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Displacement and dynamic compressors — Performance test code for electric driven low-pressure air compressor packages

Compresseurs volumétriques et turbocompresseurs — Code d'essai de performance des ensembles de compresseurs à air basse pression à entraînement électrique

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ISO copyright office
CP 401 • Ch. de Blandonnet 8
CH-1214 Vernier, Geneva
Phone: +41 22 749 01 11
Email: copyright@iso.org
Website: www.iso.org

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Contents

Page

Foreword

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This document was prepared by Technical Committee ISO/TC 118, *Compressors and pneumatic tools, machines and equipment*, Subcommittee SC 6 *Air compressors and compressed air systems*.

Any feedback or questions on this document should be directed to the user’s national standards body. A complete listing of these bodies can be found at www.iso.org/members.html.

Introduction

This document was developed in response to a recognized need to provide a methodology to correct performance of a low-pressure air compressor to guarantee conditions for positive displacement and dynamic compression types.

In dynamic compression, air is drawn between the blades on a rapid rotating compression impeller1) and accelerates to high velocity. The gas is then discharged through a diffuser, where the kinetic energy is transformed into static pressure. Dynamic low-pressure compressors are of a radial flow design, with the following typical examples:

— single-stage centrifugal (aka high speed "turbo") compressors;

— multi-stage centrifugal compressors without intercooling.

Positive displacement low-pressure compressors work on the principle of trapping a volume of air and reducing its volume, internally or externally. Two basic types are typical, as follows:

— Rotary screw positive displacement compressor where air is drawn into a compression chamber formed by intermeshing rotors1. As the rotors turn, the cavity between the rotors becomes smaller, reducing the volume of the trapped air and increasing its pressure;

— Rotary lobe positive displacement compressor where air is drawn into the case and is trapped between the rotor and the case wall. These air pockets are progressively moved to the discharge port. At the discharge port, a back flow of air into the pocket from the higher-pressure discharge line produces a constant volume pressure rise.

Existing standards (e.g. ISO 1217, ISO 5389, ISO 18740) for positive displacement compressors and dynamic compressors, do not provide clear and concise means of comparing different technologies.

This document provides simplified wire to air performance test methods that measure true performance of low-pressure air compressor packages.

1) In this document the terms "rotor" and "impeller" are used to describe the rotating element(s) which cause(s) compression, and can be considered to be interchangeable.

Displacement and dynamic compressors — Performance test code for electric driven low-pressure air compressor packages

1 Scope

This document specifies the performance test method of electrically driven low-pressure air compressor packages, where the compression is performed by positive displacement or dynamic compression; utilising atmospheric air as the compression gas. Low-pressure air compressor packages are often referred to as “blowers”.

NOTE Throughout this document, the term ‘low-pressure compressor’ is used to describe a low-pressure air compressor (“blower”) package

Low-pressure compressors with and without means of controlling flow (control may be electrical (e.g. with a variable frequency drive) or mechanical or both) are covered.

This document applies to low-pressure compressors meeting all the following limits:

— Atmospheric inlet air pressure between 0,5 bar and 1,1 bar.

— Discharge vs inlet pressure differential between 0,1 bar and 2,5 bar.

— Discharge vs inlet pressure ratio between 1,1 and 3,5.

This document is not applicable to:

— positive displacement low-pressure compressors with a liquid in the compression element (such as liquid ring pumps and liquid injected low-pressure compressor of screw type)

— multi-stage low-pressure compressors other than multistage centrifugal compressors comprised of multiple, identical or very similar uncooled sections along a single shaft (repeating stages).

— single shaft, multistage centrifugal compressors are treated from the point of measurement and calculation as a single stage

2 Normative references

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

ISO 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*

ISO 9300, *Measurement of gas flow by means of critical flow nozzles*

ISO 17089-1, *Measurement of fluid flow in closed conduits — Ultrasonic meters for gas — Part 1: Meters for custody transfer and allocation measurement*

3 Terms and definitions

For the purposes of this document, the following terms and definitions apply.

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

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

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

3.1
acceptance test
performance test carried out in accordance with this document

Note 1 to entry: See Annex C for an example of acceptance test report.

3.2
displacement compressor
packaged compressor where a static pressure rise is obtained by allowing successive volumes of gas to be aspired into and exhausted out of a closed space by means of the displacement of a moving member

[SOURCE: ISO 5390:1977, 3.1]

3.3
dynamic compressor
packaged compressor in which the fluid pressure increase is obtained by transformation of kinetic energy into potential energy with continuous flow from intake point to discharge point

[SOURCE: ISO 5390:1977, 3.2]

3.4
external coolant
medium externally supplied to the compressor to which the generated heat is finally rejected

Note 1 to entry: This is usually ambient air or cooling water

[SOURCE: ISO 1217:2009, 3.1.7]

3.5
packaged compressor
compressor with prime mover, transmission, fully piped and wired internally, including ancillary and auxiliary items of equipment where these are within the scope of supply

[SOURCE: ISO 1217:2009, 3.1.13]

3.6
isentropic compression
idealized (i.e. reversible) adiabatic thermodynamic compression process that occurs without transfer of heat into or out of a system

3.7
rotational speed
number of revolutions of the compressor drive shaft per unit of time

[SOURCE: ISO 1217:2009, 3.1.18]

3.8
process air inlet point
point upstream of any technically required component

Note 1 to entry: In the case in which a technically required component is not physically present during the test the impact of the component on performance shall be accounted for

3.9
process air discharge point
point downstream of any technically required component

Note 1 to entry: In the case in which a technically required component is not physically present during the test the impact of the component on performance shall be accounted for

3.10
guarantee conditions
site conditions for which the equipment is expected to perform. Typically, this will include atmospheric pressure and ambient temperature

3.11
absolute pressure
pressure with reference to absolute zero, i.e. with reference to an absolute vacuum

Note 1 to entry: It equals the algebraic sum of atmospheric pressure and gauge pressure (static pressure or total pressure).

[SOURCE: ISO 3857-1:1977, 1.3, modified — The second sentence was moved as a note.]

3.12
ambient pressure
absolute pressure (3.11) of the atmospheric air measured in the vicinity of the compressor

[SOURCE: ISO 1217:2009, 3.2.2]

3.13
atmospheric pressure
absolute pressure (3.11) of the atmospheric air measured at the test place

[SOURCE: ISO 1217:2009, 3.2.3]

3.14
discharge pressure
total mean *absolute pressure* (3.11) at the *process air discharge point* (3.9)

3.15
inlet pressure
total mean *absolute pressure* (3.11) at the standard *process air inlet point* (3.8)

3.16
total pressure
pressure measured at the stagnation point when a gas stream is brought to rest and its kinetic energy is converted by an *isentropic compression* (3.6) from the flow condition to the stagnation condition

[SOURCE: ISO 1217:2009, 3.2.9]

3.17
ambient temperature
total temperature (3.20) of the atmospheric air in the vicinity of the compressor, but unaffected by it

[SOURCE: ISO 1217:2009, 3.3.1]

3.18
discharge temperature
total temperature (3.20) at the *process air discharge point* (3.9)

3.19
inlet temperature
total temperature (3.21) at the standard *process air inlet point* (3.8)

3.20
total temperature
temperature that would be measured at the stagnation point if a gas stream were brought to rest and its kinetic energy converted by an *isentropic compression* (3.6) from the flow condition to the stagnation condition

[SOURCE: ISO 1217:2009, 3.3.4]

3.21
relative humidity
ratio, in humid air, expressed as a percentage, of the water vapour actual pressure to the saturated vapour pressure at the same dry bulb temperature

$$\varphi = \frac{p}{p_{\text{sat}}}$$

where p is partial pressure (ISO 80000-4:2019, item 4-14.1) of vapour and p_{sat} is its partial pressure at saturation (at the same temperature)

[SOURCE: ISO 80000-5:2019, 5-33]

3.22
isentropic exponent
ratio of the specific heat at constant pressure to the specific heat at constant volume

3.23
actual volume flow rate
volume flow rate of air, compressed and delivered at the standard discharge point, referred to conditions of *total temperature* (3.20), total pressure and composition prevailing at the standard inlet point

3.24
isentropic power
power that is theoretically required to compress an ideal gas under constant entropy, from given inlet conditions to a given *discharge pressure* (3.14)

Note 1 to entry: The term “ideal gas” is used to indicate any gas in a condition or state so that it follows closely the ideal gas law

[SOURCE: ISO 1217:2009/Amd.1:2016, 3.5.1]

3.25
isentropic efficiency
ratio of the required *isentropic power* (3.24) to measured power for the same specified boundaries with the same gas and the same inlet conditions and *discharge pressure* (3.14)

$$\eta_{\text{isen}} = \frac{P_{\text{isen}}}{P_{\text{real}}}$$

[SOURCE: ISO 1217:2009/Amd.1:2016, 3.6.1]

3.26
power input
sum of the electrical power inputs to the prime mover and any ancillaries and auxiliaries driven from the compressor shaft or by a separate prime mover at rated supply conditions, including the effect of all equipment included in the *packaged compressor* (3.5)

Note 1 to entry: Auxiliaries include oil pump, cooling fan and integral compressed air dryer

Note 2 to entry: Rated supply conditions refer to phase, voltage, frequency and ampere capability

[SOURCE: ISO 1217:2009, 3.5.3]

3.27
specific energy requirement
<of a packaged compressor> *power input* (3.26) per unit of compressor actual volume flow rate

[SOURCE: ISO 1217:2009, 3.7.2]

3.28
specific isentropic compression work
work expressed as energy per unit mass of air during *isentropic compression* (3.6)

3.29
specific isochoric compression work
work expressed as energy per unit mass of air during isochoric compression

3.30
specific combined compression work
sum of the *specific isentropic compression work* (3.28) and *specific isochoric compression work* (3.29), weighted by the internal volume ratio

3.31
internal volume ratio
ratio of the enclosed volume at moment of closure of the inlet port to the enclosed volume at the moment of opening of the discharge port for a positive *displacement compressor* (3.2)

3.32
rotor tip speed
peripheral speed at the largest rotor/impeller tip diameter

3.33
machine Mach number
ratio of the rotor tip speed to the speed of sound of the fluid inlet state at inlet conditions

3.34
accounted for value
means (measured/estimated/calculated/corrected) – a simulated or calculated substitute characteristic of components not available for the test, for example; the pressure drop of a remote air filter

3.35
idle power consumption
total consumed power when the *packaged compressor* (3.5) is not producing flow to the discharge but is rotating at significant speed. i.e. for *packaged compressor* (3.5) equipped with idling functionality

3.36
standby power consumption
power required to keep the *packaged compressor* (3.5) ready for immediate start from non-rotating state

3.37
flow coefficient
flow velocity formed from the inlet volume flow and an impeller cross-section area and rendered dimensionless by the tip speed of the impeller

3.38
work coefficient
specific compression work of the reference process rendered dimensionless by the kinetic energy of tip speed

3.39
reduced speed
alternate test speed used to achieve ratio of Mach number for contract to test equal to one

3.40
two speed test
combination of one test to determine the thermodynamic performance and one test to determine the electromechanical performance

3.41
package motor
item(s) that is a part of the *packaged compressor* (3.5) including any additional drive train components

3.42
test motor
item(s) that replaces the *package motor* (3.41) for testing

3.43
shaft power
mechanical input power at the rotor/impeller

3.44
electromechanical
part of the total losses, total power consumption or total efficiency, that is not the result of the compression work on the gas

Note 1 to entry: This shall include the impact on said values from motor(s), control(s), gear(s), bearing(s), seal(s) and all auxiliaries (e.g. fans and pumps), whether said components are mounted on or related to the driver(s), compression element(s) or part of the package.

4 Units

General use of SI units (see ISO 80000-1) as given throughout this document is recommended, see Table 1 and Table 2. However, in agreement with accepted practice in the pneumatic industry sector, some non-preferred SI units, accepted by ISO, are also used, see

Table 1 — List of symbols

Symbol	Term	SI unit
c	sonic velocity	m/s
c_p	specific heat capacity	J/(kg·K)
D	the largest rotor/impeller tip diameter	m
e	specific energy	J/m^3
h	specific enthalpy	J/kg
Ma	machine Mach number	—
M	molar mass	kg/mol
m	mass	kg
q_m	mass flow	kg/s
n	rotational speed	1/s
P	power	W
p	pressure	Pa
R	specific gas constant	J/(kg·K)
Re	Reynolds number	—
s	specific entropy	J/(kg·K)
T	thermodynamic temperature	K
t	Celsius temperature	°C
U	supply voltage	V
u	tip speed	m/s

Table 1 *(continued)*

Symbol	Term	SI unit
v	specific volume	m^3/kg
v_i	internal volume ratio	—
V	volume	m^3
q_V	volume flow	m^3/s
X_n	ratio of reduced speeds of rotation	—
x	mass ratio of water vapour to dry gas	kg/kg
y	specific compression work	J/kg
Δ	difference	—
η	efficiency	—
ϑ	ratio of (RZ1 T1) values	—
κ	ratio of specific heat capacities (isentropic exponent)	—
π	pressure ratio	—
ρ	density	kg/m^3
ϕ	ratio of volume flow rate ratios	—
φ	flow coefficient	—
φ_{rel}	relative humidity	—
ψ	work coefficient	—
σ	standard deviation	—

Table 2 — List of subscripts used in this document

Subscript	Term
0	ambient
1	inlet (suction side)
2	discharge (discharge side)
air	dry air
abs	absolute (pressure)
amb	ambient (air, temperature)
co	corrected to guarantee conditions
cog	corrected to the pressure ratio and inlet volume flow of the guarantee point
comb	combined
cool	coolant
d	dynamic
em	electromechanical
dry	dry
g	guarantee conditions or performance data at guarantee conditions
i	internal or intermediate
isoc	isochoric
ideal	according to an ideal thermodynamic process
out	output
pack	Packaged compressor boundary
Pr	reference or standard process
red	reduced speed
ref	reference value
rel	relative

Table 2 *(continued)*

Subscript	Term
s	isentropic
sat	saturated
st	static
target	target
te	test result
te1	first test in 2-speed testing
te2	second test in 2-speed testing
tol	permissible deviation
tot	total
u	tip or peripheral
vap	vapour, vapor, steam
wet	moist
idle	idle
standby	standby

5 Guarantee and measurement

5.1 Packaged compressor

The packaged compressor shall comprise all components that are necessary for the long-term functioning of the low-pressure compressor under guarantee conditions and are needed to fulfil the object of the guarantee and the preconditions of the guarantee:

— low-pressure compressor with drive system;

— variable frequency drive (as applicable);

— cooling/lubrication system;

— inlet filter;

— inlet valve/guide vanes (as applicable);

— bearing power supply (as applicable);

— fully piped and wired internally;

— including ancillary and auxiliary items of equipment and all power devices that affect power consumption.

