INTERNATIONAL STANDARD 1217

INTERNATIONAL ORGANIZATION FOR STANDARDIZATION • МЕЖДУНАРОДНАЯ ОРГАНИЗАЦИЯ ПО СТАНДАРТИЗАЦИИ • ORGANISATION INTERNATIONALE DE NORMALISATION

Displacement compressors — Acceptance tests

Compresseurs volumétriques — Essais de réception

First edition — 1975-04-01

UDC 621.51.001.41 **Ref. No. ISO 1217-1975 (E)**

Descriptors : compressors, tests, acceptability, power measurement, capacity, flow measurement, pressure measurement, testing conditions, error, definitions.

Price based on 84 pages

FOREWORD

ISO (the International Organization for Standardization) is a worldwide federation of national standards institutes (ISO Member Bodies). The work of developing International Standards is carried out through ISO Technical Committees. Every Member Body interested in a subject for which a Technical Committee has been set up 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.

International Standard ISO 1217 (originally draft No. 1115) was drawn up by Technical Committee ISO/TC 118, *Compressors, pneumatic tools and pneumatic machines.*

Draft No. 1115 (which did not include annexes C, D, E and G) was circulated to the Member Bodies in October 1967. It has been approved by the Member Bodies of the following countries :

Australia	Germany	Sweden
Belgium	Greece	Switzerland
Brazil	Ireland	Thailand
Canada	Israel	Turkey
Chile	Japan	United Kingdom
Czechoslovakia	Netherlands	U.S.S.R.
Egypt, Arab Rep. of	Poland	
France	South Africa, Rep. of	

No Member Body expressed disapproval of the document.

Annexes C, D, E and G of the document were circulated to the Member Bodies in October 1972. They have been approved by the Member Bodies of the following countries :

Belgium	Hungary	South Africa, Rep. of
Bulgaria	India	Sweden
Czechoslovakia	Ireland	Switzerland
Egypt, Arab Rep. of	Japan	Thailand
Finland	Mexico	Turkey
France	Netherlands	United Kingdom
Germany	Romania	U.S.S.R.

No Member Body expressed disapproval of these annexes.

The aim of this code for acceptance tests of displacement compressors is to standardize such tests so that results from different machines and obtained by different supervisors can be compared in a technically correct way, all to promote trade and fair competition in the field. The code can be applied entirely or in part, depending on the technical and economical conditions prevailing for every separate test (see annex C). It is recommended that type testing according to sub-clause C.2.2 be carried out and that in sales literature and tenders, measured performance values be given according to the supervisor's test report.

Printed in Switzerland

CONTENTS

Displacement compressors – Acceptance tests

1 SCOPE AND FIELD OF APPLICATION

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

This International Standard specifies methods for acceptance tests and technical conditions for the supply of displacement compressors. The absolute intake pressure should exceed approximately 100 Pa (1 mbar), thus including certain types of displacement vacuum pumps.

It gives detailed instructions on the measurement of capacity and power consumption and means of adjusting the measured values to guaranteed conditions.

NOTES

1 The main body of this International Standard gives detailed instruction regarding acceptance tests of displacement compressors. There is, however, also a practical need for somewhat simpler tests. Besides the acceptance test there exist also :

- type test,
- simplified test and
- endurance test.

Instructions for carrying out these latter classes of tests are given in annex C.

2 The International System of units (SI) is used in this International Standard. The fundamental units, metre, kilogram, second, ampere, kelvin (formerly degree Kelvin) and candela, are defined in ISO 1000.

Conversion factors for other unit systems are added to this International Standard, in order to facilitate the use of the International System. (See annex F.)

2 REFERENCES

ISO/R 541, *Measurement of fluid flow by means of orifice plates and nozzles.*

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

ISO/R 1219, *Graphical symbols for hydraulic and pneumatic equipment and accessories for fluid power transmission.*

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

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

3 DEFINITIONS

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

3.1 total pressure : The pressure measured at the stagnation point when a moving gas stream is brought to rest and its kinetic energy is converted by an isentropic compression from the flow condition to the stagnation condition. It is the pressure usually measured by a Pitot tube. In a stationary body of gas the static and the total pressures are numerically equal.

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

3.3 dynamic (velocity) pressure : The total pressure minus the static pressure.

3.4 atmospheric pressure : The absolute pressure of the atmosphere measured at the test place.

3.5 gauge (effective) pressure : The pressure measured above the atmospheric pressure.

3.6 absolute pressure : The pressure measured from absolute zero, i.e. from an absolute vacuum. It equals the algebraic sum of atmospheric pressure and gauge pressure.

3.7 vacuum : The difference between the atmospheric pressure and the absolute pressure of the gas when the latter is the smaller.

3.8 standard inlet point : The 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 (see G.2.5).

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

3.9 standard discharge point: The 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 portable air compressor is the terminal outlet valve.

3.10 inlet pressure: The average absolute total pressure at the standard inlet point.

NOTE — The absolute total pressure may be replaced by the absolute static pressure provided that the velocity and density of the gas are comparatively low.

3.11 discharge pressure: The average absolute total pressure at the standard discharge point.

NOTE — The absolute total pressure may be replaced by the absolute static pressure provided that the velocity and density of the gas are comparatively low.

3.12 pressure ratio: The ratio of the discharge pressure to the inlet pressure.

NOTES

1 Stage pressure ratio is the pressure ratio for any particular stage in a multi-stage compressor, the discharge pressure being taken before the intercooler.

2 Overall stage pressure ratio is the pressure ratio for any particular stage in a multi-stage compressor, the discharge pressure being taken after the intercooler (including separator).

3.13 total temperature: The 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.

3.14 inlet temperature: The total temperature at the standard inlet point of the compressor.

3.15 discharge temperature: The total temperature at the standard discharge point of the compressor.

3.16 intercooling: The removal of heat from a gas between stages.

NOTE — Ideal intercooling prevails when the temperature of the gas leaving the intercoolers equals the temperature of the gas at the intake of the first stage.

3.17 aftercooling: The removal of heat from the gas after the compression is completed.

3.18 displacement of a compressor: The volume displaced by the compressing elements of the first stage per unit of time.

3.19 clearance volume: The volume inside a compression space, which contains gas trapped at the end of the compression cycle.

3.20 relative clearance volume: The ratio of clearance volume to the volume swept by the compressing element.

3.21 capacity of a compressor: The actual volume rate of flow 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 (see G.2.5).

3.22 capacity of a vacuum pump: The actual volume rate of flow of gas aspired and compressed by the first stage of a vacuum pump and referred to conditions of total temperature, total pressure and composition (e.g. humidity) prevailing at the standard inlet point.

NOTE — It is normally assumed that the final stage of the vacuum pump discharges to a pressure of 1 bar absolute.

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

3.24 volumetric efficiency: The ratio of capacity to displacement of a compressor or vacuum pump.

3.25 polytropic process: A compression or expansion process of an ideal gas in which the relation between pressure and volume follows the equation

$$pv^{u} = \text{constant}$$

The exponent u can have various values. For example,

$$pv = \text{constant}$$

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

$$pv^{\kappa} = \text{constant}$$

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

NOTE — In some countries this process is called adiabatic, but to avoid confusion between adiabatic (no heat exchange with the surroundings) and reversible adiabatic (isentropic) processes it is called here isentropic.

3.26 compressibility factor Z: A factor expressing the deviation of the real gas from an ideal gas:

$$Z = \frac{pv}{RT}$$

3.27 shaft speed: The number of revolutions of the compressor drive shaft per unit of time.

3.28 shaft speed irregularity : The 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.

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

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

3.30 isothermal power consumption : The 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.

3.31 isentropic power consumption : The 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 consumption is calculated assuming ideal conditions.

3.32 shaft input (absorbed power) : The power required at the compressor drive shaft. Losses in external transmissions such as gears and belt drives are not included.

3.33 indicated power : The power calculated from indicator diagrams.

3.34 mechanical efficiency : The ratio of the indicated power to shaft input.

3.35 specific energy consumption : The shaft input per unit of compressor capacity.

3.36 isothermal efficiency : The ratio of the isothermal power consumption to shaft input.

3.37 isentropic efficiency : The ratio of the real gas isentropic power consumption to shaft input.

3.38 fuel (or steam) consumption : The mass flow rate of fuel (or steam) consumed by the prime mover.

3.29 specific fuel (or steam) consumption : The ratio of fuel (or steam) consumption to compressor capacity.

3.40 temperature-rise ratio : The ratio of the isentropic temperature rise to the measured total temperature rise during compression.

4 SYMBOLS AND ABBREVIATIONS

The use of the following letter symbols is recommended. The list is formulated in line with the following seven principles.

a) The same symbols shall be used for the same quantities regardless of the system of units.

b) For any one quantity a single symbol shall be used with subscripts to indicate readings other than the primary one.

c) The same symbols shall be used for a given concept regardless of the number of special values which occur.

d) Letter subscripts shall be used to denote values under special conditions.

e) Numerical subscripts shall be used to denote values at different points of a cycle.

f) Symbols shall be confined if possible to roman letters.

g) Where possible, capital letters shall be used for absolute quantities.

4.1 Roman letters

Symbol	Quantity	Units
A	area	m^2
c	specific heat capacity	J/(kg·K)
c	relative clearance volume	–
d	orifice diameter of the measuring device at operating condition	m, mm
d	relative density	–
D	internal diameter of measuring pipe at operating condition	m, mm
e	efficiency	–
f	frequency	Hz
F	specific fuel consumption	g/m^3
g	local gravitational acceleration	m/s^2
Ho	Hodgson's number	–
i	number of compression stages	–
j	degree of interruption	–

k	absolute roughness	mm
K	correction factor	–
m	area ratio	–
M	molecular mass	–
n	rotational frequency	rev/s, rev/min
p	pressure	Pa, bar
P	power	W, kW
q	volume rate of flow	l/s, m^3/s, m^3/h
Q	mass rate of flow	kg/s, kg/h
r	pressure ratio	–
R	gas constant	J/(kg·K)
Re	Reynolds number	–
t	temperature	°C
t	time	s, h
T	thermodynamic temperature, absolute temperature	K
u	exponent for polytropic process	–
U	reading	arbitrary
v	velocity	m/s
V	volume of conduit system between the compressor and the flow measuring device	l
x	absolute humidity	kg/kg dry gas
w	specific energy consumption	J/l
W	work	J
Z	compressibility factor	–

4.2 Greek letters

α	total flow coefficient	–
β	diameter ratio	–
ϵ	expansion factor	–
η	dynamic viscosity	Pa·s
κ	isentropic exponent NOTE – For ideal gases the ratio of specific heats and the isentropic exponent have the same value.	–
ν	kinematic viscosity	m^2/s
ρ	mass density	kg/m^3
τ	tolerance	–
φ	relative humidity	–
φ	phase angle	deg

4.3 Subscripts

0	ambient conditions
1	condition at standard inlet point
2	condition at standard discharge point
3	condition upstream of measuring device
4	condition downstream of measuring device
a	air (dry)
av	average value
B	barometric
c	contract
corr	corrected
D	diameter
fuel	fuel
g	gas
i	current
m	measured
max	maximum
min	minimum
p	pressure
r	reading
Re	Reynolds number
u	voltage
v	vapour
V	volume
w	water
w	wattmeter
wet	wet

Subscripts 1, 2, 3, I, II, III, etc., and a, b, c, etc. are further used to discriminate between different quantities of the same kind, as will appear from the following clauses.

5 MEASURING EQUIPMENT AND METHODS

The equipment and methods listed in this clause are only descriptive and not intended to restrict the use of other equipment with the same or better accuracy.

5.1 Measurement of temperature

5.1.1 Temperature shall be measured by certified or calibrated instruments such as thermometers, thermoelectric instruments, resistance thermometers or thermistors inserted into the pipe or into wells.

5.1.2 Mercury-in-glass thermometers shall have an etched stem.

5.1.3 The readings of the inlet temperature of the gas and the coolant must be with an error not exceeding ± 0,2 K.

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

5.1.4 The inlet gas temperature shall be measured near the cylinder inlet flange or connection, but sufficiently distant to avoid radiation and conduction errors from cold or hot surfaces, as well as emission of hot gas from the suction valves.

5.1.5 Thermometer wells shall be as thin, and their diameters as small, as is practical, with their outside surface substantially free from corrosion or oxide. The well shall be partially filled with a suitable fluid.

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

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

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

$$t = t_r + l\,\gamma\,(t_r - t_{av})$$

where

t is the true temperature;

t_r is the actual temperature reading;

t_{av} is the average temperature of the emergent fluid column;

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

γ is the apparent expansion coefficient of the thermometer fluid (for mercury-in-glass, $\gamma = 1/6300$).

5.1.9 Precautions shall be taken to ensure

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

b) that the sensitive part of any temperature measuring device or pocket is well swept by the medium (the sensitive part ought to point against the gas stream; in extreme cases a position perpendicular to the gas stream may be used);

c) that the average gas velocity does not exceed 30 m/s at the point of measurement;

d) that the thermometer well does not disturb the normal flow.

5.1.10 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. The electromotive force of the thermocouple shall be indicated by a potentiometer-type instrument. The cold junction shall be established by a reference temperature bath.

If thermocouples are used with thermometer wells the hot junction of the couple shall, where possible, be welded to the bottom of the well.

5.2 Measurement of pressure

5.2.1 *General*

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

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

b) Connecting piping to gauges shall be as short as possible.

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

c) Connecting piping to gauges shall be not less than 6 mm bore for pressure gauges and not less than 10 mm bore for vacuum gauges to minimize capillary effect in the piping.

Connecting piping shall also be so arranged that there are no traps where water can condense.

d) Instruments shall be mounted in a position free from harmful vibrations.

e) The diameter of the scale and the arrangement of the graduations shall permit accurate readings within ± 0,5 % of the pressure measurement. Dead-weight gauges shall have adjustable weights suitable for ± 0,2 % accuracy. Diaphragm gauges shall not be employed.

f) 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.

g) If pressure wave amplitudes measured 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 pressure waves exceed 10 % of the specified average inlet or discharge pressures, a test shall not be undertaken under the rules of this International Standard unless agreed to in writing by the parties to the test.

h) Gauges having Bourdon tubes shall be calibrated under pressure and temperature conditions similar to those prevailing during the test, using dead-weight test gauges before and after the tests.

i) 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.

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

k) Column readings shall be corrected for the ambient temperature.

l) In cases of pulsating flow, a receiver with an inlet throttling hole shall be provided between the pressure tap and the manometer.

m) Oscillations of gauges shall not be reduced by throttling with a valve.

5.2.2 *Low pressure*

a) The atmospheric pressure shall be measured with a mercury barometer, which shall be read to the nearest 0,5 mm.

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.

b) 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.

c) For all pressures of 0,2 MPa absolute, or below, manometers, columns or vacuum gauges shall be employed.

