International Standard ISO 1217

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Displacement compressors — Acceptance tests

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Foreword

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

Draft International Standards adopted by the technical committees are circulated to the member bodies for approval before their acceptance as International Standards by the ISO Council. They are approved in accordance with ISO procedures requiring at least 75 % approval by the member bodies voting.

International Standard ISO 1217 was prepared by Technical Committee ISO/TC 118, *Compressors, pneumatic tools and pneumatic machines.*

This second edition cancels and replaces the first edition (ISO 1217-1975), all clauses of which have been technically revised.

Users should note that all International Standards undergo revision from time to time and that any reference made herein to any other International Standard implies its latest edition, unless otherwise stated.

Contents

Displacement compressors — Acceptance tests

1 Scope and field of application

This International Standard specifies methods for acceptance tests, and technical conditions for the supply of displacement compressors including packaged versions (see annex C).

It gives detailed instructions on the measurement of volume flow rate and power requirement and means of adjusting the measured values to guarantee conditions.

NOTE — This International Standard may be used for full load acceptance testing of lobed rotary (Roots') blowers.

Three types of test are covered, as follows :

a) Acceptance test

This is a full performance test carried out in accordance with this International Standard.

b) Type test

This is also a full performance test carried out in accordance with this International Standard, to establish typical performance of a specific model of compressor produced in significant quantities. The manufacturer shall select at random one typical compressor from a batch of identical compressors for this type test. The test shall be witnessed by an independent expert from, or approved by, a reputable institution.

Provided the production compressors are identical with the compressor type tested, it is strongly recommended that the performance results of the type test should be used in catalogues and descriptive literature.

c) Simplified test

In annex A reference is made to a simplified compressor test. This is a test where volume flow rate and power input are measured using the manufacturer's normal test-stand instrument and equipment.

This test is normally carried out when a type test of an identical compressor has already been made. Provided the performance results obtained are within the tolerances listed in table 5, then the performance of this series-produced compressor is deemed to be the same as the type performance results. (See annex A for full particulars.)

This International Standard also specifies the operating and testing conditions which shall be agreed between the manufacturer and the purchaser (see annex B).

2 References

ISO 1000, *SI units and recommendations for the use of their multiples and of certain other units.*

ISO 1219, *Fluid power systems and components — Graphic symbols.*

ISO 2151, *Measurement of airborne noise emitted by compressor/primemover-units intended for outdoor use.*

ISO 2602, *Statistical interpretation of test results — Estimation of the mean — Confidence interval.*

ISO 2854, *Statistical interpretation of data — Techniques of estimation and tests relating to means and variances.*

ISO 2954, *Mechanical vibration of rotating and reciprocating machinery — Requirements for instruments for measuring vibration severity.*

ISO 3046, *Reciprocating internal combustion engines — Performance.*

ISO 3744, *Acoustics — Determination of sound power levels of noise sources — Engineering methods for free-field conditions over a reflecting plane.*

ISO 3857/1, *Compressors, pneumatic tools and machines — Vocabulary — Part 1 : General.*

ISO 3857/2, *Compressors, pneumatic tools and machines — Vocabulary — Part 2 : Compressors.*

ISO 3945, *Mechanical vibration of large rotating machines with speed range from* 10 *to* 200 rev/s *— Measurement and evaluation of vibration severity* in situ.

ISO 5167, *Measurement of fluid flow by means of orifice plates, nozzles and venturi tubes inserted in circular cross-section conduits running full.*

ISO 5168, *Measurement of fluid flow — Estimation of uncertainty of a flow-rate measurement.*

ISO 5388, *Stationary air compressors — Safety rules and code of practice.*

ISO 5390, *Compressors — Classification.*

ISO 5941, *Compressors, pneumatic tools and machines — Preferred pressures.*

IEC Publication 46, *Recommendations for steam turbines — Part 2 : Rules for acceptance tests.*

IEC Publication 51, *Recommendations for direct acting indicating electrical measuring instruments and their accessories.*

3 Definitions

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

3.1 General definitions

3.1.1 displacement compressor : Machine where a static pressure rise is obtained by allowing successive volumes of gas to be aspirated into and exhausted out of a closed space by means of the displacement of a moving member.

NOTE — For the definition of a liquid-ring compressor, see 7.1.1.

3.1.2 swept volume of a displacement compressor : Volume swept in one revolution by the compressing element(s) of the compressor first stage.

3.1.3 displacement of a displacement compressor : Volume swept by the compressing element(s) of the compressor first stage per unit of time.

