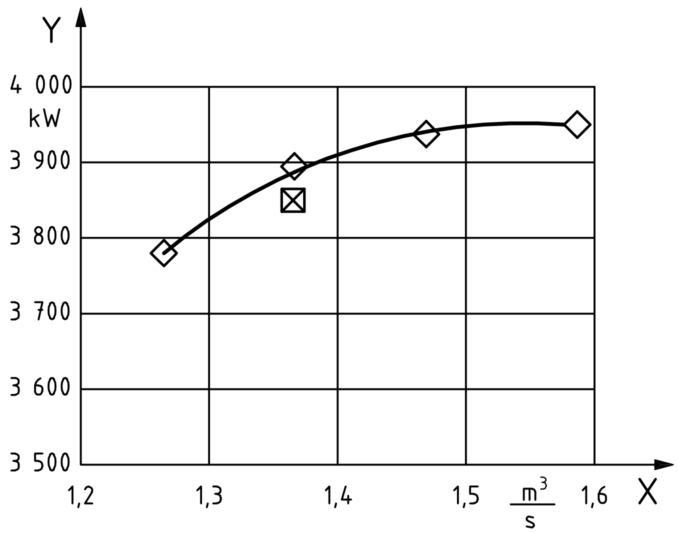
X inlet volume flow converted, $\dot{V}_{1,co,g}$

Y specific polytropic compression work converted, $y_{p,co,g}$

⊠ guarantee

Δ test

Figure F.15 — Specific polytropic compression work converted to guarantee conditions versus inlet volume flow



Key

X inlet volume flow converted, $\dot{V}_{1,co,g}$

Y coupling power converted, $P_{cou,co,g}$

⊠ guarantee

◇ test

Figure F.16 — Coupling power converted to guarantee conditions versus inlet volume flow

Measuring uncertainties

The relative measuring uncertainties were calculated for measuring point 3 using the difference method (see 6.4.4.3) applied to the computer program to calculate the test results.

As for the guarantee comparison with the main guarantee figure, the coupling power at test was determined using two different measuring methods independent of each other, the measurement result uncertainty has to be calculated as a weighted value depending on both the measurement values and their measurement uncertainties.

The equation to calculate the weighted coupling power is

$$\bar{P}_{\text{cou,co}} = \frac{\sum\left[\left(\frac{1}{V_{P\text{cou,co},i}}\right)^2 \cdot P_{\text{cou,co},i}\right]}{\sum\left(\frac{1}{V_{P\text{cou,co},i}}\right)^2} \quad \text{(F.23)}$$

where $P_{\text{cou,co},i}$ are the two results of the converted coupling power measured by the two methods being independent from each other and $P_{\text{cou,co},i}$ the individual measuring uncertainty of both result values.

The weighted average value of the measuring uncertainty is given by the equation:

$$V_{\bar{P}\text{cou,co}} = \frac{1}{\sqrt{\sum\left(\frac{1}{V_{P\text{cou,co},i}}\right)^2}} \quad \text{(F.24)}$$

The following table contains as well the measurement uncertainties of the single measurement items as well as the calculated result uncertainties.

Measurement variable	Symbol	Unit	x_i	τ_{xi} %	$f_{xi}(P_{\text{cou,co}})^2$ by temperature	$f_{xi}(P_{\text{cou,co}})^2$ by torque	$f_{xi}(\dot{V}_{1,\text{co}})^2$
Barometric pressure	b_0	MPa (mbar)	0,101 (1 010)	0,1	3,700 76·E–10	5,198 4·E–10	5,244 1·E–10
Inlet pressure	p_1	MPa (bar, gauge)	4,908 (48,07)	0,1	3,831 89·E–07	1,169 64·E–06	1,187 45·E–06
Discharge pressure	p_2	MPa (bar, gauge)	7,523 (74,22)	0,1	2,161 23·E–07	0	0
Inlet temperature	T_1	K	281,55	0,25	1,305 21·E–04	1,294 56·E–05	1,314 61·E–06
Discharge temperature	T_2	K	321,15	0,25	7,497 29 E–04	3,815 64·E–11	0
Coupling power by torque	P_{cou}	kW	3 972	1,32	0	1,742 749·E–04	0
Mechanical losses	P_{mech}	kW	30	10	2,289 81·E–07	1,000 000·E–10	0