Closed mercury columns known as absolute vacuum gauges shall not be used.

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

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

5.2.3 *Normal and high pressure*

For pressures above 0,2 MPa absolute, calibrated gauges with Bourdon tubes 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 shall be measured by a barometer.

If an intake pipe is provided, the pressure shall be measured by a suitable instrument. The intake pipe used must be identical with that of 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 also 5.2.1 l) and m)).

Vacuum pump inlet pressure may be determined by means of columns or manometers and shall be measured in a straight length of pipe as close as possible to the inlet flange of the machine.

5.2.5 *Intercooler pressure*

The intercooler pressure shall be measured after the intercooler. However, ± 1 % accuracy is sufficient.

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 connected before the manometer.

5.2.7 *Differential pressure*

The difference in pressure over the flow measuring device shall be measured with a fluid manometer. It shall be read with an accuracy of ± 0,2 %. The manometer tube shall normally have an inner diameter of at least 10 mm. For high operating pressures this dimension may, however, be diminished.

5.3 Measurement of delivered flow

If possible, the capacity shall be calculated from a measurement of the delivered flow. The best and most reliable method for this is given below.

The test shall be performed as indicated in ISO/R 541. It is necessary to ensure that all the requirements set forth in ISO/R 541 are completely fulfilled during the period of measurement. (For measurements when the flow is pulsating see annex A, clause A.1.)

5.4 Measurement of aspired flow

One object of a vacuum pump test is to determine the aspired volume. If possible, the capacity shall be calculated from a direct measurement. The best and most reliable method for this is given below.

For testing the capacity of a compressor, measurement of the aspired volume shall be used if measurement of the delivered volume is not practical and if the leakage losses can be measured separately.

Measurement of the aspired flow shall be performed as indicated in ISO/R 541. It is necessary to ensure that all the requirements set forth in ISO/R 541 are completely fulfilled during the period of measurement. (For measurements when the flow is pulsating, see annex A, clause A.5.)

5.5 Measurement of power

5.5.1 The measurement of the output of the prime mover or of the energy consumption shall be made according to existing test codes.

5.5.2 The power input shall, if possible, be measured by cradled electric dynamometers.

5.5.3 Cradled electric dynamometers shall not be used below one-tenth of their rated torque.

5.5.4 Torsion dynamometers 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. The calibration shall be conducted with the indicating means in place. Indicator 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 dynamometer is unsatisfactory.

5.5.5 In electrically driven compressors, the shaft input shall be determined by measuring the electrical power supplied and multiplying by the motor efficiency. Only precision instruments shall be used. Power as well as 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 must be used, the voltage drop shall be determined separately and taken into consideration. (See IEC Publication 51.)

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

5.5.7 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.

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

— for properly lubricated precision gears	98 % for each step
— for flat belt drive	97 %
— for Vee-rope drive	95 %

5.6 Internal combustion engines

The relevant national standard of the home country of the compressor manufacturer shall be used.

5.7 Measurement of rotational speed

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

Stroboscopes and speedometers are normally not sufficiently accurate, but may be used to ensure that the shaft speed is constant.

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.

5.8 Miscellaneous measurements

5.8.1 *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.

5.8.2 *Humidity*

If the compressed gas contains moisture, the relative humidity shall be checked during the test.

For tests with an open system, the dry and wet bulb temperatures shall be measured with a psychromoter of the Assman type or another instrument with similar accuracy. The humidity 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 or a psychrometer of the Assman type 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.

5.8.3 *Coolant flow*

The coolant flow is best 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/R 541.

5.8.4 *Condensation rate*

Before and after every test, the condensate shall be drained form the intercoolers and their separators. The separated quantitites shall be weighed for every cooler and divided by the time between the draining operations, which shall be carried out carefully, so that the steady state of the compressor in other respects will not be disturbed.

NOTE — Any oil carried over with the condensate shall be separated from the condensate before the mass of the latter is measured.

If water separators are provided, the efficiency of separation shall be determined.

The condensate collected in aftercoolers, receivers and other places after the discharge flange shall be measured and the total amount and time shall be noted.

5.8.5 *Fuel consumption*

If the compressor is driven by a combustion type engine, the fuel consumption shall be determined by weighing or by measuring the volume of the fuel consumed during the test.

5.8.6 *Steam consumption*

If the compressor is driven by a steam engine or turbine, the non-bleeding water rate shall be determined. (See IEC Publication 46 Part 2.)

6 PREPARATION OF THE MACHINE AND THE TESTING EQUIPMENT

6.1 General remarks

6.1.1 Before acceptance tests are begun, the compressor shall be examined in order 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 unused pipe connections from the compressor discharge shall be blanked. If this is not possible, the connections shall be broken at suitable points to allow constant observation of the outlets. To ensure correct test results an intake pipe shall not serve more than one compressor.

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

6.2 Installation of testing equipment

The installation of the measuring devices is specified in the respective sub-clauses of clause 5. Detailed instructions are given

- in 5.1 for temperature and
- in 5.2 for pressure.

6.3 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.

7 THE TEST

7.1 General rules for conducting the test

Preliminary tests shall be run for the purpose of

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

b) checking instruments;

c) training personnel.

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

7.1.1 During the test, all such measurements as have any bearing on the performance shall be made. In the following sub-clauses the determination of the capacity and the power consumption of the compressor are treated in detail.

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

7.1.3 The test conditions shall be as close as is reasonably possible to the guarantee conditions; deviations from these shall not exceed the limits specified in 9.1.6.

Where it is not feasible to test a machine either with the gas required by the purchaser or within the limitations specified in 9.1.6, special conditions of test or special corrections shall be agreed upon between purchaser and manufacturer.

7.1.4 The governing mechanism shall be maintained in its normal working position.

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

7.1.6 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.

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

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.

7.1.8 For each load two tests shall be made, each of them comprising sufficient readings to indicate that steady state conditions have been reached. The number of readings and the intervals shall be chosen to obtain sufficient accuracy.

7.1.9 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.

7.2 Evaluation of the readings

Before final calculations are undertaken, the recorded data shall be scrutinized for consistency of the operating conditions. The fluctuations of readings during one test shall not exceed the limits given in 9.1.6.

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

7.2.2 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.

7.2.3 The moisture content shall be determined from psychrometer readings at the standard inlet point, according to 5.8.2.

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

7.2.4 To determine the capacity, i.e. the flow through the measuring device, the moisture content there can be measured separately. This measurement will also serve as a check of the readings according to 7.2.3.

8 COMPUTATION OF TEST RESULTS

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

8.2 The mass rate of flow shall be determined according to 5.3 or 5.4.

8.3 When the measured fluid is hot or cold, expansion or contraction of the measuring device takes place depending upon the material used and the temperature difference. The flow area of the measuring device shall be corrected.

8.4 When the gas being compressed is not dry, the influence of the moisture shall be taken into account by correcting the specific energy consumption.

8.5 The actual inlet capacity 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.

8.6 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 at this point is calculated. This is the actual inlet capacity.

8.7 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 capacity apparently seems to attain a higher value. In such cases, therefore, the part load capacity is calculated with the inlet temperature valid for full load.

9 CORRECTION OF TEST RESULTS

9.1 General remarks

9.1.1 The test conditions never agree exactly with those which, according to tenders or purchase specifications, constitute the contract conditions for the capacity and the specific energy consumption. Therefore, before the test results and contract values are compared, corrections shall be introduced.

9.1.2 The test supervisor shall always try to keep the deviations of the test conditions from those specified as small as possible, as every correction adds uncertainty to the final result. But, as outlined in 9.1.1, corrections may be necessary to allow a proper comparison between test results and contract values.

9.1.3 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.

9.1.4 Within the limits specified in 9.1.6, this International Standard provides for adjustment of the capacity and specific energy consumption when the test conditions deviate from those specified. The capacity shall be adjusted for deviation in shaft speed, pressure ratio, isentropic exponent and coolant temperature. The specific energy consumption 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 entered.

9.1.5 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 consumption at the compressor shaft.

9.1.6 The maximum allowable differences between test values and specified values, and maximum allowable fluctuation from average during any set of readings are specified in table 1.

9.1.7 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 capacity. Such corrections shall be agreed upon by both parties.

9.1.8 If the deviation or fluctuation exceeds the values given in 9.1.6, the methods described in 9.1.3 shall be used if agreed to by both parties.

9.2 Corrections for capacity

9.2.1 *Correction for shaft speed;* K_1

The correction factor is

$$K_1 = \frac{n_c}{n_m}$$

where

n_c is the contract shaft speed;

n_m is the measured shaft speed during the test.

TABLE 1 – **Maximum deviations from specified values and fluctuations from average readings**

Measured variable		**Maximum allowable deviations**	**Maximum allowable fluctuation from average during any set of readings**
Inlet pressure	p_1	± 8 %	± 0,5 %
Pressure ratio	r	± 5 %	± 0,5 %
Inlet temperature	t_1	–	± 2 K
Absolute inlet humidity	x	–	± 5 %
Isentropic exponent	κ	± 3 %	–
Gas constant	R	± 2 %	–
Shaft speed	n	± 4 %	± 1 %
Difference between coolant temperature and gas temperature		± 10 K	± 2 K
Coolant flow rate		± 10 %	± 10 %
Temperature at the nozzle or orifice plate		–	± 2 K
Differential pressure at the nozzle or orifice plate		–	± 3 %
Voltage		± 5 %	± 2 %
Net frequency		± 1 %	± 0,5 %

NOTES

1 The test shall be accepted if the deviations from the specified conditions are equal to or less than the sum of the tolerances for deviations and those on measurement.

2 If the deviation from test conditions results in a deviation in specific energy consumption higher than 8 % then the test is not within the limits.

3 See also 5.2.1 g).

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.

9.2.2 *Correction for polytropic exponent and pressure ratio;* K_2

This correction applies only for single-stage reciprocating compressors. (It is negligible for two-stage and multi-stage compressors.)

A change in the ratio of specific heats and in the pressure ratio will influence the capacity 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 9.1.6 the formula below shall be used.

$$K_2 = \frac{1 - c\,(r_c{}^{1/u_c} - 1)}{1 - c\,(r_m{}^{1/u_m} - 1)}$$

where

r_m is the measured pressure ratio;

r_c is the contract pressure ratio;

c is the relative clearance volume;

u is the polytropic exponent (should be taken as 0,9 κ).

For pressure ratios below 3 the correction is simplified to

$$K_2 = 1 + c\,(r_m{}^{1/u_m} - r_c{}^{1/u_c})$$

9.2.3 *Correction 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 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.

NOTE — The influence of the coolant inlet temperature of the compressor capacity is often important. For example, a normal double-acting, 100 kW reciprocating compressor of a certain make, at a stage pressure ratio of about 3 and for an increase of the coolant inlet temperature of 10 K, at constant gas inlet temperature gave the following results :

Single-stage, completely watercooled :

0,5 % capacity decrease.

Two-stage, completely watercooled :

1,0 % capacity decrease.

Two-stage, aircooled compressor with watercooled intercooler :

0,5 % capacity decrease.

9.2.4 *Correction for change in gas constant*

A change in gas constant will affect the leakage and hence the capacity. A general expression for this influence cannot be given. For variations below those given in 9.1.6 this correction may be omitted.

9.3 Corrected capacity

The corrected capacity is

$$q = K_1 \, K_2 \, K_3 \, q_m$$

where q_m is the measured capacity calculated from observed results of the test.

9.4 Corrections of the specific energy consumption

9.4.1 *Correction for inlet pressure, isentropic exponent, and pressure ratio; K_4*

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

a) For single-stage machines, cooled and uncooled :

$$K_4 = \frac{[\kappa/(\kappa-1)]_c}{[\kappa/(\kappa-1)]_m} \times \frac{p_{1c}}{p_{1m}} \times \frac{r_c^{[(\kappa-1)/\kappa]_c} - 1}{r_m^{[(\kappa-1)/\kappa]_m} - 1}$$

b) For multi-stage compressors with intercooling :

$$K_4 = \frac{p_{1c}}{p_{1m}} \times \frac{\log r_c}{\log r_m}$$

c) As this correction is only theoretically correct, the pressure ratio during the test shall be held within ± 0,2 % of the specified value. In this case, the specific energy consumption correction for all displacement compressors can be simplified to

$$K_4 = \frac{p_{1c}}{p_{1m}}$$

9.4.2 *Correction for coolant inlet temperature; K_5*

The relation between the coolant inlet temperature and the specific energy consumption is very complicated and has many parameters.

For this correction it is advisable to follow the recommendation of 9.1.3.

NOTE — To show the size of this correction the same test as cited in the note to 9.2.3 can be mentioned. An increase of the coolant inlet temperature of 10 K at constant gas inlet temperature had the following influence :

Single-stage, completely watercooled :

0,2 % increase in specific energy consumption.

Two-stage, completely watercooled :

1,2 % increase in specific energy consumption.

Two-stage, aircooled compressor with watercooled intercooler :

0,8 % increase in specific energy consumption.

9.4.3 *Correction for humidity; K_6*

If in a multi-stage compressor vapour has condensed in the intercoolers, decreasing quantitites of vapour are compressed in the various stages. The correction is

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

where

R_v is the gas constant of the vapour;

R_g is the gas constant of the gas;

i is the number of stages;

x is the absolute humidity of the gas after any stage cooler (the absolute humidity may be calculated from the partial pressure of the vapour);

T_1 is the absolute gas inlet temperature.

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

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

$$K_6 = 1 + \left(\frac{R_v}{R_g} \times \frac{i-1}{i} \times \frac{T_{1wm}}{T_{1m}}\right)\left(x_{1m} - \frac{1}{i-1}\sum_{i=2}^{i} x_{mi}\right)$$

9.4.4 *Correction for shaft speed;* K_7

The specific energy consumption varies with the shaft speed. The size of this correction is, however, different for all compressor types. If possible the vendor shall produce, before the acceptance test, a correction curve obtained from a test on the compressor type in question, showing the influence of shaft speed on the specific energy consumption.

NOTE — For certain types of displacement compressors (for example, the screw type) operating in the vicinity of the optimum shaft speed, the aerodynamic losses and the internal leakage losses will compensate each other, which means that the correction factor is unity. If, on the other hand, the compressor operates at another shaft speed, a change in shaft speed will affect the efficiency.

For conventional reciprocating compressors with small internal leakage, the internal flow losses will dominate. These losses are proportional to the square of the velocity and hence also to the square of the shaft speed.

If no correction curve is available, the specific energy consumption of reciprocating compressors may be corrected according to the empirical formula

$$K_7 = 1 + a\left(1 - \frac{n_m}{n_c}\right)$$

where a is a constant depending on the compressor type.