5.2 Preconditions of the guarantee

If no preconditions are defined in the contract, the preconditions of the guarantee shall be applied in accordance with Table 3 below.

For testing to be possible, at least the following shall be specified as the preconditions of the guarantee:

— air inlet pressure;

— air inlet temperature;

— air inlet humidity;

— coolant inlet temperature;

— coolant flow;

— supply voltage;

— supply frequency.

NOTE Air inlet pressure, air inlet temperature, air inlet humidity and coolant inlet temperature can be taken from the default conditions in Table 3.

Table 3 — Reference conditions

Default inlet condition	Value
Inlet air pressure	100 kPa[a]
Inlet air temperature	20 °C
Inlet relative humidity	0 %
Temperature of the coolants at package inlet	20 °C
[a] 1 bar.	

Additional limits can be specified, such as:

— electromagnetic compatibility standard to be fulfilled;

— specified maximum noise level outside the packaged compressor;

— total harmonics distortion on the electrical supply;

— input current supply;

— minimum permissible starts/hour;

— minimum permissible unload cycles/hour;

— allowable pulsation level at the discharge of the packaged compressor;

— filtration grade of the air inlet filter.

5.3 Object of the guarantee

The object of the guarantee is the set of values to be guaranteed within the defined preconditions:

— inlet volume flow rate;

— discharge pressure at the discharge of the packaged compressor;

— specific energy of the packaged compressor for the delivered flow at the guaranteed discharge pressure;

— isentropic efficiency of the packaged compressor for the delivered flow at the guaranteed discharge pressure;

— idle power consumption;

— standby power consumption.

5.4 Low-pressure compressor to be tested

The low-pressure compressor configuration to be tested shall include all components required to fulfil all the preconditions.

As a general rule, the configuration of the unit under test shall be identical to the configuration of the unit to be delivered.

A packaged compressor checklist, such as given in Annex A, shall be completed by the manufacturer and shall be part of each low-pressure compressor test report. The checklist shall be used to ensure that the tested packaged compressor matches that specified.

The checklist shall indicate which components and their performance related characteristics are included, excluded, accounted for value, or not applicable for normal functioning at guarantee conditions.

If any required components are not installed in the test configuration, the correction calculations for these components shall be shown in conjunction with the checklist.

Ancillaries required for the sustainable operation of the low-pressure packaged compressor, excluding stand-by ancillaries, are to be in operation.

5.5 Low-pressure compressor specifications to be provided prior to testing

The low-pressure compressor is tested against a specified discharge pressure (at the discharge of the packaged compressor).

In addition to the preconditions, the reference inlet conditions (or the guarantee inlet conditions) and the checklist, certain data needs to be provided by the manufacturer before the test event, typically with a tender to provide the equipment:

— rotational speed at guarantee conditions when the machine is fulfilling the object of the guarantee (for variable speed machines or if the motor in testing a fixed speed machine can differ from the one to be used at the site of assembly);

— internal volume ratio for positive displacement low-pressure compressor;

— variable geometry settings (if applicable) for the low-pressure compressor.

6 Measuring equipment, methods and accuracy

6.1 General

The equipment and methods given in this document are not intended to restrict the use of other equipment and methods with the same or better accuracy. Where an international standard relating to a particular measurement or type of instrument exists, any measurements carried out or instruments used shall be in accordance with such an international standard.

All inspection, measuring, test equipment and devices that can affect the test shall be calibrated and adjusted at prescribed intervals, or prior to use, against certified equipment having a known valid relationship to nationally recognized standards. The use of data acquisition systems shall be allowed and the test logs may be print outs resulting from the system.

No measurement uncertainty tolerances are to be taken into account in corrections or acceptance. For guarantee acceptance, as tested results are treated as measured in comparison to Table 5 without additional uncertainty tolerances applied.

6.2 Measurement of pressure

6.2.1 General

Pressure taps in the pipe or receiver shall be normal to, and flush with, the inside wall. A minimum of two static or total pressure-measuring instruments shall be utilized for each measurement location spaced at 180° intervals around the pipe circumference, and 90° to temperature instrumentation

NOTE For low pressures or high flow velocities, minor irregularities such as burrs can lead to serious error.

Connecting piping shall be leak-free, as short as possible, of sufficient diameter and arranged to avoid blockage by dirt or condensed liquid. For measurement of liquid pressure or pressure of liquid-gas mixtures, the instrument shall be mounted at the same height as the measuring point and the connecting piping shall be arranged so that the height of liquid columns in the piping exerts no influence. Otherwise, account shall be taken of the difference in height.

Instruments shall be mounted so that they are not susceptible to disturbing vibrations.

The measuring instrument (analogue or digital) shall have an accuracy of ±1 % at the measured value.

The pressure measurement shall be a total measurement, or static measurement corrected to total conditions.

For definitions of static, dynamic and total measurements refer to ISO 5389:2005, 5.2 and 5.3.

6.2.2 Atmospheric pressure

The absolute atmospheric pressure shall be measured with a barometer having an accuracy better than ±0,15 %.

6.2.3 Pressure measurement for ambient inlet

The compressor package inlet pressure, p_1, is the atmospheric pressure measured by a barometer near the compressor package where the velocity is zero.

6.2.4 Pressure measurement for piped inlet

The pressure is the total pressure, p_1, measured at the process air inlet point. The pressure shall be measured at a location at least one pipe diameter upstream of the inlet.

6.2.5 Pressure measurement for piped discharge

The pressure is the total pressure, p_2, measured at the process air discharge point. The pressure shall be measured at a location at least two pipe diameters downstream of the discharge.

6.3 Measurement of temperature

6.3.1 General

Temperature shall be measured by certified or calibrated instruments such as thermometers, thermo-electrical instruments, resistance thermometers or thermistors having an accuracy of ±0,5 K inserted into the pipe or into pockets.

A minimum of two temperature-measuring instruments shall be used for each measurement location. For measurements made on piping these shall be spaced at 180° intervals around the pipe circumference.

Thermometer pockets shall be as thin, and their diameters as small, as is practical, with their outside surface substantially free from corrosion or oxide. The pocket shall be partially filled with a suitable liquid.

The thermometers or the pockets shall extend into the pipe to a distance of 100 mm, or one third the diameter of the pipe, whichever is less.

When taking readings, the thermometer shall not be lifted out of the medium being measured nor out of the pocket when one is used.

Precautions shall be taken to ensure that the:

— immediate vicinity of the insertion point and the projecting parts of the connection are well insulated so that the pocket is virtually at the same temperature as the medium being observed;

— sensor of any temperature measuring device or thermometer pocket is well swept by the medium (the sensor or thermometer pocket shall point against the gas stream; in extreme cases a position perpendicular to the gas stream may be used);

— thermometer pocket does not disturb the normal flow.

6.3.2 Temperature measurement for ambient inlet

The packaged compressor ambient temperature is the atmospheric temperature measured at the packaged compressor in the plane of the intake system.

6.3.3 Temperature measurement for piped inlet

The inlet temperature is the total temperature, T_1, measured at the process air inlet point. The temperature instrumentation shall be located at half of one pipe diameter upstream of the inlet.

6.3.4 Temperature measurement for piped discharge

The discharge temperature is the total temperature, T_2, measured at the process air discharge point. The temperature instrumentation shall be located one pipe diameter downstream of the discharge and 90° relatively rotated to the pressure measurement.

6.4 Measurement of humidity

If the gas contains moisture, the humidity shall be checked during the test. The humidity shall be measured at the process air inlet point with an instrument having an accuracy of ±3 % or better.

6.5 Measurement of rotational frequency

Rotational speed shall be determined by using methods that have an accuracy of ±0,2 % or better.

6.6 Measurement of flow rate

The actual volume flow rate is the net mass flow rate through the process connection of the packaged compressor discharge. All seal losses and side streams not delivered to the process piping connection of the packaged compressor shall be excluded from the delivered mass flow rate evaluation.

The actual volume flow rate of the compressor shall be measured by performing a test as indicated in either ISO 5167-1, ISO 9300 or ISO 17089-1 as appropriate:

— ISO 5167-1 – Pressure differential devices;

— ISO 9300 – Critical flow venturi nozzle devices;

— ISO 17089-1 – Ultrasonic devices.

Overall uncertainty of measured value shall be ±1,5 % or better.

6.7 Measurement of external coolant flow rate

The mass flow of liquid external coolant shall be determined by using a measuring method with an accuracy of ±5 % of the measured value or better.

6.8 Measurement of power and energy

6.8.1 General

Electric power of the packaged compressor shall be referred to the electrical input terminals. Factors influencing the measurement, such as voltage drop in supply cables or measurement systems, shall be taken into account.

The two-wattmeter method or some other method with similar accuracy shall be used.

Current and voltage transformers shall be chosen to operate as near to their rated load as possible so that their ratio error will be minimized.

Electrical measurement equipment shall be capable of measuring true root mean square (RMS) current, true RMS voltage, and real power up to the 40^{th} harmonic of fundamental supply source frequency.

Overall uncertainty of the measured value shall be ±0,75 % or better.

The power supply shall

— maintain the voltage greater than or equal to 95 % and less than or equal to 110 % of the rated value of the motor, and

— maintain the voltage unbalance of the power supply within ±3 % of the rated values of the motor.

For inverter applications, the wire power-measuring instrument shall be capable of handling the distorted voltage and current waveforms and phase relationship of the power factor caused by the harmonics and Electro Magnetic Interference (EMI) as a result of an inverter’s high-speed switching mode.

6.9 Calibration of instruments

Calibration records of the instruments shall be available prior to the test.

Recalibration after the test shall be carried out for those instruments of primary importance that are liable to variation in their calibration because of use during the test.

Any change in the instrument calibrations, which will create a variation exceeding the class of accuracy of the instrument, may be cause for rejecting the test.

7 Test

7.1 General test process

In order to compare the performance data of low-pressure compressors with different technologies, it is necessary to test these low-pressure compressors under the same conditions and with the same methods by applying the same principles and process steps. If a performance metric (i.e., isentropic efficiency) is based on a reference process, it shall be the same for both positive displacement and dynamic low-pressure compressors. For the derivation of correction formulae, the most appropriate ideal reference process shall be used, which in this methodology is different for different types of low-pressure compressors. For positive displacement low-pressure compressors with or without internal compression the combined isentropic and isochoric process and for dynamic low-pressure compressors of any kind the isentropic process is used.

The test shall be carried out at an appropriate test field under prevailing conditions. No changes to geometry are allowed between test and specified conditions. For the set-up, the low-pressure compressor is connected to the test equipment. Correct cooling conditions shall be established. The low-pressure compressor shall run for warm-up against test discharge pressure until steady state conditions are reached and the temperature remains constant at the inlet and discharge of the flow measuring device. The low-pressure compressor package shall operate at the steady state condition for the duration of data collection for each test point.

In Figure 1, the general process of testing is shown in a schematic.