Measurement variable	Symbol	Unit	x_i	τ_{xi} %	$f_{xi}(P_{\text{cou,co}})^2$ by temperature	$f_{xi}(P_{\text{cou,co}})^2$ by torque	$f_{xi}(\dot{V}_{\text{1,co}})^2$
Speed of rotation	N	min^{-1}	16 004	0,1	8,878 55·E–06	8,827 44·E–06	9,980 01·E–07
Mass flow	$\dot{m}$	kg/h	188 604	1,2	1,426 26·E–04	0	1,440 00·E–04
Result measurement uncertainty	$\tau_{\text{res}} = \sqrt{\sum\left(f_{xi}\right)^2}$	%	—	—	3,15	1,40	1,26

According to [6], both the values of converted coupling power and their result measurement uncertainties were taken to calculate the weighted converted coupling power:

Weighted converted coupling power	$P_{\text{cou,c}}$	kW	3 889
Guaranteed coupling power	$P_{\text{cou,g}}$	kW	3 850
Deviation	ΔP_{cou}	kW	39
<Deviation	$\dfrac{\Delta P_{\text{cou}}}{P_{\text{cou,g}}}$	%	1,01
Measurement uncertainty	τ_{res}	%	± 1,28

F.2.5.10 Guarantee comparison

All test points relevant for the guarantee comparison are plotted in Figures F.15. and F.16.

The guarantee comparison was performed graphically by taking the value of converted coupling power at the guaranteed inlet volume flow from the plotted graph in Figure F.16.

Although the converted coupling power is higher than the guarantee value, the guarantee is fulfilled with partial credit from the calculated measurement uncertainty.

Bibliography

[1] ASME PTC 10, *Power test Code — Compressors and Exhausters*

[2] VDI 2045-1, *Acceptance and Performance Tests on Turbo Compressors and Displacement Compressors (1993) — Part 1: Test Procedure and Comparison with Guaranteed Values*

[3] VDI 2045-2, *Acceptance and Performance Tests on Turbo Compressors and Displacement Compressors (1993) — Part 2: Theory and Examples*

[4] VDI/VDE 2040 Part 1, *Calculation principles for the measurement of fluid flow; using orifice plates, nozzles and venturi tubes; deviations ans supplements to DIN 1952 - 01.91*

[5] VDI/VDE 2041, *Measurement of fluid flow with primary devices; orifice plates and nozzles for special applications - 04.91*

[6] VDI/VDE 2640, *Bl. 3, Measurement of gas flow in circular; annular or rectangular sections of conduits velocity area method - 11.83*

[7] VDI 2048, *Inaccuracies at acceptance test measurements; fundamental principles - 06.78*

[8] MITTELBACH, G., *Die Berechnung von Ergebnismessspielen bei Abnahmeversuchen am Beispiel des Verdichters*, BWK 24 (1972) Nr. 3, S. 99 u. 104

[9] NEL Fluid Rep. No. 85 (Nov. 1959)

[10] FISTER, W., and KOTZUR, J., *Ungekühlte Turboverdichter bei geänderten Betriebsbedingungen*, Chem.-Ing.-Techn., 37 (1965)

[11] STRUB, R.A., BONCIANI, L., BORER, C.J., CASEY, M.V., COLE, S.L., COOK, B.B., KOTZUR, J., SIMON, H., and STRITE, M.A., *Influence of the Reynolds number on the performance of centrifugal compressors*. Transactions of the ASME, Journal of Turbomachinery, 109 (1987) pp. 541-544

[12] SCHULTZ, J.M., *The Polytropic Analysis of Centrifugal Compressors*, Transactions of the ASME, Series A. Journal of Engineering for Power, Vol. 84 (Jan. 1962), pp. 69-82

[13] HUNTINGTON, R.A., *Evaluation of Polytropic Calculation Methods for Turbomachinery Performance*, ASME Journal of Engineering for Gas Turbines and Power, Vol. 107 (Oct. 1985), pp. 872-879

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

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