For two-stage reciprocating compressors operating at a gauge pressure of 0,6 to 0,8 MPa and having specific energy consumption values in the range 300 to 500 J/l, it has been found that $a = 2,7 \times 10^{-6}\, w_m^{\,2}$, where

$$w_m = \frac{P_m}{q_m} \text{ J/l}$$

9.5 Corrected specific energy consumption

The corrected specific energy consumption is

$$w = K_4\, K_5\, K_6\, K_7\, w_m$$

9.6 Corrected power consumption

The corrected power consumption is obtained by multiplying the corrected capacity according to 9.3 by the corrected specific energy consumption :

$$P = q\, w$$

10 ACCURACY OF MEASUREMENT

10.1 General remarks

10.1.1 Due to the very nature of physical measurements it is impossible to measure a physical quantity without error, or to determine the true error in a physical measurement, i.e. the difference between the true and the measured value of a quantity.

10.1.2 However, when the conditions under which a physical quantity was measured are sufficiently known, it is possible to define and calculate a possible deviation of the measured value from the true value, such that it can be asserted that the absolute value of the true error is probably less than the said deviation.

Computation of such a deviation, or tolerance, is useful because its value constitutes a criterion of the accuracy of a measurement.

10.1.3 In this International Standard, measuring tolerance (limit of accuracy) is defined as a value equal to twice the standard deviation. This deviation shall be computed and reported under this name. See, for instance, ISO/R 541.

10.1.4 When the partial deviations, the combination of which gives the standard deviation, are independent of one another and are small and numerous their distribution approaches the Laplace-Gauss normal law. There is then 95 % probability that the absolute value of the true error does not exceed twice the standard deviation.

10.1.5 The error distribution is assumed to conform to the Laplace-Gauss normal law. If it is clear that the distribution is not of random type, the full value of the actual tolerance shall be added algebraically.

10.1.6 If repeated measurements of a quantity are not available or are so few that direct estimation of the standard deviation is likely to be unreliable, it is assumed that one is able, at least, to estimate the maximum range of the measurements under and over the adopted value of the quantity. It is then admissible that the standard deviation should be taken as 25 % of this estimated total range (i.e. half the mean maximum range under or over the adopted value of the quantity).

10.1.7 In the following clauses only the tolerances for the capacity and the specific energy consumption are considered, and then only when the capacity is measured to ISO/R 541 and the power input is measured according to the two-wattmeter method. Other measuring methods are of so varying a nature that their treatment falls outside the scope of this International Standard. However, the principles set forth and the methods described below for estimating the error may be applied in other cases as well.

10.1.8 The capacity and the specific energy consumption values are subject to tolerances which consist of the **test tolerance,** indicating the probable deviation of the unadjusted observed values from the true value, and the **adjustment tolerance,** covering the probable error of the method used for converting the observations to the value they would have had if the test conditions had agreed with the specified conditions.

If the power input fluctuates during the test, an **additional tolerance** is applicable for the specific energy consumption.

10.1.9 The test tolerance originates from a number of sources, the most important of which are

a) the imperfection of the measuring methods;

b) instrument errors such as incorrect adjustment and internal friction;

c) reading errors;

d) inaccurate figures for the efficiency of the equipment used.

Groups b) and c) are of the same kind and constitute reading errors, in group b) in adjustment and in group c) in actual testing. They are combined in the following clauses under the name "reading errors" and are treated in common.

10.1.10 The maximum value of the reading errors shall be estimated in each case, due regard being paid to

a) the graduation of the instruments and the means of reading;

b) the constancy of the values (a rapidly fluctuating liquid column or wavering pointer is more difficult to read than an adequately damped one);

c) the reading skill of the personnel, and the number of instruments per reader.

10.1.11 If the maximum reading errors have been estimated, the reading tolerances are calculated according to the formula

$$\tau = \pm\, 100 \frac{\Delta U}{U} \,\%$$

where

U is the average reading measured from the reference point used in the respective equations;

ΔU is the probable maximum reading error as estimated (expressed in the same units as U).

10.2 Tolerance for the capacity

This tolerance is a combination of the tolerances in shaft speed and flow measurement.

The tolerance in flow measurement is determined in accordance with the procedure described in clause 5 of ISO/R 541.

10.3 Tolerance for the specific energy consumption

10.3.1 The tolerance for specific energy consumption is obtained as the square root of the sum of the squares of the tolerances in flow and power measurements.

10.3.2 The tolerance τ_l for a two-wattmeter measurement is a combination of the tolerance for the instrument readings τ_{lr}, calculated in the normal way, and the tolerance for the instrument errors τ_{lm}, calculated according to the following sub-clauses.

10.3.3 For a two-wattmeter measurement, the errors of the six components (two voltage transformers, two current transformers and two wattmeters) are according to theory combined under the square root.

$$\tau_{lm} = \left[f_u^{\,2} + f_i^{\,2} + \left(\frac{2f}{\alpha_1 + \alpha_2} \right)^2 + 100^2 \left(|\vartheta_u| + |\vartheta_i| \right)^2 \operatorname{tg}^2 \varphi \right]^{1/2}$$

where

f_u is the relative maximum error of the voltage vector determined from the class of the voltage transformers, expressed as a percentage;

f_i is the relative maximum error of the current vector determined from the class of the current transformers, expressed as a percentage;

f is the relative maximum error of the wattmeters determined from the class of the wattmeters, expressed as a percentage;

α_1 and α_2 are the relative indications of the wattmeters :

$$(\alpha_1 + \alpha_2)_{max} = 2$$

φ is the phase angle;

ϑ_u and ϑ_i are the angle errors of the voltage and current transformers, expressed in radians.

10.3.4 If the components are recently calibrated, so that all the errors are certainly within the class figures, the combination of measurement and instrument errors will follow the square root formula

$$\tau_l = (\tau_{lr}^{\,2} + \tau_{lm}^{\,2})^{1/2} \,\%$$

10.3.5 If the conditions of 10.3.4 are not met, the errors shall be arithmetically added :

$$\tau_{l} = (\tau_{lr} + \tau_{lm})\ \%$$

10.3.6 With direct current, the same procedure can be followed broadly if volt-meters and ampere-meters are used and the readings are multiplied; the measuring error will have to be divided into two, τ_{lc1} and τ_{lc2}, that is to say, into an error for each instrument, these errors being combined into τ_{lc}

$$\tau_{lc} = (\tau_{lc1}{}^{2} + \tau_{lc2}{}^{2})^{1/2}\ \%$$

If duplicated measuring equipment is employed and the results agree perfectly, τ_{lc} may be decreased.

10.3.7 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 2 — **Tolerance for load fluctuation**

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

This tolerance shall be algebraically added to the resulting tolerance from 10.3.6.

10.4 Errors in efficiency

10.4.1 For the calculation of the power requirement of the compressor from the input to the motor as measured in the test, one might be faced with the necessity of using two efficiency curves that may contain errors, that is to say,

a) the efficiency curve of the electric motor, percentage tolerance τ_{lIa} and

b) the efficiency curve of the transmission, where such is to be found between the motor and the compressor, percentage tolerance τ_{lIb}.

10.4.2 It is difficult here to give any universal directions for the estimation of these tolerances. For tolerances of the efficiency of the electric motor, certain directions are given in national test codes. Otherwise, due regard should be paid to the methods of test or calculation of the efficiency.

10.4.3 The uncertainty in the efficiency figures given in 5.5.8 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.

10.5 Tolerances caused by the correction methods

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

For tests within the limits of 9.1.6 this tolerance may amount to ± 20 % of the correction.

11 TEST REPORT AND COMPARISON WITH CONTRACT VALUES

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

11.2 The test report shall state the object, the place and time of the test and name the participants.

11.3 The test report shall contain a brief description and the principal data of the compressor and the prime mover, the manufacturer's name, the manufacturing number, the guaranteed data, etc.

11.4 The test report shall contain a statement of the methods and the equipment used for the test and shall contain a diagrammatic layout of the test arrangement with all measurement points marked. In relevant cases (acceptance tests) the test report shall also contain a statement of the guarantees given by the vendor with regard to the capacity, specific energy consumption and the function of the compressor in operation and the operating conditions under which the guarantees apply. For important instruments the manufacturer's name, the manufacturing number and the construction and class, the steps taken for control and adjustments, etc., also the calibration records and correction diagrams for the various instruments, should preferably be included as appendices.

11.5 The test report shall contain data recorded during the capacity and power tests, 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 corrected values shall be entered separately in the test report.

11.6 Any special conditions shall be recorded.

11.7 The test report shall contain the formulae used for the calculation of the capacity and the power consumption with corrections, 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.

11.8 A comparison of the performance of the actual compressor with the contract values shall be included, together with conclusions from this comparison.

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

11.10 The test report shall be signed by representatives of the vendor and of the purchaser.

11.11 When the test results have been corrected to specified conditions they shall be compared with the contract values. If contract values were given for more than one load this comparison shall be made in graphical form. The tolerances according to clause 10 shall be included.

11.12 For compressors the power requirement against the capacity shall be plotted, and for vacuum pumps the power requirement and capacity against the vacuum shall be plotted. For both compressors and vacuum pumps the capacity shall be given as volume rate of flow and referred to inlet conditions. Points having the same pressure ratio and inlet pressure shall be connected by a smooth curve.

11.13 For assessing the efficiency, the specific energy consumption shall be plotted against capacity. Corresponding values of power and capacity shall be read from the curve drawn according to 11.12.

11.14 If the specific energy consumption is guaranteed for only one capacity, and the specific energy consumption according to the test meets the guarantee at a capacity that does not deviate more than ± 5 % from the capacity for which the guarantee is valid, the specific energy consumption shall be approved.

11.15 If specific energy consumption figures are guaranteed for more than one capacity or pressure, then weighted averages shall be used both for the test results and for the guarantees. If no agreement has been made, the weight of every point is taken as 1.

ANNEX A

ALTERNATIVE METHODS FOR DETERMINING THE FLOW

For the following methods no basis for judging the accuracy of the measurements is given while the supervisor of the test must make a special study before he can make the error calculation according to the general principles laid down in clause 10.

A.1 MEASUREMENT OF PULSATING DELIVERED FLOW

a) If pulsating flow in the measuring pipe cannot be avoided, a generous receiver shall be provided between the discharge and the measuring pipes.

b) ISO/R 541 does not include the measurement of pulsating flows, but pulsations often occur during compressor tests, where they cause errors. The most important source of error is that the measuring device will damp the nozzle pressure difference whereby its average value is read instead of the average value of the square root of the pressure difference. This error is always positive. A simple way to diminish the pulsations is to arrange a supercritical expansion between the compressor and the measuring device. Another and complementary way is to insert an extra volume in the pipe line before the measuring device. By choosing a volume big enough the average error can be kept below any desired limit.

c) There are two categories of pulsating flow to distinguish between :

1) Interrupted pulsating flow as from a single-acting, single-cylinder compressor.

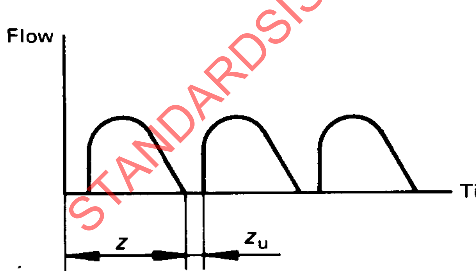


FIGURE 1 — **Interrupted pulsating flow**

This flow is characterized by the degree of interruption

$$j_x = \frac{z_u}{z}$$

where

z_u is the time during which the flow is zero;

z is the time of each period.

2) Pure pulsating flow as from a double-acting, two-cylinder compressor.

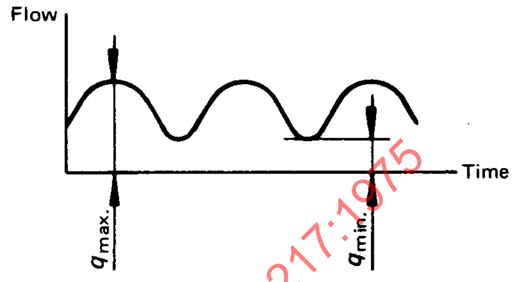


FIGURE 2 — **Pure pulsating flow**

This flow is characterized by the ratio

$$j_y = \frac{q_{max}}{q_{min}}$$

where

q_{max} is the maximum gas flow;

q_{min} is the minimum gas flow.

d) The dimensionless Hodgson number is defined as

$$Ho = \frac{V f \Delta p}{p_{av} q_{av}}$$

where

V is the volume of pipe system between the compressor and the flow measuring device, expressed in litres;

f is the frequency of the pulsations, expressed in hertz;

Δp is the total pressure drop up to the orifice plate or nozzle, expressed in pascals;

p_{av} is the average absolute pressure in the receiver, expressed in pascals;

q_{av} is the average flow at nozzle conditions, expressed in litres per second.

e) The Hodgson number necessary to keep the average error of the capacity measurement below a certain desired limit is given in tables 3 and 4.

TABLE 3 — **Hodgson number for interrupted pulsating flow**

$\frac{z_u}{z}$	*Ho* for error ⩽ ± 0,5 %	*Ho* for error ⩽ ± 1,0 %	*Ho* for error ⩽ ± 2,0 %
1,0	2,50	1,70	1,20
0,8	1,85	1,25	0,90
0,6	1,35	0,90	0,60
0,4	0,95	0,65	0,45
0,2	0,65	0,45	0,30
0	0,50	0,35	0,25

TABLE 4 – **Hodgson number for pure pulsating flow**

$\frac{q_{max}}{q_{min}}$	*Ho* for error $\leqslant \pm 0{,}5$ %	$\leqslant \pm 1{,}0$ %	$\leqslant \pm 2{,}0$ %
1	0,00	0,00	0,00
2	0,14	0,09	0,04
3	0,24	0,15	0,08
4	0,30	0,19	0,11
6	0,35	0,23	0,14
8	0,38	0,26	0,16
10	0,40	0,27	0,16

When the Hodgson number and the total pressure drop are known, the necessary intermediate volume is obtained.

f) In order to avoid abnormal velocity distribution, vortices, etc. in the measuring pipe, ISO/R 541 contains instructions on the minimum length of straight pipe that should precede the measuring device. Sometimes, the local conditions do not permit the fulfilment of these requirements. In that case, an arrangement with a perforated baffle plate and straightening vanes with an approximate length of 2 *D* shall be used. The upstream section of the pipe shall have a minimum length of 10 *D*, and the downstream section 5 *D*.

g) If the gas contains moisture, the humidity at inlet and the liquid condensed after each cooler shall be measured. Condensate shall be prevented from entering the measuring pipe.

h) If the gas used for the test differs from the gas specified in the contract then the difference in densities and in the ratios of specific heats shall be taken into account.

i) If the test is performed on air, the test arrangement may be simplified. After the measuring pipe, the air can be blown out to the atmosphere and suction can take place from the atmosphere.

