# International Standard ISO 1217

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## Displacement compressors — Acceptance tests

*Compresseurs volumétriques — Essais de réception*

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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.





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## 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.

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

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 | $\omega$ | $T^{-1}$ | rad/s | — |
| Correction factor | $K$ | — | pure number | — |
| Rotational frequency (shaft-speed) | $N$ | $T^{-1}$ | $s^{-1}$ | $min^{-1}$ |
| Mass density | $\varrho$ | $ML^{-3}$ | $kg/m^3$ | kg/l |
| Celsius temperature | $\theta$ | $\Theta$ | °C | — |
| Thermodynamic temperature | $T$ | $\Theta$ | 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 | $\eta$ | $ML^{-1}T^{-1}$ | Pa·s | — |
| Efficiency | $\eta$ | — | pure number | — |
| Isentropic exponent | $\varkappa$ | — | pure number | — |
| Relative vapour-pressure | $\varphi$ | — | pure number | — |

M = mass L = length T = time $\Theta$ = 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

$\theta$ is the true temperature in degrees Celsius;

$\theta_R$ is the actual temperature reading in degrees Celsius;

$\theta_{av}$ is the average temperature of the emergent fluid column in degrees Celsius;

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

$\gamma$ 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, $\theta_1$ | 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 bypass 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

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

#### 6.7.3 Correction factor for humidity, multi-stage compressors, $K_6$

If in a multi-stage compressor vapour has condensed in the intercoolers, decreasing quantities of vapour are compressed in the various stages. The correction factor (see annex I) is

$$K_6 = 1 + \left[\frac{R_v}{R_g} \times \frac{n-1}{n}\right] \times \left[\frac{T_{1wR}}{T_{1R}}\right.$$

$$\left.\left(x_{1R} - \frac{1}{n-1}\sum_{i=2}^{n} x_{iR}\right) - \frac{T_{1wc}}{T_{1c}}\left(x_{1c} - \frac{1}{n-1}\sum_{i=2}^{n} x_{ic}\right)\right]$$

where

$R_v$ is the gas constant of the vapour in joules per mole kelvin;

$R_g$ is the gas constant of the gas in joules per mole kelvin;

$n$ is the number of stages;

$x$ is the absolute humidity of the gas at the inlet of any stage (the absolute humidity may be calculated from the partial pressure of the vapour);

$T_1$ is the absolute gas inlet temperature in kelvins;

$T_{1w}$ is the absolute coolant inlet temperature in kelvins.

When the specified gas is dry and at the same temperature as the coolant, the formula is simplified to

$$K_6 = 1 + \frac{R_v}{R_g} \times \frac{n-1}{n} \times \frac{T_{1wR}}{T_{1R}}\left(x_{1R} - \frac{1}{n-1}\sum_{i=2}^{n} x_{iR}\right)$$

#### 6.7.4 Correction factor for coolant inlet temperature, $K_7$

The relation between the coolant inlet temperature and the power absorbed is very complicated and has many parameters.

For this correction it is advisable to follow the recommendation of 6.5.3.

For liquid-injected displacement type rotary compressors the inlet air temperature, the temperature of the injected liquid, and the differential between them all affect the absorbed power, because the heating of the inlet air and changes in injected liquid viscosity will affect the internal leakage and the hydraulic losses. Correction factors shall be based on tests by the manufacturer.

### 6.8 Corrected absorbed power

The corrected absorbed power is

$$P = K_4 K_5 K_6 K_7 P_R$$

### 6.9 Corrected specific energy requirement

The corrected specific energy requirement is obtained by dividing the corrected absorbed power by the corrected volume flow rate (see 6.6 and 6.8).

## 7 Acceptance test for liquid-ring compressors

Liquid-ring compressors are of the displacement type but due to their specific design characteristics they are covered separately in this clause.

This clause gives rules for the measurement of the volume flow rate referred to inlet conditions, of the power absorbed and rules for conversion of the performance figures to specified conditions. (See also ISO 2602 and ISO 2854.)

### 7.1 Definition

**liquid-ring compressor** : Machine with a rotating impeller with protruding blades eccentrically mounted in a stationary round housing or centrally mounted in a stationary elliptical housing. A liquid ring rotating together with the bladed impeller creates one or two crescent-shaped working spaces.

The volumes trapped between each pair of blades, the hub and the liquid ring will vary periodically, thereby creating a change in pressure that will generate a flow from the suction to the discharge side of the compressor.

### 7.2 Measuring equipment and methods

See clause 5.

### 7.3 Test

See 6.1.1 to 6.3.3; however, table 1 is replaced by table 2.

### 7.4 Conversion of test results to specified conditions

**Table 2 — Maximum deviations from the specified values during an acceptance test**

| Measured variable | Maximum allowable deviation |
|---|---|
| Inlet pressure, $p_1$ | ± 5 % |
| Discharge pressure, $p_2$ | ± 5 % |
| Rotational frequency, $N$ | ± 3 % |
| Working liquid flow rate | ± 10 % |
| Working liquid temperature in degrees Celsius | ± 5 % |

NOTES

1 Whenever the pressure ratio is used as basis for conversion, it must agree within ± 2 % with the specified value.

2 Whenever the relative vapour pressure is used as basis for conversion, it must agree as closely as possible with the specified value.

Whenever the test conditions deviate from the specified conditions, a conversion of the intake volume flow rate and the power absorbed or specific energy requirement must be made. The intake volume flow rate is influenced by deviations in shaft-speed, suction- and discharge pressures, moisture content and temperature of the liquid.

The power input is influenced by deviations in shaft-speed, inlet and discharge pressures.

If the acceptance test is not carried out with the gas specified, the parties shall agree beforehand upon the conversion method to be used.

If the deviation or fluctuation of the measured variables exceeds the limits of table 2, the parties shall agree beforehand upon the conversion method to be used.

## 7.5 Inlet volume flow rate corrections

### 7.5.1 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.

### 7.5.2 Correction factor for working liquid temperature, $K_2$

The correction factor is

$$K_2 = \frac{p_{1c} - p_{Lc}}{p_{1R} - p_{LR}} \times \frac{T_{LR}}{T_{Lc}}$$

where

$p_{1c}$ is the specified absolute inlet pressure in bar;

$p_{Lc}$ is the partial pressure of the working liquid, in bar, at its specified temperature;

$p_{1R}$ is the measured absolute inlet pressure in bar;

$p_{LR}$ is the partial pressure of the working liquid, in bar, at its actual temperature;

$T_{LR}$ is the measured absolute temperature of the working liquid in kelvins;

$T_{Lc}$ is the specified absolute temperature of the working liquid in kelvins.

NOTE — This correction factor is valid on condition that the heat transfer between the liquid and the gas allows the gas to attain the same temperature as the liquid before the compression process commences.

### 7.5.3 Correction factor for gas inlet temperature, $K_3$

The correction factor is

$$K_3 = \frac{T_{1c}}{T_{1R}}$$

where

$T_{1c}$ is the specified absolute inlet gas temperature, in kelvins;

$T_{1R}$ is the measured absolute inlet gas temperature, in kelvins.

NOTE — This correction factor is valid on condition that the heat transfer between the liquid and the gas allows the gas to attain the same temperature as the liquid before the compression process commences.

## 7.6 Corrected inlet volume flow rate

The corrected volume flow rate is

$$q_{V1} = K_1 K_2 K_3 q_{R1}$$

where $q_{R1}$ is the measured volume flow rate calculated from observed results of the test.

## 7.7 Power correction

### 7.7.1 Correction factor for shaft-speed, $K_4$

The correction factor is

$$K_4 = \left(\frac{N_c}{N_R}\right)^2$$

where

$N_c$ is the specified shaft-speed;

$N_R$ is the measured shaft-speed during the test.

## 7.8 Corrected absorbed power

The corrected absorbed power is

$$P = K_4 P_R$$

## 7.9 Corrected specific energy requirement

The corrected specific energy requirement is obtained by dividing the corrected absorbed power by the corrected volume flow rate (see 7.6 and 7.8).

### 7.10 Accuracy of measurement

See clause 8.

### 7.11 Test report and comparison with specified values

See clause 9.

## 8 Accuracy of measurement

NOTE — A calculation of the probable error, according to this clause, is not always necessary.

### 8.1 General

Due to the very nature of physical measurements, it is impossible to measure a physical quantity without error or, in fact, to determine the true error of any one particular measurement. However, if the conditions of the measurement are sufficiently well-known, it is possible to estimate or calculate a characteristic deviation of the measured value from the true value, such that it can be asserted with a certain degree of confidence that the true error is less than the said deviation. The value of such a deviation (normally the 95 % confidence limit) constitutes a criterion of the accuracy of the particular measurement.

It is assumed that all systematic errors that may occur in the measurement of the individual quantities measured and of the characteristics of the gas may be compensated for by corrections. A further assumption is that the confidence-limits in errors in reading and integration errors may be negligible if the number of readings is sufficient. The (small) systematic errors that may occur are covered by the inaccuracy of measurements. Quality classifications and limits of error are often invoked for ascertaining the inaccuracy of individual measurements because apart from the exceptions (e.g. electrical transducers), they constitute only a fraction of the quality class or the limit of error.

The information about ascertaining the inaccuracy of the measurement of the individual quantities measured and on the confidence limits of the gas properties are approximations. These approximations can only be improved at a disproportionate expense. (See ISO 2602 and ISO 2854.)

### 8.2 Inaccuracy of individual measurements

#### 8.2.1 Inaccuracy of pressure measurements

**8.2.1.1** Precision Bourdon-type pressure gauges

The relative inaccuracy, $\tau_{\Delta p}$, of measurement of pressure difference as a percentage, using a precision Bourdon pressure gauge is

$$\tau_{\Delta p} = \frac{\widetilde{V}_{\Delta p}}{\Delta p}$$

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

where

$G$ is the quality class as a percentage (if the quality class of the pressure gauge is < 0,2, $G$ is still be taken as 0,2 in the above formula to allow for errors in mounting);

$p_{\mathrm{En}}$ is the full scale value of the instrument;

$p_{\mathrm{R}}$ is the measured pressure.

**8.2.1.2** Liquid columns

When liquid columns are used, the inaccuracy of the measurement depends primarily on the ease with which the difference in level, $\Delta h$, can be read. If no special aid is used, an inaccuracy of measurement, $\widetilde{V}_{\Delta h}$, of ± 1 mm can be reached.

In the range from 0,1 m < $h$ < 1,0 m the relative inaccuracy of measurement, $\tau_{\Delta h}$, as a percentage, is

$$\tau_{\Delta h} = \frac{\widetilde{V}_{\Delta h}}{\Delta h}$$

$$\tau_{\Delta h} = \frac{1}{\Delta h}$$

At $\Delta h$ > 1 m the relative inacurracy of measurement as a percentage is :

$$\tau_{\Delta h} = \pm 0,1$$

**8.2.1.3** Other measuring instruments

All other measuring instruments for pressure with the quality classification referred to the final value of the range of measurement are dealt with according to 8.2.2.3. As with these measuring instruments the measured value most often becomes apparent at the end of a series of measurements, the rules for series of measurements are to be observed.

#### 8.2.2 Inaccuracy of temperature measurements

**8.2.2.1** Liquid-in-glass thermometers

The inaccuracy of measurement, $\widetilde{V}_\theta$, to be inserted is the extended temperature limit obtained by calibration and supplemented by the addition of the failure margins. In most cases the inaccuracy of measurement may be taken from table 3.

**Table 3 — Inaccuracy of measurement, $\widetilde{V}_\theta$, for calibrated liquid-in-glass thermometers**

| Temperature range | Scale divisions, K | | | | |
|---|---|---|---|---|---|
| °C | 0,1 | 0,1 | 0,5 | 1 | 2 |
| − 50 to − 5 | 0,6 | 0,8 | 1,7 | 2,0 | 4,0 |
| − 5 to 60 | 0,3 | 0,4 | 1,0 | 1,4 | 2,0 |
| 60 to 110 | 0,5 | 0,6 | 1,0 | 2,0 | 3,0 |
| 110 to 210 | — | 1,0 | 2,0 | 3,0 | 4,0 |
| 210 to 310 | — | — | 3,0 | 4,0 | 6,0 |

#### 8.2.2.2 Thermocouples

If a works test certificate of recent date of a calibration of the entire equipment used for the fixed points on the temperature scale is available and measurement has been made with precision compensation instruments (quality class 0,1), an inaccuracy of measurement, $\tilde{V}_\theta$, of ± 1,0 K may be used up to temperatures of 300 °C.

By using special combinations of instruments, considerably smaller inaccuracies of measurement can be reached, especially in the measurement of small temperature differences.

#### 8.2.2.3 Resistance thermometers

If a works test certificate of recent date is available for a calibration of the entire range of measurement at the fixed temperature points, an inaccuracy of measurement, $\tilde{V}_\theta$, of ± 1,0 K can be applied for temperatures up to 300 °C.

### 8.2.3 Inaccuracy of flow measurements

The tolerance, $\tau_q$, on flow measurements with standardized throttling devices shall be calculated according to ISO 5168. If the suppression of pulsations was unsuccessful, corrections should be applied. In that case, the tolerance should be increased by 20 % of the correction.

### 8.2.4 Inaccuracy of shaft-speed measurements

With shaft-speed measured by means of calibrated analogue measuring instruments, the relative inaccuracy of measurement, $\tau_N$, as a percentage, is

$$\tau_N = \frac{\tilde{V}_N}{N}$$

$$\tau_N = \pm G \frac{N_{\mathrm{En}}}{N_{\mathrm{R}}}$$

where

$G$ is the quality class as a percentage;

$N_{\mathrm{En}}$ is the full scale value of the instrument;

$N_{\mathrm{R}}$ is the measured shaft-speed.

With shaft-speed measured by means of digital measuring instruments, the relative inaccuracy of measurement, as a percentage, is

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

where

$S$ is the digital step;

$A$ is the measured value read from the scale.

### 8.2.5 Inaccuracy of power measurements

#### 8.2.5.1 Torque measurements

With torque measurements by means of calibrated torsion dynamometers, the relative inaccuracy of measurement, $\tau_{M_\mathrm{d}}$, as a percentage, is

$$\tau_{M_\mathrm{d}} = \frac{\tilde{V}_{M_\mathrm{d}}}{M_\mathrm{d}}$$

$$\tau_{M_\mathrm{d}} = \pm G \frac{M_{\mathrm{dEn}}}{M_{\mathrm{dR}}}$$

where

$G$ is the quality class, as a percentage;

$M_{\mathrm{dEn}}$ is the full scale value of the instrument;

$M_{\mathrm{dR}}$ is the measured value.

When torque measurements are made with an electric dynamometer, the inaccuracy of measurement indicated by the manufacturer may be used.

#### 8.2.5.2 Inaccuracy of the two-wattmeter method

The tolerance for a two-wattmeter measurement is a combination of the tolerance for the instrument readings calculated in the normal way, and the tolerance for the instrument errors calculated according to the following.