3.1.4 shaft-driven reciprocating compressor : Displacement compressor in which gas intake and compression are achieved by the straightforward alternating movement of a moving element in a space constituting a compression chamber due to a shaft rotation.

3.1.5 rotary compressor : Displacement compressor in which the element is one or more rotors operating in a casing, the displacement being effected by vanes, meshing elements, or by displacement of the rotor itself.

3.1.6 packaged compressor : Compressor unit supplied by the manufacturer, fully piped and wired internally (see annex C). These may be stationary or mobile (portable) units.

3.1.7 clearance volume : Volume inside the compression space, which contains gas trapped at the end of the compression cycle.

3.1.8 relative clearance volume : Ratio of clearance volume of the stage under consideration to the swept volume of the compressing element of this stage.

3.1.9 standard inlet point : Inlet point considered representative for each compressor. This point varies with compressor design and type of installation.

NOTES

1 The standard inlet point of a stationary compressor is generally at the inlet flange.

2 The standard inlet point of a packaged air compressor is a point close to the compressor, chosen so that the thermometer is unaffected by the compressor operation.

3.1.10 standard inlet condition : Condition of the aspirated gas at the standard inlet point of the compressor.

3.1.11 standard discharge point : Discharge point considered representative for each compressor. This point varies with compressor design and type of installation.

NOTES

1 The standard discharge point of a stationary compressor is generally at the compressor discharge flange.

2 The standard discharge point of a packaged air compressor is the terminal outlet.

3.1.12 standard discharge condition : Condition of the compressed gas at the standard discharge point of the compressor.

3.1.13 intercooling : Removal of heat from a gas between stages.

3.1.14 aftercooling : Removal of heat from the gas after the compression is completed.

3.1.15 polytropic process : Compression or expansion process of an ideal gas in which the relation between pressure and volume follows the equation

$$pV^n = \text{constant}$$

The exponent n can have various values. For example :

$$pV = \text{constant}$$

describes an isothermal process, i.e. the gas temperature remains constant.

$$pV^x = \text{constant}$$

describes an isentropic process, i.e. the gas entropy remains constant.

NOTE — Sometimes this process is called adiabatic, but to avoid confusion between adiabatic (no heat exchange with the surroundings) and reversible adiabatic (isentropic) processes, the expression isentropic is preferred.

3.1.16 ideal multi-stage compression : Process when a perfect gas is isentropically compressed and the gas inlet temperature as well as the amount of work spent is the same for each stage.

3.1.17 shaft rotational speed : Number of revolutions of the compressor drive shaft per unit of time.

3.1.18 shaft-speed irregularity : Dimensionless number obtained when the difference between maximum and minimum instantaneous shaft-speeds during one period is divided by the arithmetic mean of these two.

$$\text{Shaft-speed irregularity} = 2\,\frac{n_{max} - n_{min}}{n_{max} + n_{min}}$$

3.2 Pressures

3.2.1 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 from the flow condition to the stagnation condition.

3.2.2 static pressure : Pressure measured in a gas in such a manner that no effect on measurement is produced by the gas velocity.

In stationary gas the static and the total pressures are numerically equal.

3.2.3 dynamic (velocity) pressure : Total pressure minus the static pressure.

3.2.4 atmospheric pressure : Absolute pressure of the atmosphere measured at the test place.

3.2.5 effective (gauge) pressure : Pressure measured above the atmospheric pressure.

3.2.6 absolute pressure : Pressure measured from absolute zero, i.e. from an absolute vacuum. It equals the algebraic sum of atmospheric pressure and effective pressure.

3.2.7 inlet pressure : Total mean absolute pressure at the standard inlet point.

NOTE — The total absolute pressure may be replaced by the static absolute pressure provided that the dynamic pressure is less than 0,5 % of the static pressure.

3.2.8 discharge pressure : Total mean absolute pressure at the standard discharge point.

NOTE — The total absolute pressure may be replaced by the static absolute pressure provided that the dynamic pressure is less than 0,5 % of the static pressure.

3.3 Temperatures

3.3.1 total temperature : Temperature which would be measured at the stagnation point if a gas stream were brought to rest and its kinetic energy converted by an isentropic compression from the flow condition to the stagnation condition.

3.3.2 inlet temperature : Total temperature at the standard inlet point of the compressor.

3.3.3 discharge temperature : Total temperature at the standard discharge point of the compressor.

3.4 Flow rates

3.4.1 actual volume flow rate of a compressor : Actual volume flow rate of gas compressed and delivered at the standard discharge point, referred to conditions of total temperature, total pressure and composition (e.g. humidity) prevailing at the standard inlet point.

NOTE — The expression "actual capacity" should be avoided as it may be confusing.