A.2 MEASUREMENT OF THE DELIVERED FLOW 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.

A.3 MEASUREMENT OF THE DELIVERED FLOW BY FILLING A RECEIVER (See figure 3)

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 items below shall be observed.

a) The shut-off valves and all pipes and fittings shall be checked for leaks.

b) 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.

c) The receiver size shall correspond to a charging time of 5 min or more.

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

e) The compressor shall first pump up the pulsation damper. The discharge valve shall be left slightly open to 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) 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.

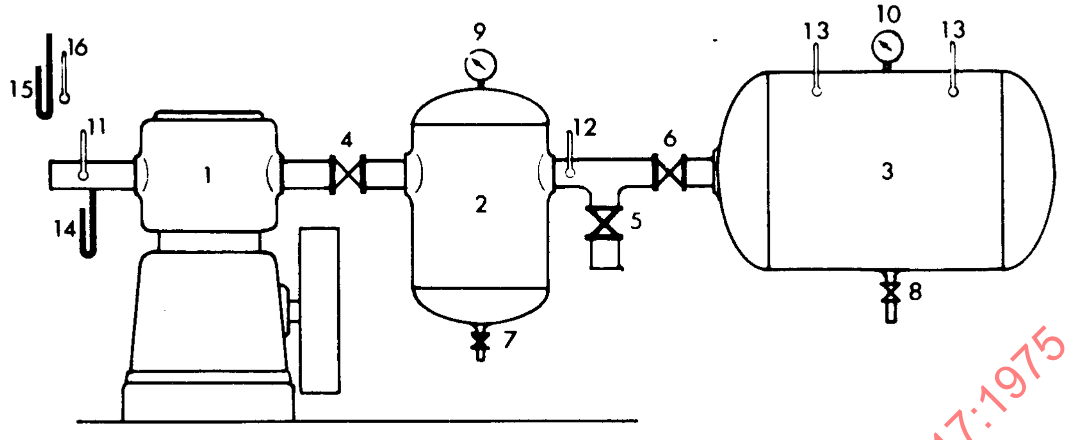
g) The compressor charges the pulsation damper. When the working pressure is reached, the communicating valve (6) to the main receiver shall be opened slowly so that the correct working pressure is maintained in the pulsation damper.

h) Pressure and temperature readings shall be taken for the gas in the receiver during the charging period.

i) 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.

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

k) 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.



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 3 – **Test layout**

l) The shaft speed shall be measured during the test.

m) The receiver shall now be left until thermal equilibrium is reached and the condensate has collected.

n) Pressure and temperature readings shall be made, after which the condensate shall be drained and weighed.

o) The compressor capacity is

$$q_m = \frac{V_r\,T_1}{z\,p_1}\left(\frac{p_{4z}}{T_{4z}} - \frac{p_{40}}{T_{40}}\right)$$

where

q_m is the measured capacity, expressed in litres per second;

V_r is the receiver volume, expressed in litres;

z is the charging time, expressed in seconds;

p_1 is the absolute pressure at the standard inlet point, expressed in pascals;

p_{40} is the absolute pressure in the receiver at the beginning of the charging period, expressed in pascals;

p_{4z} is the absolute pressure in the receiver at the end of the charging period, expressed in pascals;

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

T_{40} is the absolute temperature in the receiver at the beginning of the charging period, expressed in kelvins;

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

A.4 MEASUREMENT OF THE DELIVERED FLOW 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 capacities.

A.5 MEASUREMENT OF PULSATING ASPIRED FLOW

a) In cases of pulsating flow, a receiver volume with inlet throttling shall be provided between the measuring and inlet pipe.

b) The throttling of the intake pressure by the measuring device shall be taken into account.

c) If pulsating flow in the measuring pipe cannot be avoided, the influence of the Hodgson number on the error shall be considered (see clause A.1).

d) If the gas contains moisture, the humidity content shall be measured.

e) If the gas used for the test differs from the gas specified in the contract then the difference in densitites and in the ratios of specific heats shall be taken into account.

A.6 MEASUREMENT OF THE ASPIRED FLOW BY GAS-HOLDER

The aspired 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 otherwise 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 that reason, the test shall be performed on a cloudy day or, even better, at night.

A.7 MEASUREMENT OF THE ASPIRED 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 capacity.

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.

When testing vacuum pumps of the rotary displacement type, the meter shall be located on the intake side with a restricting valve between the meter and the vacuum side so that the meter measures atmospheric air.

A.8 OTHER METHODS FOR DETERMINING THE FLOW

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.

A.8.1 Determination of the flow 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 capacity based on such diagrams always gives inaccurate figures, because the heat transfer to the aspired gas is unknown and cannot be taken into account.

A.8.2 Determination of the flow 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 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 capacity and for high pressures. Using a good insulation and precise instruments, an arbitrary required accuracy may be assured.

A.8.3 Determination of the flow from the velocity distribution of the gas stream

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

ANNEX B

OTHER MEASUREMENTS OF INTEREST

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

B.1 LUBRICATING OIL CONSUMPTION

The consumption of lubricating oil shall be measured under normal operating conditions and after the compressor has been run-in. The oil consumption in lubricating pumps or lubricators shall be measured over the period of the test. If possible, the oil consumption test shall cover at least 50 h. Before the test the compressor shall be warmed up to its operating temperature, whereupon the oil is drained from the crankcase or sump and permitted to flow for a period of at least 30 min. The oil is then weighed and poured back into the compressor. After the test, the emptying is repeated observing exactly the same temperature and procedure. The difference in mass plus the mass of any oil added during the test is the consumption of lubricating oil.

B.2 LUBRICATING OIL PRESSURE

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

B.3 PERFORMANCE OF THE 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.

B.4 PERFORMANCE OF THE GAS COOLERS

To check the efficiency of the gas coolers, the coolant flow as well as 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.

B.5 MEASUREMENTS ON THE PRIMEMOVER

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

B.6 TEMPERATURE MEASUREMENTS

Besides 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 oil temperature in the crankcase.

ANNEX C

CONDITIONS FOR DIFFERENT CLASSES OF COMPRESSOR TESTS

C.1 DEFINITIONS OF CLASSES OF COMPRESSOR TESTS

C.1.1 acceptance test of a compressor : A performance test carried out in full accordance with the stipulations in the main text of this International Standard.

C.1.2 type test of a compressor : A performance test according to C.1.1 and supervised by an independent expert from, or authorized by, a reputable institution.

C.1.3 simplified test of a compressor : A simplified test, applicable to series produced compressors, where only the capacity and specific energy consumption are measured.

NOTE – This test shall only be carried out if a type test has already been made on a compressor of the same type. When a compressor is modified in a manner which is likely to affect its performance characteristics it can no longer be regarded as being of the same type.

C.1.4 endurance test of a compressor : An extended type test according to C.1.2 comprising a series of tests, starting with an acceptance test according to C.1.1 followed by long-time operation, according to agreement for each compressor-type and case, whereafter a new acceptance test is carried out followed by dismantling and inspection of the compressor.

C.2 COMMENTS ON DIFFERENT CLASSES OF COMPRESSOR TESTS

C.2.1 Acceptance test

The main text of this International Standard applies.

C.2.2 Type test

a) It is recommended that for series-produced compressors, the manufacturer arranges for one compressor of each series to be type tested.

b) The independent expert shall take out the test specimen at random from the compressors ready for delivery. The number of compressors in the batch shall be stated in the test report. The manufacturer shall guarantee that they are produced to the same standard.

c) The independent expert responsible for the type test shall, in the test report, clearly give the results obtained together with the tolerances. Only the results of the type test with tolerances may be used in tenders and sales literature for compressors of the same series as the type tested specimen.

d) The name and address of the authorized institution and the independent expert shall always be stated when reference is made to the type test.

e) The manufacturer shall not make reference to the type test, unless the actual compressor is identical in performance with the type tested specimen.

C.2.3 Simplified test

a) When a compressor has been type tested, the manufacturer has the right to use a simplified test procedure to prove that the capacity and specific energy consumption figures of the actual compressor comply with the type tested specimen.

b) For a simplified test the tolerance (twice standard deviation) of the measurements shall be within ± 4 %. The tolerance class of each measuring instrument must be stated.

c) When a simplified test is made in the manufacturer's shop, the normal test-stand instruments and equipment can be used provided the accuracy stipulated in C.2.3 b) is met.

d) For a simplified test on site, the control-board instruments delivered with the compressor can be used provided the accuracy stipulated in C.2.3 b) is met.

e) A compressor shall be considered to have passed the simplified test if the measured capacity and specific energy consumption do not deviate from the nominal capacity and specific energy consumption obtained in the type test by more than the figures given in table 5.

f) The percentages in table 5 cover

- type test tolerances;
- manufacturing tolerances;
- measuring tolerances for the simplified test.

TABLE 5 – **Acceptable deviation from type test nominal values**

Compressor shaft input at normal load	100 % capacity		50 % capacity		No-load power
	Capacity	Specific energy requirement	Capacity	Specific energy requirement	
below 10 kW	± 6 %	± 7 %	–	–	± 20 %
10 to 100 kW	± 5 %	± 6 %	± 7 %	± 7 %	± 20 %
above 100 kW	± 4 %	± 5 %	± 5 %	± 6 %	± 20 %

NOTE – The need for a simplified test at other than full capacity is to be the subject of agreement between manufacturer and purchaser.

g) The test report of a simplified test may be short and simple, without tolerance calculation and with only the essential corrections mentioned.

C.2.4 Endurance test

An endurance test is an exceptional test which is only made when one of the parties find such a test is justified.

a) For an endurance test made at the request of a purchaser the extent and the program of the test are matters of agreement between the manufacturer and the purchaser.

b) An endurance test made at the request of the manufacturer can be regarded as an extended type test according to C.1.2. The extent and the program of the test are matters of agreement between the manufacturer and the independent expert. The result of the inspection according to C.1.4 shall be stated in the test report.

ANNEX D

TYPICAL TEST REPORTS

This annex gives seven examples of tests, as listed below and in table 6, together with a list of symbols used in the diagrams.

TABLE 6 – **Specification of test examples**

Test No.	Type of gas	Type of compressor	Number of stages	Cooling	Capacity l/s	Absolute pressure Inlet		Absolute pressure Discharge	
						MPa	bar	MPa	bar
1	air	reciprocating	2	water	354,2	0,1	1	0,8	8
2	air	reciprocating	2	air	68,1	0,1	1	0,8	8
3	air	reciprocating	2	air	68,1	0,1	1	0,8	8
4	air	vane	2	oil injection	283,3	0,1	1	0,8	8
5	air	screw	2	water	675	0,1	1	0,8	8
6	ammonia synthesis gas	reciprocating	3	water	324,2	1,4	14	30,1	301
7	natural gas	reciprocating	2	water	2 778	0,1	1	0,9	9

D.1 TEST EXAMPLE No. 1

Type of gas :	air
Type of compressor :	reciprocating
Number of stages :	2
Cooling :	water
Capacity :	354,2 l/s
Inlet abs. pressure :	0,1 MPa (= 1 bar)
Discharge abs. pressure :	0,8 MPa (= 8 bar)

Certificate of Compressor Test according to International Standard ISO 1217

D.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 (Lloyds, etc.) . . . . Certificate (for pressure vessels) . . . .

Short description of compressor : . . . .

Type of gas : Air

Primemover : Asynchronous slipring Make . . . . . . Type . . . . . . No. . . . .

Transmission : Direct drive Make . . . . . . Type . . . . . . No. . . . .

TABLE 7 — **Specified conditions for the guarantee**

Item	Symbol	Figure	Unit
Absolute inlet pressure	p_{1c}	0,1	MPa
Inlet temperature	t_{1c}	not specified	°C
Inlet humidity	φ_c	none	—
Absolute discharge pressure	p_{2c}	0,8	MPa
Shaft speed	n_c	419	rev/min
Cooling water thermal difference	$(t_{2w} - t_{1w})_c$	21	K
Cooling water flow	q_{wc}	3,1	m^3/h
Water inlet temperature	t_{1wc}	equal to ambient air	°C

TABLE 8 — **Guaranteed performance**

Item	Symbol	Figure	Unit
Capacity	q_c	354,2	l/s
Power requirement	P_c	101	kW
Specific energy consumption	w_c	285,8	J/l

Capacity and specific energy consumption are guaranteed for free suction from the atmosphere.

D.1.2 Methods and equipment used for the test

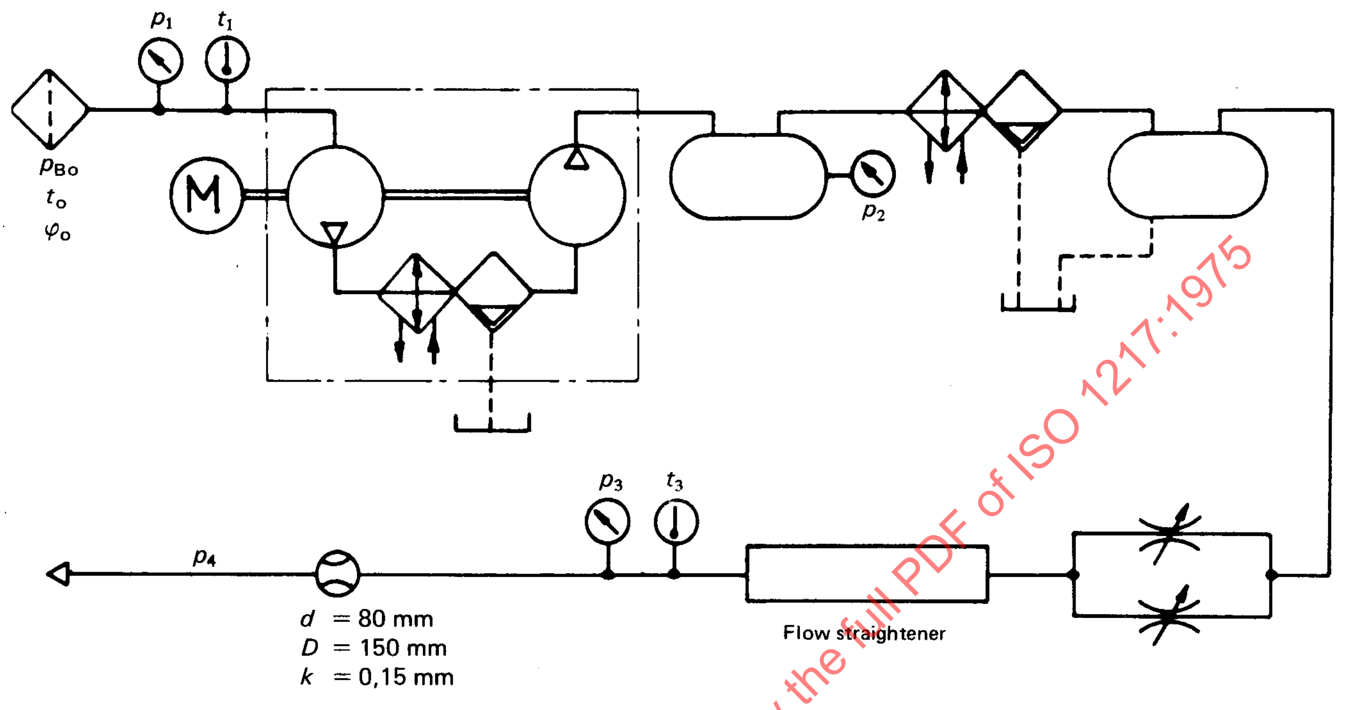


FIGURE 4 – **Test layout**

Flow measurement : With nozzle according to ISO/R 541.