For a two-wattmeter measurement, the relative inaccuracy of the measurement of the electric power input is

$$\tau_{P_{\mathrm{el}}} = \frac{\tilde{V}_{P_{\mathrm{el}}}}{P_{\mathrm{el}}}$$

$$\tau_{P_{\mathrm{el}}} = \pm \left[ G_\mathrm{u}^2 + G_\mathrm{i}^2 + G_\mathrm{W}^2 \left( \frac{\alpha_\mathrm{E}}{\alpha_1 + \alpha_2} \right)^2 f_t + (\Gamma_\mathrm{u}^2 + \Gamma_\mathrm{i}^2) f_\Gamma \right]^{1/2}$$

with

$$f_t = \left(1 + \frac{\theta - 20}{10} \times \frac{\alpha_1}{\alpha_\mathrm{E}}\right)^2 + \left(1 + \frac{\theta - 20}{10} \times \frac{\alpha_2}{\alpha_\mathrm{E}}\right)^2$$

$$f_\Gamma = 3{,}3 \times 10^3 \times [(3K - 1)^2 + (3K - 2)^2]$$

where

$G_\mathrm{u}$ is the quality class of the voltage transformers, as a percentage;

$G_\mathrm{i}$ is the quality class of the current transformers, as a percentage;

$G_\mathrm{W}$ is the quality class of the wattmeters, as a percentage;

$\theta$ is the temperature of instruments, in degrees Celsius;

$\Gamma_u$ are the limits of the angle errors of the voltage transformers, in radians;

$\Gamma_i$ are the limits of the angle errors of the current transformers, in radians;

$\alpha$ is the actual reading of the wattmeters;

$\alpha_E$ is the full scale reading of one wattmeter;

$K$ is the part load of the first wattmeter $\left(K = \dfrac{P_{el1}}{P_{el2}}\right)$.

It is assumed that $G$ and $\Gamma$ are equal in both measuring circuits. If for the wattmeter preresistors with the error limits $G_R$ are used, the term $G_R^2\,[1 + 2K\,(K - 1)]$ shall be added inside the brackets of $\tau_{P_{el}}$.

**8.2.5.3** Inaccuracy of direct current measurements

With direct current, a similar procedure can be followed broadly if voltmeters and amperemeters are used and the readings are multiplied.

The measuring errors for tension, $\tau_u$, and current, $\tau_i$, are combined to give

$$\tau_{1c} = \pm\sqrt{\tau_u^2 + \tau_i^2}$$

**8.2.5.4** Inaccuracy of the results of measurement at the coupling of the prime mover

When measurements of the power at the coupling are made through the medium of the power drawn by an electric motor, the relative inaccuracy of measurement, $\tau_{P_{Co}}$, as a percentage, is

$$\tau_{P_{Co}} = \frac{\tilde{V}_{P_{Co}}}{P_{Co}}$$

$$\tau_{P_{Co}} = \pm\left[\left(\frac{\tilde{V}_{P_{el}}}{P_{el}}\right)^2 + \left(\frac{\tilde{V}_{\eta M}}{\eta_M}\right)^2\right]$$

$$\tau_{P_{Co}} = \pm\sqrt{\tau_{P_{el}}{}^2 + \tau_{\eta M}{}^2}$$

where

$\tilde{V}_{P_{el}}$ is the inaccuracy of measurement of the electric power;

$\tilde{V}_{\eta M}$ is the uncertainty in the determination of the efficiency of the motor. The supplier of the electric motor shall provide curves showing the variation of the efficiency of the motor with the load, and shall indicate its uncertainty.

**8.2.5.5** The uncertainty in the efficiency figures given in 5.7.7 can be estimated to ± 0,5 % for precision gears and ± 1 % for belt drives.

If there is no test record for the establishment of these efficiencies, the conditions for application of the rule of error distribution according to Gauss are not met. The tolerances for the efficiencies shall then be algebraically added to the tolerance for the power requirement of the compressor.

**8.2.5.6** The measured input will be affected by load fluctuations occurring during each measurement. To cover this influence an additional tolerance is required. This additional tolerance is determined from the actual, absolute fluctuation of the load above and below the average value.

**Table 4 — Tolerance for load fluctuation**

Values in percentage

| Average fluctuation in power input | Additional tolerance |
|---|---|
| ± 2 | ± 0,0 |
| ± 3 | ± 0,5 |
| ± 4 | ± 1,0 |
| ± 5 | ± 1,5 |

This tolerance shall be algebraically added to the resulting tolerance from 8.3.3.

### 8.2.6 Confidence limits of gas properties

When the composition of the gas fluctuates, special care shall be taken to secure a suitable sample. If the fluctuations exceed the measure that can thus be compensated for, the confidence limits shall be increased.

The following information presumes, further, that suitable chemical or physical analyses will be undertaken to determine the composition of the gas.

**8.2.6.1** Gas constant

If the gas constant is taken from standard tables, its confidence limits may be neglected.

If the conditions of 8.2.6 are adhered to, the confidence limits, $V_R$, of the gas constant may be neglected. If the gas constant is determined from a density measurement made with accurate instruments, a relative confidence limit, $V_R/R$, of ± 0,5 % shall be used.

**8.2.6.2** Compressibility factor

The confidence limits of the compressibility factors of the pure gases most often compressed may be taken from the relevant literature.

For gas mixtures the greatest accuracy is attainable by measuring the compressibility.

The calculation of the compressibility factor yields only approximate values. In the estimation of the confidence limits of the compressibility factor of the mixture, the confidence limits of the compressibility factor of the components occupying the greatest proportion by volume and the confidence limits of the components whose compressibility factors diverge most from unity are the data chiefly used.

**8.2.6.3** Isentropic exponent

If the isentropic exponent for approximately ideal gases is taken from standard tables, the confidence limits of the isentropic exponent may be neglected.

No exact information is available about the confidence limits of the isentropic exponents of gases that differ to a large extent from the ideal state. The confidence limits shall be estimated.

**8.2.7 Inaccuracies caused by the correction methods**

To the tolerances given above, further tolerances shall be added expressing the uncertainty of the correction methods used to convert the measured values to contract conditions.

For these tests within the limits of table 1, this tolerance may amount to ± 20 % of the correction.

## 8.3 Inaccuracy of results of measurement

**8.3.1 Relative inaccuracy of the result of measurement of the volume flow rate**

The effects of the isentropic exponent, the pressure ratio, the difference between the inlet temperatures of the gas and the cooling medium, and the clearance space on the inaccuracy of measurement may usually be neglected.

The relative inaccuracy, $\tau_{\text{ErgR}}$, of the result of measurement, as a percentage, is then :

$$\tau_{\text{ErgR}} = \pm \sqrt{\tau_{q\text{R}}^2 + \tau_{N\text{R}}^2 + \tau_{p1\text{R}}^2 + \tau_{T1\text{R}}^2 + \tau_{Z1\text{R}}^2}$$

**8.3.2 Relative inaccuracy of the result of measurement of the pressure ratio**

The relative inaccuracy, $\tau_{\text{ErgR}}$, of the result of measurement of the pressure ratio, as a percentage, is

$$\tau_{\text{ErgR}} = \pm \sqrt{\tau_{p1\text{R}}^2 + \tau_{p2\text{R}}^2}$$

**8.3.3 Relative inaccuracy of the result of measurement of the power absorbed, the specific energy requirement, and the efficiency**

The effect of the viscosity of the lubricant on the inaccuracy of measurement may usually be neglected.

If the power at the coupling with the prime mover is measured, the relative inaccuracy, $\tau_{\text{Erg}P_{\text{Co}}}$, of the result of measurement, as a percentage, is

$$\tau_{\text{Erg}P_{\text{Co}}} = \tau_{\text{Erg}(P_{\text{Co}}/V)}$$

$$\tau_{\text{Erg}P_{\text{Co}}} = \tau_{\text{Erg}\,\eta_{\text{Co}}}$$

$$\tau_{\text{Erg}P_{\text{Co}}} = \pm \left[\tau_{P_{\text{Co}}\text{R}}^2 + \tau_{q\text{R}}^2 + \tau_{T1\text{R}}^2 + \left(\frac{Z-1}{Z}\,\tau_{T\text{w}1}\right)^2 + \varepsilon_2^2 \times \tau_K^2\right]^{1/2}$$

with $\varepsilon_2 = \dfrac{1}{1-K_{\text{R}}} + \dfrac{1}{K_{\text{R}}} \times \ln r_{\text{R}} \Big/ \left[1 - r_{\text{R}}^{(1-K_{\text{R}})/K_{\text{R}}}\right]$

If the power at the coupling, $P_{\text{Co}}$, is determined by measuring the torque, $M_{\text{d}}$, and the shaft-speed, $N$, then $(\tau_{M_{\text{d}}\text{R}}^2 + \tau_{N\text{R}}^2)$ replaces $\tau_{P_{\text{Co}}\text{R}}^2$ in the equation above.

**8.3.4 Relative inaccuracy of the result of measurement on a single stage of a multi-stage compressor**

In the process of building up the total characteristic curve from the curves of stages measured singly, or of groups of curves, the relative inaccuracy, $\tau_{\text{comb}}$, in the measurement below is as indicated, in percentages.

**8.3.4.1** Volume flow rate

$$\tau_{\text{comb}q} = \pm\,[1 + 0{,}2\,(Z-1)^{1/2}] \times \frac{\Sigma\,\tau_{qj}}{Z}$$

**8.3.4.2** Pressure ratio

$$\tau_{\text{comb}r} = \pm\,[1 + 0{,}2\,(Z-1)^{1/2}] \times \Sigma\,\tau_{rj} \times \frac{Y_j}{\Sigma\,Y_j}$$

**8.3.4.3** Power consumption

$$\tau_{\text{comb}P} = \pm\,[1 + 0{,}2\,(Z-1)^{1/2}] \times \Sigma\,\tau_{Pj} \times \frac{P_j}{\Sigma\,P_j}$$

The factors 0,2 and $(Z-1)$ allow for the inevitable inaccuracies in the measurement of the separate groups of stages and in the compounding of the results.

# 9 Test report and comparison with specified values

**9.1** A test report shall be made by the supervisor of the test and shall contain everything needed for evaluating the test.

**9.2** The test report shall state the object, the place and time of the test.

**9.3** The test report shall contain :

a) a brief description with the principal data of the compressor and the prime mover;

b) the manufacturer's name and the manufacturing number;

c) a statement of the methods and the equipment used for the test with a diagrammatic layout of the test arrangement with all measurement points marked;

d) a statement of the values specified by the vendor with regard to the volume flow rate, the input power and the function of the compressor in operation and the operating conditions under which the specifications apply;

e) for important instruments the manufacturer's name, the manufacturing number and the construction and quality class, the steps taken for control and adjustments, etc. The calibration records and correction diagrams for the various instruments should preferably be included as appendices.

**9.4** The test report shall contain data recorded during the test, a table of the average values of the readings, information of interest for the estimation of the accuracy of the readings, etc. Instrument indications or readings shall be recorded as observed. Original log sheets shall remain in the custody of the test supervisor. Corrections and converted values shall be entered separately in the test report.

**9.5** Any special conditions or occurrences shall be recorded.

**9.6** The test report shall contain the evaluation of the test and the results of calculations. Calculation of errors showing the degree of accuracy of the results shall also be included. Extensive calculations should preferably be incorporated as appendices, in order not to burden the report. The results shall be arranged in the form of clear tables.

**9.7** A comparison of the performance of the actual compressor with the specified values shall be included, together with conclusions from this comparison.

**9.8** A summary of the results of the test shall be included, together with general conclusions relative to the equipment tested, and a statement as to whether the specified values have been met or not.

**9.9** The test report shall be signed by the vendor or his representative.

**9.10** When the test results have been converted to specified conditions, they shall be compared with the specified values.

**9.11** If the specific energy requirement is specified for only flow rate and pressure ratio, and the specific energy requirement according to the test meets the specification at a flow rate that does not deviate by more than ± 5 % from the specified flow rate, the specific energy requirement shall be approved. For liquid-ring compressors, see 7.11.

**9.12** If specific energy requirement figures are specified for more than one flow rate or pressure ratio, then weighted averages shall be used both for the test results and for the specification. If no agreement has been made, the weight of every point is taken as 1.

# Annex A

## Simplified test of a compressor

This is a performance test applicable to series-produced compressors. The following conditions apply.

**A.1** When a compressor has been type-tested, the manufacturer has the right to use a simplified test procedure to prove that the volume flow rate and specific energy requirement figures of the actual production compressors comply with the type-tested specimen.

**A.2** When a simplified test is carried out in the manufacturer's shop, the normal test-stand instruments and equipment can be used provided the measurement tolerances are within those of table 5.

**A.3** For a simplified test on site, the instrument panel delivered with the compressor can be used on agreement by both parties.

**A.4** A compressor shall be considered to have passed the simplified test if the measured volume flow rate and specific energy requirement do not deviate from guaranteed values by more than the figures given in table 5.

**A.5** The figures in table 5 cover :

— type test tolerances;

— manufacturing tolerances;

— measuring tolerances for the simplified test.

**A.6** The test report of a simplified test may be short and simple, without tolerance calculation and with only the essential corrections cited.



**Table 5 — Acceptable deviation in percent from type test nominal values**

| Compressor shaft input at normal load<br>kW | Full load (flow) | | Half load (flow) | | No-load power |
|---|---|---|---|---|---|
| | Volume flow rate | Specific energy requirement | Volume flow rate | Specific energy requirement | |
| < 10 | ± 6 | ± 7 | — | — | ± 20 |
| 10 to 100 | ± 5 | ± 6 | ± 7 | ± 7 | ± 20 |
| > 100 | ± 4 | ± 5 | ± 6 | ± 6 | ± 20 |

NOTES

1 This test is normally carried out when a type test has already been made on a compressor of the same type. When a compressor is modified in a manner likely to affect its performance, it shall no longer be regarded as being of the same type.

2 A simplified test can take place also when a type test has not been run, if accepted by both parties.

3 Any test at part load or overload is subject to agreement between manufacturer and purchaser.

# Annex B

## Specification of operating and testing conditions

### B.1 Scope

This annex covers the specifications upon which agreement shall be reached between the manufacturer (or his representative) and the purchaser at the time of signing the contract or before testing in order to define the technical supply conditions and the main features of the test procedure.

Whenever the word "contract" is used in this annex, it shall be understood as any agreement accepted in writing by both parties.

### B.2 General

Clauses B.2.1 to B.2.6 apply when no formal written contract has been set up between manufacturer and purchaser.

**B.2.1** If the purchaser requires a compressor test, he shall notify the manufacturer, at the time of enquiry, and state the kind of test and whether the test is to be in the manufacturer's works or on site and the extent of auxiliaries, if any, to be included in the test.