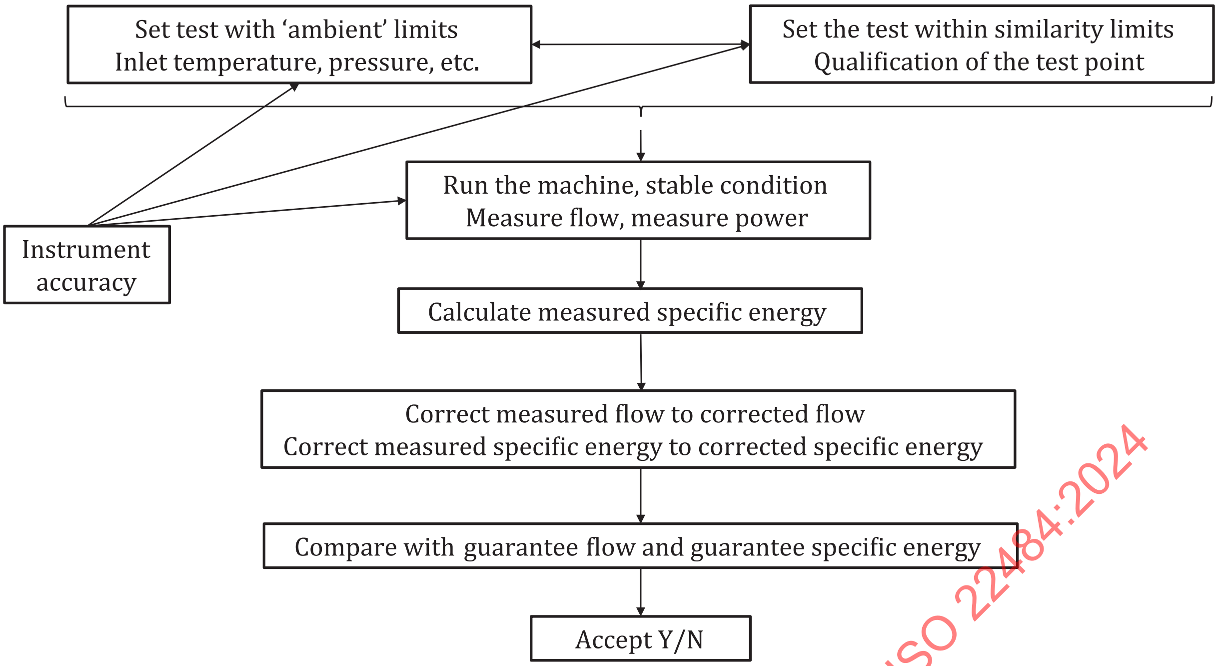

Figure 1 — Overview of test process

For variable flow low-pressure compressor this method can be repeated at several flow rates to establish the performance over the operating range of the compressor at the specified discharge pressure. For the purpose of general data sheets, or in case there is no specific agreement with the client on how to test variable flow compressors, 5 flow rates shall be tested. The minimum and maximum flow rates that can be achieved continuously under guarantee conditions shall be specified and tested by the manufacturer. Furthermore, three additional flow rates evenly spread within the total flow rate range of the low-pressure compressor shall be specified and tested. For each test point, the adequate test loads shall be determined in the same way as for a single working point.

7.2 Allowed deviation of rotational speed between test and guarantee

The manufacturer shall identify the expected rotational speed for the specified point while operating at guarantee conditions prior to conducting the test.

When allowed, the deviation for rotational speed between test and guarantee is (±5 %).

- Fixed speed compressors:
  - displacement:
    - package motor: no speed correction allowed;
    - test motor: speed correction allowed;
  - dynamic:
    - package motor: no speed correction allowed;
    - test motor: speed correction allowed;
- variable speed compressors:
  - displacement:
    - package motor: no speed correction allowed;

  — test motor: speed correction allowed;

— dynamic:

  — package motor: speed correction allowed together with reduced speed calculations;

  — test motor: speed correction allowed together with reduced speed calculations.

In the case the reduced speed calculations results in a rotational speed set point that deviates by more than ±5 % from the expected rotational speed, then a second test at the guarantee speed is to be conducted, to verify the electromechanical efficiency. See procedure for two speed testing in 7.10.

7.3 Allowed deviation of ambient conditions

7.3.1 Testing against general performance data

The following restrictions apply to the ambient conditions if the low-pressure air compressor is tested against general performance data, published by the vendor and being independent from an individual customer order. Examples include but are not limited to, data sheets, machine brochures, and the vendor's website. In such case, the contract of supply shall refer to the respective public data source. For guarantee conditions, 5.2 applies.

The ambient conditions shall not deviate compared to the guarantee conditions by more than

— inlet temperature: ±10 K, and

— inlet pressure: ±10 %.

7.3.2 Testing against customer specified data sheets

If ambient conditions are specified by the customer, the vendor provides customized performance data for these particular conditions, e.g. in the contract of supply or customized data sheets. In such case, there is no limit for the deviations between the test conditions and the guarantee conditions.

7.4 Allowed deviation of preconditions

Below are the allowed deviations of the precondition data, if applicable:

— liquid external coolant temperature: ±15 K;

— mass flow of liquid external coolant: ±10 %.

7.5 Allowed deviation of machine Mach number

It is essential to keep the dimensionless numbers as similar as possible between the guarantee conditions and the test conditions.

Calculation of the machine Mach number can be done as follows by Formula (1):

$$Ma = \frac{u}{c_1} = \frac{u}{\sqrt{\kappa_1 \cdot R \cdot T_1}} \tag{1}$$

For dynamic low-pressure compressors, the deviation of machine Mach number shall be between -5 % and +5 %.

For positive displacement low-pressure compressors there is no applicability on the deviation of machine Mach number.

7.6 Selection of test flow

7.6.1 Selection of flow setting

7.6.1.1 Fixed flow, positive displacement and dynamic low-pressure compressors

For low-pressure compressors with no possibility to adjust the flow, the flow will result from the actual speed at which the low-pressure compressor is running (e.g. constant speed low-pressure compressors with no flow-adjusting device).

7.6.1.2 Variable flow, positive displacement low-pressure compressors

The specified volume flow shall be matched within the given tolerance by adjusting the low-pressure compressor (by speed or by positive displacement per revolution, if adjustable).

7.6.1.3 Variable speed, dynamic compressors

The test point of the low-pressure, variable-speed, dynamic compressor shall be achieved by maintaining the dimensionless numbers within the limits established in 7.8.1.1. To establish the operating point, the following method is recommended, but not mandatory:

An initial rotational speed can be determined by matching machine Mach numbers at guarantee and test conditions. Next, the target flow and discharge pressure will result from maintaining both the flow and work coefficients respectively. These coefficients should be matched between test and guarantee conditions with the least possible deviations. This is done by adapting rotational speed without exceeding the dimensionless limits given in 7.8.1.1.

The manufacturer shall provide the reference rotational speed and settings of variable geometry for the specified operating point at the guarantee conditions in advance of test.

The speed setting for the test shall be achieved by maintaining the dimensionless numbers constant as follows:

Rotational speed is determined from the machine Mach numbers at specified and test conditions keeping machine Mach number constant, flow will result from the rotational speed and the discharge pressure setting as explained below.

7.6.1.4 Fixed speed, variable flow, dynamic compressors

The manufacturer shall identify the reference rotational speed and settings of variable geometry for the specified operating point at the guarantee conditions in advance of the test. Rotational speed is determined by the speed of the drive. Test setting for flow will result from the discharge pressure setting as explained below.

7.7 Selection of test pressure

7.7.1 Note: The following terms are required for R_{te} and K_{te} with formulae and source.

7.7.2 For positive displacement low-pressure compressors with or without internal compression

The discharge pressure for positive displacement low-pressure compressors with internal compression (internal volume ratio $V_i > 1$) or without internal compression (internal volume ratio $V_i = 1$) shall be set such that the specific combined compression work in the prevailing test conditions matches those in the guarantee conditions, as shown in Formula (2):

$$y_{comb,te} = y_{comb,g} \quad (2)$$

See Annex D for explanation of combined compression work.

The target discharge pressure can then be calculated as follows by Formula (3):

$$p_{2,\text{target}} = \left[\frac{y_{\text{comb,g}}}{R_{1,te} \cdot T_{1,\text{te}}} - \frac{\kappa_{\text{te}}}{\kappa_{\text{te}} - 1} \left(\frac{1}{\kappa_{\text{te}}} \cdot V_{\text{i}}^{\kappa_{\text{te}} - 1} - 1 \right) \right] \cdot V_{\text{i}} \cdot p_{1,\text{te}} \tag{3}$$

7.7.3 For dynamic low-pressure compressors

The discharge pressure for dynamic low-pressure compressors shall be set in such a way that the specific isentropic compression work in the prevailing test conditions matches the one in the guarantee conditions multiplied with the square of the speed ratio, in order to achieve similarity, as shown in Formula (4):

$$y_{\text{s,te}} = y_{\text{s,g}} \cdot \left(\frac{u_{\text{te}}}{u_{\text{g}}} \right)^2 \tag{4}$$

The target discharge pressure can then be calculated as given in Formula (5):

$$p_{2,\text{target}} = p_{1,\text{te}} \cdot \left\{ 1 + \left(\frac{\kappa_{\text{te}} - 1}{\kappa_{\text{te}}} \right) \left[\frac{y_{\text{s,g}} \cdot \left(\frac{u_{\text{te}}}{u_{\text{g}}} \right)^2}{\left(R_{1,\text{te}} \cdot T_{1,\text{te}} \right)} \right] \right\}^{\left(\frac{\kappa_{\text{te}}}{\kappa_{\text{te}} - 1} \right)} \tag{5}$$

7.8 Allowed deviation of flow and work coefficient

7.8.1 Allowed deviations to be checked for test validity

7.8.1.1 For dynamic low-pressure compressors

It is essential to keep the dimensionless numbers as close as possible between the guarantee conditions and the test conditions, the deviations are as follows:

— work coefficient: - 2 %, +2 %;

— flow coefficient: -2 %, +2 %;

— machine Mach number: -5 %, +5 %.

7.8.1.2 For positive displacement low-pressure compressors

It is essential to keep the following numbers as close as possible between the guarantee conditions and the test conditions, the deviations being:

— work y_{comb}: -2 %, +2 %;

— flow coefficient: -2 %, +2 %.

For typical positive displacement low-pressure compressors with fixed geometry, where the flow is linear to speed, this latter condition shall be fulfilled.

7.9 Fluctuations on the specific test readings/results during test at steady state

Readings are to be taken at steady state which is defined as the state in which the variation in the difference between inlet and discharge temperatures is within 1 K for a period of three minutes or more.

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

For each load, enough readings shall be taken to indicate that steady-state conditions have been reached. The number of readings and the intervals shall be chosen to demonstrate repeatability.

For individual readings, the limits on fluctuations in Table 4 below apply:

Table 4 — Permissible fluctuations of test readings

Measurement (symbol)	Symbol	Maximum permissible fluctuation from average during any set of readings
Inlet pressure	$p_{1,\mathrm{te}}$	1 %
Inlet temperature	T_1	
Discharge pressure	$p_{2,\mathrm{te}}$	0,5 %
Flow	$q_{V,\mathrm{te}}$	1 %
Speed (rotational speed)	n_{te}	0,5 %
Electrical power	P_{te}	1 %
Supply voltage	U	2 %

7.10 Two-speed test

7.10.1 General

If the conditions at the test site differ significantly from the preconditions of the guarantee, the situation may arise for variable speed dynamic low-pressure compressors where the limits on the work coefficient, flow coefficient and machine Mach number as defined in 7.8.1.1. exceed the limits imposed on rotational speed as defined in 7.2. Although compression efficiency is still comparable, this may result in a difference in electromechanical efficiency of the packaged low-pressure compressor between test and guarantee conditions, which cannot be neglected.

7.10.2 First test

The test is conducted as 7.6.1.3, 7.7.3 and 7.8.1.1. The test is corrected according to 8.2, 8.3.1, 8.4 and 8.5; corrections of test results as the standard procedure.

In addition to the standard procedure then the shaft power shall be measured/calculated. This will require the discharge temperature $T_{2,\mathrm{te1}}$ to be measured as given in Formula (6):

$$P_{\mathrm{shaft,te1}} = \dot{q}_{\mathrm{m,te1}} \cdot \frac{\kappa_{\mathrm{te1}}}{\kappa_{\mathrm{te1}} - 1} \cdot R_{\mathrm{te1}} \cdot \left(T_{2,\mathrm{te1}} - T_{1,\mathrm{te1}}\right) \tag{6}$$

The electromechanical efficiency is calculated as Formula (7):

$$\eta_{\mathrm{em,te1}} = \frac{P_{\mathrm{shaft,te1}}}{P_{\mathrm{pack,te1}}} \tag{7}$$

7.10.3 Second test

The second test is made to measure the electromechanical efficiency at the customer specified conditions as guaranteed. For the specific test point.

Set point:

Set the test speed to be equal to guaranteed rotational speed $n_{\mathrm{te2}} = n_{\mathrm{g}}$, within the limit of ±2 %.

Load the drive train shaft to be equal to the corrected shaft power $P_{\text{shaft,co,g,te1}}$ of test 1, within the limit of ±2 %, as shown in Formula (8):

$$P_{\text{shaft,te2}} = \dot{q}_{\text{m,te2}} \cdot \frac{\kappa_{\text{te2}}}{\kappa_{\text{te2}} - 1} \cdot R_{\text{te2}} \cdot \left(T_{2,\text{te2}} - T_{1,\text{te2}}\right) \tag{8}$$

The electromechanical efficiency is calculated as Formula (9):

$$\eta_{\text{em,te2}} = \frac{P_{\text{shaft,te2}}}{P_{\text{pack,te2}}} \tag{9}$$

The final correction of packaged power is made accordingly to 8.6.

8 Correction of test results

8.1 General

The test results measured at the test bench (subscript $_{\text{te}}$) shall be recalculated to the corrected values (subscript $_{\text{co}}$) with the formulas in the sections that follow. The equations consider the guarantee conditions (subscript $_{\text{g}}$) to calculate these corrected values.

NOTE In this calculation scheme, there is no correction for the difference in Reynolds number.