The humidity of the ambient air was measured with a psychrometer of the Assman 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 primemover 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)

D.1.3 TABLE 9 – **Average of test readings**

(The primary readings have already been scrutinized in accordance with 7.2 of ISO 1217.)

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_{2m}	MPa	0,804 ±0,010	0,805 ± 0,010	0,8
7 Barometric pressure	p_{Bm}	MPa	0,105 0	0,105 0	0,105 0
8 Site temperature	t_{om}	°C	18,2	18,1	18,3
9 Inlet temperature	t_{1m}	°C	20,0 ± 1,0	20,3 ± 0,9	–
10 Inlet wet bulb temperature	$t_{o\ wet\ m}$	°C	17,7	17,8	–
11 Shaft speed	n_m	rev/min	418 ± 1	418 ± 1	424
12 Inlet pressure	p_{1m}	MPa	0,104 0	0,104 0	–
13 Water flow, compressor		m³/h	3,1	3,1	–
14 Cooling water inlet temperature	t_{1wm}	°C	9,0	9,0	–
15 Cooling water outlet temperature	t_{2wm}	°C	30,2	30,4	–
16 Absolute pressure before the nozzle	p_{3m}	MPa	0,107 2	0,107 2	–
17 Differential pressure over nozzle	$p_{3m} - p_{4m}$	Pa	2 970 ± 40	2 978 ± 40	–
18 Temperature at the nozzle	t_{3m}	°C	21,1 ± 0,4	21,1 ± 0,4	–
19 Motor input[1)]		kW	109,5 ± 1,2	109,5 ± 1,2	14,5
20 Interstage gauge pressure		MPa	0,185	0,185	0,11
21 Lubricating oil gauge pressure		MPa	0,12	0,12	0,12
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		m³/h	–	–	–
29 Mass flow of condensate in intercooler		kg/h	3,25	3,35	–
30 Mass flow of condensate in aftercooler and air receiver		kg/h	5,80	5,70	–

1) Tolerance from readings only.

The tolerance figures are twice the standard deviations and based on calculated reading errors with due consideration to reading and instrument errors. (For the instrument errors in the two-wattmeter measurement see 10.3.2 to 10.3.5.)

D.1.4 Capacity calculation

The volume rate of flow (q_3) through the nozzle at p_3, T_3 is calculated according to the following formula from ISO/R 541 :

$$q_3 = 1\,000\ \alpha\epsilon\ \frac{\pi}{4}\ d^2 \left[\frac{2\ (p_3 - p_4)}{\rho_3}\right]^{1/2} \quad ; \quad \alpha\epsilon = 0{,}999\ 9$$

$$\rho_3 = \frac{p_3 - p_{3v}}{R_a\ T_3} + \frac{p_{3v}}{R_v\ T_3}$$

$R_a = 287{,}1$ (air) J/(kg·K) $R_v = 461{,}5$ (vapour) J/(kg·K)

Referred 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 capacity correction must be added to q_1 for the volume q_v of the condensed vapour referred to the inlet conditions. The capacity q_m of the compressor is then according to 3.21

$$q_m = q_1 + q_v$$

To enable a first determination to be made of α and ϵ, which factors depend on the Reynold's 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.

$$q_{1\,approx} = 1\,000\ \alpha\epsilon\ \frac{\pi}{4}\ d^2\ \frac{T_1}{p_1} \left[\frac{2\ (p_3 - p_4)\ p_3\ R_a}{T_3}\right]^{1/2} \quad \text{l/s}$$

D.1.5 TABLE 10 — Calculated figures

Item	Symbol	Unit	Calculated figure		
31 Capacity at actual shaft speed	q_m	l/s	356,1 [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 capacity**	q_{corr}	l/s	353,1	353,9	–
36 Shaft input	P_m	kW	102,3	102,3	9,6
37 Specific energy consumption	w	J/l	287,28	286,56	–
38 Correction for inlet pressure, pressure ratio and isentropic exponent	K_4	–	0,977 7	0,977 7	–
39 Correction for coolant temperature	K_5	–	1,013 2	1,013 2	–
40 Correction for humidity	K_6	–	0,998 6	0,998 6	–
41 Correction for shaft speed	K_7	–	1,000 5	1,000 5	–
42 **Corrected specific energy consumption**	w_{corr}	J/l	287,64	286,92	–

1) The calculation is shown in D.1.8.

D.1.6 Tolerance calculations for Test No. 1

In clause 10 of ISO 1217, the general rules for tolerance calculations are given. In this example only the results from Test No. 1 are subject to tolerance calculations.

Tolerance for corrected capacity

Base for the calculation is the formula in D.1.4 for $q_{1\ \mathrm{approx}}$

Tolerances for

α	(according to ISO/R 541)	± 1,02 %
ϵ	(according to ISO/R 541)	± 0,06 %
T_1	1,0/293	± 0,34 %
T_3	0,4/294,1	± 0,14 %
$p_3 - p_4$	40/297 0	± 1,35 %
n	1/418	± 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 10.4, is 20 % of ± 0,011 or ± 0,22 %, which latter figure shall be algebraically added.

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

$$= \pm [1{,}30 + 0{,}22] = \pm 1{,}52$$

Tolerance for specific energy consumption

Tolerance for flow

When calculating the tolerance for specific energy consumption according to 10.3, the part of the tolerance coming from the flow measurement shall not include the tolerance in the shaft speed measurement. Thus the square root for the flow influence shall be :

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

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.

Tolerance for wattmeter reading 1,2/109,5 (see table 9) . . . . 1,10 %

Tolerance τ_{Im} for the two-wattmeter instrument errors

The method for calculating this tolerance is given in 10.3.2 to 10.3.5.

The classes of instrument used were the following :

	Class	Error in percent	Angle error arc minutes
voltage transformer	0,1	$\pm f_u = 0{,}1$	$\vartheta_u = 5$
current transformer	0,1	$\pm f_i = 0{,}1$	$\vartheta_i = 5$
wattmeter	0,2	$\pm f = 0{,}2$	–

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

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

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

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

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 consumption. 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 of 9.4.1.

The tolerances for the correction factors are estimated as 20 % according to 10.4.

Correction for inlet pressure	K_4	$0{,}2 \times 2{,}23$	0,45 %
Correction for coolant temperature	K_5	$0{,}2 \times 1{,}32$	0,26 %
Correction for humidity	K_6	$0{,}2 \times 0{,}14$	0,03 %
Correction for shaft speed	K_7	$0{,}2 \times 0{,}05$	0,01 %

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

$$\tau_w = \pm [(1{,}49^2 + 0{,}64^2 + 1{,}10^2 + 0{,}55^2 \times 1{,}24^2)^{1/2} + (0{,}45^2 + 0{,}26^2 + 0{,}03^2 + 0{,}01^2)^{1/2} + 0{,}35]$$

$$\tau_w = \pm [2{,}07 + 0{,}52 + 0{,}35] = \pm 2{,}94\ \%$$

D.1.7 Test data for compressor type . . . . working at 419 rev/min and an absolute discharge pressure of 0,8 MPa (8 bar). Average of tests 1 and 2.

Capacity :1) 353,5 (± 1,5 %) l/s

Specific energy consumption :1) 287,3 (± 2,9 %) J/l (= 0,079 8 kWh/m³)

Conclusion : The specified performance has been met.

Date : 19 . . . .

. . . .

Test engineer Purchaser's representative

Further participants :

. . . .

. . . .

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 capacity tolerance should have been ± 1,3 instead of ± 1,5 % and the tolerance for specific energy consumption ± 2,1 instead of ± 2,9 %.

D.1.8 Numerical calculation in detail for Test No. 1

Calculation of q_m at test No. 1

a) Basic data : (See D.1.3)

$p_B = 0{,}105$ MPa

$p_o = 0{,}105$ MPa

$p_1 = 1{,}04 \times 10^5$ Pa

$p_2 = 8{,}04 \times 10^5$ Pa

$p_3 = 1{,}072 \times 10^5$ Pa

$p_3 - p_4 = 2\,970$ Pa

$d = 0{,}08$ m; $D = 0{,}15$ m

$R_g = 287{,}1$ J/(kg·K)

$R_v = 461{,}5$ J/(kg·K); $R_g/R_v = 0{,}622$

$t_o = 18{,}2$ °C

$t_{o\ wet} = 17{,}7$ °C

$t_1 = 20{,}0$ °C

$t_3 = 21{,}1$ °C

Mass flow of condensate : $Q_w = 3{,}25 + 5{,}80 = 9{,}05$ kg/h

b) The relative humidity φ_o of the atmosphere and φ_1 in the suction pipe :

Psychrometer difference : $18{,}2 - 17{,}7 = 0{,}5$ K and $t_o = 18{,}2$ °C giving $\varphi_o = 0{,}96$

Pressure of saturated vapour : $t_o = 18{,}2$ °C giving $p_{vo}'' = 2\,110$ Pa

$t_1 = 20{,}0$ °C giving $p_{v1}'' = 2\,338$ Pa

$$x = \frac{R_g}{R_v} \times \frac{\varphi\, p_v''}{p - \varphi\, p_v''}$$

Constant absolute humidity of the aspirated air gives $x_1 = x_o$.

Hence

$$\varphi_1 = \varphi_o \times \frac{p_1}{p_o} \times \frac{p_{vo}''}{p_{v1}''} = 0{,}96 \times \frac{1{,}04 \times 10^5}{1{,}05 \times 10^5} \times \frac{2\,110}{2\,338}$$

which gives : $\underline{\varphi_1 = 0{,}86}$

c) Mass flow of water vapour Q_{v1} sucked in by the compressor :

$$Q_{v1} = \frac{\varphi_1\, p_v''\, q_{v1}}{R_v\, T_1}$$

here $p_v'' = 2\,338$ Pa is the vapour pressure of the water at saturation and at $t_o = 20{,}0$ °C

$\varphi_1 = 0{,}86$, $R_v = 461{,}5$, $T_1 = 293{,}2$ K, $q_{v1} = q_{1\ approx}$

Assuming $\alpha\epsilon = 1{,}0$ (to be verified later) we obtain (from D.1.4)

$$q_{1\,approx} = 1\,000 \times 1{,}0 \times \frac{\pi \times 0{,}08^2}{4} \times \frac{293{,}2}{1{,}04 \times 10^5} \times \left(\frac{2 \times 2\,970 \times 1{,}072 \times 10^5 \times 287{,}1}{294{,}3}\right)^{1/2} = 353{,}2 \text{ l/s}$$

$\varphi\, p_{v1}'' = 0{,}86 \times 2\,338 = 2\,011$ Pa

$$Q_{v1} = \frac{0{,}86 \times 2\,338 \times 353{,}2}{461{,}5 \times 293{,}2} = 5{,}247 \text{ g/s} = 18{,}89 \text{ kg/h}$$

d) Mass flow of water vapour Q_{v3} passing the nozzle :

$$Q_{v3} = Q_{v1} - Q_w \text{ where } Q_w \text{ is the vapour condensed in the plant kg/h}$$

$$Q_{v3} = 18{,}89 - 9{,}05 = 9{,}84 \text{ kg/h}$$

e) Approximate mass flow of dry air $Q_{3\ \text{approx}}$ passing the nozzle :

$$Q_{3\ \text{approx}} = Q_{1\ \text{approx}} = \frac{(p_1 - \varphi_1\, p_{v1}'')}{R_g\, T_1}\, q_{1\ \text{approx}}$$

$$= \frac{(1{,}04 - 0{,}020\ 11) \times 10^5 \times 353{,}2 \times 3{,}6}{287{,}1 \times 293{,}2} = 1\ 541 \text{ kg/h}$$

f) Approximate absolute humidity $x_{3\ \text{approx}}$ of the gas passing the nozzle :

$$x_{3\ \text{approx}} = \frac{Q_{v3}}{Q_{3\ \text{approx}}} = \frac{9{,}84}{1\ 541} = 6{,}39 \times 10^{-3} \text{ kg/kg dry air}$$

g) Approximate partial pressure ($p_{3v\ \text{approx}}$) of the water vapour passing the nozzle :

but $$x_{3\ \text{approx}} = \frac{R_g}{R_v} \times \frac{\varphi_{3\ \text{approx}}\, p_v''}{p_3 - \varphi_{3\ \text{approx}}\, p_v''}$$

After elimination of p_v we obtain

$$p_{3v\ \text{approx}} = \frac{x_{3\ \text{approx}}\, p_3}{\dfrac{R_g}{R_v} + x_{3\ \text{approx}}}$$

Here $x_{3\ \text{approx}} = 6{,}39 \times 10^{-3}$; $p_3 = 1{,}072 \times 10^5$

$$p_{3v\ \text{approx}} = \frac{6{,}39 \times 10^{-3} \times 1{,}072 \times 10^5}{0{,}622 + 0{,}006\ 39} = 1\ 090 \text{ Pa}$$

h) Approximate density $\rho_{3\ \text{approx}}$ of the gas passing the nozzle :

$$\rho_{3\ \text{approx}} = \frac{p_3 - p_{3v}}{R_g\, T_3} + \frac{p_{3v}}{R_v\, T_3} = \frac{(1{,}072 - 0{,}010\ 9)\ 10^5}{287{,}1 \times 294{,}3} + \frac{0{,}010\ 9 \times 10^5}{461{,}5 \times 294{,}3} = 1{,}256 + 0{,}008 = 1{,}264 \text{ kg/m}^3$$

i) The expansion factor ϵ (see ISO/R 541, page 37) :

$$\epsilon = \left\{ (1-x)^{2/\kappa} \left(\frac{\kappa}{\kappa - 1}\right) \left[\frac{1 - (1-x)^{(\kappa - 1)/\kappa}}{x}\right] \left[\frac{1 - \beta^4}{1 - \beta^4\ (1-x)^{2/\kappa}}\right] \right\}^{1/2}$$

Here $x = \dfrac{p_3 - p_4}{p_3} = 0{,}027\ 71$; $1 - x = 0{,}972\ 3$

$$\beta = \frac{d}{D} = \frac{0{,}080}{0{,}150} = 0{,}533\ 33; \quad \beta^4 = 0{,}080\ 91; \quad 1 - \beta^4 = 0{,}919\ 09$$

$$\frac{2}{\kappa} = \frac{2}{1{,}40} = 1{,}429 \qquad \frac{\kappa - 1}{\kappa} = 0{,}285\ 7 \qquad \frac{\kappa}{\kappa - 1} = 3{,}50$$

$(1-x)^{2/\kappa} = 0{,}972\ 3^{1{,}429} = 0{,}960\ 61$

$\beta^4\ (1-x)^{2/\kappa} = 0{,}080\ 91 \times 0{,}960\ 61 = 0{,}077\ 719$

$(1-x)^{(\kappa-1)/\kappa} = 0{,}972\ 3^{0{,}285\ 7} = 0{,}991\ 916$

$$\text{Thus } \epsilon = \left[0{,}960\ 61 \times 3{,}5 \left(\frac{1-0{,}991\ 916}{0{,}027\ 71}\right)\left(\frac{0{,}919\ 09}{1-0{,}077\ 719}\right)\right]^{1/2}$$

$\epsilon = 0{,}989$

k) The total flow coefficient α (see ISO/R 541, pages 35 and 36) :

$\alpha = \alpha'' r_{Re}$ where $\alpha'' = f\ (\beta^4, Re_D)$ according to 7.1.7.1 in ISO/R 541.