**B.2.2** The purchaser has the right to decide whether the competent supervisor for the test shall come from the manufacturer, the purchaser or be an independent outside person. In the two latter cases the person in question shall be subject to agreement between the purchaser and the manufacturer.

**B.2.3** If no other time limit is stated in the contract, a site test shall be performed within three months of starting up.

**B.2.4** Normal inlet conditions for air compressors shall be as follows.

If the conditions for the air at the standard inlet point are not specified in the contract and the air is taken from the free atmosphere, it is assumed that these conditions are

— dry air,

— the temperature measured at the test,

— an absolute pressure of 1 bar.

NOTE — Where a purchaser requires a compressor to deliver a certain mass flow and specifies this as either Standard cubic metres (STD $m^3$) or Normal cubic metres (NORM $m^3$) per unit of time, since there is no universally accepted definition of standard conditions, it is necessary to define the standard or normal conditions referred to and also to state the intake conditions of pressure, temperature and humidity at which the compressor is to deliver the required mass flow rate.

**B.2.5** Normal coolant conditions for compressors shall be as follows.

If no figures are stated in the contract, the inlet temperature of the coolant is assumed to be the same as the gas inlet temperature.

**B.2.6** If a test shows that the compressor does not meet the specification, then suitable alterations shall be made and a separate test shall be carried out to prove the effect of the alterations.

A repeated full test may not be necessary to check that the compressor complies with the contract.

### B.3 Operating conditions and safety regulations

Before signing a contract, the two parties shall exchange the necessary information to reach agreement on the clauses below (for operating conditions table 1 shall be observed).

**B.3.1** Regulations for pressure vessels of the country in which the equipment will be used.

**B.3.2** Any relevant safety regulations which apply in the country in which the equipment will be used, particularly those relating to its use in an explosive, flammable or otherwise dangerous atmosphere.

**B.3.3** Any other regulations concerning health or safety which apply to the installation where the equipment will be used.

**B.3.4** Altitude and ambient air temperature at the compressor.

**B.3.5** Nature and composition of the gas to be compressed. Nature and amount of any dust, solid or liquid matter which may be present in the gas. If these are likely to carry an electric charge this shall be stated.

**B.3.6** Suction conditions; temperature, absolute pressure and relative vapour pressure.

**B.3.7** Discharge conditions at the terminal outlet of the equipment supplied; for example, absolute pressure, and temperature if the latter should be specified.

**B.3.8** Volume flow rate and conditions under which it is to be expressed.

**B.3.9** Additional tolerances, if any, above the errors of measurement, allowed on the volume flow rate and the power input.

**B.3.10** Temperature or temperature range, nature and flow rate of the coolant.

**B.3.11** The power absorbed at the drive-shaft of the equipment supplied, or the specific energy requirement of the driver, if the latter is included in the equipment supplied, at the volume flow rate and conditions specified in B.3.7 and B.3.8.

**B.3.12** Rotational frequency and direction of rotation of the compressor shaft.

**B.3.13** Method of flow rate control.

**B.3.14** Maximum allowable amplitude of pressure pulsations allowed in the discharge piping, and possibly also in the suction piping.

**B.3.15** Allowable shaft-speed irregularity.

**B.3.16** Type of compressor service (continuous duty, intermittent, etc.).

**B.3.17** Anticipated variation, during operation, of any of the numerical values listed above.

**B.3.18** Static and dynamic forces acting on the foundation. Seismic conditions.

**B.3.19** Compression space to be lubricated or not.

**B.3.20** The installation conditions of the compressors (outdoors, with or without protection, indoors, etc.).

**B.3.21** For compressors handling a special gas, the condition in which the parts in contact with the gas should be delivered (surfaces specially cleaned, degreased or not, etc.).

**B.3.22** For compressors handling a special gas, the type of metal that can be used in the zones being in contact with the gas.

**B.3.23** For compressors handling a special gas, any specifications for tightness between compression chamber and mechanical part (crankcase, gear-box, etc.) as well as to the outside, and test methods to be applied.

**B.3.24** Conditions under which the equipment may be modified or refused after a test.

## B.4 Test procedure

Before undertaking the tests, the two parties shall reach agreement on the following points.

**B.4.1** Responsibility for operating the complete installation during the test, and general conditions relating to these.

**B.4.2** Definition of any running-in procedure which may be required before the tests.

**B.4.3** Nature and composition of the gas used during the tests, if it is different from that specified in B.3.5.

**B.4.4** Conversions which should be made to the test results if the operating conditions are not identical with those defined in the contract (see 6.4.8).

**B.4.5** Deviation limits of the various operating characteristics beyond which the test should not be undertaken (see table 2).

**B.4.6** Details of the prime mover to be used for the test including its performance (efficiency, etc.).

**B.4.7** Programme of the test including, if necessary, any test of accessories supplied with the machine.

**B.4.8** Test methods to be used.

**B.4.9** Measuring instruments necessary, and a statement defining who is responsible for supplying them.

**B.4.10** Location of the principal measuring instruments.

**B.4.11** Number and timing of the readings.

# Annex C

## Performance statements for packaged air compressor of displacement type

### C.1 Definitions

**C.1.1 packaged compressor :** Unit supplied by the manufacturer, fully piped and wired internally. It will include power transmission, prime mover, filters and flow rate control.

A canopy may be provided for sound insulation and/or weather protection.

Packaged compressors may also include starting equipment, intercoolers, aftercoolers, silencers, moisture separators, dryers, outlet filters, minimum pressure devices, outlet valves, check valves, etc.

**C.1.2 volume flow rate :** Flow rate of inlet air delivered at the terminal outlet of the package, but converted to the reference conditions (see C.3).

**C.1.3 absorbed power :** 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. voltage, frequency).

The power absorbed shall include the effect of any equipment such as flow rate controls, intake filters, silencers, intercooler, aftercooler, moisture separators, liquid separation equipment including their return systems, dryers, outlet shut-off valves, etc., included in the package.

### C.2 Performance statements

**C.2.1** The manufacturer shall state the make-up of the package unit, i.e. list the major components, such as those listed in C.1.1.

**C.2.2** The manufacturer shall state the performance of the total package, i.e. take into account the effect on volume flow rate and absorbed power of all the components contained in the package.

**C.2.3** The manufacturer shall state the actual shaft power load on the main prime mover.

**C.2.4** Performance figures shall be based on ambient air conditions at the air inlet of the package, measured at a spot where the pressure and the temperature are virtually unaffected by compressor operation (e.g. heat radiation and convection), and converted to reference conditions (see clause C.3).

Packaged units without enclosures shall have the intake temperature measured at the air inlet filter.

NOTE — The performance (volume flow rate and power input) of a packaged compressor should be stated as though it were operated at reference conditions (see clause C.3). As operation at the reference conditions is difficult to accomplish in production, a suitable correction factor should be determined by test for the models involved, to permit accurate interpolation.

**C.2.5** The effective (gauge) delivery pressure and temperature of the package refer to a point at the customer's connection at the package.

### C.3 Reference inlet conditions

Unless otherwise stated all performance figures for packaged compressors shall refer to the following conditions :

— Absolute pressure : 1 bar

— Air humidity : zero

— Air temperature : + 20 °C

— Cooling water temperature : + 20 °C

NOTE — Packaged compressors with radiator type enclosed liquid cooling system are regarded as air-cooled.

# Annex D

## Flow measurement with a flow straightener

### D.1 General

Whenever the recommendations of ISO 5167 for the installation of the primary measuring device cannot be followed, shortened installation lengths upstream of the primary measuring device can be used, on condition that a suitable flow straightener is used.

This annex describes such a flow straightener for use in combination with ISA 1932 nozzles.

With the arrangement described, an uncertainty of ± 2 % in the flow measurement has been shown to be practical under steady flow conditions and using an uncalibrated nozzle.

If the nozzle is calibrated, an uncertainty of ± 1,3 % can be expected.

### D.2 Scope

See ISO 5167, clause 1.

### D.3 Symbols and definitions

See ISO 5167, clause 2 excluding 2.3.2 and 2.3.4.



### D.4 Principles of the method of measurement and computation

See ISO 5167, clause 3 excluding 3.2.

### D.5 General requirements for the measurements

See ISO 5167, clause 5.

### D.6 Installation requirements

#### D.6.1 General

**D.6.1.1** The measuring process applies only to fluids flowing through a pipeline of circular cross-section.

**D.6.1.2** The pipe shall run full at the measuring section.

**D.6.1.3** The primary device shall be installed in a straight pipeline consisting of the following elements commencing at the upstream side :

a) a valve of the nominal pipe diameter;

b) a length of pipe equal to twice the pipe diameter;

c) a perforated plate according to figure 1;

d) a length of pipe equal to the pipe diameter;

e) a perforated plate according to figure 1;

f) a length of pipe equal to twice the pipe diameter;

g) an ISA 1932 nozzle;

h) a length of pipe equal to six times the pipe diameter.

**D.6.1.4** The value for the pipe diameter, $D$, to be used in the computation of the diameter ratio shall be the mean of the internal diameter over a length of 0,5 $D$ upstream of the upstream pressure tapping. This internal mean diameter shall be the arithmetic mean of measurements at four diameters at least, distributed over a length of 0,5 $D$, two of these sections being at distances 0,0 $D$ and 0,5 $D$ from the upstream tapping.

**D.6.1.5** The pipe bore shall be circular over the entire minimum length of straight pipe required. The cross-section is taken to be circular if it appears so by mere visual inspection. The circularity of the outside of the pipe may be taken as a guide, except in the immediate vicinity of the primary device where special requirements shall apply according to the type of primary device used (see D.5.3).

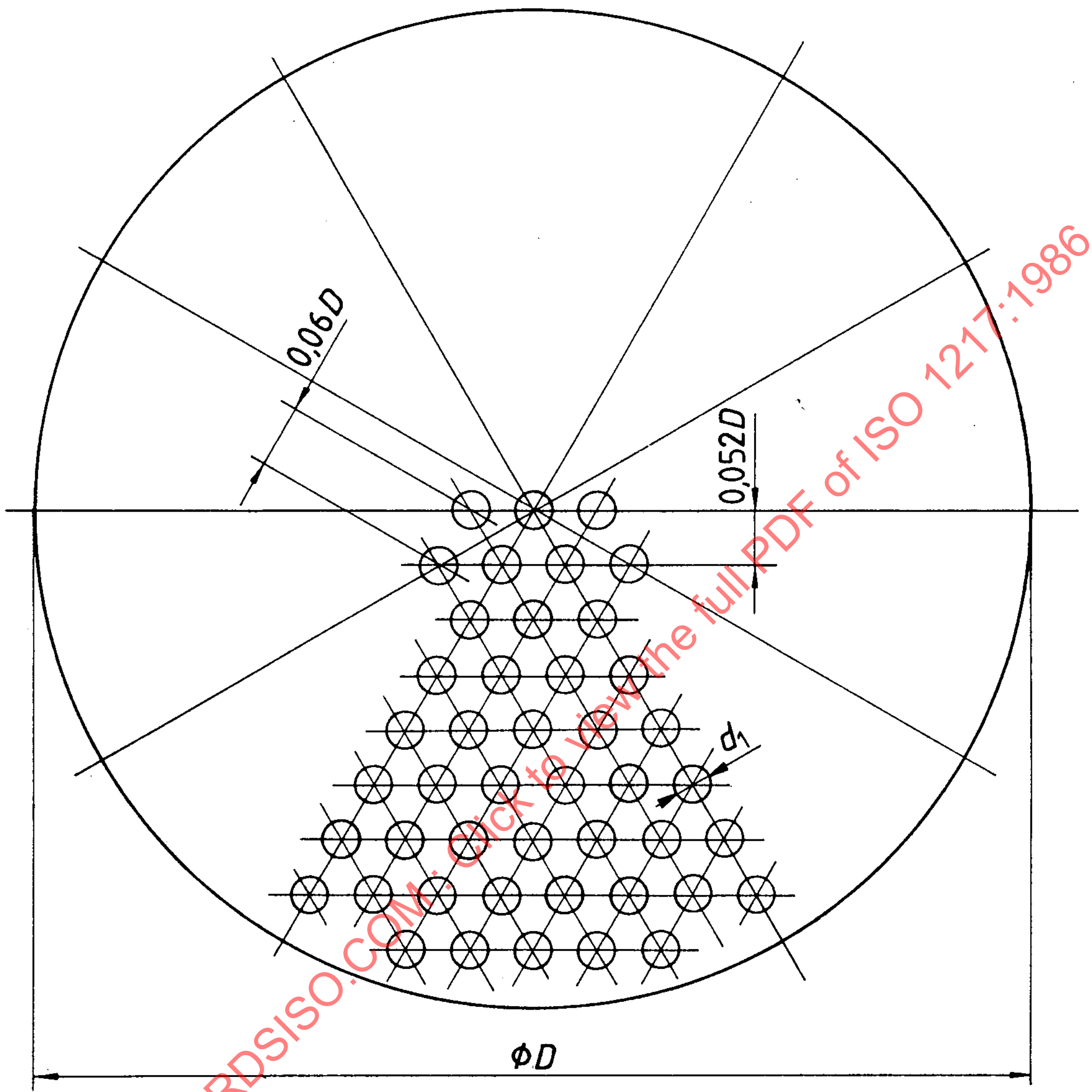
**D.6.1.6** The internal diameter $D$ of the measuring pipe shall comply with the values given for each size of nozzle.

**D.6.1.7** The inside surface of the measuring pipe shall be clean, free from pitting or deposits.

#### D.6.2 Perforated plates

The two perforated plates separated by one pipe diameter act as a flow straightening device.

The plates shall have the dimensions given in figure 1.






$d_1 = 0{,}04\,D$

$t = d_1$

where

$d_1$ is the hole diameter;

$D$ is the pipe diameter;

$t$ is the thickness of the plate.

**Figure 1 — Perforated plate for flow straightener**

### D.6.3 Additional specific installation requirements

See ISO 5167, clause 6.5 excluding 6.5.1.2 to 6.5.1.4 and 6.5.3.4.

### D.6.4 ISA 1932 nozzle

#### D.6.4.1 Dimensions

Table 6 gives the dimensions of the nozzles selected for this International Standard.

With these nozzles the pipe diameters have been fixed to give a diameter ratio approximately equal to 0,4.

#### D.6.4.2 General shape

The part of the nozzle inside the pipe is circular. The nozzle consists of a convergent portion of rounded profile and a cylindrical throat.

#### D.6.4.3 Pressure tappings

The pressure tappings are integral with the nozzle body as shown in figures 2 and 3.

When installed in the test pipe the pressure tapping should be at 180° to the opening of the valve when cracked open, that is the valve external spindle and the centre-line at the pressure tappings should lie at the same angle.

#### D.6.4.4 Discharge coefficient

Every nozzle used in conjunction with this International Standard shall be calibrated by a recognised authority and the value of the discharge coefficient thus obtained shall be used.