8.2 Correction of measured flow (variable speed packaged compressors, only)

Calculate the corrected volume flow as in Formula (10):

$$q_{V1,\text{co}} = q_{V1,\text{te}} \cdot \frac{u_{\text{g}}}{u_{\text{te}}} \tag{10}$$

NOTE Tip speed ratio $\left(\frac{u_{\text{g}}}{u_{\text{te}}}\right)$ is equal to the rotational speed ratio $\left(\frac{n_{\text{g}}}{n_{\text{te}}}\right)$.

Using Formula 10 to compensate differences in rotational speed is only allowed for dynamic variable speed packaged compressors or if the motor is different between test and the site of assembly. In all other cases rotational speed and tip speed ratio shall be set to 1. Scaled tests are not allowed.

8.3 Correction of measured pressure

As the discharge pressure during testing differs from the target test discharge pressure, this pressure with its deviations shall be corrected to the guaranteed conditions. The corrected pressure compared to the guarantee pressure shall be within the limits as listed in Table 5.

8.3.1 For dynamic low-pressure packaged compressors

First, calculation of the corrected compression work shall be made with Formula (11):

$$y_{\text{s,co}} = y_{\text{s,te}} \cdot \left(\frac{u_{\text{g}}}{u_{\text{te}}}\right)^2 \tag{11}$$

NOTE Tip speed ratio $\left(\frac{u_{\text{g}}}{u_{\text{te}}}\right)$ is equal to the rotational speed ratio $\left(\frac{n_{\text{g}}}{n_{\text{te}}}\right)$.

Using Formula (11) to compensate differences in rotational speed is only allowed for dynamic variable speed compressors or if the motor is different between test and the site of assembly. In all other cases rotational speed and tip speed ratio shall be set to 1.

Then, the corrected pressure ratio is calculated as Formula (12):

$$\pi_{co} = \left[1 + \left(\frac{\kappa_g - 1}{\kappa_g}\right)\left(\frac{y_{s,co}}{R_{1,g} \cdot T_{1,g}}\right)\right]^{\left(\frac{\kappa_g}{\kappa_g - 1}\right)} \tag{12}$$

8.3.2 For positive displacement low-pressure packaged compressors

For positive displacement low-pressure compressors, the corrected pressure ratio shall be calculated as Formula (13):

$$\pi_{co} = \left[\frac{y_{comb,te}}{R_1 \cdot T_1} - \frac{\kappa_g}{\kappa_g - 1}\left(\frac{1}{k} \cdot V_i^{\kappa_g - 1} - 1\right)\right] \cdot v_i \tag{13}$$

8.3.3 For positive displacement and dynamic low-pressure packaged compressors

To calculate corrected discharge pressure, Formula (14) shall be used using the appropriate corrected pressure ratio from above:

$$p_{2,co} = \pi_{co} \cdot p_{1,g} \tag{14}$$

8.4 Correction of specific energy demand

The tested specific energy demand e_{te} is the ratio of the measured power during test P_{te} and the measured flow $q_{V1,te}$, calculated as Formula (15):

$$e_{te} = \frac{P_{te}}{q_{V1,te}} \tag{15}$$

The specific energy demand is then corrected for the differences in density ρ and work y between test and guarantee conditions. Although the work is set in the test according to the work in guarantee conditions, any difference between actual test conditions and the set point is considered in the correction formula.

Depending on the type of low-pressure compressor, a different reference process for the specific compression work shall be used.

For positive displacement low-pressure compressors with internal compression (internal volume ratio $V_i > 1$) or without internal compression (internal volume ratio $V_i = 1$) the specific energy demand shall be corrected as follows by Formulae (16) and (17):

$$e_{CO} = \frac{\rho_{1,g}}{\rho_{1,te}} \cdot \frac{y_{comb,co}}{y_{comb,te}} \cdot e_{te} \tag{16}$$

$$e_{CO,g} = \frac{\rho_{1,g}}{\rho_{1,te}} \cdot \frac{y_{comb,g}}{y_{comb,te}} \cdot e_{te} \tag{17}$$

For dynamic low-pressure compressors, the correction is as follows by Formulae (18) and (19):

$$e_{CO} = \frac{\rho_{1,g}}{\rho_{1,te}} \cdot \frac{y_{s,co}}{y_{s,te}} \cdot e_{te} \tag{18}$$

$$e_{CO,g} = \frac{\rho_{1,g}}{\rho_{1,te}} \cdot \frac{y_{s,g}}{y_{s,te}} \cdot e_{te} \tag{19}$$

For the guaranteed specific compression work ($y_{comb,g}$ or $y_{s,g}$), the stated guarantee values of pressure and inlet temperature are to be used. For the tested specific compression work ($y_{comb,te}$ or $y_{s,te}$), the measured test values of pressure and inlet temperature are to be used.

8.5 Corrected packaged compressor power consumption

The power consumption of the packaged compressor in the guarantee conditions can be expressed in two different ways.

The power consumption of the tested packaged compressor at guarantee conditions (effectively delivering the corrected flow $q_{V1,co}$) will be as follows by Formula (20):

$$P_{co} = e_{co} \cdot q_{V1,co} \tag{20}$$

The power consumption of the tested packaged compressor at guarantee conditions and at the guarantee flow (thus this is the case in which the packaged compressor matches the guarantee flow $q_{V1,g}$) will be as follows by Formula (21):

$$P_{co,g} = e_{co,g} \cdot q_{V1,g} \tag{21}$$

8.6 Power correction of the two-speed test

First test corrections

Correction of measured shaft power at test 1, calculated as Formula (22):

$$P_{shaft,co,gte1} = P_{co,g,te1} \cdot \eta_{em,te1} \tag{22}$$

Second test corrections

Correction of corrected shaft power and input power at test 1, based on electromechanical efficiency measured at test 2.

Final corrected packaged compressor power input, calculated as Formula (23):

$$P_{co,g,te1,te2} = \frac{P_{shaft,co,g,te1}}{\eta_{em,te2}} \tag{23}$$

Final corrected specific energy of a packaged compressor, calculated as Formula (24):

$$e_{co,g,te1,te2} = \frac{P_{co,g,te1,te2}}{q_{V1,g}} \tag{24}$$

8.7 Calculated package isentropic efficiency

Isentropic efficiency of the compressor package is the ratio of power required for an ideal isentropic compression process to the actual packaged compressor power input used at a given load point.

The isentropic efficiency of the compressor package at guarantee conditions will be as follows by Formula (25):

$$\eta_{s,co} = \frac{P_{s,co}}{P_{co}} \tag{25}$$

where the isentropic power input for the compressor package at the guarantee condition is calculated by Formula (26):

$$P_{s,co} = q_{V1,co} \cdot p_{1,g} \cdot \frac{k_g}{k_g - 1} \cdot \left[\left(\frac{p_{2,co}}{p_{1,g}} \right)^{\frac{k_g - 1}{k_g}} - 1 \right] \tag{26}$$

8.8 Comparison of corrected values with guaranteed values

The test results are corrected to the specified operating conditions with the purpose of comparability with the guaranteed or specified performance.

The comparisons shall include:

— corrected specific energy e_{co} with the guaranteed specific energy e_g;

— corrected volume flow rate $q_{V1,co}$ with the guaranteed volume flow rate $q_{V1,g}$;

— corrected discharge pressure $p_{2,co}$ with the guaranteed $p_{2,g}$;

— overall packaged compressor power consumption P_{co} with the guarantee value P_g.

Table 5 — Acceptance tolerances

Volume flow rate at specified conditions $q_{V1,g}$ $m^3/s \times 10^{-3}$	Volume flow rate at specified conditions $q_{V1,g}$ m^3/min	Volume flow rate at specified conditions $q_{V1,g}$ m^3/h	Volume flow rate $q_{V1,g}$ %	Specific energy[a] e %	Discharge pressure p_2 %	Idle power P_{idle} %	Standby power $P_{standby}$ %
0	0	0	n/a	n/a	n/a	±10	±10
$0 < q_{V1,g} \leq 8,3$	$0 < q_{V1,g} \leq 0,5$	$0 < q_{V1,g} \leq 30$	±7	±8	0÷ +1	n/a	n/a
$8,3 < q_{V1,g} \leq 25$	$0,5 < q_{V1,g} \leq 1,5$	$30 < q_{V1,g} \leq 90$	±6	±7			
$25 < q_{V1,g} \leq 250$	$1,5 < q_{V1,g} \leq 15$	$90 < q_{V1,g} \leq 900$	±5	±6			
$250 < q_{V1,g}$	$15 < q_{V1,g}$	$900 < q_{V1,g}$	±4	±5			

[a] The tolerance band on packaged compressor power is defined by the tolerance on specific energy consumption.

The corrected values are to be compared to the guaranteed values. If the values are within the limits as defined in this standard, then the low-pressure compressor passes the test and is accepted. If the values are not within the limits as defined in this standard, then the low-pressure compressor fails the test and is not accepted.

8.9 Examples of calculations

Examples of calculations, comparison and test reports are given in Annex C.

9 Test report

9.1 Test report content

At a minimum, the test report shall include the following:

- test piping arrangement indicating pipe sizes and lengths, pressure and temperature measurement locations, flow measurement arrangement, valve location(s), and verification of compliance with ISO measurement standards;
- a reference to this document, i.e. ISO 22484:2024;
- original test logs including all recorded data required for calculations;
- detailed representative calculation for one test point;
- instrument calibration certificates;
- date of test;
- test report number;
- low-pressure compressor type, manufacturer, model, serial number, date of manufacture;
- manufacturer's packaged compressor checklist per Annex A.

9.2 Test results summary

Test results shall be summarised, an example of the format is given in Annex B.

Annex A
(normative)

Equipment checklist

A packaged compressor checklist shall be presented which clearly defines the scope of testing, see Table A.1. Where items are not included or not necessary, the column "Not applicable" should be marked.

Table A.1 — Equipment checklist

Section	Item	Included	Accounted for value (units)	Not applicable
Process air in/out	Process air filter		Δp (Pa)	
	Inlet silencer		Δp (Pa)	
	Inlet guide vanes		------------	
	Inlet unload valve or throttled valve		Δp (Pa)	
	Variable diffuser		------------	
	Blow off valve[a]		Δp (Pa)	
	Check valve		Δp (Pa)	
	Discharge silencer		Δp (Pa)	
	Additional inlet losses[b]		Δp (Pa)	
	Additional discharge losses[b]		Δp (Pa)	
Drive train	Main drive motor efficiency		η (%)	
	Frequency inverter		η (%)	
	EMC filter		η (%)	
	Choke		η (%)	
	Starter		η (%)	
	Sinus filter		η (%)	
	Power transmission (gear box, belts)		η (%)	
	Additional components[c]		η (%)	
Ancillaries, electrical power input	Local control system		P (kW)	
	Cooling circulation (liquid)		P (kW)	
	Lubrication system		P (kW)	
	Main drive cooling fan		P (kW)	
	Cooling air fans		P (kW)	
	Heat exchanger fans		P (kW)	
	Additional components[c]		P (kW)	

[a] Pressure loss when closed.

[b] The cause of the additional losses shall be detailed.

[c] Additional components shall be identified and accounted for.

Annex B
(informative)

Test result summary

Equipment checklist as defined in Annex A should also be included with the test results summary, see Table B.1.

Table B.1 — Example of test result summary

Test data	Symbol	Unit (metric)	Reference section	Numerical values		
				guarantee	test	corrected
Test number	#			1g	1te	1Co
Test period start/end	time	min				
Barometric pressure	$p_{0.amb.abs}$	Pa	6.2.2			
Inlet pressure	$p_{1.abs}$	Pa	6.2.3/6.2.4			
Inlet temperature	T_1	K	6.3			
Relative humidity	$\varphi_{rel,1}$	ratio	3.22			
Isentropic exponent	κ_{wet}	ratio	3.23			
Gas constant	R_{wet}	J/(kg*K)	7.7.1			
Inlet density	ρ_1	kg/m^3	7.7.1			
Discharge temperature	T_2	K	7.10			
Compressor speed	n	1/s	7.2			
Supply voltage	U	V	5.2			
Supply frequency	f	Hz	5.2			
External coolant inlet temperature	$T_{1,cool}$	K	5.2/6.3			
External coolant flow	$q_{m,cool}$	kg/s	5.2/6.7			
Specific isentropic work	y_s	J/kg	3.29			
Specific package work	y	J/kg	-			
Package isentropic efficiency	η	%	3.26			
Rotor tip speed	u	m/s	3.33			
machine Mach number	Ma	-	3.34/7.5			
Key performance indicators						
Inlet volume flow rate	q_{V1}	m^3/s	6.6/8.2			
Discharge pressure	p_2	Pa	8.3			
Package input power	P	W	8.5			
Specific energy	e	W/m^3/s	3.28/8.4			

Annex C
(informative)

Examples of acceptance test reports/calculations

C.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 manufacturer. In order to conduct a correct guarantee comparison, the schedule for the acceptance tests, the variables to be measured, the measuring methods to be used and, possibly, the gas data equations and evaluation systems and procedures should be agreed upon between the purchaser and the manufacturer and/or any third party also involved at a sufficiently early stage (if possible, during the actual contract negotiations) on the basis of this standard.

C.2 Test examples

Test example number	Compressor type	Is flow and pressure adjustable?
1	Dynamic compressor	The example is valid for fixed and variable flow packaged compressors
2	Positive displacement compressor	The example is valid for fixed and variable flow packaged compressors
3	Dynamic compressor	two speed test: The example is valid for fixed and variable flow packaged compressors using variable speed

C.2.1 The guaranteed performance point

The performance is defined at the package level. The air inlet condition is the same for all examples and the guarantee is the same for examples 1 and 2.