3.4.2 standard volume flow rate : Actual volume flow rate of compressed gas as delivered at the standard discharge point, but referred to standard conditions (for temperature, pressure, and inlet gas composition).

NOTE — The expression "standard capacity" should be avoided as it may be confusing.

3.4.3 free air : Air at the atmospheric conditions of the site and unaffected by the compressor.

3.5 Powers

3.5.1 isothermal power required : Power which is theoretically required to compress an ideal gas under constant temperature, in a compressor free from losses, from a given inlet pressure to a given discharge pressure.

3.5.2 isentropic power required : Power which is theoretically required to compress an ideal gas under constant entropy, from a given inlet pressure to a given discharge pressure. In multi-stage compression, the theoretical isentropic power required is the sum of the isentropic power required at all the stages.

3.5.3 shaft power : Power required at the compressor drive-shaft. It is the sum of the mechanical losses and the internal power. Losses in external transmissions such as gear drives or belt drives are not included unless part of the scope of supply.

3.5.4 packaged compressor power input : Sum of the power input to the prime mover and any accessories (e.g. oil-pump, cooling fan, etc.) driven from the compressor shaft or by a separate prime mover at rated supply conditions (e.g. phase, voltage, frequency and ampere capability). The power input shall include the effect of any equipment such as flow rate controls, intake filters, silencers, liquid separation equipment including their return systems, dryer, outlet shut-off valves, etc., included in the package (see annex C).

NOTE — The power input of a packaged compressor is always higher than the shaft power due to the motor losses and the power taken up by the accessories. These two concepts cannot therefore be compared.

3.6 Efficiencies

3.6.1 isentropic overall efficiency : Ratio of the required isentropic power to the power input for the scope of supply.

3.6.2 isothermal efficiency : Ratio of the isothermal power required to shaft power.

3.6.3 isentropic efficiency : Ratio of the isentropic power required to shaft power.

3.6.4 volumetric efficiency : Ratio of the actual volume flow rate to the displacement of the compressor.

3.7 Specific energy requirements

3.7.1 theoretical specific energy requirement : Work necessary to compress a unit mass of gas (mass specific energy) or unit volume of gas (volume specific energy) according to the specified process (isothermal, isentropic, polytropic).

3.7.2 actual specific energy requirement of a bare compressor : Shaft input power per unit of actual compressor volume flow rate.

3.7.3 actual specific energy requirement of a packaged compressor : Packaged compressor input power per unit of actual compressor volume flow rate.

3.7.4 specific fuel (or steam) consumption : Fuel (or steam) mass flow per unit of compressor actual volume flow rate.

3.8 Gas properties

3.8.1 compressibility factor, Z : Factor expressing the deviation of the real gas from an ideal gas.

3.8.2 relative vapour pressure : Ratio of the partial pressure of a vapour to its saturation pressure at the same temperature.

NOTE — In the case of water, the expression "relative humidity" was used previously.

4 Symbols, units and subscripts

4.1 Symbols and units

Quantity	Symbol	Dimensions	SI unit	Other practical units
Area	A	L^2	m^2	mm^2
Volume	V	L^3	m^3	l
Time	t	T	s	h, min
Velocity	c	LT^{-1}	m/s	—
Angular velocity	ω	T^{-1}	rad/s	—
Correction factor	K	—	pure number	—
Rotational frequency (shaft-speed)	N	T^{-1}	s^{-1}	min^{-1}
Mass density	ϱ	ML^{-3}	kg/m^3	kg/l
Celsius temperature	θ	Θ	°C	—
Thermodynamic temperature	T	Θ	K	—
Pressure	p	$ML^{-1}T^{-2}$	Pa	MPa, bar, kPa, mbar
Pressure ratio	r	—	pure number	—
Work	W	ML^2T^{-2}	J	MJ, kJ, kWh
Power	P	ML^2T^{-3}	W	MW, kW
Mass specific energy	W_m	L^2T^{-2}	J/kg	kJ/kg
Volume specific energy	W_V	$ML^{-1}T^{-2}$	J/m^3	J/l, kWh/m^3
Mass rate of flow	q_m	MT^{-1}	kg/s	kg/h
Volume rate of flow	q_V	L^3T^{-1}	m^3/s	m^3/h, m^3/min, l/s
Relative clearance volume	e	—	pure number	—
Exponent for polytropic process in pV diagram	n	—	pure number	—
Molar gas constant	R	$ML^2T^{-2}\Theta^{-1}N^{-1}$	J/(K·mol)	kJ/(K·mol)
Absolute humidity	x	—	pure number	—
Compressibility factor	Z	—	pure number	—
Dynamic viscosity	η	$ML^{-1}T^{-1}$	Pa·s	—
Efficiency	η	—	pure number	—
Isentropic exponent	$\varkappa$	—	pure number	—
Relative vapour-pressure	φ	—	pure number	—