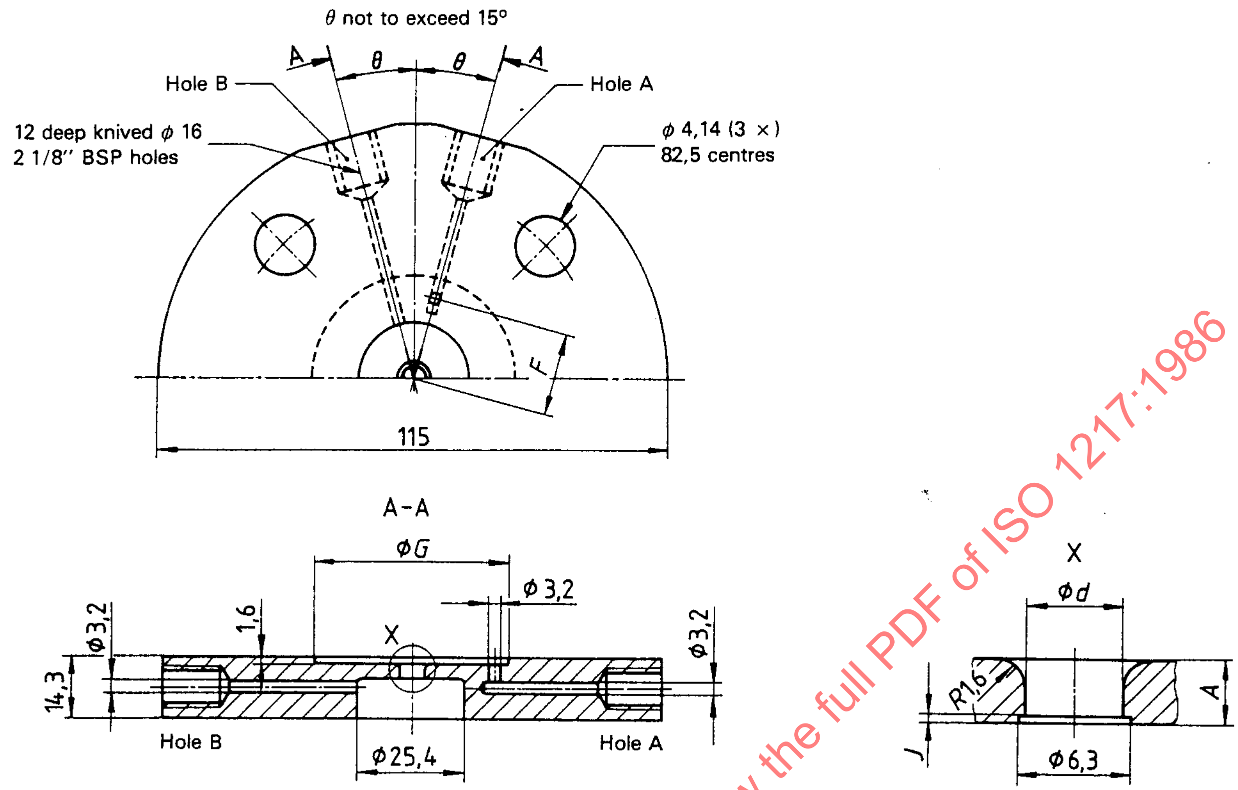
**Table 6 — Dimensions of nozzles and tolerances**
(see figures 2 and 3)

All dimensions are in millimetres, unless otherwise stated, at a temperature of 20 °C.

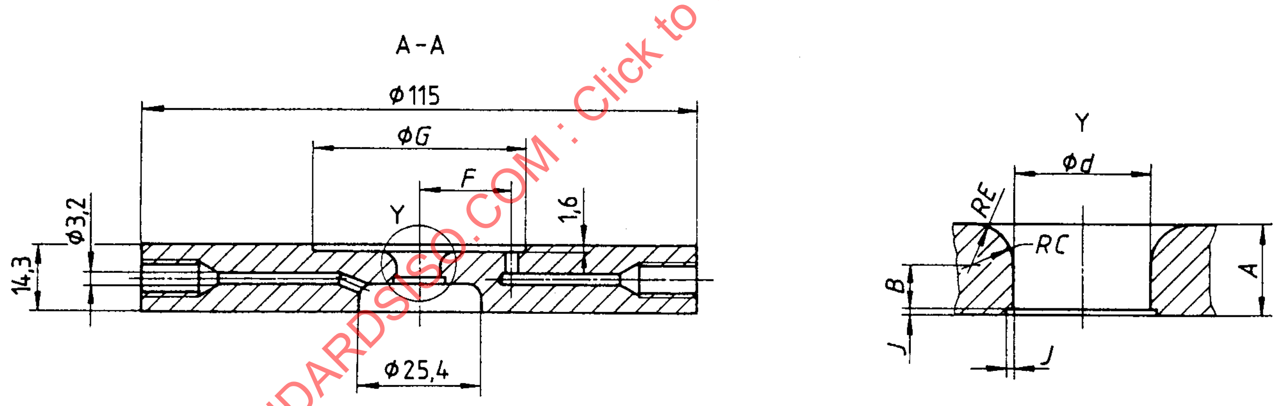
| Pipe bore $D$ | Nozzle diameter $d$ | $A$ | $B$ | $C$ | $E$ | Tolerance* on profile | $F$ | $G$ | $H$ | $J$ | $K$ |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 25 | 5,56 ± 0,01 | 3,18 ± 0,06 | — | — | — | ± 0,13 | 19,05 | 44,45 | — | 0,40 | — |
| 25 | 9,53 ± 0,01 | 5,77 ± 0,06 | 2,87 | 3,15 | 1,91 | ± 0,13 | 19,05 | 44,45 | — | 0,40 | — |
| 40 | 15,88 ± 0,01 | 9,60 ± 0,06 | 4,78 | 5,23 | 3,18 | ± 0,13 | 26,19 | 60,33 | — | 0,40 | — |
| 65 | 25,4 ± 0,03 | 15,37 ± 0,08 | 7,62 | 8,38 | 5,08 | ± 0,13 | 38,89 | 85,73 | 11,11 | 0,80 | 29,77 |
| 90 | 38,1 ± 0,03 | 23,06 ± 0,10 | 10,29 | 12,57 | 7,62 | ± 0,18 | 51,59 | 111,13 | 11,11 | 1,60 | 44,45 |
| 150 | 63,5 ± 0,05 | 38,43 ± 0,13 | 19,05 | 20,96 | 12,70 | ± 0,25 | 84,93 | 177,80 | 11,11 | 2,40 | 74,22 |
| 270 | 101,6 ± 0,05 | 61,47 ± 0,13 | 30,48 | 33,53 | 20,32 | ± 0,25 | 141,29 | 292,10 | 12,7 | 3,20 | 117,67 |
| 375 | 152,4 ± 0,10 | 92,20 ± 0,18 | 45,72 | 50,29 | 30,48 | ± 0,25 | 206,38 | 425,45 | 15,88 | 4,80 | 177,80 |
| 600 | 254,0 ± 0,18 | 153,67 ± 0,25 | 76,20 | 83,82 | 50,80 | ± 0,25 | 323,85 | 660,4 | 19,05 | 8,00 | 293,10 |
| 900 | 381,0 ± 0,25 | 230,51 ± 0,38 | 114,30 | 125,73 | 76,20 | ± 0,25 | 497,43 | 971,55 | 19,05 | 12,20 | 419,1 |

* The curved surface shall not depart from the nominal profile by more than this tolerance at any point between the cylindrical throat and the upstream face of the nozzle.

Dimensions in millimetres

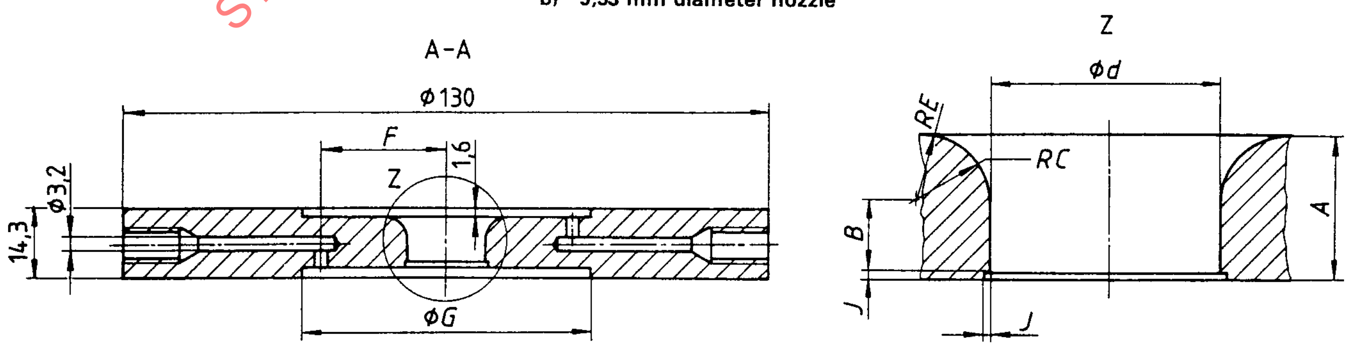



a) **5,56 mm diameter nozzle**




NOTE — Angle between pressure tapping as in figure 3.

b) **9,53 mm diameter nozzle**




c) **15,88 mm diameter nozzle**

**Figure 2 — 5,56, 9,53 and 15,88 diameter nozzles**

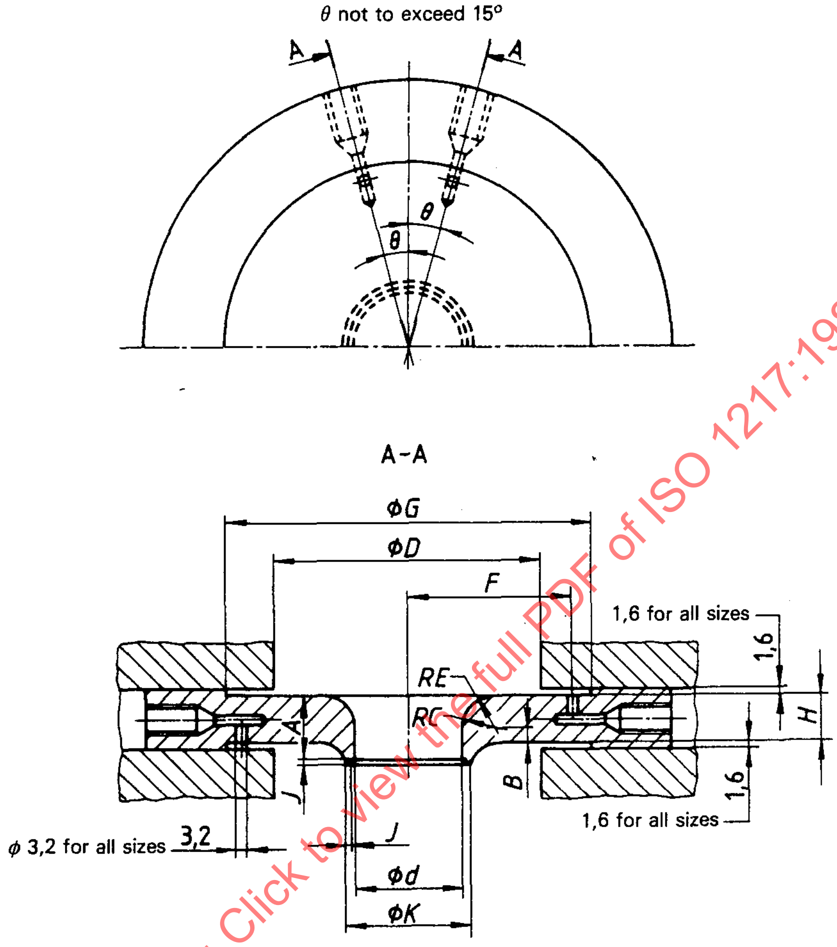



**Figure 3 — Nozzles 25,4 mm diameter and larger**



### D.6.5 Material and manufacture

See ISO 5167, clause 7.1.9 excluding 7.1.9.3.

## D.7 Expansibility (expansion) factor

The expansibility (expansion) factor is calculated by means of the following formula :

$$\varepsilon = \left[ \frac{\varkappa \tau^{2/\varkappa}}{\varkappa - 1} \times \frac{1 - \beta^4}{1 - \beta^4 \tau^{2/\varkappa}} \times \frac{1 - \tau^{(\varkappa - 1)/\varkappa}}{1 - \tau} \right]^{1/2}$$

Test results for determination of $\varepsilon$ are known for air, steam and natural gas only. However, there is no known objection to using the same formula for other gases and vapours the isentropic exponent of which is known.

Moreover the formula is applicable only if

$$\frac{p_2}{p_1} > 0{,}75$$

## D.8 Uncertainties in the measurement of flow rate

See ISO 5168.

With the test pipe-line as described and using a calibrated nozzle the uncertainty of the effect of any control valve induced flow asymmetry is ± 0,5 %.

The tolerance, $\tau_\varepsilon$, as a percentage, is

$$\tau_\varepsilon = \pm 2 \frac{\Delta p}{p_1}$$

### D.8.1 Definition of uncertainty

See ISO 5167, clause 10.1.

### D.8.2 Practical computation of the uncertainty

See ISO 5167, clause 10.2 excluding 10.2.2.2.

# Annex E

## Simplified method for air volume flow rate measurement by means of circular arc venturi nozzles at critical flow conditions

### E.1 General

The purpose of this annex is to provide a simple, quick and economical method of measuring the flow rate of air compressors.[1)]

This method has an accuracy of ± 2,5 %.

### E.2 Test arrangement

The nozzle diameter shall be chosen so as to ensure that the pressure ratio across the nozzle produces sonic velocity in the throat.

The nozzle shall be inserted in a pipe with a diameter equal to or greater than four times the nozzle throat diameter. Upstream of the nozzle shall be a length of pipe equal to at least two pipe diameters, in the wall of which are fitted means for measuring the pressure and temperature of the air flowing through the pipe. At the upstream end of this pipe, a flow straightener shall be fitted consisting of two perforated plates mounted one pipe diameter apart. See figures 2 and 4. Downstream of the nozzle a pipe and silencer can be fitted provided the pressure drop across this downstream piping does not invalidate the critical flow conditions across the nozzle.