C.2.1.1 Purpose of tests

Verification of performance guarantees for one specific data point.

C.2.1.2 Test arrangement

The boundary of the low-pressure compressor is as default set to cover any part that will have influence on the performance. All parts shall be covered by the equipment checklist.

C.2.1.3 Equipment checklist

Section	Item	Included	Accounted for value (units)	Not applicable
Process air in/out	Process air filter	x	Δp (Pa)	
	Inlet silencer	x	Δp (Pa)	
	Inlet guide vanes		------------	x
	Inlet unload valve or throttled valve		Δp (Pa)	x
	Variable diffuser vanes		------------	x
	Blow off valve[a]	x	Δp (Pa)	
	Check valve	x	Δp (Pa)	
	Discharge silencer		Δp (Pa)	x
	Additional inlet losses[b]		Δp (Pa)	x
	Additional discharge losses[b]		Δp (Pa)	x
Drive train	Main drive motor efficiency	x	η (%)	
	Frequency inverter	x	η (%)	
	EMC filter	x	η (%)	
	Choke	x	η (%)	
	Starter	x	η (%)	
	Sinus filter	x	η (%)	
	Power transmission (gear box, belts… transformers etc.)		η (%)	x
	Additional components		η (%)	x
Ancillaries, electrical power input	Local control system	x	P (kW)	
	Cooling circulation (liquid)	x	P (kW)	
	Lubrication system	x	P (kW)	
	Main drive cooling fan	x	P (kW)	
	Cooling air fans	x	P (kW)	
	Heat exchanger fans		P (kW)	x
	Additional components		P (kW)	x

[a] Pressure loss when closed.

[b] The cause of the additional losses should be detailed.

C.3 Advanced test calculation example 1 (dynamic compressor)

C.3.1 Guarantee conditions

	Symbol	Numerical value	Units
Inlet pressure	$p_{1,g}$	100 000	Pa
Inlet temperature	$T_{1,g}$	293,15	K
Inlet relative humidity	$\varphi_{rel,g}$	0,0	%

C.3.2 Object of guarantee

	Symbol	Numerical value	Units		
Inlet volume flow	$q_{V1,g}$	0,413 3	m^3/s		
Discharge pressure	$p_{2,g}$	140 000	Pa		
Electric power at the electric input terminals	P_g	23 000	W		

Acceptance tolerances (refer to table 5, section 8.7) for this specific test point:

e_{pack}=±5 %; q_{v1}=±4 %; p_2=+1 %

C.3.3 General calculation on guarantee data

General calculation made on the guaranteed performance data used for verification of similarity and acceptance comparison: As the guarantee data in these examples is the same the calculation is valid for all compressor types.

Step 1: Preconditions of the guarantee

— Ambient pressure $p_{1,g} = 100\,000$ Pa

— Ambient temperature $T_{1,g} = 293{,}15$ K

— Ambient humidity $\varphi_{rel,g} = 0$ %

Step 2: Mixture of air and water vapor (humid air) on guarantee data

— Ambient temperature $t_g = (T_{1,g} - 273{,}15)$ °C

— Calculate the vapor pressure:

The partial pressure of water vapor is found using the steam tables: (ref. NBS/NRC steam tables ISBN 0-89116-353-0 paper)

$$p_{vap.sat,g} = \left(\begin{array}{l} 6{,}251\,080\,56 \cdot 10^{-3} + 4{,}338\,188\,3 \times 10^{-4} \cdot t + 1{,}734\,107\,7 \times 10^{-5} \cdot t^2 + 1{,}210\,936 \times 10^{-7} \cdot \\ \quad t^3 + 6{,}292\,408 \times 10^{-9} \times t^4 \end{array} \right) \cdot 98\,066{,}5$$

Vapor pressure $p_{vap.sat,g} = 2\,338$ Pa

Relative humidity

$$\varphi_{rel,g} = \frac{p_{vap,g}}{p_{vap,sat,g}} = 0\ \%$$

Therefore: $p_{vap,g} = 0$ Pa

— Vapor content $x_{air,g} = 0{,}622 \cdot \frac{\varphi_{rel,g} \cdot p_{vap,sat,g}}{p_{1,g} - \varphi_{rel,g} \cdot p_{vap,sat,g}} = 0$ $\frac{kg}{kg}$

— Isentropic exponent dry $\kappa_{dry} \approx 1{,}4$

— Isentropic exponent $\kappa_{wet,g} = \kappa_{dry} \cdot (1 - 0{,}11 \cdot x_{air,g}) = 1{,}4$

— Gas constant (dry) $R_{air} = 287{,}1$ $\frac{J}{kg \cdot K}$

— Gas constant (humid air) $R_{wet,g} = \left[R_{air} \cdot \left(1 + \frac{x_{air,g}}{x_{air,g} + 1} \cdot 0{,}6081 \right) \right] = 287{,}1$ $\frac{J}{kg \cdot K}$

— Ambient air density $\rho_{1,g} = \left(\frac{p_{1,g}}{R_{wet,g} \cdot T_{1,g}} \right) = 1{,}188$ $\frac{kg}{m^3}$

— Inlet specific volume $V_{1,g} = \left(\frac{1}{\rho_{1,g}} \right) = 0{,}842$ $\frac{m^3}{kg}$

— Inlet volume flow rate $q_{V1,g} = \frac{q_{m2,g}}{\rho_{1,g}} = 0{,}4133$ $\frac{m^3}{s}$

— Mass flow rate $q_{m2,g} = q_{V1,g} \cdot \rho_{1,g} = 0{,}04911$ $\frac{kg}{s}$

— Discharge pressure $p_{2,g} = 140\,000$ Pa

— Measured package input power $P_g = 23\,000$ W

— Impeller speed $n_g = 495{,}83$ 1/s

— Impeller diameter $D = 0{,}157$ m

Step 3: Calculate key performance indicators for guarantee data

— Specific isentropic work

$$y_{s,g} = \frac{\kappa_{wet,g}}{\kappa_{wet,g} - 1} \cdot R_{wet,g} \cdot T_{1,g} \cdot \left[\left(\frac{p_{2,g}}{p_{1,g}} \right)^{\left(\frac{\kappa_{wet,g} - 1}{\kappa_{wet,g}} \right)} - 1 \right] = 29\,725 \quad \frac{J}{kg}$$

— Specific energy $e_g = \frac{P_g}{q_{V1,g}} = 55\,650$ $\frac{W \cdot s}{m^3}$

— Specific package work $y_g = \frac{P_g}{q_{m2,g}} = 46\,837$ $\frac{J}{kg}$

— Package isentropic efficiency $\eta_g = \frac{y_{s,g}}{y_g} \times 100 = 63{,}46$ %

— Rotor tip speed $u_g = \pi \cdot D \cdot n_g = 244{,}6$ $\frac{m}{s}$

— Machine Mach number $Ma_g = \frac{u}{c_1} = \frac{u_g}{\sqrt{\kappa_{wet,g} \cdot R_{wet,g} \cdot T_{1,g}}} = 0{,}7125$

C.3.4 General calculations on inlet test data

General calculation made on the tested performance data used for verification of similarity and acceptance comparisons. The tested inlet data in these examples is the same, so the calculations are valid for all compressor types.

Test inlet conditions

— Ambient pressure $P_{1,\text{te}} = 101\,000$ Pa

— Ambient temperature $T_{1,\text{te}} = 291{,}88$ K

— Ambient humidity $\varphi_{\text{rel,te}} = 33$ %

Step 4: Mixture of air and water vapor (humid air) on test data

— Ambient temperature $t_{\text{te}} = (T_{1,\text{te}} - 273{,}15)$ °C

— Calculate the vapor pressure:

The partial pressure of water vapor is found using the steam tables: (ref. NBS/NRC steam tables ISBN 0-89116-353-0 paper)

$$p_{\text{vap.sat,te}} = \begin{pmatrix} 6{,}251\,080\,56 \cdot 10^{-3} + 4{,}338\,188\,3 \times 10^{-4} \cdot t_{\text{te}} + 1{,}734\,107\,7 \times 10^{-5} \cdot t_{\text{te}}{}^{2} \\ + 1{,}210\,936 \times 10^{-7} \cdot t_{\text{te}}{}^{3} + 6{,}292\,408 \times 10^{-9} \times t_{\text{te}}{}^{4} \end{pmatrix} \cdot 98\,066{,}5$$

— Vapor pressure $p_{\text{vap.sat,te}} = 2\,160$ Pa

— Relative humidity

$$\varphi_{\text{rel,te}} = \frac{p_{\text{vap,te}}}{p_{\text{vap,sat,te}}} = 33\ \%$$

$$p_{\text{vap,te}} = 712{,}8 \quad \text{Pa}$$

— Vapor content

$$x_{\text{air,te}} = 0{,}622 \cdot \frac{\varphi_{\text{rel,g}} \cdot p_{\text{vap,sat,te}}}{p_{1,\text{te}} - \varphi_{\text{rel,g}} \cdot p_{\text{vap,sat,te}}} = 0{,}004\,42 \quad \frac{\text{kg}}{\text{kg}}$$

— Isentropic exponent dry $\kappa_{\text{dry}} \approx 1{,}4$

— Isentropic exponent $\kappa_{\text{wet,te}} = \kappa_{\text{dry}} \cdot (1 - 0{,}11 \cdot x_{\text{wet,te}}) = 1{,}399$

— Gas constant (dry) $R_{\text{air}} = 287{,}1$ $\frac{\text{J}}{\text{kg} \cdot \text{K}}$

— Gas constant (humid air)

$$R_{\text{wet,te}} = \left[R_{\text{air}} \cdot \left(1 + \frac{x_{\text{air,te}}}{x_{\text{air,te}} + 1} \cdot 0{,}608\,1 \right) \right] = 287{,}9 \quad \frac{\text{J}}{\text{kg} \cdot \text{K}}$$

— Ambient air density

$$\rho_{1,\text{te}} = \left(\frac{p_{1,\text{te}}}{R_{\text{wet,te}} \cdot T_{1,\text{te}}} \right) = 1{,}202 \quad \frac{\text{kg}}{\text{m}^3}$$

— Inlet specific volume

$$V_{1,\text{te}} = \left(\frac{1}{\rho_{1,\text{te}}} \right) = 0{,}832 \quad \frac{\text{m}^3}{\text{kg}}$$

Step 5: Calculate discharge pressure of the test setup (dynamic compressors only)

Reduced test speed

$$Ma_{\text{te}} = Ma_{\text{g}}$$

therefore:

$$u_{\mathrm{te,red}} = \frac{u_{\mathrm{g}}}{\sqrt{k_{\mathrm{wet,g}} \cdot R_{\mathrm{wet,g}} \cdot T_{1,\mathrm{g}}}} \cdot \sqrt{k_{\mathrm{wet,te}} \cdot R_{\mathrm{wet,te}} \cdot T_{1,\mathrm{te}}} = 244{,}30\ \mathrm{m/s}$$

$$n_{\mathrm{te,red}} = \frac{u_{\mathrm{te,red}}}{\pi \cdot D} = 495{,}30 \qquad 1/\mathrm{s}$$

$$n_{\mathrm{te}} = n_{\mathrm{te,red}} = 495{,}30 \qquad 1/\mathrm{s}$$

Target discharge pressure:

$$p_{2,\mathrm{target}} = p_{1,\mathrm{te}} \cdot \left\{ 1 + \left(\frac{\kappa_{\mathrm{wet,te}} - 1}{\kappa_{\mathrm{wet,te}}} \right) \left[\frac{y_{\mathrm{s,g}} \cdot \left(\frac{u_{\mathrm{te}}}{u_{\mathrm{g}}} \right)^2}{\left(R_{\mathrm{wet,te}} \cdot T_{1,\mathrm{te}} \right)} \right] \right\}^{\left(\frac{\kappa_{\mathrm{te}}}{\kappa_{\mathrm{te}} - 1} \right)} = 141\,381 \qquad \mathrm{Pa}$$

Both flow and work coefficients shall be within allowable deviations. Adjustments to speed and test pressure may be required as needed to maintain similarity per 7.6.1.3.