Here $\beta = 0{,}533\ 33$; $\beta^4 = 0{,}080\ 91$; $Re_D = \frac{v_3\ D}{\upsilon} = \frac{v_3\ D\ \rho_3}{\mu}$

$D = 0{,}150$ m, $\eta = 17{,}2 \times 10^{-6} \left(\frac{T_3}{273{,}2}\right)^{0{,}76} = 18{,}2 \times 10^{-6}$ Pa·s (according to "Hütte", I, 28th edition)

$$v_3 = \frac{q_{3\ \text{approx}}}{1\ 000\ \pi\ D^2/4};\ q_{3\ \text{approx}}\ \frac{T_3}{T_1} \times \frac{p_1}{p_3} = 353{,}2 \times \frac{294{,}3}{293{,}2} \times \frac{1{,}040}{1{,}072}$$

$$q_{3\ \text{approx}} = 343{,}9\ \text{l/s}$$

$$v_3 = \frac{343{,}9}{1\ 000\ \pi \times 0{,}15^2/4} = 19{,}5\ \text{m/s}$$

$$Re_D = \frac{19{,}5 \times 0{,}15 \times 1{,}264}{18{,}2 \times 10^{-6}} = 2{,}03 \times 10^5$$

According to table 10 in ISO/R 541 : $\alpha'' = 1{,}011$

$r_{Re} = (r_o - 1)\left(\frac{\log Re_D}{5{,}5}\right)^2 + 1$ (from ISO/R 541, 7.1.7.2)

Here $\frac{D}{k} = \frac{150}{0{,}15} = 1\ 000$ and $\beta^2 = 0{,}284\ 4$

giving $r_{Re} = 1{,}00$

and $\alpha = \alpha'' = 1{,}011$

l) The combined coefficient $\alpha\epsilon$

$\alpha\epsilon = 1{,}011 \times 0{,}989 = 0{,}999\ 9$

m) Calculation of q_3 :

$$q_3 = 1\ 000\ \alpha\epsilon \frac{\pi \times 0{,}08^2}{4}\left[\frac{2\ (p_3 - p_4)}{\rho_3}\right]^{1/2} = 1\ 000 \times 0{,}999\ 9 \times 16\ \pi \times 10^{-4}\left(\frac{2 \times 2\ 970}{1{,}264}\right)^{1/2}$$

$q_3 = 344{,}6$ l/s

n) Calculation of q_1

$$q_1 = q_3 \frac{T_1}{T_3} \times \frac{p_3}{p_1} = 343{,}9 \times \frac{293{,}2}{294{,}3} \times \frac{1{,}072}{1{,}040} = 353{,}2 \text{ l/s}$$

o) Calculation of the capacity influence from the condensed water vapour $Q_w = 9{,}05$ kg/h

The size of this influence is calculated on the assumption that the aspired volume flow is independent of the composition of the gas.

$$q_{v1} = \frac{Q_w R_v T_1}{p_1} = \frac{9{,}05 \times 1\,000 \times 461{,}5 \times 293{,}2}{3\,600 \times 1{,}04 \times 10^5} = 3{,}3 \text{ l/s}$$

p) Calculation of q_m :

$$q_m = q_1 + q_{v1} = 353{,}2 + 3{,}3 = 356{,}5 \text{ l/s}$$

q) Corrections for capacity

$$K_1 = \frac{n_c}{n_m} = \frac{419}{418} = 1{,}002\,4 \qquad (9.2.1)$$

$$K_2 = 1{,}000\,0 \qquad (9.2.2)$$

K_3 : In this case an increase in coolant temperature of 10 K will reduce the capacity by 1 % (9.2.3).

$$K_3 = 1 - 0{,}01 \frac{t_{lwc} - t_{lwm}}{10} = 1 - 0{,}001\,(20{,}0 - 9{,}0) = 0{,}989\,0$$

r) Tolerance for corrected capacity

According to ISO/R 541, page 37, the standard deviation of α is

$$\pm 0{,}25\,[1 + 3\,\beta^4 + 100\,(r_{Re} - 1) + (\log_{10} Re_D - 6)^2 + 50/D]\ \%$$

Here, $\beta = \frac{80}{150} = 0{,}533\,3$ $\qquad \beta^4 = 0{,}080\,9$

$r_{Re} = 1{,}0$ as before

$Re_D = 2{,}03 \times 10^5$ as before $\qquad D = 150$ mm which gives

$$\pm 0{,}25\,[1 + (3 \times 0{,}080\,9) + 0 + (5{,}322 - 6)^2 + 0{,}333\,3] = \pm 0{,}51\ \%$$

According to ISO/R 541, page 37, the standard deviation of ϵ is

$$\pm \frac{p_3 - p_4}{p_3} = \frac{0{,}029\,70 \times 10^5}{1{,}072 \times 10^5} = \pm 0{,}027\,7\ \%$$

Thus $\tau_\alpha = 2 \times 0{,}51 = 1{,}02\ \%$ (twice the standard deviation)

$\tau_\epsilon = 2 \times 0{,}03 = 0{,}06\ \%$ $\qquad$ (twice the standard deviation)

s) Corrections for specific energy consumption

$$K_4 = \frac{p_{1c}}{p_{1m}} \times \frac{\log r_c}{\log r_m} \qquad (9.4.1)$$

$$K_4 = \frac{1,000}{1,040} \times \frac{\log 8}{\log (8,04/1,04)} = 0,977\,7$$

$$K_5 = 1 + 0,012 \frac{t_{1wc} - t_{1wm}}{10} \qquad (9.4.2)$$

$$K_5 = 1 + 0,012 \frac{20 - 9}{10} = 1,013\,2$$

$$K_6 = 1 + \frac{R_v}{R_g} \times \frac{i-1}{i} \left[\frac{T_{1wm}}{T_{1m}} \left(x_{1m} - \frac{1}{i-1} \sum_{i=2}^{i} x_{im} \right) - \frac{T_{1wc}}{T_{1c}} \left(x_{1c} - \frac{1}{i-1} \sum_{i=2}^{i} x_{ic} \right) \right] \qquad (9.4.3)$$

$R_v = 461,5$ J/(kg·K) $\qquad R_g = 287,1$ J/(kg·K) $\qquad \frac{R_g}{R_v} = 0,622$

$i = 2$

$T_{1wm} = 282,2$ K $\qquad T_{1m} = 293,2$ K

The following indices are used below :

21 outlet stage 1

22 outlet stage 2

$$x_{1m} = \frac{R_g}{R_v} \frac{\varphi_{21}\, p_{21}''}{p_{21} - \varphi_{21}\, p_{21}''}$$

$\varphi_{21} = 1,00$

$p_{21} = 2,90 \times 10^5$ Pa the total pressure after the first stage.

$p_{21}'' = 117\,0$ Pa obtained from steam tables (saturation pressure at $t_{1wm} = 9,0$ °C)

$$x_{1m} = 0,622 \frac{0,011\,7}{2,90 - 0,011\,7} = 0,002\,52 \text{ kg/kg dry air}$$

$$x_{2m} = \frac{R_g}{R_v} \frac{\varphi_{22}\, p_{22}''}{p_{22} - \varphi_{22}\, p_{22}''}$$

$\varphi_{22} = 1,00$

$p_{22} = 8,04 \times 10^5$ Pa

$p_{22}'' = p_{21}''$ as the air is assumed to be cooled to t_{1wm}

$$x_{2m} = 0{,}622 \frac{0{,}011\ 7}{8{,}04 - 0{,}011\ 7} = 0{,}000\ 91 \text{ kg/kg dry air}$$

According to the guarantee

$$T_{1wc} = T_{1m} = T_{1c} = 293{,}2 \text{ K}$$

$$x_{1c} = \frac{R_g}{R_v} \frac{\varphi_{21c}\, p_{21c}''}{p_{21c} - \varphi_{21c}\, p_{21c}''}$$

$$\varphi_{21c} = \varphi_{21} = 1{,}00$$

$$p_{21c} = p_{21} = 2{,}90 \times 10^5 \text{ Pa}$$

$p_{21c}'' = 233\ 8$ Pa obtained from steam tables (saturation pressure at $t_{1wc} = 20{,}0\ ^\circ\text{C}$)

$$x_{1c} = 0{,}622 \frac{0{,}023\ 38}{2{,}90 - 0{,}023\ 38} = 0{,}005\ 05 \text{ kg/kg dry air}$$

$$x_{2c} = \frac{R_g}{R_v} \frac{\varphi_{22c}\, p_{22c}''}{p_{22c} - \varphi_{22c}\, p_{22c}''}$$

$$\varphi_{22c} = 1{,}00$$

$$p_{22c} = p_{21c} = 8 \times 10^5 \text{ Pa}$$

$$p_{22c}'' = p_{21c}''$$

$$x_{2c} = 0{,}622 \frac{0{,}023\ 38}{8{,}0 - 0{,}023\ 38} = 0{,}001\ 82 \text{ kg/kg dry air}$$

$$K_6 = 1 + \left[\frac{461{,}5}{287{,}1} \times \frac{2 - 1}{2}\right] \left[\frac{282{,}2}{293{,}2} (0{,}002\ 52 - 0{,}000\ 91) - (0{,}005\ 05 - 0{,}001\ 82)\right]$$

$$= 1 + 0{,}803\ 7\ (0{,}962\ 5 \times 0{,}001\ 61 - 0{,}003\ 23)$$

$$= 0{,}997\ 1$$

$$K_7 = 1 + a \left(1 - \frac{n_m}{n_c}\right) \qquad (9.4.4)$$

where $a = 2{,}7 \times 10^{-6} w_m{}^2 = 2{,}7 \times 10^{-6} \left(\frac{p_m}{q_m}\right)^2 = 2{,}7 \times 10^{-6} \left(\frac{102{,}3 \times 10^3}{356{,}5}\right)^2 = 0{,}223$

$$K_7 = 1 + 0{,}223 \left(1 - \frac{418}{419}\right) = 1{,}000\ 5$$

D.2 TEST EXAMPLE No. 2

Type of gas :	air
Type of compressor :	reciprocating
Number of stages :	2
Cooling :	air
Capacity :	68,6 l/s
Inlet abs. pressure :	0,1 MPa (= 1 bar)
Discharge abs. pressure :	0,8 MPa (= 8 bar)

Certificate of Compressor Test[1] according to International Standard ISO 1217

D.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.) . . . .

Short description of compressor . . . .

Medium to be compressed : Air

Primemover : Squirrel cage, asynchronous motor Make . . . . Type . . . . No. . . . .

Transmission : Direct drive Make . . . . Type . . . . No. . . . .

TABLE 11 — **Specified conditions for the guarantee**

Item	Symbol	Figure	Unit
Absolute inlet pressure	p_{1c}	0,1	MPa
Inlet temperature	t_{1c}	not specified	°C
Inlet humidity	x_c	none	—
Absolute discharge pressure	p_{2c}	0,8	MPa
Shaft speed	n_c	1 460	rev/min

TABLE 12 — **Guaranteed performance**

Item	Symbol	Figure	Unit
Capacity	q_c	68,6	l/s
Power requirements	P_c	25,0	kW
Specific energy consumption	w_c	367,2	J/l

Capacity and specific energy consumption are guaranteed for free suction from the atmosphere including intake filter.

1) This test serves as a Type Test supervised by an independent expert. A corresponding Simplified Test in the workshop is found as Test No. 3.

D.2.2 Methods and equipment used for the test

The test layout is illustrated below.

Flow measurement : With nozzle according to ISO/R 541.

The humidity of the ambient air was measured with a psychrometer of the Assman type. No water condensed during the test.

Instruments : . . . .

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

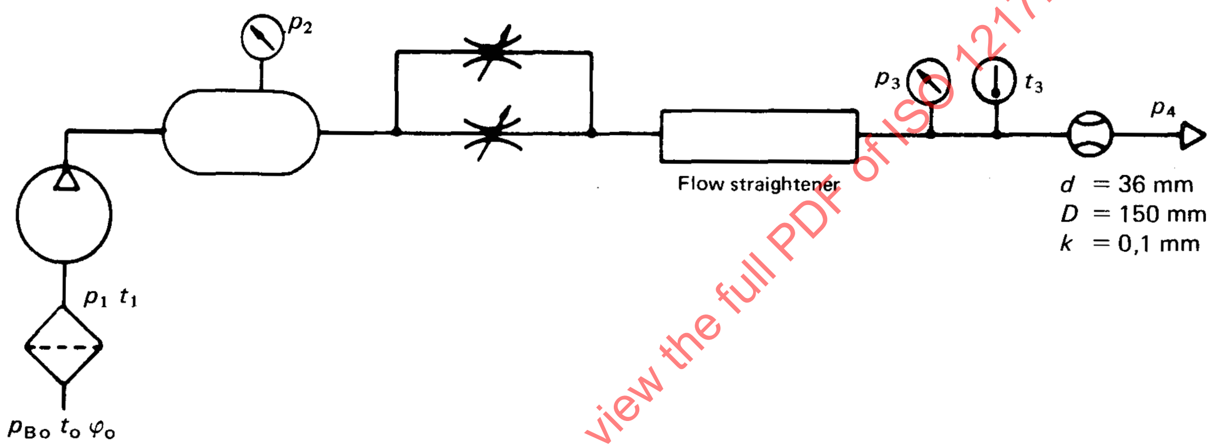


FIGURE 5 – **Test layout**

Motor efficiency

(determined from test) : Full compressor load 0,900 ± 0,007 (twice standard deviation)

No compressor load 0,68

D.2.3 TABLE 13 – **Average of test readings**

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_{2m}	MPa	0,798 ± 0,010	0,803 ± 0,010	0,8
7 Barometric pressure	$p_{Bm} = p_{1m}$	MPa	0,101 0	0,101 0	0,101 0
8 Site temperature	$t_{om} \approx t_{1m}$	°C	22,5 ± 1,1	22,6 ± 1,2	22,3
9 Inlet wet bulb temperature	$t_{1\,wetm}$	°C	19,1	19,2	–
10 Shaft speed	n_m	rev/min	1 465 ± 4	1 465 ± 4	1 475
11 Pressure before nozzle	p_{3m}	MPa	0,103 4	0,103 4	–
12 Differential pressure over the nozzle	$p_{3m} - p_{4m}$	Pa	3 890 ± 25	3 925 ± 20	–
13 Temperature at the nozzle	t_{3m}	°C	148,4 ± 1,5	150,3 ± 1,5	–
14 Motor power input		kW	28,56 ± 0,4[1)]	28,32 ± 0,4[1)]	5,15
15 Interstage gauge pressure		MPa	0,22	0,22	0,11
16 Lubricating oil gauge pressure		MPa	0,17	0,17	0,17
17 Oil sump temperature		°C	67	68	63
18 Air temperature after 2nd stage		°C	158	160	–

1) Tolerance only from readings.