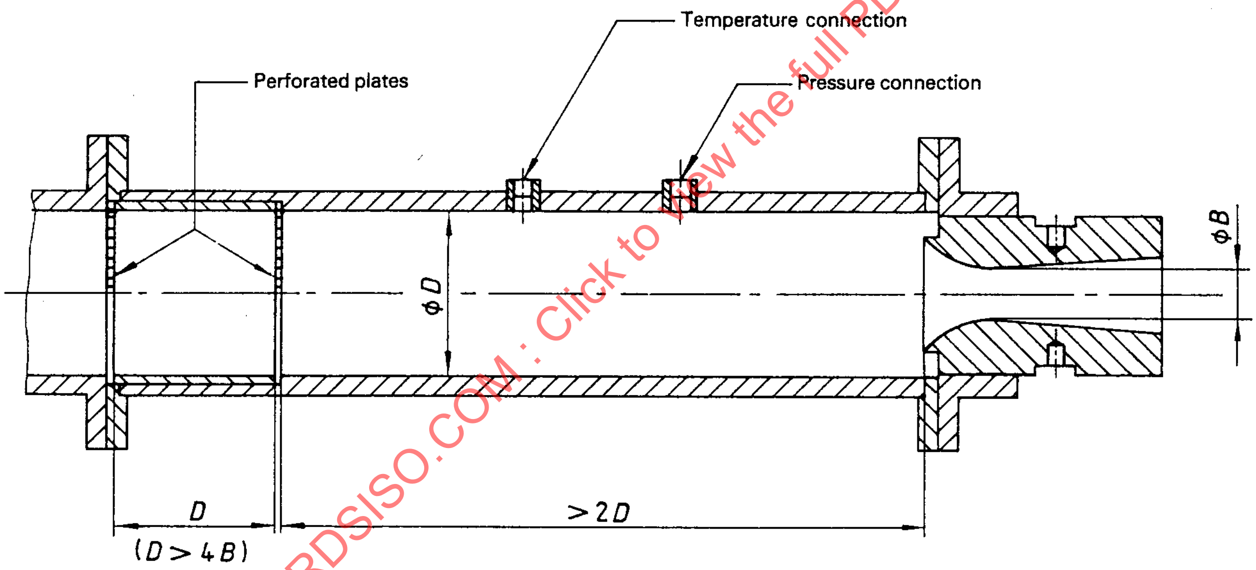



**Figure 4 — Measuring pipe**

1) The method is based on work carried out by the National Engineering Laboratory (British Department of Industry), East Kilbride, Scotland. See also BLAKE K.A., KINGHORN F.C., and STEVENSON R., The design of flow straightener/nozzle packages for acceptance testing of air compressors and exhausters. Paper C 33/78 Institution of Mechanical Engineers, International Conference on Design and Operation of Industrial Compressors, March 1978, and BRAIN T.J.S. and REID J., Primary calibrations of critical flow venturi nozzles in high pressure gas. FLOMEKO Conference, Groningen, Netherlands Sept. 1978.

## E.3 Circular arc venturi

The design shall be as shown in figure 5. The internal surface shall be polished and the throat diameter measured accurately. Suggested dimensions are given in table 7.

## E.4 Pressure and temperature readings

The pressure shall be read with an accuracy of ± 0,5 % and the temperature with an accuracy of ± 1 K.

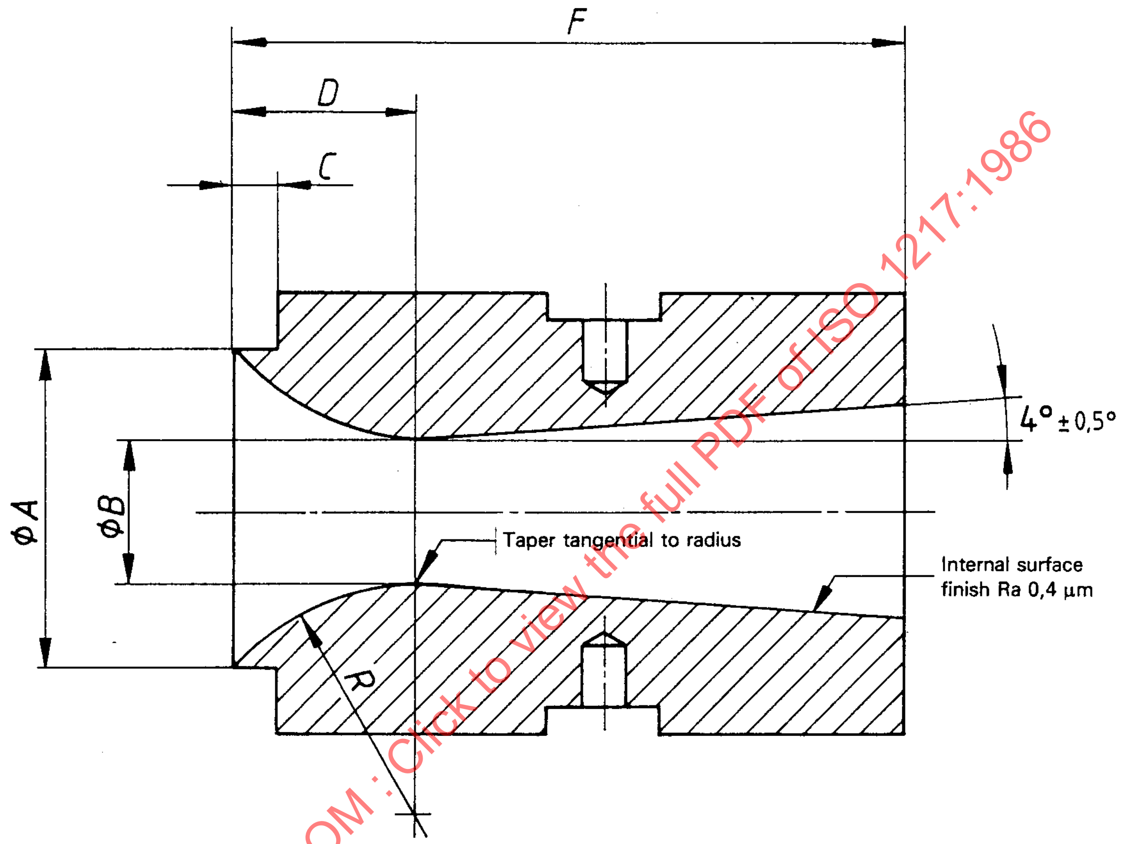





**Figure 5 — Circular arc venturi nozzle**

**Table 7 — Nozzle dimensions**

| **Flow rate** l/s | *A* mm | *B* mm | *C* mm | *D* mm | *R* mm | *F* mm | **Conical thread** in |
|---|---|---|---|---|---|---|---|
| 12 to 40 | 16,00 | 6,350 | 2,40 | 9,96 | 12,70 | 60,5 | 1.0 |
| 24 to 90 | 24,00 | 9,525 | 3,60 | 14,95 | 19,05 | 91,0 | 1.5 |
| 50 to 160 | 32,00 | 12,700 | 4,60 | 19,93 | 25,40 | 121,5 | 2.0 |
| 100 to 360 | 48,00 | 19,050 | 7,10 | 29,89 | 38,10 | 182,0 | 2.5 |
| 180 to 650 | 64,00 | 25,400 | 9,60 | 39,85 | 50,80 | 243,0 | 3.0 |
| 280 to 1 000 | 80,00 | 31,750 | 12,00 | 49,82 | 63,50 | 303,5 | 3.5 |
| 400 to 1 500 | 95,00 | 38,100 | 14,20 | 59,38 | 76,20 | 364,0 | 4.0 |

## E.5 Test

When steady flow conditions have been reached, the following readings shall be taken :

a) atmospheric pressure, $p_b$;

b) nozzle upstream pressure, $p_N$;

c) nozzle upstream temperature, $\theta_N$;

d) temperature, $\theta_o$, and pressure, $p_o$, at which volume flow rate is required.

## E.6 Flow rate calculations

The mass flow rate, $q_m$, in kilograms per second is calculated as follows :

$$q_m = \frac{0{,}1\ \pi\ B^2\ C_D\ C^*\ p_N}{4\sqrt{R\ T_N}}$$

where

$B$ is the nozzle diameter in millimetres;

$C_D$ is the discharge coefficient;

$C^*$ is the critical flow factor;

$p_N$ is the absolute pressure upstream of the nozzle in bar;

$T_N$ is the absolute temperature upstream of the nozzle in kelvins;

$R$ is the gas constant in joules per kilogram kelvin (for air, $R$ = 287,1).

The critical flow factor is given by

$$C^* = 0{,}684\,858 + (3{,}705\,75 - 4{,}769\,02 \times 10^{-2} \times \theta_N + 2{,}630\,19 \times 10^{-4} \times \theta_N^2)\ p_N \times 10^{-4}$$

where $\theta_N$ is the temperature upstream of the nozzle in degrees Celsius.

Based on test results and for the accuracy stipulated

$$C_D = 0{,}988\,8$$

When used at the discharge of portable or packaged air compressors, $\theta_N$ will vary from 20 to 70 °C and $p_N$ from 2 to 8 bar. $C^*$ will therefore vary from 0,687 1 to 0,685 2; an average value of 0,686 2 can be used. Under these conditions the equation, giving a result in kilograms per second, can be simplified to :

$$q_m = \frac{0{,}1\ \pi\ B^2\ 0{,}988\,8 \times 0{,}686\,2 \times p_N}{4\sqrt{287{,}1\ T_N}}$$

$$= \frac{3{,}143 \times 10^{-3}\ B^2\ p_N}{\sqrt{T_N}}$$

or converted to volume flow rate, $q_V$, giving a result in litres per second, at the reference conditions :

$$q_V = \frac{9 \times 10^{-3} \times B^2\ p_N\ T_o}{p_o\sqrt{T_N}}$$

where

$p_o$ is the absolute reference pressure in bar;

$T_o$ is the absolute reference temperature in kelvins.

# Annex F

## Alternative methods for determining volume flow rates

### F.1 Measurement of the volume flow rate by gas meter

If a wet or dry gas meter is used, precautions shall be taken to avoid pulsating flow.

By this method, the volume is measured directly and the density and flow disturbances are relatively unimportant. However, it shall be used only on the condition that the gas meter is in good condition and recently calibrated.

The gas meter shall be checked for leaks. For instruments with a sealing liquid, it shall be checked that the liquid is saturated with the gas to be measured.

The accuracy shall be ± 1 % or better.

### F.2 Measurement of the volume flow rate by filling a receiver (see figure 6)

This method can lead to errors due to the difficulty of measuring the gas temperature in the receiver and due to leaks in the shut-off valves. It shall be used only for small compressors and the conditions specified below shall be observed.

**F.2.1** The shut-off valves and all pipes and fittings shall be checked for leaks.

**F.2.2** A pulsation damper (2) having a discharge valve (5) to atmosphere shall be fitted between the compressor and the receiver. The size shall correspond to a charging time of 30 s or more.

**F.2.3** The receiver size shall correspond to a charging time of 5 min or more.

**F.2.4** The volume of the receiver shall be determined with an accuracy of ± 0,2 %. The best way is to fill it with water.

**F.2.5** The compressor shall first pump up the pulsation damper. The discharge valve shall be left slightly open to the atmosphere so that the correct working pressure is maintained. The communicating valve (6) between the pulsation damper and the main receiver shall be closed.

The compressor shall operate until steady conditions are reached. During this time the pulsation damper and the receiver shall be carefully drained of any condensate.

**F.2.6** The discharge valve (5) on the pulsation damper shall now be opened a little more so that the pressure in the pulsation damper drops. The discharge valve shall then be closed.

**F.2.7** The compressor charges the pulsation damper. When the working pressure is reached, the communicating valve (6) to the main receiver shall be opened slowly so that the correct working pressure is maintained in the pulsation damper.

**F.2.8** Pressure and temperature readings shall be taken for the gas in the receiver during the charging period.

**F.2.9** When the receiver pressure is about 90 % of the working pressure, the valve (6) between the pulsation damper and the receiver shall be opened further so that the pressure in the pulsation damper drops a little, after which the valve shall be fully closed.

**F.2.10** The pressure in the pulsation damper will now increase. The compressor shall be stopped or unloaded when the working pressure is reached.

**F.2.11** The time to fill the receiver shall be measured from the opening of the communicating valve between the pulsation damper and the receiver to the moment when pressure in the damper rises to working pressure after the valve is closed.

**F.2.12** The shaft-speed shall be measured during the test.

**F.2.13** The receiver shall now be left until thermal equilibrium is reached and the condensate has collected.

**F.2.14** Pressure and temperature readings shall be made, after which the condensate shall be drained and weighed.

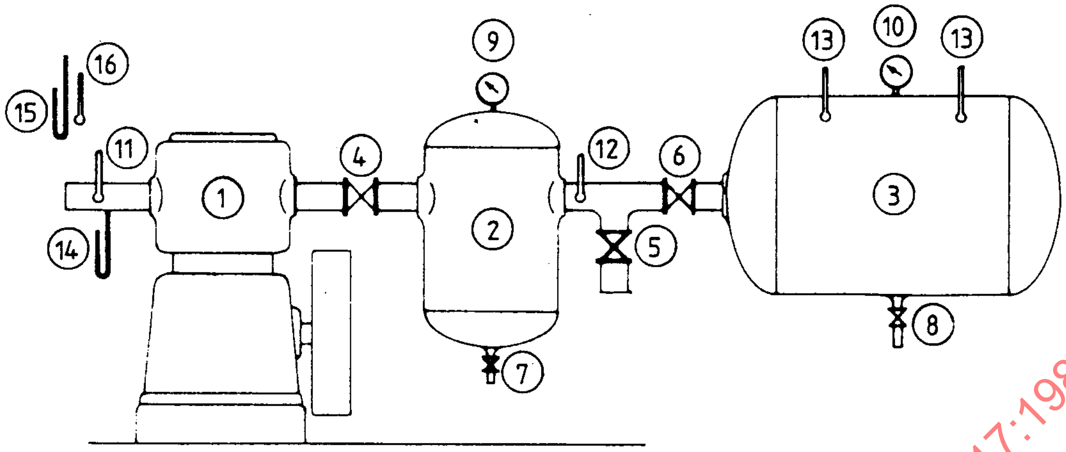
**F.2.15** The compressor volume flow rate, $q_V$, in litres per second, is given by

$$q_V = \frac{V_r T_1}{z \ p_1} \left( \frac{p_{4z}}{T_{4z}} - \frac{p_{4o}}{T_{4o}} \right)$$

where

$V_r$ is the receiver volume in litres;

$z$ is the charging time in seconds;




1 Compressor
2 Pulsation damper
3 Main air receiver
4 Communicating valve
5 Discharge
6 Communication valve
7 Drain valve
8 Drain valve
9 Manometer
10 Manometer
11 Thermometer
12 Thermometer
13 Thermometer
14 Open manometer
15 Barometer
16 Thermometer

**Figure 6 — Test layout**



$p_1$ is the absolute pressure at the standard inlet point in bar;

$p_{4o}$ is the absolute pressure in the receiver at the beginning of the charging period in bar;

$p_{4z}$ is the absolute pressure in the receiver at the beginning of the charging period in bar;

$T_1$ is the absolute temperature at the standard inlet point in kelvins;

$T_{4o}$ is the absolute temperature in the receiver at the beginning of the charging period in kelvins;

$T_{4z}$ is the absolute temperature in the receiver at the end of the charging period in kelvins.