C.3.5 Test example 1 (dynamic compressor)

Volume flow and speed is adjustable

Recorded performance data from the test (te)

— Mass flow	$q_{m,\mathrm{te}} = 0{,}497\,4$	$\frac{\mathrm{kg}}{\mathrm{s}}$
— Inlet volume flow rate	$q_{V1,\mathrm{te}} = \frac{q_{m,\mathrm{te}}}{\rho_{1,\mathrm{te}}} = 0{,}413\,8$	$\frac{\mathrm{m}^3}{\mathrm{s}}$
— Measured discharge pressure	$p_{2,\mathrm{te}} = 141\,400$	Pa
— Measured package input power	$P_{\mathrm{pack,te}} = 23\,500$	W
— Impeller speed	$n_{\mathrm{te}} = 495{,}3$	1/s
— Impeller diameter	$D = 0{,}157$	m

Step 6: Calculate and confirm limit indicators on the test conditions

— Limits on test speed:

$$-5\,\% < \mathrm{lim}_{\mathrm{n,te}} < 5\,\% \qquad \mathrm{lim}_{\mathrm{n,te}} := \frac{n_{\mathrm{g}}}{n_{\mathrm{te}}} - 1 = 0{,}107\,8 \qquad \%$$

Step 7: Calculate test machine Mach number

— Rotor tip speed $\quad u_{\mathrm{te}} = \pi \cdot D \cdot n_{\mathrm{te}} = 244{,}30 \quad \frac{\mathrm{m}}{\mathrm{s}}$

$$Ma_{\mathrm{te}} = \frac{u}{c_1} = \frac{u_{\mathrm{te}}}{\sqrt{\kappa_{\mathrm{wet,te}} \cdot R_{\mathrm{wet,te}} \cdot T_{1,\mathrm{te}}}} = 0{,}712\,5$$

Step 8: Calculate key performance indicator for test

(dynamic compressors only)

— Specific isentropic compression work:

$$y_{s,te} = \frac{\kappa_{wet,te}}{\kappa_{wet,te} - 1} \cdot R_{wet,te} \cdot T_{1,te} \cdot \left[\left(\frac{p_{2,te}}{p_{1,te}} \right)^{\left(\frac{\kappa_{wet,te} - 1}{\kappa_{wet,te}} \right)} - 1 \right] = 29\,673 \frac{\text{J}}{\text{kg}}$$

— Set test within similarity limits to qualify the test point

— **Work coefficient: -2 %, +2 %**

— **Flow coefficient: -2 %, +2 %**

— **Machine Mach number: -5 %, +5 %**

— Work coefficient

$$\frac{y_{s,te}}{y_{s,g}} \cdot \left(\frac{u_g}{u_{te}} \right)^2 - 1 = 0{,}042\,71 \qquad \%$$

— Flow coefficient

$$\frac{q_{v1,te}}{q_{v1,g}} \cdot \left(\frac{u_g}{u_{te}} \right) - 1 = 0{,}227\,1 \qquad \%$$

— Machine Mach number

$$\left(\frac{Ma_{te}}{Ma_g} \right) - 1 = 0 \qquad \%$$

Step 9: Calculate package work/efficiency

— Specific package work $y_{pack,te} = \frac{P_{pack,te}}{q_{m2,te}} = 47\,246 \qquad \frac{\text{J}}{\text{kg}}$

— Package isentropic efficiency $\eta_{te} = \frac{y_{s,te}}{y_{pack,te}} \cdot 100 = 62{,}81 \qquad \%$

Step 10: Correction of test results to precondition and object of the guarantee

— Correction of volume flow

$$q_{v1,co} = q_{v1,te} \cdot \frac{u_g}{u_{te}} = 0{,}414\,2 \qquad \frac{\text{m}^3}{\text{s}}$$

— Correction of reference work, dynamic low pressure (8.3.1)

$$y_{s,co} = y_{s,te} \left(\frac{u_g}{u_{te}} \right)^2 = 29\,737 \qquad \frac{\text{J}}{\text{kg}}$$

— Corrected pressure ratio

$$\pi_{co} = \left[1 + \left(\frac{\kappa_g - 1}{\kappa_g} \right) \cdot \left(\frac{y_{s,co}}{R_{1,g} \cdot T_{1,g}} \right) \right]^{\left(\frac{\kappa_g}{\kappa_g - 1} \right)} = 1{,}400\,19$$

— Correction for discharge pressure

$$p_{2,co} = \pi_{co} \cdot p_{1,g} = 140\,019 \qquad \text{Pa}$$

— Correction of specific energy

$$e_{te} = \frac{P_{pack,te}}{q_{v1,te}} = 56\,792 \qquad \frac{W \cdot s}{m^3}$$

(dynamic packaged compressor)

$$e_{co} = \frac{\rho_{1,g}}{\rho_{1,te}} \cdot \frac{y_{s,co}}{y_{s,te}} \cdot e_{te} = 56\,257 \qquad \frac{W \cdot s}{m^3}$$

$$e_{co,g} = \frac{\rho_{1,g}}{\rho_{1,te}} \cdot \frac{y_{s,g}}{y_{s,te}} \cdot e_{te} = 52\,233 \qquad \frac{W \cdot s}{m^3}$$

— Correction for package power consumption

$$P_{co} = e_{co} \cdot q_{v1,co} = 23\,304 \qquad \mathrm{W}$$

$$P_{co,g} = e_{co,g} \cdot q_{v1,g} = 23\,241 \qquad \mathrm{W}$$

— Correction for isentropic efficiency

$$\eta_{co,g} = \frac{y_{s,g}}{\frac{P_{co,g}}{q_{m2,g}}} = 62{,}81 \qquad \%$$

Step 11: Compared deviations of guarantee to corrected test value

— Package specific energy $$\frac{e_{co,g}}{e_g} - 1 = 1{,}048 \qquad \%$$

— Volume flow rate $$\frac{q_{v1,co}}{q_{v1,g}} - 1 = 0{,}2271 \qquad \%$$

— Discharge pressure $$\frac{p_{2,co}}{p_{2,g}} - 1 = 0{,}0137 \qquad \%$$

— Package power $$\frac{P_{co,g}}{P_g} - 1 = 1{,}048 \qquad \%$$

Test results summary – example 1 (dynamic):

	Symbol	Unit (metric)	Reference section	Numerical values		
TEST DATA				**guarantee**	**test**	**corrected**
Test number	#			1g	1te	1co/cog*
Test period start/end	time	min		-	10	
Barometric pressure	$p_{0.amb.abs}$	Pa	6.2.2	100 000	101 000	
Inlet pressure	$p_{1.abs}$	Pa	6.2.3/6.2.4	100 000	101 000	
Inlet temperature	T_1	K	6.3	293,15	291,88	
Relative humidity	$\varphi_{rel,1}$	ratio	3.22	0	0,33	
Isentropic exponent	κ_{wet}	ratio	3.23	1,4	1,399	
Gas constant	R_{wet}	J/(kg·K)	7.7.1	287,1	287,9	
Inlet density	ρ_1	kg/m^3	7.7.1	1,188	1,202	
Discharge temperature	T_2	K	7.10	-	-	
Compressor speed	n	1/s	7.2	495,83	495,30	
Supply voltage	U	V	5.2	400	400	
Supply frequency	f	Hz	5.2	50	50	

	Symbol	Unit (metric)	Reference section	Numerical values		
TEST DATA				guarantee	test	corrected
External coolant inlet temperature	$T_{1,\text{cool}}$	K	5.2/6.3	-	-	
External coolant flow	$q_{m,\text{cool}}$	kg/s	5.2/6.7	-	-	
Specific isentropic compression work	y_s	J/kg	3.29	29 725	29 673	29 737
Specific package work	Y	J/kg	-	46 837	47 246	
Package isentropic efficiency	H	%	3.26	63,46	62,81	62,81*
Rotor tip speed	U	m/s	3.33	244,6	244,3	
Machine Mach number	Ma	-	3.34/7.5	0,712 5	0,712 5	
Key performance indicators						
Inlet volume flow rate	q_{V1}	m^3/s	6.6/8.2	0,413 3	0,413 8	0,414 2
Discharge pressure	p_2	Pa	8.3	140 000	141 400	140 019
Package input power	P	W	8.5	23 000	23 500	23 304
Specific energy	E	W/m^3/s	3.28/8.4	55 650	56 792	56 257

C.4 Advanced test calculation example 2 (positive displacement compressor)

C.4.1 Guarantee conditions

	Symbol	Numerical value	Units
Inlet pressure	$p_{1,g}$	100 000	Pa
Inlet temperature	$T_{1,g}$	293,15	K
Inlet relative humidity	$\varphi_{\text{rel},g}$	0,0	%

C.4.2 Object of guarantee

	Symbol	Numerical value	Units
Inlet volume flow	$q_{V1,g}$	0,413 3	m^3/s
Discharge pressure	$p_{2,g}$	140 000	Pa
Electric power @ the electric input terminals	P_g	23 000	W
Rotor speed	n_g	83,333	1/s

Acceptance tolerances (refer to Table 5 in 8.7) for this specific test point:

e_{pack}=±5 %; q_{v1}=±4 %; p_2=+1 %

C.4.3 General calculation on guarantee data

General calculation made on the tested performance data used for verification of similarity and acceptance comparisons. The tested inlet data in these examples is the same, so the calculations are valid for all compressor types.

Step 1: Preconditions of the guarantee

— Ambient pressure $p_{1,g} = 100\,000$ Pa

— Ambient temperature $T_{1,g} = 293{,}15$ K

— Ambient humidity $\varphi_{rel,g} = 0$ %

Step 2: Mixture of air and water vapor (humid air) on test data

— Ambient temperature $t_g = (T_{1,g} - 273{,}15)$ °C

— Calculate the vapor pressure

The partial pressure of water vapor is found using the steam tables: (ref. NBS/NRC Steam Tables ISBN 0-89116-353-0 paper)

$$p_{vap.sat,g} = \left(6{,}251\,080\,56 \cdot 10^{-3} + 4{,}338\,188\,3 \times 10^{-4} \cdot t + 1{,}734\,107\,7 \times 10^{-5} \cdot t^2 + 1{,}210\,936 \times 10^{-7} \cdot t^3 + 6{,}292\,408 \times 10^{-9} \times t^4 \right) \cdot 98\,066{,}5$$

— Vapor pressure $p_{vap.sat,g} = 2338$ Pa

— Relative humidity $\varphi_{rel,g} = \dfrac{p_{vap,g}}{p_{vap,sat,g}} = 0\%$

Therefore $p_{vap,g} = 0$ Pa

— Vapor content $x_{air,g} = 0{,}622 \cdot \dfrac{\varphi_{rel,g} \cdot p_{vap,sat,g}}{p_{1,g} - \varphi_{rel,g} \cdot p_{vap,sat,g}} = 0$ $\dfrac{kg}{kg}$

— Isentropic exponent dry $\kappa_{dry} \approx 1{,}4$

— Isentropic exponent $\kappa_{wet,g} = \kappa_{dry} \cdot (1 - 0{,}11 \cdot x_{air,g}) = 1{,}4$

— Gas constant (dry) $R_{air} = 287{,}1$ $\dfrac{J}{kg \cdot K}$

— Gas constant (humid air) $R_{wet,g} = \left[R_{air} \cdot \left(1 + \dfrac{x_{air,g}}{x_{air,g} + 1} \cdot 0{,}608\,1 \right) \right] = 287{,}1$ $\dfrac{J}{kg \cdot K}$

— Ambient air density $\rho_{1,g} = \left(\dfrac{P_{1,g}}{R_{wet,g} \cdot T_{1,g}} \right) = 1{,}188$ $\dfrac{kg}{m^3}$

— Inlet specific volume $V_{1,g} = \left(\dfrac{1}{\rho_{1,g}} \right) = 0{,}842$ $\dfrac{m^3}{kg}$

— Inlet volume flow rate $q_{V1,g} = \dfrac{q_{m2,g}}{\rho_{1,g}} = 0{,}413\,3$ $\dfrac{m^3}{s}$

— Mass flow rate $\therefore q_{m2,g} = q_{V1,g} \cdot \rho_{1,g} = 0{,}491\,1$ $\dfrac{kg}{s}$

— Discharge pressure $p_{2,g} = 140\,000$ Pa

— Measured package input power $P_g = 23\,000$ W

— Driver speed $n_g = 83{,}33$ 1/s

Step 3: Calculate key performance indicators for guarantee data

— Specific isentropic compression work

$$y_{s,g} = \frac{\kappa_{wet,g}}{\kappa_{wet,g} - 1} \cdot R_{wet,g} \cdot T_{1,g} \cdot \left[\left(\frac{p_{2,g}}{p_{1,g}} \right)^{\left(\frac{\kappa_{wet,g} - 1}{\kappa_{wet,g}} \right)} - 1 \right] = 29\,725 \qquad \frac{\mathrm{J}}{\mathrm{kg}}$$

— Reference process work for screw type. The specific value of $\boldsymbol{v}_{\mathrm{i}}$ shall be calculated for the specific packaged compressor

$$v_{i,g} = 1{,}36$$

$$y_{comb,g} = R_{wet,g} \cdot T_{1,g} \cdot \left[\frac{\frac{p_{2,g}}{p_{1,g}}}{v_{i,g}} + \frac{\kappa_{wet,g}}{\kappa_{wet,g} - 1} \left(\frac{1}{\kappa} \cdot v_{i,g}{}^{\kappa - 1} - 1 \right) \right] = 30\,013 \qquad \frac{\mathrm{J}}{\mathrm{kg}}$$

— Specific energy $\quad e_g = \frac{P_g}{q_{v1,g}} = 55\,650 \quad \frac{W \cdot \mathrm{s}}{\mathrm{m}^3}$

— Specific package work $\quad y_g = \frac{P_g}{q_{m2,g}} = 46\,837 \quad \frac{\mathrm{J}}{\mathrm{kg}}$

— Package isentropic efficiency $\quad \eta_g = \frac{y_{s,g}}{y_g} = 63{,}46 \quad \%$

C.4.4 General calculations on inlet test data

General calculation made on the tested performance data used for verification of similarity and acceptance comparisons. The tested inlet data in these examples is the same, so the calculations are valid for all compressor types.