M = mass   L = length   T = time Θ = temperature   N = quantity of matter

4.2 Letters and figures used as subscripts

Subscript	Meaning	Remarks
0	ambient condition	
1	inlet	Indicates the quantities measured at the standard inlet point of the compressor.
2	discharge	Indicates the quantities measured at the standard discharge point of the compressor.
a	absolute	
ab	absorbed	
ap	approximate	
av	average	
b	atmospheric	Characterizes the atmospheric pressures and temperatures.
c	contractual	Indicates the quantities specified in the contract.
cd	condensate	
corr	corrected	
cr	critical	Characterizes the critical pressures and temperatures.
d	dynamic	Characterizes the dynamic pressures and temperatures.
e	effective	
g	gas	
in	internal	
m	mass	Characterizes the mass specific rates of flow, energies and volumes.
me	mechanical	
N	normal	
pol	polytropic	Characterizes a polytropic process.
r	reduced	Characterizes the reduced pressures and temperatures.
R	reading	Indicates the quantities read during the test or predetermined as test conditions.
s	saturated	
t	total	
T	isothermal	Characterizes an isothermal process.
th	theoretical	
v	vapour	
V	volume	
w	coolant	

5 Measuring equipment and methods

5.1 General

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

5.2 Measurement of pressure

5.2.1 General

5.2.1.1 Pressure taps in the pipe or receiver shall be normal to, and flush with, the inside wall.

NOTE — For low pressures or high flow velocities, it should be noted that minor irregularities such as burrs can give serious errors.

5.2.1.2 Connecting piping to gauges shall be as short as possible.

Tightness shall be tested (for example with soap solution) and all leaks eliminated.

5.2.1.3 Connecting piping to gauges shall be not less than 6 mm bore.

Connecting piping shall be arranged so that there are no traps where liquid can collect.

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

5.2.1.5 The measuring instrument (analogue or digital) shall have an accuracy of ± 1 %.

5.2.1.6 The total pressure is the sum of the static and the dynamic pressures. It shall be measured with a Pitot tube having the axis parallel to the flow. When the dynamic pressure is less than 5 % of the total pressure, it shall be calculated on the basis of a calculated average velocity.

5.2.1.7 If the amplitudes of low frequency (< 1 Hz) pressure waves in the inlet pipe or the discharge pipe are found to exceed 10 % of the prevailing average absolute pressure, the piping installation shall be corrected before proceeding with the test.

Where the amplitudes of such pressure waves exceed 10 % of the specified average inlet or discharge pressures, a test shall not be undertaken under the requirements of this International Standard unless agreed to in writing by the parties to the test.

5.2.1.8 Gauges of the Bourdon type shall be calibrated under pressure and temperature conditions similar to those prevailing during the test, using dead-weight test gauges.

5.2.1.9 Dead-weight gauges shall be examined to ensure that the piston moves freely. The diameter of the piston shall be measured and the weights shall be compared with authentic standards.

5.2.1.10 Column readings and dead-weight gauges shall be corrected for the gravitational acceleration at the location of the instrument.

5.2.1.11 Column readings shall be corrected for ambient temperature.

5.2.1.12 In case of low frequency (< 1 Hz) flow pulsations, a receiver with inlet throttling shall be provided between the pressure tap and the manometer.

5.2.1.13 Oscillations of gauges shall not be reduced by throttling with a valve. However, a restricting orifice may be used.

5.2.2 Pressure less than or equal to 2 bar[1)]

5.2.2.1 The atmospheric pressure shall be measured with a mercury barometer, reading to the nearest millimetre.

The temperature for correcting the barometer reading shall be read with an accuracy of ± 1 K.

A boiling manometer or a precision aneroid barometer may also be used, but the accuracy shall be checked.

If a reliable barometer is not available, an approximation shall be obtained by using records of the nearest meteorological station, and correcting for the difference in altitude between the station and the compressor.

5.2.2.2 For sloping-limb and other amplifying instruments, the relation between the scale readings and the true water column length shall be determined previously by calibration against an absolute manometer of suitable sensitivity.

The inclination to the horizontal and the density of the manometer liquid shall be the same as for the calibration.

Manometers or columns for low pressure measurement shall comprise glass tubing of not less than 10 mm bore for the single-limb type and not less than 6 mm for the double-limb U-type, with a scale clearly graduated to allow the column to be read to within 1 mm.