The tolerance figures are twice the standard deviations and based on calculated reading errors with due consideration to reading and instrument errors. (For the instrument errors in the two-wattmeter measurement, see 10.3.2 to 10.3.3.)

Capacity calculation : see Test No. 1.

D.2.4 TABLE 14 – **Calculated figures**

Item	Symbol	Unit	Calculated figure		
19 Capacity at actual shaft speed	q_m	l/s	68,75	68,79	–
20 Correction for shaft speed	K_1	–	0,996 6	0,996 6	–
21 Correction for polytropic exponent and pressure ratio (does not apply)	K_2	–	–	–	–
22 Correction for coolant temperature (does not apply)	K_3	–	–	–	–
23 **Corrected capacity**	q_{corr}	l/s	68,72	68,56	–
24 Shaft input	P_m	kW	25,70	25,49	3,50
25 Specific energy consumption	w_m	J/l	372,6	370,4	–
26 Correction for inlet pressure, pressure ratio, isentropic exponent	K_4	–	0,990 1	0,990 1	–
27 Correction for coolant temperature (does not apply)	K_5	–	–	–	–
28 Correction for humidity (does not apply)	K_6	–	–	–	–
29 Correction for shaft speed (negligible)	K_7	–	–	–	–
30 **Corrected specific energy consumption**	w_{corr}	J/l	368,9	366,7	–

D.2.5 Tolerance calculations for Test No. 2

(Full test : see Test No. 1)

Tolerance for corrected capacity

Tolerances for

α	(according to ISO/R 541)	± 1,94 %
ϵ	(according to ISO/R 541)	± 0,08 %
T_3	1,5/421	± 0,36 %
$p_3 - p_4$	25/389 0	± 0,64 %
T_1	1,1/295,5	± 0,37 %
n	4/146 5	± 0,27 %

$$\tau_q = [1{,}94^2 + 0{,}08^2 + 0{,}37^2 + 0{,}5^2 \ (0{,}36^2 + 0{,}64^2) + 0{,}27^2]^{1/2} \ \%$$

$$\tau_q = \pm 2{,}03 \ \%$$

Tolerance for specific energy consumption

Tolerance for flow . . . . ± 2,03 %

Tolerance for motor efficiency ± 0,007/0,90 . . . . ± 0,78 %

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

Tolerance for wattmeter reading ± 0,40/28,56 (See table 13) . . . . ± 1,40 %

Tolerance τ_m for the two-wattmeter instrument errors

The classes of instrument used were the following :

	Class	Error in percent	Angle error, arc minutes
voltage transformer	0,2	$f_u = \pm 0{,}2$	$\vartheta_u = 10$
current transformer	0,2	$f_i = \pm 0{,}2$	$\vartheta_i = 10$
wattmeter	0,5	$f = \pm 0{,}5$	–

the power factor $\cos \varphi = 0{,}87$

$\text{tg}\, \varphi = 0{,}57$

The instruments were recently calibrated and therefore the square root formula should be used.

$$\tau_{lm} = \pm \left\{ 0{,}2^2 + 0{,}2^2 + \left(\frac{2 \times 0{,}5}{1{,}50}\right)^2 + [0{,}57 \times 2{,}9 \times 10^{-2}\,(10+10)]^2 \right\}^{1/2}$$

$= \pm\,(0{,}040 + 0{,}040 + 0{,}444 + 0{,}110)^{1/2} = \pm\,0{,}80$ . . . . ± 0,80 %

The advantage of having one stage better class for the transformers than for the wattmeter should be recognized. If the class had been 0,5 also for the transformers, the tolerance 0,8 % would have gone up to ± 1,3 %.

The tolerance for discharge pressure is ± 0,010/0,798 . . . . ± 1,25 %

Earlier tests on the same type of compressor have shown that the specific energy consumption increases with the discharge pressure raised to 0,50 (see page 30).

$$\tau_w = \pm\,(2{,}0^2 + 1{,}4^2 + 0{,}8^2 + 0{,}50^2 \times 1{,}25^2 + 0{,}78^2)^{1/2} = \pm\,2{,}76\,\%$$ [1)]

D.2.6 Type test data for compressor type . . . . working at 1 460 rev/min and an absolute discharge pressure of 0,8 MPa (8 bar).

Capacity : 68,6 (± 2,0 %) l/s

Specific energy consumption : 367,8 (± 2,8 %) J/l (= 0,102 kWh/m^3)

Independent expert :

. . . .

Name

. . . .

Address

Manufacturer's representative :

. . . .

Name

. . . .

Address

Date : 19 . . - . . - . .

1) If the tolerance for the motor efficiency had been a catalogue figure and not a measured value, the tolerance should have been algebraically added. The tolerance for specific energy consumption should then have been

$$\tau_w = \pm\,[(2{,}0^2 + 1{,}4^2 + 0{,}8^2 + 0{,}5^2 \times 1{,}25^2)^{1/2} + 0{,}78] = \pm\,3{,}42\,\%$$

which should be regarded as too high a figure for the tolerance of a Type Test.

D.3 TEST EXAMPLE No. 3

Type of gas :	air
Type of compressor :	reciprocating
Number of stages :	2
Cooling :	air
Capacity :	68,6 l/s
Inlet abs. pressure :	0,1 MPa (= 1 bar)
Discharge abs. pressure :	0,8 MPa (= 8 bar)

Certificate of Compressor Simplified Test according to Annex C of International Standard ISO 1217

D.3.1 Basic data

Place of test : The workshop of the Manufacturer . . . . . . 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.) . . . .

Short description of compressor . . . .

Medium to be compressed : Air

Primemover : Squirrel cage asynchronous motor Make . . . .Type . . . . No. . . . .

Transmission : Direct drive Make . . . .Type . . . . No. . . . .

The compressor has been Type-Tested according to annex C of ISO 1217.

Result of Type Test : See Test Example No. 2

Capacity :	68,6 (± 2,0 %) l/s
Specific energy consumption :	367,2 (± 2,8 %) J/l (0,102 kWh/m^3)

D.3.2 Methods and equipment used for the test

The manufacturer's normal test stand instruments were used.

The airflow was measured with a nozzle according to ISO/R 541.

The electrical input was measured by the two-wattmeter method.

After the compressor had reached steady state conditions one reading was taken at full load.

D.3.3 TABLE 15 – **Test readings**

Item	Symbol	Unit	Reading
1 Discharge absolute pressure	p_2	MPa	0,800
2 Barometric pressure	$p_B \equiv p_1$	MPa	0,103
3 Site temperature	$t_0 \approx t_1$	°C	21,0
4 Shaft speed	n	rev/min	1 470
5 Pressure before nozzle (d = 36 mm) Flow coefficient $\alpha\epsilon$ = 0,985		MPa	0,103 2
6 Differential pressure over the nozzle	$p_3 - p_4$	Pa	3 860
7 Temperature before the nozzle	t_3	°C	142
8 Power consumption of motor (Motor efficiency 0,90)	P_m	kW	28,7
9 Inter-stage gauge pressure		MPa	0,22
10 Oil gauge pressure		MPa	0,18

D.3.4 TABLE 16 – **Result of actual test**

Item	Symbol	Unit	Calculated figures
11 Capacity at actual shaft speed	q_m	l/s	68,6
12 **Capacity at 1 460 rev/min**	q_{corr}	l/s	68,1
13 Specific energy consumption	w_m	J/l	374,4
14 **Corrected specific energy consumption**	w_{corr}	J/l	363,2

NOTE – Corrected capacity

$$K_1 = \frac{1\,460}{1\,470} = 0{,}993 \text{ (shaft speed)}$$

$$q_{corr} = 0{,}993 \times 68{,}6 = 68{,}1 \text{ l/s}$$

Corrected specific energy consumption

$$K_4 = \frac{1{,}00}{1{,}03} = 0{,}970 \text{ (inlet pressure)}$$

$$w_{corr} = 0{,}97 \times 374{,}4 = 363{,}2 \text{ J/l} \quad (= 0{,}101 \text{ kWh/m}^3)$$

Conclusion : The compressor meets the specifications of the Type Test within the tolerances given in annex C to ISO 1217.

. . . . Test engineer

. . . . Customer's representative

Date : 19 . . - . . - . .

D.4 TEST EXAMPLE No. 4

Type of gas :	air
Type of compressor :	vane, portable
Number of stages :	2
Cooling :	oil injected
Capacity :	283,3 l/s
Inlet abs. pressure :	0,1 MPa (= 1 bar)
Discharge abs. pressure :	0,8 MPa (= 8 bar)

Certificate of Compressor Test according to International Standard ISO 1217

D.4.1 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 (Lloyds, etc.) . . . . Certificate (for pressure vessels) . . . .
Short description of compressor . . . .
Medium to be compressed : Air . . . .
Primemover : Diesel engine . . . . Make . . . . . Type . . . . . No. . . . .
Transmission : Direct drive . . . . Make . . . . . Type . . . . . No. . . . .

TABLE 17 — **Specified conditions for the guarantee**

Item	Symbol	Figure	Unit
Absolute inlet pressure	$p_B \equiv p_{1c}$	0,1	MPa
Inlet temperature	t_{1c}	–	°C
Inlet humidity	x_{1c}	none	–
Absolute discharge pressure	p_{2c}	0,8	MPa
Shaft speed	n_c	1 800	rev/min

Capacity and specific fuel consumption are guaranteed according to ISO 1217.

TABLE 18 — **Guaranteed performance**

Item	Symbol	Figure	Unit
Capacity	q_c	283,3	l/s
Specific fuel consumption	F_c	0,028	g/l
Maximum discharge temperature above ambient	Δt_{2c}	61	K

D.4.2 Methods and equipment used for the test

The test layout is illustrated below.

Flow measurement : With nozzle according to ISO/R 541.

The humidity was measured at a distance from the compressor with a psychrometer of the Assman type. No water condensed during the test.

Instruments : . . . .

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

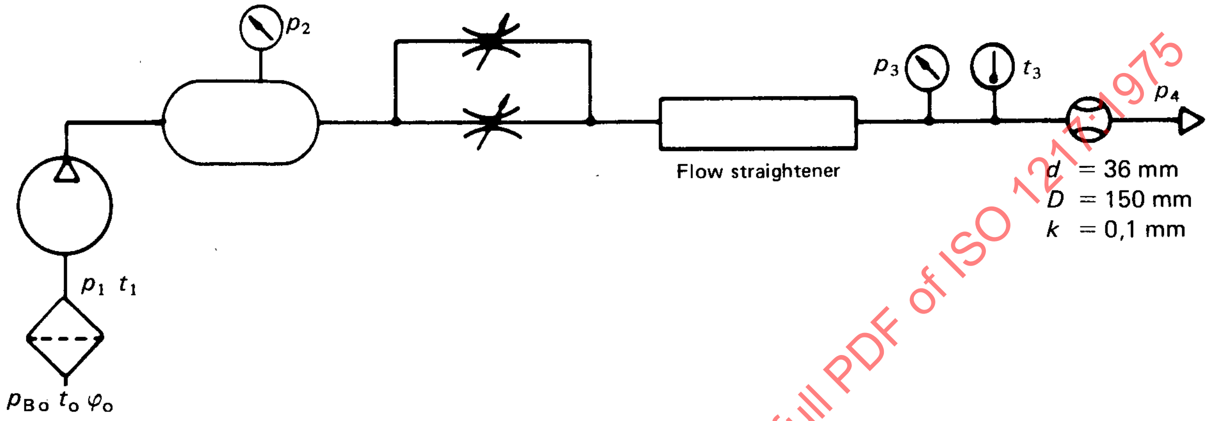


FIGURE 6 – **Test layout**

D.4.3 TABLE 19 – **Average of test readings**

Item	Symbol	Unit	Test figure		
1 Date	–	–	–	–	–
2 Test number	–	–	1	2	3
3 Number of readings used	–	–	6	6	1
4 Duration of test	–	min	35	35	15
5 Compressor load	%	–	100	100	0
6 Absolute discharge pressure	p_{2m}	MPa	0,796 ± 0,015	0,802 ± 0,015	0,75 – 0,80
7 Discharge temperature	t_{2m}	°C	79	80	–
8 Barometric pressure	$p_{Bm} \equiv p_{1m}$	MPa	0,104	0,104	0,104
9 Site temperature	$t_{om} \approx t_{1m}$	°C	22,0 ± 2,5	22,6 ± 2,4	22,1
10 Inlet wet bulb temperature	$t_{1\ wetm}$	°C	19,0	19,2	–
11 Shaft speed	n_m	rev/min	1 811 ± 18	1 808 ± 20	1 100
12 Pressure before nozzle	p_{3m}	MPa	0,108	0,108	–
13 Differential pressure over the nozzle	$p_{3m} - p_{4m}$	Pa	6 320 ± 50	6 340 ± 60	–
14 Temperature at the nozzle	t_{3m}	°C	63,3 ± 2,0	63,8 ± 1,8	–
15 Fuel consumption	–	kg/h	30,7 ± 0,1	30,5 ± 0,1	11,0
16 Inter-stage gauge pressure	–	MPa	0,21	0,21	–
17 Lubricating oil gauge pressure	–	MPa	0,77	0,76	–
18 Oil gauge pressure, engine	–	MPa	0,31	0,30	0,20
19 Water inlet temperature to engine radiator	–	°C	83	84	75
20 Engine oil sump temperature	–	°C	81	83	73

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

Capacity calculations : see Test example No. 1.