## F.3 Measurement of the volume flow rate by weighing a receiver

This method avoids the error due to incorrect temperature readings. The difference in mass between the empty and the filled receiver shall not be too small to allow a reasonably accurate reading of the scale. This method shall consequently be used only for high-pressure compressors with small volume flow rates.

## F.4 Measurement of the aspirated flow by gas-holder

The aspirated volume can be measured with good accuracy by measuring the descent of a gas-holder, provided that the following items are observed :

a) The gas-holder shall be efficiently isolated during the test, and the shut-off valves shall be checked for leaks. It shall be made certain that no gas enters the gas-holder and that no gas escapes other than through the compressor.

b) The descent of the gas-holder shall be measured at least at three different points around the circumference. The average value shall be used.

c) The diameter of the gas-holder shall be known from a certified working drawing.

d) The gas-holder pressure shall be checked with a water gauge during the test.

e) The temperature of the gas shall be measured in the suction pipe immediately after the gas-holder.

f) The ambient conditions shall be such that the temperature of the gas in the gas-holder can be considered as being equal to the measured ambient temperature. For this reason, the test shall be performed on a cloudy day or, even better, at night.

## F.5 Measurement of the aspirated flow by gas meter

If a wet or dry gas meter is used, a receiver of sufficient volume shall be installed between the gas meter and the compressor. The gas meter shall be in good condition and recently calibrated.

The pressure drop caused by the gas meter shall be taken into account for the calculation of the volume flow rate.

The gas meter shall be checked for leaks. For instruments with a sealing liquid, it shall be checked that the liquid is saturated with the gas to be measured.

## F.6 Other methods for determining the flow rate

If, for practical reasons, none of the methods so far mentioned can be used, other methods may become necessary. The methods given in the following sub-clauses normally have less accuracy but may give useful information about the performance of the machine.

### F.6.1 Determination of the flow rate by indicator diagrams

Indicator diagrams can give very valuable information on the behaviour of the valves and the general performance of a compressor. However, a calculation of the volume flow rate based on such diagrams always gives inaccurate figures, because the heat transfer to the aspirated gas is unknown and cannot be taken into account.

### F.6.2 Determination of the flow rate by heat balance

The total amount of the heat extracted in each intercooler can be calculated from the coolant flow and the temperature difference of the coolant. If the inlet and outlet temperatures of the gas are measured, and if the specific heat capacity of the gas is known (or an enthalpy/entropy diagram available) the mass flow of gas can be calculated from these data.

The latent heat of any entrained liquid that is condensed in the intercooler and aftercooler shall be taken into account, as well as heat losses by radiation or convection and any heat dissipated with the lubricating oil.

This method sometimes presents the only possible way for testing compressors for high flow rates and for high pressures. Using a good insulation and precise instruments, an arbitrary required accuracy may be ensured.

### F.6.3 Determination of the flow rate from the velocity distribution of the gas stream

If the mass density is uniform, the flow rate in the measuring pipe can be calculated when the velocity distribution is known.

# Annex G

## Other measurements of interest

When agreed upon, one or more of the following measurements shall be made.

### G.1 Lubricant consumption

The consumption of lubricant shall be measured under normal operating conditions and after the compressor has been run-in.

### G.2 Lubricant pressure

If the compressor is provided with a pressure lubricating system, a lubricant pressure gauge is normally supplied. The readings of this gauge shall be noted in the test report.

### G.3 Performance of regulating and safety devices

Various measurements can be made for checking the performance of the regulating and safety devices. The pressure limits can be determined for the engagement and disengagement of the various stages of unloading. In installations with automatic start and stop, the operating pressures can be measured and the time established for starting up, for taking up load, etc.

### G.4 Performance of gas coolers

To check the efficiency of gas coolers, the coolant flow and inlet and outlet temperatures of the coolant and the compressed gas shall be measured. The gas side pressure drop over the cooler shall be measured, preferably with water or mercury manometers, the legs of which are connected to the cooler.



### G.5 Measurements on the primemover

If measurements are desired they shall be carried out as described in appropriate test codes.

### G.6 Temperature measurements

Apart from the compulsory temperature measurements, other measurements may also be of interest. These include, for example :

- the temperature of the gas leaving the compressor;
- the temperature in the gas receiver;
- the temperature of the coolant supplied to and discharged from the cooling jackets;
- the temperature of the coolant supplied to and discharged from the intercoolers;
- the temperature of the bearings;
- the temperature of the lubricant in the crankcase.

### G.7 Sound level

See ISO 2151, ISO 3744 and ISO 4872.

### G.8 Vibrations

See ISO 2954 and ISO 3945.

# Annex H

## Method for measuring specific energy requirement

### H.1 Scope and field of application

This annex governs the use of this International Standard when declaring the specific energy requirement of a displacement compressor.

### H.2 Unit

The specific energy requirement shall be expressed in kilowatt hours per cubic metre.[1]

### H.3 State of the compressor

Measurements shall only be carried out on new, properly run-in compressors. Only readings taken under steady-state conditions shall be used.

### H.4 Standard inlet and discharge points

| Type of compressor | Inlet point | Discharge point |
|---|---|---|
| Bare stationary | Inlet flange of the compressor | Discharge flange of the compressor |
| Packaged stationary or skid-frame mounted or portable | A point close to the compressor, but chosen so that the temperature readings are not influenced by the compressor, when working | Terminal outlet of the compressor |

NOTE — When aftercoolers or air dryers or pipeline filters are integral with the package, they should be included in the measurement.

### H.5 Specific energy requirement

The specific energy requirement figure shall be based on :

a) Power input measured according to sub-clause 5.7 and corrected according to sub-clause 6.7 or 7.7 as appropriate :

1) for bare stationary compressors the power input shall be measured at the compressor drive shaft including any built-in gear transmission; this applies also to turbine drive;

2) for electrically driven packaged compressors the power input shall be measured at the input terminals of the package;

3) for combustion-engine or gas-turbine driven compressors the energy requirement shall be calculated from the measured fuel consumption according to sub-clause 5.8; the fuel shall be specified;

b) Air flow rate measured according to sub-clause 5.6 and corrected according to sub-clauses 6.5 or 7.5 as appropriate.

### H.6 Reference conditions

The compressor shall operate at full load.

The compressor reference conditions shall be as follows :

| | |
|---|---|
| Absolute inlet pressure : | 1 bar |
| Air temperature : | + 20 °C |
| Air humidity : | 0 |
| Cooling water temperature : | + 20 °C |
| Effective (gauge) working pressure : | 7 bar |

NOTE — When the intended service of the compressor requires a different pressure, the test should be carried out at that pressure, which should be stated together with the declared specific energy requirement figure.

Such pressures should agree with ISO 5941.

1) 1 kWh/m$^3$ = 3 600 J/l

# Annex I

## Derivation of the humidity correction formula

When the compressed gas contains a vapour, a certain quantity of this vapour will condense in each intercooler.

This means that the measured volume flow rate is smaller than if the aspirated gas had been dry. Power has been spent on the compression of the vapour.

The power needed, $P_w$, to compress a humid gas is given by the formula

$$P_w = P(1 + \xi)$$

where $\xi$ is a factor expressing the influence of the humidity.

Let us assume

a) isentropic compression,

b) the gas-vapour mixture is after each stage cooled to a temperature equal to the entering temperature ($T_{1w}$) of the coolant and the gas is saturated with moisture from the second to the last stage,

c) the condensate is fully removed after each stage,

d) the number of gas molecules remains constant.

The fact that a certain quantity of vapour is condensed in each intercooler implies that the volume flow rate measured on the discharge side will be smaller than if the aspirated gas had been dry. Power has been used to compress the vapour. The correction factor for the power requirement is

$$K_6 = \frac{1 + \xi_c}{1 + \xi_R}$$

or, disregarding terms of the second degree,

$$K_6 = 1 + \xi_c - \xi_R$$

Based on the four assumptions above, the inlet volume flow rate to the $i$th stage is

$$q_i = q_1 \times \frac{T_{1w}}{T_1} \times \frac{p_1 - \varphi_1 p''_{1vs}}{p_{1i} - p''_{1wvs}}$$

where

$q_1$ is the inlet volume flow rate in litres per second;

$T_1$ is the absolute gas inlet temperature in kelvins;

$T_{1w}$ is the absolute coolant inlet temperature in kelvins;

$p_1$ is the absolute inlet pressure in bar;

$\varphi_1$ is the inlet relative vapour pressure;

$p''_{1vs}$ is the inlet saturation pressure of the vapour of the first stage at $T_1$ in bar;

$p''_{1wvs}$ is the inlet saturation pressure of the vapour of the second and following stages at $T_{1w}$ in bar.

Simplified for $T_1 = T_{1w}$ the total compression power for a dry gas is

$$P = p_1 q_1 \times \frac{\varkappa}{\varkappa - 1} \times \sum_{i=1}^{n} \left[ \left( \frac{p_{2i}}{p_{1i}} \right)^{(\varkappa - 1)/\varkappa} - 1 \right]$$

where $n$ is the number of stages.

The compression power for a humid gas in the $i$th stage is

$$P_{wi} = p_1 q_1 \times \frac{\varkappa}{\varkappa - 1} \times \frac{T_{1w}}{T_1} \times \frac{p_1 - \varphi_1 p''_{1v}}{p_{1i} - p''_{1wv}} \times \frac{p_{1i}}{p_1} \times \left[ \left( \frac{p_{2i}}{p_{1i}} \right)^{(\varkappa - 1)/\varkappa} - 1 \right]$$

The compression power for a humid gas with no water separation is for the $i$th stage

$$P_{wi} = p_1 q_1 \times \frac{\varkappa}{\varkappa - 1} \times \frac{T_{1w}}{T_1} \times \left[ \left( \frac{p_{2i}}{p_{1i}} \right)^{(\varkappa - 1)/\varkappa} - 1 \right]$$

The moisture separation affects only the compression power in the second stage and onwards.

The influence factor is thus :

$$\xi = \frac{\dfrac{T_{1w}}{T_1}\left\{\left(1 - \dfrac{\varphi_1 p''_{1v}}{p_1}\right) \times \sum_{i=2}^{n}\left(1 + \dfrac{p''_{1wv}}{p_{1i} - p''_{1wv}}\right)\left[\left(\dfrac{p_{2i}}{p_{1i}}\right)^{(x-1)/x} - 1\right] - \sum_{i=2}^{n}\left[\left(\dfrac{p_{2i}}{p_{1i}}\right)^{(x-1)/x} - 1\right]\right\}}{\sum_{i=1}^{n}\left[\left(\dfrac{p_{2i}}{p_{1i}}\right)^{(x-1)/x} - 1\right]}$$

If the stage pressure ratios do not deviate more than 3 % from their theoretical value, this expression can be simplified to

$$\xi = \frac{T_{1w}}{T_1}\left[\left(1 - \frac{\varphi_1 p''_{1v}}{p_1}\right)\frac{1}{n} \times \sum_{i=2}^{n} \frac{p''_{1wv}}{p_{1i} - p''_{1wv}} - \left(\frac{n-1}{n} \times \frac{\varphi_1 p''_{1v}}{p_1}\right)\right]$$

As the factor $\varphi_1 p''_{1w}/p_1$ is much smaller than unity, the expression can further be reduced to

$$\xi = \frac{T_{1w}}{T_1} \times \frac{n-1}{n} \times \left(\frac{1}{n-1} \times \sum_{i=2}^{n} \frac{p''_{1wv}}{p_{1i} - p''_{1wv}} - \frac{\varphi_1 p''_{1v}}{p_1}\right)$$

The ratio of the partial vapour pressure to the total pressure of the humid gas can, with a very small error, be replaced by the ratio of the partial pressures of the vapour and the gas :

$$\frac{\varphi_1 p''_{1v}}{p_1} \approx \frac{\varphi_1 p''_{1v}}{p_1 - \varphi_1 p''_{1v}}$$

The ratio of the partial pressures can also be expressed as absolute humidity :

$$\frac{\varphi_1 p''_{1v}}{p_1 - \varphi_1 p''_{1v}} = \frac{R_v}{R_g} x_i$$

where

$R_v$ is the gas constant of vapour in joules per mole kelvin;

$R_g$ is the gas constant of the gas in joules per mole kelvin;

$x_i$ is the absolute humidity of the gas at the inlet of any stage (the absolute humidity may be calculated from the partial pressure of the vapour).

With the assumptions made above, the influence factor can finally be simplified to

$$\xi = \frac{T_{1w}}{T_1} \times \frac{R_v}{R_g} \times \frac{n-1}{n}\left(x_1 - \frac{1}{n-1} \times \sum_{i=2}^{n} x_i\right)$$

After inserting this into the expression for the correction factor $K_6$ we obtain

$$K_6 = 1 + \frac{R_v}{R_g} \times \frac{n-1}{n}\left[\frac{T_{1wR}}{T_{1R}}\left(x_{1R} - \frac{1}{n-1} \times \sum_{i=2}^{n} x_{Ri}\right) - \frac{T_{1wc}}{T_{1c}}\left(x_{1c} - \frac{1}{n-1} \times \sum_{i=2}^{n} x_{ci}\right)\right]$$

When the specified gas is dry and of the same temperature as the coolant the correction factor is

$$K_6 = 1 + \frac{R_v}{R_g} \times \frac{n-1}{n} \times \frac{T_{1wR}}{T_{1R}}\left(x_{1R} - \frac{1}{n-1} \times \sum_{i=2}^{n} x_{iR}\right)$$

# Annex J

# Typical test reports

## J.1 Test example No. 1

Type of gas : air

Type of compressor : reciprocating

Number of stages : 2

Coolant : water

Volume flow rate : 354,2 l/s

Absolute inlet pressure : 1 bar

Absolute discharge pressure : 8 bar

**Certificate of compressor test according to ISO 1217**

### J.1.1 Basic data

Place of test .......................................... Date of test ..........................................

Manufacturer .......................... Type .......................... Serial No. ..........................

Purchaser ..........................................................................................

Tender No. .......................................... Order acknowledgement ..........................................

Manufacturer's order No. .......................... Purchaser's order No. ..........................

Documents (catalogues, instruction books, etc.) ..........................................................

Classification (Lloyd's, etc.) .......................... Certificate (for pressure vessels) ..........................

Short description of compressor ..........................................................................

Type of gas : air

Prime mover : asynchronous slipring Make .................... Type .................... No. ....................

Transmission : direct drive Make .................... Type .................... No. ....................