The manometers shall be filled with a stable liquid of known mass density.

5.2.3 Pressure greater than 2 bar

For absolute pressure above 2 bar, calibrated gauges of Bourdon type or dead-weight gauges, mercury manometers or their equivalent shall be employed.

5.2.4 Inlet pressure

The inlet pressure of an air compressor operating without intake pipe or filter shall be measured by a barometer.

If an intake pipe is provided for the test, it shall be as close in size and shape as possible to the actual installation.

In cases of pulsating flow, a receiver volume with inlet throttling shall be provided between the manometer and the intake pipe (see 5.2.1.12 and 5.2.1.13).

5.2.5 Intercooler pressure

The intercooler pressure shall be measured after the intercooler.

5.2.6 Discharge pressure

The pressure tap shall be placed close to the standard discharge point of the compressor, if necessary on a pulsation damper with a throttling device before the manometer.

5.3 Measurement of temperature

5.3.1 Temperature shall be measured by certified or calibrated instruments such as thermometers, thermo-electrical instruments, resistance thermometers or thermistors inserted into the pipe or into pockets.

5.3.2 The measurement of the inlet temperature of the gas and the coolant shall be made with an accuracy of ± 1 K.

Commercial or industrial metal-encased thermometers shall not be used for temperatures that will influence the fulfilment of the guarantee.

5.3.3 The inlet gas temperature shall be measured near the inlet flange or connection, but sufficiently distant to avoid radiation and conduction errors.

5.3.4 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 fluid.

5.3.5 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.

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

1) 1 bar = 10^5 Pa

5.3.7 The thermometer reading shall be corrected for the emergent stem according to the following formula :

$$\theta = \theta_R + l\gamma(\theta_R - \theta_{av})$$

where

θ is the true temperature in degrees Celsius;

θ_R is the actual temperature reading in degrees Celsius;

θ_{av} is the average temperature of the emergent fluid column in degrees Celsius;

l is the length of the emerging fluid column expressed in kelvins;

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

5.3.8 Precautions shall be taken to ensure

a) 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;

b) that the 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);

c) that the thermometer pocket does not disturb the normal flow.

5.3.9 Thermocouples shall have a welded hot junction and shall be calibrated together with their wires for the anticipated operating range. They shall be made of materials suitable for the temperature and the gas being measured. If thermocouples are used with thermometer pockets, the hot junction of the couple shall, where possible, be welded to the bottom of the pocket.

5.4 Measurement of humidity

If the compressed air or gas contains moisture, the relative vapour pressure shall be checked during the test.

For tests with an open system, the dry and wet bulb temperatures shall be measured with a psychrometer of the Assmann type or another instrument with similar accuracy. The moisture content is then found from psychrometric tables or from an enthalpy/humidity chart.

For tests with a closed system, the humidity shall be measured with a dew point instrument, a psychrometer or another instrument with similar accuracy.

The humidity shall, if possible, be measured at the standard inlet point. If this is not possible the humidity shall be estimated.

5.5 Measurement of rotational frequency

If possible, the total number of compressor revolutions during the test run shall be measured with a revolution counter free from slip and the time of the test run shall simultaneously be accurately measured.

If a synchronous motor is used, a synchronous clock may replace the revolution counter.

If an asynchronous motor is used, the net frequency and the slip may be measured.

5.6 Measurement of flow rate

5.6.1 If possible, the actual volume flow rate of the compressor shall be calculated from a measurement of the delivered flow rate.

The test should be performed as indicated in ISO 5167.

It is necessary to ensure that all the requirements of ISO 5167 are completely fulfilled during the measurement period.

For testing the volume flow rate of a compressor, measurement of the aspirated volume shall be used if measurement of the delivered volume is not practical and if the leakage losses can be measured separately and with sufficient accuracy.

NOTES

1 When the demands on length of straight pipe upstream of the measuring device as given in ISO 5167 cannot be met, an alternative measuring pipe design is given in annex D.

2 Alternative methods for determining the actual volume flow rate are given in annex F.

5.6.2 The coolant flow rate may be determined with the aid of a vessel of known volume and a stop-watch or with a calibrated flow meter. The measurement may also be made with an orifice or nozzle according to ISO 5167.

5.7 Measurement of power and energy

5.7.1 The measurement of the output of the prime mover shall be made according to a recognized test code.

5.7.2 The power input to the compressor may be measured directly by reaction mounted drivers, or a torque meter, or indirectly determined from measurements of electrical input to the driving motor and the motor efficiency.