Test inlet conditions

— Ambient pressure $\quad P_{1,te} = 101\,000 \quad$ Pa

— Ambient temperature $\quad T_{1,te} = 291{,}88 \quad$ K

— Ambient humidity $\quad \varphi_{rel,te} = 33 \quad$ %

Step 4: Mixture of air and water vapor (humid air) on test data

— Ambient temperature $\quad t_{te} = (T_{1,te} - 273{,}15) \quad$ °C

— Calculate the vapor pressure:

The partial pressure of water vapor is found using the steam tables: (ref. NBS/NRC Steam Tables ISBN 0-89116-353-0 paper)

$$p_{vap.sat,te} = \left(\begin{array}{l} 6{,}251\,080\,56 \cdot 10^{-3} + 4{,}338\,188\,3 \times 10^{-4} \cdot t_{te} + 1{,}734\,107\,7 \times 10^{-5} \cdot t_{te}{}^2 + 1{,}210\,936 \times 10^{-7} \\ \cdot t_{te}{}^3 + 6{,}292\,408 \times 10^{-9} \times t_{te}{}^4 \end{array} \right) \cdot 98\,066{,}5$$

— Vapor pressure $\quad p_{vap.sat,te} = 2\,160 \quad$ Pa

— Relative humidity $$\varphi_{rel,te} = \frac{p_{vap,te}}{p_{vap,sat,te}} = 33\ \%$$

Therefore $$p_{vap,te} = 712{,}8 \quad \text{Pa}$$

— Vapor content $$x_{air,te} = 0{,}622 \cdot \frac{\varphi_{rel,g} \cdot p_{vap,sat,te}}{p_{1,te} - \varphi_{rel,g} \cdot p_{vap,sat,te}} = 0{,}004\,42 \quad \frac{\text{kg}}{\text{kg}}$$

— Isentropic exponent dry $$\kappa_{dry} \approx 1{,}4$$

— Isentropic exponent $$\kappa_{wet,te} = \kappa_{dry} \cdot (1 - 0{,}11 \cdot x_{air,te}) = 1{,}399$$

— Gas constant (dry) $$R_{air} = 287{,}1 \quad \frac{\text{J}}{\text{kg} \cdot \text{K}}$$

— Gas constant (humid air) $$R_{wet,te} = \left[R_{air} \cdot \left(1 + \frac{x_{air,te}}{x_{air,te} + 1} \cdot 0{,}608\,1 \right) \right] = 287{,}9 \quad \frac{\text{J}}{\text{kg} \cdot \text{K}}$$

— Ambient air density $$\rho_{1,te} = \left(\frac{p_{1,te}}{R_{wet,te} \cdot T_{1,te}} \right) = 1{,}202 \quad \frac{\text{kg}}{\text{m}^3}$$

— Inlet specific volume $$V_{1,te} = \left(\frac{1}{\rho_{1,te}} \right) = 0{,}832 \quad \frac{\text{m}^3}{\text{kg}}$$

Step 5: Calculate discharge pressure of the test setup (positive displacement compressors only)

— Target discharge pressure:

$v_{i,te}$ = 1,36 (given for specific internal compression unit)

$$p_{2,target} = p_{1,te} \cdot v_{i,te} \cdot \left[\frac{y_{comb,g}}{R_{wet,te} \cdot T_{1,te}} - \frac{\kappa_{wet,te}}{\kappa_{wet,te} - 1} \left(\frac{1}{\kappa_{wet,te}} \cdot V_{i,te}{}^{\kappa_{wet,te} - 1} - 1 \right) \right] = 141\,487 \quad \text{Pa}$$

C.4.5 Test example 2 (positive displacement compressor)

Volume flow and speed is adjustable

Recorded performance data from the test (te)

— Mass flow $$q_{m,te} = 0{,}497\,4 \quad \frac{\text{kg}}{\text{s}}$$

— Inlet volume flow rate $$q_{V1,te} = \frac{q_{m,te}}{\rho_{1,te}} = 0{,}413\,8 \quad \frac{m^3}{\text{s}}$$

— Measured discharge pressure $$p_{2,te} = 142\,000 \quad \text{Pa}$$

— Measured package input power $$P_{pack,te} = 24\,100 \quad \text{W}$$

— Driver speed $$n_{te} = 85 \quad 1/\text{s}$$

Step 6: Calculate and confirm limit indicators on the test conditions

— Limits on test speed:

$$-5\ \% < lim_{n,te} < 5\ \% \qquad lim_{n,te} := \frac{n_g}{n_{te}} - 1 = -1{,}96 \quad \%$$

Step 7: Calculate test machine Mach number

— Not utilized for positive displacement packaged compressors

Step 8: Calculate key performance indicator for test

(positive displacement compressors only)

— Specific isentropic compression work

$$y_{\mathrm{s,te}} = \frac{\kappa_{\mathrm{wet,te}}}{\kappa_{\mathrm{wet,te}} - 1} \cdot R_{\mathrm{wet,te}} \cdot T_{1,\mathrm{te}} \cdot \left[\left(\frac{p_{2,\mathrm{te}}}{p_{1,\mathrm{te}}} \right)^{\left(\frac{\kappa_{\mathrm{wet,te}} - 1}{\kappa_{\mathrm{wet,te}}} \right)} - 1 \right] = 30\,065 \qquad \frac{\mathrm{J}}{\mathrm{kg}}$$

— Specific isentropic compression work for screw type. The specific value of V_i shall be calculated for the specific packaged compressor

$$y_{\mathrm{comb,te}} = R_{\mathrm{wet,te}} \cdot T_{1,\mathrm{te}} \cdot \left[\frac{\frac{p_{2,\mathrm{te}}}{p_{1,\mathrm{te}}}}{V_{\mathrm{i,te}}} + \frac{\kappa_{\mathrm{wet,te}}}{\kappa_{\mathrm{wet,te}} - 1} \left(\frac{1}{\kappa_{\mathrm{wet,te}}} \cdot V_{\mathrm{i,te}}{}^{\kappa_{\mathrm{wet,te}} - 1} - 1 \right) \right] = 30\,327 \qquad \frac{\mathrm{J}}{\mathrm{kg}}$$

— Set test within similarity limits to qualify the test point

  — **Work coefficient: -2 %, +2 %**

  — **Flow coefficient: -2 %, +2 %**

— Work coefficient

$$\frac{y_{\mathrm{comb,te}}}{y_{\mathrm{comb,g}}} - 1 = 1{,}046 \qquad \%$$

— Flow coefficient

$$\frac{q_{V1,\mathrm{te}}}{q_{V1,\mathrm{g}}} \cdot \left(\frac{n_\mathrm{g}}{n_{\mathrm{te}}} \right) - 1 = -1{,}844 \qquad \%$$

Step 9: Calculate package work/efficiency

— Specific package work

$$y_{\mathrm{pack,te}} = \frac{P_{\mathrm{pack,te}}}{q_{m2,\mathrm{te}}} = 48\,452 \qquad \frac{\mathrm{J}}{\mathrm{kg}}$$

— Package isentropic efficiency

$$\eta_{\mathrm{te}} = \frac{y_{\mathrm{s,te}}}{y_{\mathrm{pack,te}}} = 62{,}05 \qquad \%$$

Step 10: Correction of test results to precondition and object of the guarantee

— Correction of volume flow

$$q_{V1,\mathrm{co}} = q_{V1,\mathrm{te}} \cdot \frac{n_\mathrm{g}}{n_{\mathrm{te}}} = 0{,}405\,7 \qquad \frac{\mathrm{m}^3}{\mathrm{s}}$$

— Correction of reference work, positive displacement low pressure

$$y_{\mathrm{comb,co}} = y_{\mathrm{comb,te}} = 30\,327 \qquad \frac{\mathrm{J}}{\mathrm{kg}}$$

— Corrected pressure ratio

$$\pi_{co} = V_{i,g} \cdot \left[\frac{y_{comb,te}}{R_{wet,g} \cdot T_{1,g}} - \frac{\kappa_{wet,g}}{\kappa_{wet,g} - 1} \left(\frac{1}{\kappa_{wet,g}} \cdot V_{i,g}{}^{\kappa_{wet,g} - 1} - 1 \right) \right] = 1{,}405\,07$$

— Correction for discharge pressure

$$p_{2,co} = \pi_{co} \cdot p_{1,g} = 140\,507 \qquad \text{Pa}$$

— Correction of specific energy

$$e_{te} = \frac{P_{pack,te}}{q_{V\,1,te}} = 58\,242 \qquad \frac{W \cdot s}{m^3}$$

(positive displacement packaged compressor)

$$e_{co} = \frac{\rho_{1,g}}{\rho_{1,te}} \cdot \frac{y_{comb,co}}{y_{comb,te}} \cdot e_{te} = 57\,569 \qquad \frac{W \cdot s}{m^3}$$

$$e_{co,g} = \frac{\rho_{1,g}}{\rho_{1,te}} \cdot \frac{y_{comb,g}}{y_{comb,te}} \cdot e_{te} = 56\,973 \qquad \frac{W \cdot s}{m^3}$$

— Correction for package power consumption

$$P_{co} = e_{co} \cdot q_{V\,1,co} = 23\,355 \qquad \text{W}$$

$$P_{co,g} = e_{co,g} \cdot q_{V\,1,g} = 23\,547 \qquad \text{W}$$

— Correction for isentropic efficiency

$$\eta_{co,g} = \frac{y_{s,g}}{\dfrac{P_{co,g}}{q_{m2,g}}} = 61{,}93 \qquad \%$$

Step 11: Compared deviations of guarantee to corrected test value

— Package specific energy

$$\frac{e_{co,g}}{e_g} - 1 = 2{,}38 \qquad \%$$

— Volume flow rate

$$\frac{q_{V\,1,co}}{q_{V\,1,g}} - 1 = -1{,}85 \qquad \%$$

— Discharge pressure

$$\frac{p_{2,co}}{p_{2,g}} - 1 = 0{,}36 \qquad \%$$

— Package power

$$\frac{P_{co,g}}{P_g} - 1 = 2{,}38 \qquad \%$$

Test results summary – example 2 (positive displacement):

	Symbol	**Unit (metric)**	**Reference section**	**Numerical values**		
TEST DATA				**guarantee**	**test**	**corrected**
Test number	#			1g	1te	1co/cog*
Test period start/end	time	min		-	10	
Barometric pressure	$p_{0.amb.abs}$	Pa	6.2.2	100 000	101 000	
Inlet pressure	$p_{1.abs}$	Pa	6.2.3/6.2.4	100 000	101 000	
Inlet temperature	T_1	K	6.3	293,15	291,88	
Relative humidity	$\varphi_{rel,1}$	ratio	3.22	0	0,33	

	Symbol	Unit (metric)	Reference section	Numerical values		
TEST DATA				**guarantee**	**test**	**corrected**
Isentropic exponent	κ_{wet}	ratio	3.23	1,4	1,399	
Gas constant	R_{wet}	J/(kg·K)	7.7.1	287,1	287,9	
Inlet density	ρ_1	kg/m^3	7.7.1	1,188	1,202	
Discharge temperature	T_2	K	7.10	-	-	
Compressor speed	n	1/s	7.2	83,33	85,00	
Supply voltage	U	V	5.2	400	400	
Supply frequency	f	Hz	5.2	50	50	
External coolant inlet temperature	$T_{1,cool}$	K	5.2/6.3	-	-	
External coolant flow	$q_{m,cool}$	kg/s	5.2/6.7	-	-	
Specific combined work	Y_{comb}	J/kg	3.31/7.7.2	30 013	30 327	30 327
Specific isentropic compression work	y_s	J/kg	3.29	29 725	30 065	-
Specific package work	y	J/kg	-	46 833	48 451	
Package isentropic efficiency	η	%	3.26	63,46	62,05	61,93
Rotor tip speed	u	m/s	3.33	-	-	
Machine Mach number	Ma	-	3.34/7.5	-	-	
Key performance indicators						
Inlet volume flow rate	q_{V1}	m^3/s	6.6/8.2	0,413 3	0,413 8	0,405 7
Discharge pressure	p_2	Pa	8.3	140 000	142 000	140 507
Package input power	P	W	8.5	23 000	2 410	23 355
Specific energy	e	W/m^3/s	3.28/8.4	55 650	58 242	57 569

C.5 Advanced test calculation example 3 (dynamic, two-speed test)

The 2-speed test consists of 3 parts.

1. Test1 running at a calculated rotational speed based on the ratio of machine Mach numbers for guarantee.
2. Test2 running at specified rotational speed $n_{te} = n_g$
3. Test 1 and Test 2 are combined to evaluate the corrected package power.

C.5.1 Guarantee conditions

	Symbol	Numerical value	Units
Inlet pressure	$p_{1,g}$	101 325	Pa
Inlet temperature	$T_{1,g}$	313,15	K
Inlet relative humidity	$\varphi_{rel,g}$	60	%

C.5.2 Object of guarantee

	Symbol	Numerical value	Units
Inlet volume flow	$q_{V1,g}$	0,413 3	m^3/s
Discharge pressure	$p_{2,g}$	140 000	Pa
Electric power at the electric input terminals	P_g	19 760	W

Acceptance tolerances (refer to Table 5 in 8.7) for this specific test point:

e_{pack}=±5 %; q_{v1}=±4 %; p_2=+1 %

C.5.3 General calculation on guarantee data

General calculation made on the guaranteed performance data used for verification of similarity and acceptance comparison: As the guarantee data in these examples is the same the calculation is valid for all compressor types.