**Table 8 — Specified conditions for the guarantee**

| Item | Symbol | Figure | Unit |
|---|---|---|---|
| Absolute inlet pressure | $p_{1c}$ | 1 | bar |
| Inlet temperature | $\theta_{1c}$ | not specified | °C |
| Inlet humidity | $\varphi_c$ | 0 | — |
| Absolute discharge pressure | $p_{2c}$ | 8 | bar |
| Shaft-speed | $N_c$ | 419 | $min^{-1}$ |
| Cooling water thermal difference | $(\theta_{2w} - \theta_{1w})_c$ | 21 | K |
| Cooling water flow | $q_{wc}$ | 3,1 | $m^3/h$ |
| Water inlet temperature | $\theta_{1wc}$ | equal to ambient air | °C |

**Table 9 — Guaranteed performance**

| Item | Symbol | Figure | Unit |
|---|---|---|---|
| Volume flow rate | $q_c$ | 354,2 | l/s |
| Shaft power requirement | $P_c$ | 101 | kW |

Volume flow rate and shaft input power are guaranteed for free suction from the atmosphere.

### J.1.2 Methods and equipment used for the test

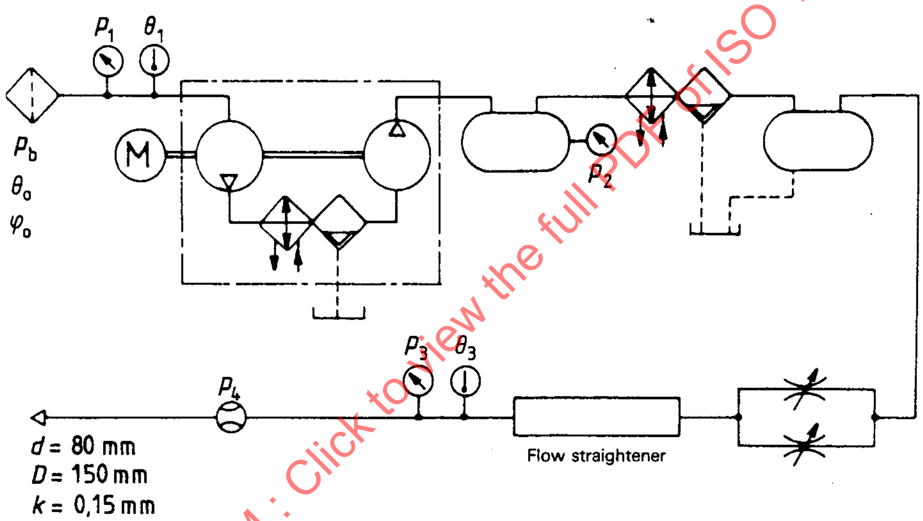



**Figure 7 — Test layout**



Flow measurement : with nozzle according to ISO 5167.

The humidity of the ambient air was measured with a psychrometer of the Assmann type.

The condensate collected during the test was measured in the intercooler, the aftercooler and the second air receiver. The aftercooler was not subject to test.

The two-wattmeter method was used for determining the prime mover input.

Motor efficiency (determined from test) :

— full compressor load : 0,934 ± 0,006 (twice standard deviation)

— no compressor load : 0,74

Instruments : . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . .

Calibration of instruments : (shall be described but is not included in this example).

**J.1.3 Table 10 — Average of test readings**
(the primary readings have already been scrutinized in accordance with 6.3.1)

| | Item | Symbol | Unit | Test figure | | |
|---|---|---|---|---|---|---|
| 1 | Date | — | — | — | — | — |
| 2 | Test number | — | — | 1 | 2 | 3 |
| 3 | Number of readings | — | — | 6 | 6 | 2 |
| 4 | Duration of test | — | min | 35 | 35 | 15 |
| 5 | Compressor load | — | % | 100 | 100 | 0 |
| 6 | Absolute discharge pressure | $p_{2R}$ | bar | 8,04 ± 0,10 | 8,05 ± 0,10 | 8,0 |
| 7 | Barometric pressure | $p_{bR}$ | mbar | 1 050 | 1 050 | 1 050 |
| 8 | Site temperature | $\theta_{oR}$ | °C | 18,2 | 18,1 | 18,3 |
| 9 | Inlet temperature | $\theta_{1R}$ | °C | 20,0 ± 1,0 | 20,3 ± 0,9 | — |
| 10 | Inlet wet bulb temperature | $\theta_{owR}$ | °C | 17,7 | 17,8 | — |
| 11 | Shaft-speed | $N_R$ | $min^{-1}$ | 418 ± 1 | 418 ± 1 | 424 |
| 12 | Inlet absolute pressure | $p_{1R}$ | bar | 1,040 | 1,040 | — |
| 13 | Water flow, compressor | $q_{wR}$ | $m^3/h$ | 3,1 | 3,1 | — |
| 14 | Cooling water inlet temperature | $\theta_{1wR}$ | °C | 9,0 | 9,0 | — |
| 15 | Cooling water outlet temperature | $\theta_{2wR}$ | °C | 30,2 | 30,4 | — |
| 16 | Absolute pressure before the nozzle | $p_{3R}$ | bar | 1,072 | 1,072 | — |
| 17 | Differential pressure over nozzle | $(p_3 - p_4)_R$ | mbar | 29,70 ± 0,40 | 29,78 ± 0,40 | — |
| 18 | Temperature at the nozzle | $\theta_{3R}$ | °C | 21,1 ± 0,4 | 21,1 ± 0,4 | — |
| 19 | Motor input* | $P_R$ | kW | 109,5 ± 1,2 | 109,5 ± 1,2 | 14,5 |
| 20 | Interstage effective pressure | — | bar | 1,85 | 1,85 | 1,1 |
| 21 | Lubricating oil effective pressure | — | bar | 1,2 | 1,2 | 1,2 |
| 22 | Oil sump temperature | — | °C | 47 | 48 | 48 |
| 23 | Air temperature after 1st stage | — | °C | 115 | 116 | — |
| 24 | Air temperature after intercooler | — | °C | 18 | 18 | — |
| 25 | Air temperature after 2nd stage | — | °C | 120 | 121 | — |
| 26 | Air temperature after aftercooler | — | °C | — | — | — |
| 27 | Water temperature after aftercooler | — | °C | — | — | — |
| 28 | Water flow, aftercooler | — | $m^3/h$ | — | — | — |
| 29 | Mass flow of condensate in intercooler | — | kg/h | 3,25 | 3,25 | — |
| 30 | Mass flow of condensate in aftercooler and air receiver | — | kg/h | 5,80 | 5,70 | — |

* Tolerance from readings only.

The tolerance figures are twice the standard deviations and based on estimated reading errors with due consideration to reading and instrument errors.

(For detailed numerical calculations see test No. 2.)

### J.1.4 Volume flow rate calculation

The volume rate of flow, $q_3$, through the nozzle at $p_3$, $T_3$ is calculated according to the following formula (see ISO 5167).

$$q_3 = 1\,000\, \alpha\, \varepsilon\, \frac{\pi}{4}\, d^2 \left[\frac{2\,(p_3 - p_4)}{\varrho_3}\right]^{1/2}$$

where $\alpha\,\varepsilon = 0{,}999\,9$ and $\varrho_3$ is given by the formula

$$\varrho_3 = \frac{p_3 - p_{3\mathrm{v}}}{R_\mathrm{a}\, T_3} + \frac{p_{3\mathrm{v}}}{R_\mathrm{v}\, T_3}$$

$$R_\mathrm{a} = 287{,}1 \text{ J/(kg·K) (air)}$$

$$R_\mathrm{v} = 461{,}5 \text{ J/(kg·K) (vapour)}$$

Converted to inlet conditions $p_1$, $T_1$ by the formula :

$$q_1 = q_3\, \frac{T_1\, p_3}{T_3\, p_1}$$

When water has condensed in coolers and receivers, a flow rate correction shall be added to $q_1$ for the volume $q_\mathrm{V}$ of the condensed vapour referred to the inlet conditions. The flow rate $q_\mathrm{m}$ of the compressor is then :

$$q_\mathrm{m} = q_1 + q_\mathrm{V}$$

To enable a first determination to be made of $\alpha$ and $\varepsilon$, which factors depend on Reynolds' number, an estimate of the air flow should be made. For this purpose the following formula is used. The same formula is further used for the error calculation :

$$q_{1\mathrm{ap}} = 1\,000\, \alpha\, \varepsilon\, \frac{\pi}{4}\, d^2\, \frac{T_1}{p_1} \left[\frac{2\,(p_3 - p_4)\, p_3\, R_\mathrm{a}}{T_3}\right]^{1/2}$$



### J.1.5 Table 11 — Calculated figures

| Item | | Symbol | Unit | Calculated figures | | |
|---|---|---|---|---|---|---|
| 31 | Flow rate at actual shaft-speed | $q_\mathrm{R}$ | l/s | 356,1 | 357,0 | — |
| 32 | Correction for shaft-speed | $K_1$ | — | 1,002 4 | 1,002 4 | — |
| 33 | Correction for polytropic exponent and pressure ratio (does not apply) | $K_2$ | — | — | — | — |
| 34 | Correction for coolant temperature | $K_3$ | — | 0,989 | 0,989 | — |
| 35 | Corrected flow rate | $q_\mathrm{c}$ | l/s | 353,1 | 353,9 | — |
| 36 | Shaft input power | $P_\mathrm{R}$ | kW | 102,3 | 102,3 | 9,6 |
| 37 | Corrected power | $P$ | kW | 99,02 | 99,02 | — |
| 38 | Specific energy requirement | $w$ | J/l | 287,28 | 286,56 | — |
| 39 | Correction for inlet pressure, pressure ratio and isentropic exponent | $K_5$ | — | 0,977 7 | 0,977 7 | — |
| 40 | Correction for coolant temperature | $K_7$ | — | 0,989 | 0,989 | — |
| 41 | Correction for humidity | $K_6$ | — | 0,998 6 | 0,998 6 | — |
| 42 | Correction for shaft-speed | $K_4$ | — | 1,002 4 | 1,002 4 | — |
| 43 | Corrected specific energy requirement | $W_\mathrm{c}$ | J/l | 278,06 | 277,4 | — |

### J.1.6 Tolerance calculations for test No. 1

In 8.1, the general rules for tolerance calculations are given. In this example only the results from test No. 1 are subject to tolerance calculations.

#### J.1.6.1 Tolerance for corrected volume flow rate

Base for the calculation is the formula for $q_{1ap}$. Tolerances for

$\tau_{\alpha} = \pm$ 1,02 % (according to ISO 5167)

$\tau_{\varepsilon} = \pm$ 0,06 % (according to ISO 5167)

$$\tau_{T_1} = \pm \frac{1}{0,293} = \pm 0,34 \%$$

$$\tau_{T_3} = \pm \frac{0,4}{294,1} = \pm 0,14 \%$$

$$\tau_{(p_3 - p_4)} = \pm \frac{40}{2\,970} = \pm 1,35 \%$$

$$\tau_N = \pm \frac{1}{418} = \pm 0,24 \%$$

The errors in $d$, $D$, $p_1$ and $p_3$ are negligible.

The tolerance for the coolant temperature correction $K_3$, according to 6.5.3 is according to 8.2.7 20 % of $1 - K_3 = 0,011\,0$ or ± 0,22 % which latter figure shall be algebraically added.

$$\tau_{q\text{corr}} = \pm \left[\left\{1,02^2 + 0,06^2 + 0,34^2 + 0,5^2 \times (0,14^2 + 1,35^2) + 0,24^2\right\}^{1/2} + 0,22\right]$$

$$= \pm [1,30 + 0,22]$$

$$= \pm 1,52$$

#### J.1.6.2 Tolerance for specific energy requirement

**J.1.6.2.1** Tolerance for flow rate

When calculating the tolerance for specific energy requirement, the part of the tolerance coming from the flow measurement shall not include the tolerance in the shaft-speed measurement. Thus the tolerance for the flow rate is :

$$\pm \left\{[1,02^2 + 0,06^2 + 0,34^2 + 0,5^2 \times (0,14^2 + 1,35^2)]^{1/2} + 0,22\right\} = \pm 1,49 \ldots 1,49 \%$$

**J.1.6.2.2** Tolerance for motor efficiency : 0,006/0,934 ... 0,64 %

For this test a vector diagram has been available for determining the motor losses and the probable error in the efficiency figure.

**J.1.6.2.3** Tolerance for wattmeter reading : 1,2/109,5 (see table 10) ... 1,10 %

**J.1.6.2.4** Tolerance $\tau_R$ for the two-wattmeter instrument errors

The classes of instrument used were the following :

| Instrument | Class | Error, % | Angle error, arc minutes |
|---|---|---|---|
| Voltage transformer | 0,1 | $f_u = \pm 0,1$ | $\nu_u = 5$ |
| Current transformer | 0,1 | $f_i = \pm 0,1$ | $\nu_i = 5$ |
| Wattmeter | 0,2 | $f = \pm 0,2$ | — |

The power factor $\cos \varphi = 0{,}866$, $\operatorname{tg} \varphi = 0{,}58$

$$\tau_R = \pm \left\{ 0{,}1^2 + 0{,}1^2 + \left( \frac{2 \times 0{,}2}{1{,}50} \right)^2 + [0{,}58 \times 2{,}9 \times 10^{-2} \times (5 + 5)]^2 \right\}^{1/2}$$

$$= \pm (0{,}010 + 0{,}010 + 0{,}071 + 0{,}028)^{1/2} = \pm 0{,}35 \ldots 0{,}35 \%$$

Tolerance in inlet pressure is neglected due to the accuracy of the mercury barometer.

**J.1.6.2.5** Tolerance for discharge pressure : 0,010/0,804 ... 1,24 %

Earlier tests on the same type of compressor have shown that 1 % increase in discharge pressure corresponds to 0,55 % increase in specific energy requirement.

This can be written as

$$\frac{\partial W}{W} = 0{,}55 \frac{\partial p_2}{p_2}$$

which after integration yields

$$\ln W = \ln p_2^{0{,}55} + \ln \text{Const}$$

or

$$W = \text{Const} \times p_2^{0{,}55}$$

Thus the exponent 0,55 is the "weight" of the error in $p_2$ under the square root. Further error calculation is based on this figure as this is more accurate than a calculation based on the theoretical formula.

**J.1.6.2.6** The tolerances for the correction factors are estimated as 20 % according to 8.2.7.

| | | |
|---|---|---|
| Correction for inlet pressure | $K_5$ | 0,1 × 2,23 ... 0,45 % |
| Correction for coolant temperature | $K_7$ | 0,2 × 1,10 ... 0,22 % |
| Correction for humidity | $K_6$ | 0,2 × 0,14 ... 0,03 % |
| Correction for shaft-speed | $K_4$ | 0,2 × 0,05 ... 0,01 % |

**J.1.6.2.7** As the electrical instruments have not been recently calibrated the tolerance ± 0,35 % caused by the two-wattmeter method shall be algebraically added.

**J.1.6.2.8** The tolerance on inlet pressure is

$$\tau_W = \pm \left[ (1{,}49^2 + 0{,}64^2 + 1{,}10^2 + 0{,}55^2 \times 1{,}24^2)^{1/2} + (0{,}45^2 + 0{,}22^2 + 0{,}03^2 + 0{,}01^2)^{1/2} + 0{,}35 \right]$$

$$= \pm (2{,}07 + 0{,}51 + 0{,}35) = \pm 2{,}93 \%$$

**J.1.7 Test data for compressor type** ... working at 419 min$^{-1}$ and an absolute discharge pressure of 8 bar.

Average of tests Nos 1 and 2.