5.7.3 Precision torque meters shall not be used below one-third of their rated torque. They shall be calibrated after the test with the torsion member at the same temperature as during the test. Readings shall be made with a series of increasing loads with the precaution that, during the taking of readings with increasing loads, the load shall at no time be decreased.

Similarly, when readings are made with decreasing loads, the load shall at no time be increased. The calculation of output shall be based on the average of the increasing and decreasing loads as determined by the calibration. If the torque difference between increasing and decreasing loads exceeds 1 %, the torque meter is unsatisfactory.

5.7.4 The shaft power of an electrically-driven compressor shall be determined by measuring the electrical power supplied and multiplying by the motor efficiency. Only precision instruments shall be used. Power, voltage and current shall be measured. The voltage coils of the instruments shall be connected immediately before the terminals of the motor, so that voltage drop in cables will not affect the measurement. If remote instruments are used, the voltage drop shall be determined separately and taken into consideration (see IEC Publication 51).

5.7.5 For three-phase motors, the two-wattmeter method or some other method with similar accuracy shall be used.

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

For checking purposes it may be convenient to have a recently adjusted kWh-meter connected to the circuit during the test.

5.7.7 As a basis for the efficiency of the transmission, the following figures shall be used, at nominal load, unless other reliable information is available :

- for properly lubricated precision gears : 98 % for each stage,
- for belt drive : 95 % for each stage.

5.8 Miscellaneous measurements

5.8.1 Fuel consumption

If the compressor is driven by a combustion type engine or a gas turbine, the mean fuel consumption shall be determined by weighing or by measuring the volume of the fuel consumed during the test. (See ISO 3046.)

5.8.2 Steam consumption

If the compressor is driven by a steam engine or turbine, the non-bleeding steam rate shall be determined. (See IEC Publication 46.)

5.8.3 Gas composition

When tests are performed with gases other than air, the chemical composition and the physical properties of the gas entering the compressor during the tests shall be determined and if necessary checked at regular intervals.

5.8.4 Condensation rate

Before and after every test, the condensate shall be drained from the intercoolers and their separators in such a way that the steady state of the compressor is not disturbed. The separated quantities shall be weighed for every cooler and divided by the time between the draining operations.

NOTE — Any oil carried over with the condensate should be separated from the condensate before the mass of the latter is measured. If water separators are provided, the efficiency of separation can be determined.

The condensate collected in aftercoolers, receivers and other places after the discharge flange, but before the flow measuring device, shall be measured.

5.9 Calibration of instruments

Initial calibration records of the instruments shall be available prior to the test.

Recalibration after the test shall be made for those instruments of primary importance which are liable to variation in their calibration as a result of use during the test.

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

6 Test method

6.1 General

6.1.1 Before acceptance tests begin, the compressor shall be examined to ascertain whether it is in suitable condition to conduct an acceptance test. All external leakage shall be eliminated : in particular the pipe system shall be checked for leakage.

6.1.2 All parts likely to accumulate deposits, and particularly the coolers, shall be clean both on the gas and coolant sides.

6.2 Test arrangements

6.2.1 Preliminary tests shall be run for the purpose of

- checking instruments;
- training personnel.

A preliminary test may, by agreement, be considered the acceptance test, provided that all requirements for an acceptance test have been met.

6.2.2 During the test, all such measurements as have any bearing on the performance shall be made. In the following clauses the determination of the flow rate and the power absorbed by the compressor are covered in detail.

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

6.2.4 The test conditions shall be as close as reasonably possible to the guarantee conditions; deviations from these shall not exceed the limits specified in table 1.

6.2.5 Where it is not feasible to test a machine with the gas specified by the purchaser or within the limitations specified in table 1, special conditions of test or special corrections shall be agreed upon between purchaser and manufacturer.

6.2.6 The governing mechanism shall be maintained in its normal operating condition.

6.2.7 During the test, the lubricant and the rate of feed shall comply with the operating instructions.

6.2.8 During the test, no adjustments other than those required to maintain the test conditions and those required for normal operation as given in the instruction manual shall be made.

6.2.9 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.

However, should the test conditions be such that systematic changes cannot be avoided, or if individual readings are subject to great variations, then the number of readings shall be increased and due regard paid to this in the calculation of the tolerances.

6.2.10 For each load, a sufficient number of readings shall be taken to indicate that steady-state conditions have been reached. The number of readings and the intervals shall be chosen to obtain the required accuracy.

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

6.3 Evaluation of readings

6.3.1 Before final calculations are undertaken, the recorded data shall be scrutinized for consistency of operating conditions. The fluctuations of readings during one test shall not exceed the limits given in table 1.