Step 1: Preconditions of the guarantee

— Ambient pressure $p_{1,g} = 101\,325$ Pa

— Ambient temperature $T_{1,g} = 313{,}15$ K

— Ambient humidity $\varphi_{rel,g} = 60$ %

Step 2: Mixture of air and water vapor (humid air) on guarantee data

— Ambient temperature $t_g = \left(T_{1,g} - 273{,}15\right)$ °C

— Calculate the vapor pressure:

The partial pressure of water vapor is found using the steam tables: (ref. NBS/NRC Steam Tables ISBN 0-89116-353-0 paper)

$$p_{vap.sat,g} = \left(\begin{array}{l} 6{,}251\,080\,56 \cdot 10^{-3} + 4{,}338\,188\,3 \times 10^{-4} \cdot t + 1{,}734\,107\,7 \times 10^{-5} \cdot t^2 + 1{,}210\,936 \times 10^{-7} \\ \cdot t^3 + 6{,}292\,408 \times 10^{-9} \times t^4 \end{array}\right) \cdot 98\,066{,}5$$

— Vapor pressure $p_{vap.sat,g} = 7\,375$ Pa

Relative humidity

$$\varphi_{rel,g} = \frac{p_{vap,g}}{p_{vap,sat,g}} = 60\ \%$$

Therefore: $p_{vap,g} = 4\,425$ Pa

— Vapor content $x_{air,g} = 0{,}622 \cdot \frac{\varphi_{rel,g} \cdot p_{vap,sat,g}}{p_{1,g} - \varphi_{rel,g} \cdot p_{vap,sat,g}} = 0{,}028\,4$ $\frac{kg}{kg}$

— Isentropic exponent dry $\kappa_{dry} \approx 1{,}4$

— Isentropic exponent $\kappa_{wet,g} = \kappa_{dry} \cdot \left(1 - 0{,}11 \cdot x_{air,g}\right) = 1{,}396$

— Gas constant (dry) $R_{air} = 287{,}1$ $\frac{J}{kg \cdot K}$

— Gas constant (humid air) $R_{wet,g} = \left[R_{air} \cdot \left(1 + \frac{x_{air,g}}{x_{air,g} + 1} \cdot 0{,}6081 \right) \right] = 291{,}9$ $\frac{J}{kg \cdot K}$

— Ambient air density $\rho_{1,g} = \left(\frac{p_{1,g}}{R_{wet,g} \cdot T_{1,g}} \right) = 1{,}108$ $\frac{kg}{m^3}$

— Inlet specific volume $V_{1,g} = \left(\frac{1}{\rho_{1,g}} \right) = 0{,}902$ $\frac{m^3}{kg}$

— Inlet volume flow rate $q_{V1,g} = \frac{q_{m2,g}}{\rho_{1,g}} = 0{,}4133$ $\frac{m^3}{s}$

— Mass flow rate $\therefore q_{m2,g} = q_{V1,g} \cdot \rho_{1,g} = 0{,}4581$ $\frac{kg}{s}$

— Discharge pressure $p_{2,g} = 140\,000$ Pa

— Measured package input power $P_g = 19\,760$ W

— Impeller speed $n_g = 495$ 1/s

— Impeller diameter $D = 0{,}159\,7$ m

Step 3: Calculate key performance indicators for guarantee data

— Specific isentropic compression work

$$y_{s,g} = \frac{\kappa_{wet,g}}{\kappa_{wet,g} - 1} \cdot R_{wet,g} \cdot T_{1,g} \cdot \left[\left(\frac{p_{2,g}}{p_{1,g}} \right)^{\left(\frac{\kappa_{wet,g} - 1}{\kappa_{wet,g}} \right)} - 1 \right] = 30\,952 \quad \frac{J}{kg}$$

— Specific energy $e_g = \frac{P_g}{q_{V1,g}} = 47\,810$ $\frac{W \cdot s}{m^3}$

— Specific package work $y_g = \frac{P_g}{q_{m2,g}} = 43\,134$ $\frac{J}{kg}$

— Package isentropic efficiency $\eta_g = \frac{y_{s,g}}{y_g} \times 100 = 71{,}76$ %

— Rotor tip speed $u_g = \pi \cdot D \cdot n_g = 248{,}3$ $\frac{m}{s}$

— Machine Mach number $Ma_g = \frac{u}{c_1} = \frac{u_g}{\sqrt{\kappa_{wet,g} \cdot R_{wet,g} \cdot T_{1,g}}} = 0{,}6953$

C.5.4 General calculations on inlet test data (test 1)

General calculation made on the tested performance data used for verification of similarity and acceptance comparisons. The tested inlet data in these examples is the same, so the calculations are valid for all compressor types.

Test inlet conditions

— Ambient pressure $P_{1,te1} = 99\,800$ Pa

— Ambient temperature $T_{1,te1} = 268{,}15$ K

— Ambient humidity $\varphi_{rel,te1} = 60$ %

Step 4: Mixture of air and water vapor (humid air) on test data

— Ambient temperature $t_{te} = (T_{1,te} - 273{,}15)$ °C

— Calculate the vapor pressure

The partial pressure of water vapor is found using the steam tables: (ref. NBS/NRC Steam Tables ISBN 0-89116-353-0 paper)

$$p_{vap.sat,te1} = \begin{pmatrix} 6{,}251\,080\,56 \cdot 10^{-3} + 4{,}338\,188\,3 \times 10^{-4} \cdot t_{te} + 1{,}734\,107\,7 \times 10^{-5} \\ \cdot t_{te}{}^{2} + 1{,}210\,936 \times 10^{-7} \cdot t_{te}{}^{3} + 6{,}292\,408 \times 10^{-9} \times t_{te}{}^{4} \end{pmatrix} \cdot 98\,066{,}5$$

— Vapor pressure $p_{vap.sat,te1} = 441{,}7$ Pa

— Relative humidity $\varphi_{rel,te1} = \dfrac{p_{vap,te1}}{p_{vap,sat,te1}} = 60\ \%$

Therefore $p_{vap,te1} = 265{,}2$ Pa

— Vapor content $x_{air,te1} = 0{,}622 \cdot \dfrac{\varphi_{rel,g} \cdot p_{vap,sat,te1}}{p_{1,te1} - \varphi_{rel,g} \cdot p_{vap,sat,te1}} = 0{,}001\,656$ $\dfrac{kg}{kg}$

— Isentropic exponent dry $\kappa_{dry} \approx 1{,}4$

— Isentropic exponent $\kappa_{wet,te1} = \kappa_{dry} \cdot (1 - 0{,}11 \cdot x_{wet,te1}) = 1{,}4$

— Gas constant (dry) $R_{air} = 287{,}1$ $\dfrac{J}{kg \cdot K}$

— Gas constant (humid air) $R_{wet,te1} = \left[R_{air} \cdot \left(1 + \dfrac{x_{air,te1}}{x_{air,te1} + 1} \cdot 0{,}608\,1 \right) \right] = 287{,}4$ $\dfrac{J}{kg \cdot K}$

— Ambient air density $\rho_{1,te1} = \left(\dfrac{p_{1,te1}}{R_{wet,te1} \cdot T_{1,te1}} \right) = 1{,}295$ $\dfrac{kg}{m^3}$

— Inlet specific volume $v_{1,te1} = \left(\dfrac{1}{\rho_{1,te1}} \right) = 0{,}772$ $\dfrac{m^3}{kg}$

Step 5: Calculate discharge pressure of the test setup (dynamic compressors only)

Reduced test speed

$$Ma_{te1} = Ma_g$$

therefore:

$$u_{te1,red} = \frac{u_g}{\sqrt{k_{wet,g} \cdot R_{wet,g} \cdot T_{1,g}}} \cdot \sqrt{k_{wet,te1} \cdot R_{wet,te1} \cdot T_{1,te1}} = 228{,}36 \quad \text{m/s}$$

$$n_{te1,red} = \frac{u_{te1,red}}{\pi \cdot D} = 455{,}16 \quad 1/\text{s}$$

$$n_{te1} = n_{te1,red} = 455{,}16 \quad 1/\text{s}$$

Target discharge pressure:

$$p_{2,\text{target}} = p_{1,\text{te1}} \cdot \left[1 + \left(\frac{\kappa_{\text{wet,te1}} - 1}{\kappa_{\text{wet,te1}}} \right) \left(\frac{y_{\text{s,g}} \cdot \left(\frac{u_{\text{te1}}}{u_{\text{g}}} \right)^2}{\left(R_{\text{wet,te1}} \cdot T_{1,\text{te1}} \right)} \right) \right]^{\left(\frac{\kappa_{\text{te1}}}{\kappa_{\text{te1}} - 1} \right)} = 138\,004 \quad \text{Pa}$$

Both flow and work coefficients shall be within allowable deviations. Adjustments to speed and test pressure may be required as needed to maintain similarity per section 7.6.1.3.

C.5.5 Test example 1 (dynamic compressor)

Volume flow and speed is adjustable

Recorded performance data from the test (te)

- Mass flow $q_{m,\text{te1}} = 0{,}491\,5$ $\frac{\text{kg}}{\text{s}}$
- Inlet volume flow rate $q_{V1,\text{te1}} = \frac{q_{m,\text{te1}}}{\rho_{1,\text{te1}}} = 0{,}379\,5$ $\frac{\text{m}^3}{\text{s}}$
- Measured discharge pressure $p_{2,\text{te1}} = 137\,878$ Pa
- Measured package input power $P_{\text{pack,te1}} = 18\,030$ W
- Impeller speed $n_{\text{te1}} = 455{,}16$ 1/s
- Impeller diameter $D = 0{,}159\,7$ m

Step 6: Calculate and confirm limit indicators on the test conditions

Limits on test speed:

$$-5\,\% < lim_{\text{n,te}} < 5\,\% \qquad lim_{\text{n,te}} := \frac{n_{\text{g}}}{n_{\text{te1}}} - 1 = 8{,}754 \quad \%$$

NOTE Speed exceeds allowable limits from section 7.2. A machine two speed test is required.

Step 7: Calculate test machine Mach number

- Rotor tip speed $u_{\text{te1}} = \pi \cdot D \cdot n_{\text{te1}} = 228{,}36$ $\frac{\text{m}}{\text{s}}$

$$Ma_{\text{te1}} = \frac{u}{c_1} = \frac{u_{\text{te1}}}{\sqrt{\kappa_{\text{wet,te1}} \cdot R_{\text{wet,te1}} \cdot T_{1,\text{te1}}}} = 0{,}695\,3$$

Step 8: Calculate key performance indicator for test (dynamic compressors only)

- Specific isentropic compression work:

$$y_{\text{s,te1}} = \frac{\kappa_{\text{wet,te1}}}{\kappa_{\text{wet,te1}} - 1} \cdot R_{\text{wet,te1}} \cdot T_{1,\text{te1}} \cdot \left[\left(\frac{p_{2,\text{te1}}}{p_{1,\text{te1}}} \right)^{\left(\frac{\kappa_{\text{wet,te1}} - 1}{\kappa_{\text{wet,te1}}} \right)} - 1 \right] = 26\,093 \quad \frac{\text{J}}{\text{kg}}$$

— Set test within similarity limits to qualify the test point

— **Work coefficient: -2 %, +2 %**

— **Flow coefficient: -2 %, +2 %**

— **Machine Mach number: -5 %, +5 %**

— Work coefficient

$$\frac{y_{s,te1}}{y_{s,g}} \cdot \left(\frac{u_g}{u_{te1}}\right)^2 - 1 = -0{,}2947 \quad \%$$

— Flow coefficient

$$\frac{q_{v1,te1}}{q_{v1,g}} \cdot \left(\frac{u_g}{u_{te1}}\right) - 1 = -0{,}1334 \quad \%$$

— Machine Mach number

$$\left(\frac{Ma_{te1}}{Ma_g}\right) - 1 = 0{,}0 \quad \%$$

Step 9: Calculate package work/efficiency

Specific package work

$$y_{pack,te1} = \frac{P_{pack,te1}}{q_{m2,te1}} = 36\,684 \quad \frac{\mathrm{J}}{\mathrm{kg}}$$

— Package isentropic efficiency

$$\eta_{te1} = \frac{y_{s,te1}}{y_{pack,te1}} \cdot 100 = 71{,}13 \quad \%$$

Step 10: Correction of test results to precondition and object of the guarantee

— Correction of volume flow

$$q_{V1,co,te1} = q_{V1,te1} \cdot \frac{u_g}{u_{te1}} = 0{,}4127 \quad \frac{\mathrm{m}^3}{\mathrm{s}}$$

— Correction of reference work, dynamic low pressure (8.3.1)

$$y_{s,co,te1} = y_{s,te1} \cdot \left(\frac{u_g}{u_{te1}}\right)^2 = 30\,861 \quad \frac{\mathrm{J}}{\mathrm{kg}}$$

— Corrected pressure ratio

$$\pi_{co,te1} = \left[1 + \left(\frac{\kappa_g - 1}{\kappa_g}\right) \cdot \left(\frac{y_{s,co,te1}}{R_{1,g} \cdot T_{1,g}}\right)\right]^{\left(\frac{\kappa_g}{\kappa_g - 1}\right)} = 1{,}39873$$

— Correction for discharge pressure

$$p_{2,co,te1} = \pi_{co,te1} \cdot p_{1,g} = 139\,873 \quad \mathrm{Pa}$$

— Correction of specific energy

$$e_{te1} = \frac{P_{pack,te1}}{q_{v1,te1}} = 47\,507 \quad \frac{W \cdot \mathrm{s}}{\mathrm{m}^3}$$

(dynamic packaged compressor)