Volume flow rate[1) ]: 353,5 (± 1,5 %) l/s

Specific energy requirement[1)] : 277,73 (± 2,9 %) J/l = 0,077 1 kWh/m$^3$

1) It should be observed that if no coolant temperature and inlet pressure corrections had to be made and the electrical instruments had recently been checked, the flow rate tolerance should have been ± 1,3 % instead of ± 1,5 % and the tolerance for specific energy requirement ± 2,1 % instead of ± 2,9 %.

**J.1.8** Conclusion : the specified performance has been met.

Date : ............................................

............................................ Test engineer

............................................ Purchaser's representative

Further participants :

............................................

............................................

## J.2 Test example No. 2

| | |
|---|---|
| Type of gas : | air |
| Type of compressor : | packaged oil-flooded rotary screw |
| Number of stages : | 2 |
| Coolant : | water |
| Volume flow rate : | 350 l/s |
| Absolute inlet pressure : | 1 bar |
| Absolute discharge pressure : | 8 bar |

**Certificate of compressor test according to ISO 1217**

### J.2.1 Basic data

Place of test ........................................ Date of test ........................................

Manufacturer .......................... Type .......................... Serial No. ..........................

Purchaser ........................................................................

Tender No. .................................... Order acknowledgement ....................................

Manufacturer's order No. .................................... Purchaser's order No. ....................................

Documents (catalogues, instruction books, etc.) ........................................................................

Classification (Lloyd's, etc.) .................................... Certificate (for pressure vessels) ....................................

Short description of compressor ........................................................................

Type of gas : air

Prime mover : .................................... Make : ........... Type : ........... No. : ...........

Transmission : .................................... Make : ........... Type : ........... No. : ...........



**Table 12 — Specified conditions for the guarantee**

| Item | Symbol | Figure | Unit |
|---|---|---|---|
| Absolute ambient pressure | $p_{oc}$ | 1 | bar |
| Ambient air temperature | $\theta_{oc}$ | 20 | °C |
| Ambient air humidity | $\varphi_{oc}$ | 0 | — |
| Absolute discharge pressure | $p_{3c}$ | 8 | bar |
| Shaft-speed | $N_c$ | 2 950 | min$^{-1}$ |
| Maximum discharge temperature | $\theta_{3c}$ | 50 | °C |

**Table 13 — Guaranteed performance**

| Item | Symbol | Figure | Unit |
|---|---|---|---|
| Volume flow rate | $q_c$ | 350 | l/s |
| Power input | $P_c$ | 140 | kW |

## J.2.2 Method and equipment used for the test

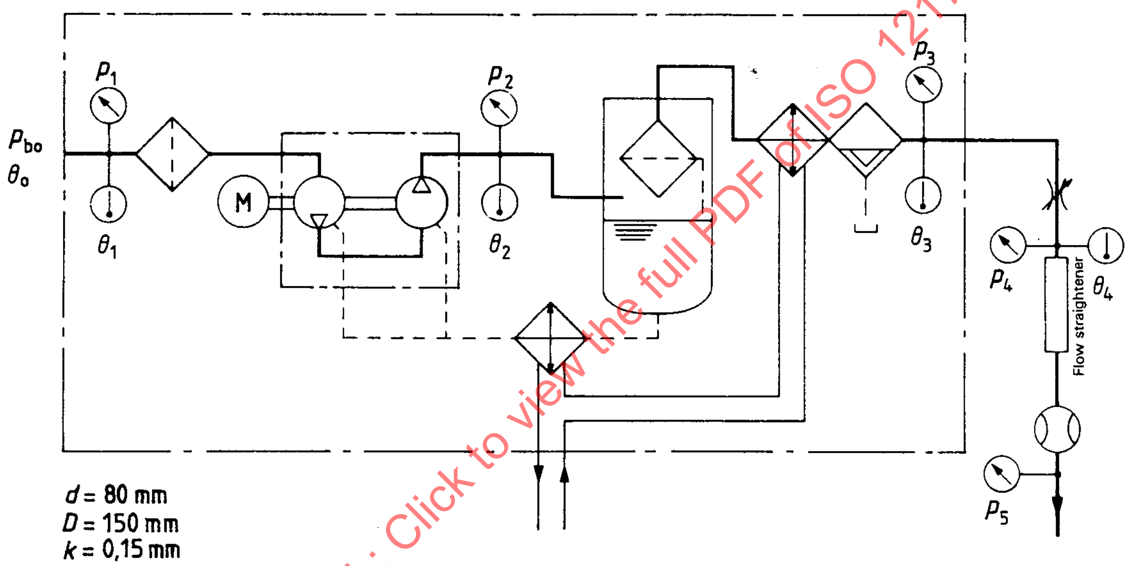



**Figure 8 — Test layout**

Flow measurement : with nozzle according to ISO 5167.

The humidity of the ambient air was measured with a psychrometer of the Assmann type.

Motor efficiency at full load (determined by a separate test) : 0,934 ± 0,006.

### J.2.3 Table 14 — Average of test readings

| Item | Symbol | Unit | Test figures | |
|---|---|---|---|---|
| 1 Date | — | — | — | — |
| 2 Test number | — | — | 1 | 2 |
| 3 Number of readings | — | — | 4 | 6 |
| 4 Duration of test | — | min | 60 | 60 |
| 5 Compressor load | — | % | 100 | 100 |
| 6 Ambient pressure | $p_{oR}$ | mbar | 1 050 | 1 050 |
| 7 Ambient temperature | $\theta_{oR}$ | °C | 18,2 | 18,1 |
| 8 Wet bulb temperature | $\theta_{oWR}$ | °C | 16,6 | 16,9 |
| 9 Inlet temperature | $\theta_{1R}$ | °C | 20,0 ± 2 | 20,3 ± 2 |
| 10 Absolute inlet pressure | $p_{1R}$ | bar | 1,04 | 1,04 |
| 11 Absolute discharge pressure | $p_{3R}$ | bar | 8,00 | 8,00 |
| 12 Discharge temperature | $\theta_{3R}$ | °C | 47 | 48 |
| 13 Condensate flow rate | $q_{medR}$ | kg/h | 3,25 | 3,28 |
| 14 Absolute pressure before the nozzle | $p_{4R}$ | bar | 1,072 | 1,072 |
| 15 Differential pressure over nozzle | $(p_4 - p_5)_R$ | mbar | 29,7 ± 0,5 | 29,8 ± 0,5 |
| 16 Temperature before nozzle | $\theta_{4R}$ | °C | 21,1 ± 2 | 21,1 ± 2 |
| 17 Electric input* | $P_R$ | kW | 132,03 ± 1,5 | 132,03 ± 1,5 |
| 18 Shaft-speed | $N_R$ | $min^{-1}$ | 2 936 ± 5 | 2 936 ± 5 |

* Including fan motor.



### J.2.4 Detailed numerical calculations (test No. 1 only)

#### J.2.4.1 Humidity content of inlet air

Relative vapour pressure (according to W. Ferrel) :

$$\varphi = \frac{p_{sw} - 0{,}660\,(1 + 0{,}001\,15\,\theta_w)\,(\theta_d - \theta_w)\,p_o}{p_{sd}}$$

where

$p_{sw}$ is the saturation pressure in millibars at $\theta_w$;

$p_o$ is the absolute ambient pressure in bars;

$p_{sd}$ is the saturation pressure in millibars at $\theta_d$;

$\theta_d$ is the dry bulb temperature in degrees Celsius;

$\theta_w$ is the wet bulb temperature in degrees Celsius.

At 18,2 °C the vapour pressure $p_{sodR}$ = 21,1 mbar.

At 16,6 °C the vapour pressure $p_{sowR}$ = 19,1 mbar.

Relative vapour pressure of the ambient air is :

$$\varphi_{oR} = [19{,}1 - 0{,}660 \times (1 + 0{,}001\,15 \times 16{,}6) \times \times (18{,}2 - 16{,}6) \times 1{,}05]/21{,}1$$

$$= (19{,}1 - 0{,}660 \times 1{,}019\,1 \times 1{,}6 \times 1{,}05)/21{,}1$$

$$= 0{,}852$$

Absolute humidity content on the inlet air is :

$$x_{oR} = \frac{0{,}622\,\varphi_{oR}\,p_{sodR}}{p_o \times 10^3 - \varphi_{oR}\,p_{sodR}}$$

$$x_{oR} = 0{,}622 \times 0{,}852 \times 21{,}1/(1{,}05 \times 10^3 - 0{,}852 \times 21{,}1)$$

$$0{,}622 \times 0{,}852 \times 21{,}1/1\,032$$

$$0{,}010\,84 \text{ kg/kg}$$

At 20 °C the vapour pressure $p_{s1R}$ = 23,38 mbar

$$x_{1R} = x_{oR}$$

Relative vapour pressure at the inlet is :

$$\varphi_{1R} = 1{,}04 \times 10^3/[23{,}38\,(1 + 0{,}622/0{,}010\,84)]$$

$$1\,040/(23{,}38 \times 58{,}38)$$

$$0{,}762$$

#### J.2.4.2 Approximate volume flow rate

$$q_{1Rap} = \alpha\,\varepsilon\,\frac{\pi}{4}\,d^2\,\frac{T_{1R}}{p_{1R}}\left[\frac{2\,(p_{4R} - p_{5R}) \times 10^2\,p_{4R}\,R_a}{T_{4R}}\right]^{1/2}$$

$$T_{1R} = 20{,}0 + 273{,}2 = 293{,}2 \text{ K}$$

$$T_{4R} = 21{,}1 + 273{,}2 = 294{,}3 \text{ K}$$

$R_a = 287{,}1\ \mathrm{J/(kg \cdot K)}$

$R_v = 461{,}5\ \mathrm{J/(kg \cdot K)}$

Assume $\alpha\,\varepsilon = 1{,}000$

$$q_{1Rap} = 1 \times \frac{\pi}{4} \times 0{,}08^2 \times$$

$$\times \frac{293{,}2}{1{,}04 \times 10^5}\left(\frac{2 \times 29{,}7 \times 10^2 \times 1{,}072 \times 10^5 \times 287{,}1}{294{,}3}\right)^{1/2}$$

$$= 0{,}353\ \mathrm{m^3/s}$$

**J.2.4.3** Mass flow rate of water vapour at the inlet

$$q_{mv1R} = \frac{\varphi_{1R}\, p_{s1R}\, 10^2\, q_{1Rap}}{R_v\, T_{1R}}$$

$$q_{mv1R} = \frac{0{,}762 \times 23{,}38 \times 10^2 \times 0{,}353}{461{,}5 \times 293{,}2}$$

$$= 0{,}004\ 65\ \mathrm{kg/s} = 16{,}73\ \mathrm{kg/h}$$

**J.2.4.4** Mass flow rate of water vapour at the nozzle

$$q_{ms4R} = q_{mv1R} - q_{mcdR}$$

$$q_{ms4R} = 16{,}74 - 3{,}22 = 13{,}52\ \mathrm{kg/h} = 0{,}003\ 76\ \mathrm{kg/s}$$

**J.2.4.5** Approximate mass flow of dry air at the nozzle

$$q_{m4Rap} = \frac{(p_{1R}\, 10^5 - \varphi_{1R}\, p_{s1R})\, q_{1Rap}}{R_a\, T_{1R}}$$

$$q_{m4Rap} = \frac{(1{,}04 \times 10^5 - 0{,}762 \times 0{,}023\ 38) \times 0{,}353}{287{,}1 \times 293{,}2}$$

$$= 0{,}436\ \mathrm{kg/s}$$

**J.2.4.6** Approximate absolute humidity content at the nozzle

$$x_{4Rap} = \frac{q_{ms4R}}{3\ 600\, q_{m4Rap}}$$

$$x_{4Rap} = \frac{0{,}003\ 76}{0{,}436}$$

$$= 0{,}008\ 6\ \mathrm{kg/kg}$$

**J.2.4.7** Approximate partial pressure of the water vapour at the nozzle

$$p_{4vRap} = \frac{x_{4Rap}\, p_{4R}}{0{,}622 + x_{4Rap}}$$

$$x_{4Rap} = \frac{0{,}008\ 6 \times 1{,}072}{0{,}622 + 0{,}008\ 6}$$

$$= 0{,}014\ \mathrm{bar}$$

**J.2.4.8** Approximate density of the air at the nozzle

$$\varrho_{4Rap} = \frac{(p_{4R} - p_{4vRap})\, 10^5}{R_a\, T_{4R}} + \frac{p_{4vRap}\, 10^5}{R_v\, T_{4R}}$$

$$\varrho_{4Rap} = \frac{(1{,}072 - 0{,}014) \times 10^5}{287{,}1 \times 294{,}3} + \frac{0{,}014 \times 10^5}{461{,}5 \times 294{,}3}$$

$$= 1{,}262\ \mathrm{kg/m^3}$$

**J.2.4.9** Determination of the final value of $\alpha\,\varepsilon$ (see ISO 5167)

$$\frac{p_{5R}}{p_{4R}} = \frac{p_{4R} - p_{5R}\, 10^{-3}}{p_{4R}}$$

$$\frac{p_{5R}}{p_{4R}} = \frac{1{,}072 - 0{,}029\ 7}{1{,}072}$$

$$= 0{,}972$$

$$m = \left(\frac{d}{D}\right)^2$$

$$m = \frac{0{,}08^2}{0{,}15^2}$$

$$= 0{,}284\ 4$$

$$m^2 = 0{,}080\ 9$$

$$\varepsilon = 0{,}983\ 2$$

**J.2.4.10** Dynamic viscosity of air

$$\eta = 17{,}2 \times 10^{-6}\left(\frac{T_{4R}}{273{,}2}\right)^{0{,}76}$$

$$\eta = 17{,}2 \times 10^{-6} \times \left(\frac{294{,}3}{273{,}2}\right)^{0{,}76}$$

$$= 18{,}2 \times 10^{-6}\ \mathrm{Pa \cdot s}$$

$$q_{4Rap} = \frac{q_{1Rap}\, T_{4R}\, p_{1R}}{T_{1R}\, p_{4R}}$$

$$q_{4Rap} = \frac{0{,}353 \times 294{,}3 \times 1{,}04}{293{,}2 \times 1{,}072}$$

$$= 0{,}344\ \mathrm{m^3/s}$$

$$c_{4R} = q_{4Rap}\, \frac{4}{\pi\, D^2}$$

$$c_{4R} = 0{,}344 \times \frac{4}{\pi \times 0{,}15^2}$$

$$= 19{,}46\ \mathrm{m/s}$$

$$Re = \frac{c_{4R}\, D\, \varrho_{4Rap}}{\eta}$$