Table 1 — Maximum deviations from specified values and fluctuations from average readings
(for liquid-ring compressors, see also table 2)

Measured variable	Maximum allowable deviations	Maximum allowable fluctuation from average during any set of readings
Inlet pressure, p_1	± 10 %	± 0,5 %
Pressure ratio, r	± 5 %	± 5 %
Inlet temperature, θ_1	not specified	± 2 K
Absolute inlet humidity, x_1	not specified	± 5 %
Isentropic exponent, $\varkappa$	± 3 %	not specified
Gas constant, R	± 5 %	not specified
Shaft-speed, N	± 4 %	± 1 %
Temperature difference between coolant and gas	± 10 K	± 2 K
Coolant flow rate	± 10 %	± 10 %
Temperature at the nozzle or orifice plate	not specified	± 2 K
Differential pressure over nozzle or orifice plate	not specified	± 3 %
Voltage	± 5 %	± 2 %
Net frequency	± 1 %	± 0,5 %

NOTES

1 The test can be performed if the deviations from the specified conditions are equal to or less than the deviation tolerances.

2 If the deviation from test conditions results in a deviation in absorbed power higher than ± 10 %, then the test is not within the limits.

3 See 5.2.1.7.

4 For outdoor tests with portable compressors, the allowable inlet temperature fluctuation is increased to ± 3 K.

5 A test at a shaft-speed different from the specified value is not accepted if unpermitted resonant pressure pulsations occur.

6 For the test of a gas compressor with a gas different from the actual one, a bigger variation in gas properties often occurs. This should be agreed upon by both parties.

6.3.2 All accepted readings from any test run shall be consecutive.

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

6.3.4 The moisture content shall be determined from psychrometer readings at the standard inlet point, according to 5.4.

The moisture content for the different compression stages and at the flow measuring device shall then be determined from condensate measurements.

6.4 Computation of test results

6.4.1 Test results, except those for flow measurements, shall be calculated from the arithmetic average values of the accepted readings.

6.4.2 The mass flow rate shall be determined according to 5.6.

6.4.3 When the gas being compressed is not dry, the influence of the moisture shall be taken into account by correcting the absorbed power.

6.4.4 The actual volume flow rate at the inlet is obtained by converting the gas flow measured through the measuring device from the condition there to the condition at the standard inlet point, due consideration being paid to any separated moisture.

Any vapour condensed between the standard inlet point and the measuring device shall be added to the measured mass flow to obtain the mass flow at the standard inlet point. Then from the mass flow at the standard inlet point the volume flow rate at this point is calculated. This is the actual inlet volume flow rate.

6.4.5 Some unloading systems exhaust warm gas from the unloaded side of the piston to the inlet at part load. The inlet temperature thus becomes higher at part load than at full load, whereby the volume flow rate apparently seems to attain a higher value. In such cases, therefore, the part load flow rate is calculated with the inlet temperature valid for full load.

6.4.6 Test conditions never agree exactly with specified conditions. Therefore, before test results and specified values are compared, corrections shall be applied to volume flow rate and absorbed power.

6.4.7 When the specified operating conditions cannot be met, the influence of the operating conditions on the performance of the actual compressor shall be determined by a method of variation, so that the size of each correction to the specified operating conditions can be determined by interpolation or, in extreme cases, by extrapolation.

If this is difficult to arrange, the correction methods given in this clause shall be used.

6.4.8 Within the limits specified in table 1, this International Standard provides for adjustment of the volume flow rate and the absorbed power when the test conditions deviate from those specified. The volume flow rate shall be adjusted for deviation in shaft-speed, pressure ratio, isentropic exponent and coolant temperature. The absorbed power shall be adjusted for deviation in inlet pressure, isentropic exponent, pressure ratio, coolant temperature, humidity and shaft-speed.

NOTE — Other corrections, such as correction for the compressibility factor, may have to be made.

6.4.9 For process compressors where certain amounts of compressed medium are injected or extracted between the stages, the specific energy concept is meaningless and shall be replaced by the power input to the compressor shaft.

6.4.10 If the test is carried out with a gas different from the one specified, a correction shall be made. A change in the gas constant will affect the leakage and hence the flow rate. Such corrections shall be agreed upon by the parties concerned.

6.4.11 If the deviation or fluctuation exceeds the value given in table 1, the methods described in 6.4.8 shall be used if agreed to by the parties concerned.

6.5 Volume flow rate corrections

6.5.1 Correction factor for shaft-speed, K_1

The correction factor is

$$K_1 = \frac{N_c}{N_R}$$

where

N_c is the specified shaft-speed;

N_R is the measured shaft-speed during the test.

6.5.2 Correction factor for polytropic exponent and pressure ratio, K_2

This correction factor can generally be neglected except for when testing single stage reciprocating compressors.

A change in the ratio of specific heat-capacities and in the pressure ratio will influence the volume flow rate as the expansion of the gas trapped in the clearance volume is affected. The degree of this influence is not fully known, so that the test supervisor should strive to operate as near the specified pressure ratio as possible. For differences within the limits given in table 1, the formula below shall be used :

$$K_2 = [1 - e\,(r_c^{1/n_c} - 1)]/[1 - e\,(r_R^{1/n_R} - 1)]$$

where

r_R is the measured pressure ratio;

r_c is the specified pressure ratio;

e is the relative clearance volume;

n is the polytropic exponent (should be taken as 0,9 $\varkappa$, in which $\varkappa$ is the isentropic exponent).

For pressure ratios below 3 the correction is simplified to :

$$K_2 = 1 + e\,(r_R^{1/n_R} - r_c^{1/n_c})$$

6.5.3 Correction factor for coolant temperature, K_3

The temperature difference between the coolant and the gas at their intake points will affect the gas temperature in the compressor cylinders as well as in the intercoolers. As this influence varies with compressor type, size and shaft-speed, no general correction formula can be given. If the specified conditions cannot be met, the compressor shall be operated at two different coolant inlet temperatures and at constant gas inlet temperature and then the required value shall be obtained by interpolating or extrapolating to the specified conditions with a straight line through the two test points.

For liquid injected displacement-type rotary compressors, the volume flow rate is affected by the temperature of the liquid injected into the compressor. This effect is due to heat transfer between the air and the liquid in the intake passages before compression begins and changes in sealing due to changes in viscosity. The system will be affected by the action of any thermostatic valve which may be fitted to enable the liquid to by-pass the cooler until a given liquid temperature is reached. For a given air inlet temperature, the injection of colder liquid normally gives higher volume flow rate due to less preheating of incoming air and more efficient cooling during compression. The magnitude of this influence depends upon compressor design, internal clearances, rotor tip speed and also on liquid flow rate, liquid viscosity, etc.

The correction factor, K_3, should be based on tests with the specific type of unit.

6.5.4 Correction for deviation in gas constant and compressibility factor

A change in gas constant or compressibility factor will affect the leakage and hence the volume flow rate. A general expression for this influence cannot be given. For deviations smaller than those given in table 1 this correction may be omitted.

6.6 Corrected volume flow rate

The corrected volume flow rate is

$$q_V = K_1 K_2 K_3 q_{VR}$$

where q_{VR} is the measured volume flow rate calculated from observed results of the test.

6.7 Power corrections

6.7.1 Correction factor for shaft-speed, K_4

The absorbed power is affected by the shaft-speed. It may be assumed that the compressor efficiency remains unchanged for deviations in test shaft-speed from the specified shaft-speed of ± 4 %.

The correction factor is then

$$K_4 = \frac{N_c}{N_R}$$

where

N_c is the specified shaft-speed;

N_R is the measured shaft-speed during the test.

6.7.2 Correction factor for inlet pressure, polytropic exponent and pressure ratio, K_5

The corrections for variation in inlet pressure and pressure ratio are fairly easy to obtain, with great accuracy, from practical tests.

The specified pressure ratio can normally be maintained within ± 1 % by adjusting the discharge pressure. If correction curves have not been established from earlier tests with the specific compressor type, the correction shall be based on a comparison of the work of compression using an appropriate polytropic exponent.

NOTE — The actual polytropic exponent varies during the compression process. If no test results are available, the isentropic exponent should be used. (For air $\varkappa$ = 1,40.)

If the inlet pressure, polytropic exponent and the pressure ratio deviate from the figures specified in the contract, then the correction methods below shall be used.

6.7.2.1 For single-stage machines, cooled and uncooled :

$$K_5 = \frac{[\varkappa/(\varkappa - 1)]_c}{[\varkappa/(\varkappa - 1)]_R} \times \frac{p_{1c}}{p_{1R}} \times \frac{r_c^{[(\varkappa - 1)/\varkappa]_c} - 1}{r_R^{[(\varkappa - 1)/\varkappa]_R} - 1} \times K_2$$

6.7.2.2 For multi-stage compressors with intercoolers :

$$K_5 = \frac{p_{1c}}{p_{1R}} \times \frac{\lg r_c}{\lg r_R}$$

6.7.2.3 If the pressure ratio during the test is held within ± 0,2 %, the power input correction for all displacement compressors can be simplified to

$$K_5 = \frac{p_{1c}}{p_{1R}}$$