D.4.4 TABLE 20 – Calculated figures

Item	Symbol	Unit	Calculated figure	
21 Capacity at actual shaft speed	q_m	l/s	287,0	288,4
22 Correction for shaft speed	K_1	–	0,994	0,996
23 Correction for pressure ratio and polytropic exponent (does not apply)	K_2	–	–	–
24 Correction for coolant temperature (does not apply)	K_3	–	–	–
25 **Corrected capacity at 1 800 rev/min**	q_{corr}	l/s	**285,2**	**287,2**
26 Specific fuel consumption	F_m	g/l	0,029 7	0,029 5
27 Correction for inlet pressure, pressure ratio and isentropic exponent	K_4	–	0,982 4	0,978 8
28 Correction for coolant temperature (does not apply)	K_5	–	–	–
29 Correction for humidity (does not apply)	K_6	–	–	–
30 Correction for shaft speed (does not apply)	K_7	–	–	–
31 **Corrected specific fuel consumption**	F_{corr}	g/l	0,029 2	0,028 9

NOTE – Correction K_4 is based on 9.4.1 b) of ISO 1217.

D.4.5 Tolerance calculations for Test No. 4

(Full text : see Test example No. 1)

Tolerance for corrected capacity

Tolerances for

α	(according to ISO/R 541)	± 1,10 %
ϵ	(according to ISO/R 541)	± 0,12 %
T_3	2,0/336,3	± 0,59 %
$p_3 - p_4$	50/6 320	± 0,79 %
T_1	2,3/295	± 0,78 %
n	18/1 811	± 1,00 %

$$\tau_{q\,corr} = \pm [1,10^2 + 0,12^2 + 0,78^2 + 0,5^2\ (0,79^2 + 0,59^2) + 1,00^2]^{1/2}$$

$$\tau_{q\,corr} = \pm\ 1,75\ \%$$

Tolerance for specific fuel consumption

Tolerance for flow

According to 10.2 the tolerance for the flow measurement should be taken into consideration. Thus the flow tolerance influence will be

$\pm [1{,}10^2 + 0{,}12^2 + 0{,}78^2 + 0{,}5^2\ (0{,}79^2 + 0{,}59^2)]^{1/2} = \pm 1{,}44\ \%$ . . . . $\pm 1{,}44\ \%$

Tolerance in fuel consumption 0,1/30,7 . . . . $\pm 0{,}33\ \%$

Tolerance for discharge pressure is 0,015/0,079 6 . . . . $\pm 1{,}88\ \%$

Separate tests on the same type of compressor have shown that the specific fuel consumption increases with the discharge pressure raised to 0,60 (see page 30).

$\tau_w = \pm [1{,}44^2 + 0{,}33^2 + (0{,}60^2 \times 1{,}88^2)]^{1/2} = \pm 1{,}86\ \%$

D.4.6 Test data for compressor type . . . . at 1 800 rev/min and 0,8 MPa abs. discharge pressure.

Capacity : 286,2 (± 1,8 %) l/s

Specific fuel consumption : 0,029 2 (± 1,9 %) g/l

Conclusion : The capacity guarantee has been met but not the specific fuel consumption guarantee.

Date : 19 . . - . . - . .

. . . . Test engineer

. . . . Purchaser's representative

Further participants :

. . . .

. . . .

D.5 TEST EXAMPLE No. 5

Type of gas :	air
Type of compressor :	screw
Number of stages :	2
Cooling :	water
Capacity :	676,7 l/s
Inlet abs. pressure :	0,1 MPa (= 1 bar)
Discharge abs. pressure :	0,8 MPa (= 8 bar)

Certificate of Compressor Test according to International Standard ISO 1217

D.5.1 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 . . . . Certificate . . . .
Short description of compressor . . . .
Medium to be compressed : Air . . . .
Primemover : squirrel cage asynchronous motor . . . . . Make . . . . . Type . . . . . No. . . . .
Transmission : Drive via twin gear . . . . Make . . . . . Type . . . . . No. . . . .

TABLE 21 — **Specified conditions for the guarantee**

Item	Symbol	Figure	Unit
Absolute inlet pressure	p_{1c}	0,1	MPa
Temperature at standard inlet point	t_{1c}	20	°C
Relative inlet humidity	φ_c	60	%
Absolute discharge pressure	p_{2c}	0,8	MPa
Shaft speed	n_c	2 960	rev/min
Cooling water inlet temperature	t_{1wc}	15	°C
Cooling water consumption	q_{wc}	3,8	m³/h

TABLE 22 — **Guaranteed performance**

Item	Symbol	Figure	Unit
Capacity	q_c	675	l/s
Power requirement including gear losses	P_c	242	kW
Specific energy consumption	w_c	358,6	J/l

Capacity and specific energy consumption are guaranteed according to ISO 1217.

D.5.2 Methods and equipment used for the test

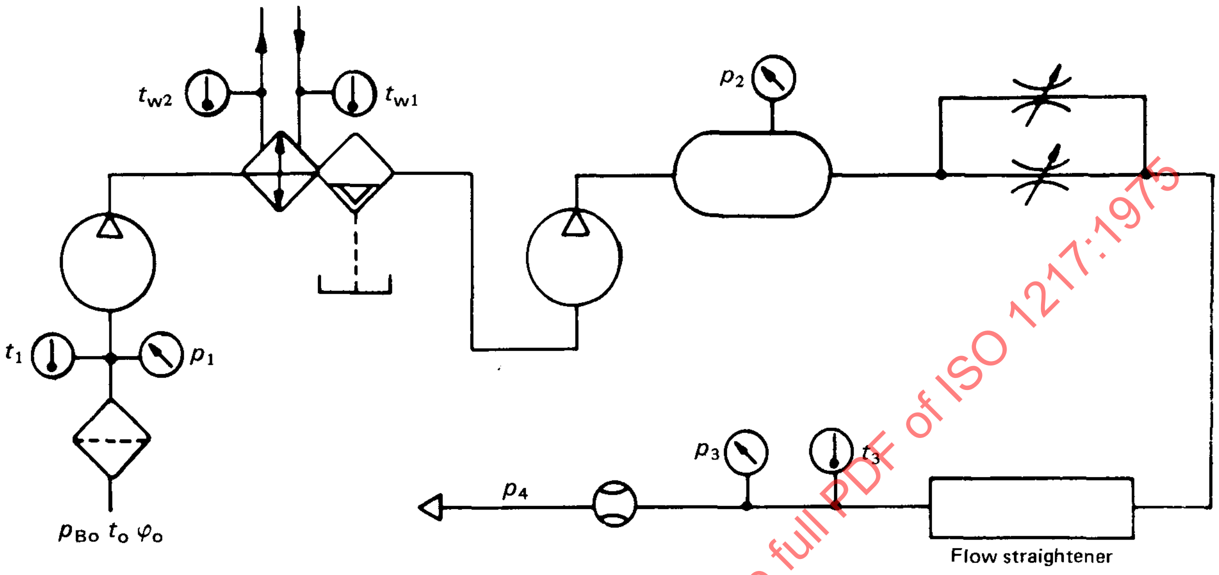


FIGURE 7 – **Test layout**

Flow measurement : At full pressure after the pulsation damper and with orifice plate according to ISO/R 541.

diameter of orifice plate	$d =$ 96,63 mm
diameter of pipeline	$D =$ 150,0 mm
flow coefficient	$\alpha =$ 0,67

The humidity of the ambient air was measured with a psychrometer of the Assman type. The condensate collected during the test was measured in the intercooler.

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

Motor efficiency

(losses checked by tests) : Compressor full load 0,940 ± 0,004 (twice standard deviation)

Instruments : . . . .

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

D.5.3 TABLE 23 – **Average of test readings**

Item	Symbol	Unit	Reading
1 Date		–	3.2.1964
2 Test number		–	1
3 Number of readings used		–	8
4 Duration of test		min	25
5 Compressor load		%	100
6 Absolute discharge pressure	p_{2m}	MPa	0,800 ± 0,008
7 Barometric pressure	p_B	MPa	0,102
8 Site temperature	t_{om}	°C	15,0
9 Inlet temperature	t_{1m}	°C	15,8 ± 0,6
10 Absolute humidity at inlet	x_{om}	kg/kg	0,007 6
11 Shaft speed, motor	n_m	rev/min	2 930 ± 6
12 Inlet pressure	p_{1m}	MPa	0,101
13 Water consumption		m³/h	3,68
14 Cooling water inlet temperature	t_{w1m}	°C	8,0
15 Cooling water outlet temperature	t_{w2m}	°C	41
16 Absolute pressure before orifice	p_{3m}	MPa	0,802 4
17 Orifice plate differential pressure	$p_{3m} - p_{4m}$	Pa	2 070 ± 15
18 Temperature at the orifice	t_{3m}	°C	136,0 ± 0,5
19 Motor input[1]	P_m	kW	260 ± 2,6
20 Inter-stage gauge pressure		MPa	0,211
21 Air temperature after 1st stage		°C	145,5
22 Air temperature after intercooler		°C	24
23 Air temperature after 2nd stage		°C	144
24 Condensate collected in intercooler		kg/h	4,8

1) Tolerance only from wattmeter readings.

The tolerance figures are twice the standard deviations and based on calculated reading errors with due consideration to reading and instrument errors. (For the instrument errors in the two-wattmeter measurement see 10.3.2 to 10.3.13.)

Capacity calculation : see Test No. 1.

D.5.4 TABLE 24 — **Calculated figures**

Item	Symbol	Unit	Calculated figure
25 Expansion factor	ϵ	–	0,999
26 Capacity at actual shaft speed	q_m	l/s	671,4
27 Correction for shaft speed	K_1	–	1,010 1
28 Correction for polytropic exponent and pressure ratio (does not apply)	K_2	–	–
29 Correction for coolant temperature[1)]	K_3	–	0,997 6
30 **Corrected capacity**	q_{corr}	l/s	676,7
31 Shaft input, including gear losses	P_m	kW	244,4
32 Specific energy consumption	w_m	J/l	364,1
33 Correction for inlet pressure, pressure ratio and isentropic exponent	K_4	–	0,994 8
34 Correction for coolant temperature[1)]	K_5	–	1,003 1
35 Correction for humidity	K_6	–	0,997
36 Correction for shaft speed[2)]	K_7	–	–
37 **Corrected specific energy consumption**	w_{corr}	J/l	362,2

NOTES

1 From tests with this type of screw compressor it has been found that an increase of 1 K in cooling water inlet temperature at constant air inlet temperature gives a decrease in capacity of 0,086 % and an increase in specific energy consumption of 0,11 %.

2 The compressor operated close to the optimum speed where the effeciency is nearly constant for small variations in speed.

D.5.5 *Tolerance calculations*

In clause 10 general rules for tolerance calculations are given.

Tolerance for corrected capacity

Base for the calculation is the formula for $q_{1\ approx}$ in D.1.4.

Tolerances for

α	(according to ISO/R 541)	± 0,90 %
ϵ	(according to ISO/R 541)	± 0,01 %
T_3	0,5/409	± 0,12 %
$p_3 - p_4$	15/2 070	± 0,72 %
T_1	0,6/288,8	± 0,21 %
n	6/2 930	± 0,20 %

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

$$\tau_{q\,corr} = \pm\ [0{,}90^2 + 0{,}01^2 + 0{,}21^2 + 0{,}5^2\ (0{,}12^2 + 0{,}72^2) + 0{,}20^2]^{1/2}$$

$$\tau_{q\,corr} = \pm\ 1{,}01\ \%$$

Tolerances for specific energy consumption

Tolerance for flow

When calculating the tolerance for specific energy consumption according to 10.3 the part of the tolerance coming from the flow measurement should not include the tolerance in the shaft speed measurement. Thus the square root for the flow influence should be

$\pm\ [0{,}90^2 + 0{,}01^2 + 0{,}21^2 + 0{,}5^2\ (0{,}12^2 + 0{,}72^2)]^{1/2} = \pm\ 0{,}99\ \%$ . . . . ± 0,99 %

Tolerance for motor efficiency ± (0,4/94) . . . . ± 0,43 %

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

Tolerance for wattmeter reading ± (2,6/260) . . . . ± 1,00 %

Tolerance for two-wattmeter instrument error

The same class of instruments were used as in Test No. 1.

The tolerance will then be as calculated in 10.3.3 . . . . ± 0,35 %

The instruments had recently been calibrated, so the tolerance should stand under the square root.

Tolerance for inlet pressure in neglected due to the exactness of the mercury barometer.

Tolerance for pressure ratio is ± 0,008/0,800 . . . . ± 1,00 %

Earlier tests have shown that the specific energy consumption increases with the discharge pressure raised to 0,53.

The tolerances for the correction factors are estimated to

Correction for inlet pressure, etc.	K_4	. . . .	± 0,01 %
Correction for coolant temperature	K_5	(0,2 × 0,31)	± 0,06 %
Correction for humidity	K_6	(0,2 × 0,30)	± 0,06 %
Correction for shaft speed	K_7	. . . .	± 0,00 %

$$\tau_w = \pm \left\{ [0{,}99^2 + 0{,}43^2 + 1{,}00^2 + 0{,}35^2 + (0{,}53^2 \times 1{,}00^2)]^{1/2} + (0{,}01^2 + 0{,}06^2 + 0{,}06^2)^{1/2} \right\}$$

$$\tau_w = \pm\ [1{,}60 + 0{,}09] = \pm\ 1{,}69\ \%$$

D.5.6 Test data for compressor type . . . . working at 2 960 rev/min and an absolute discharge pressure of 0,8 MPa (8 bar).

Capacity : 676,7 (± 1,1 %) l/s

Specific energy consumption : 362,2 (± 1,7 %) J/l (= 0,100 6 kWh/m^3)

Conclusion : The specified performance has been met.

Date : 19 . . - . . - . .

. . . .
Test engineer

. . . .
Purchaser's representative

Further participants :

. . . .

. . . .

TEST EXAMPLE No. 6

Type of gas :	ammonia synthesis gas
Type of compressor :	reciprocating
Number of stages :	3
Cooling :	water
Capacity :	324,2 l/s
Inlet abs. pressure :	1,4 MPa (= 14 bar)
Discharge abs. pressure :	30,1 MPa (= 301 bar)

Certificate of Compressor Test according to International Standard ISO 1217

D.6.1 Basic data

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

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

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

Purchaser . . . .

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

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

Classification . . . . Certificate . . . .

Short description of compressor . . . .

Medium to be compressed : ammonia synthesis gas

Gas analysis :

Percent by volume :

	Section 1 to 2 1st and 2nd stage	Section 3 to 7 3rd stage
H_2	70,0	74,3
N_2	23,4	24,7
CO	6,0	0,4
CO_2	0,3	0,3
Ar	0,3	0,3
	100,0	100,0

Primemover : Synchronous motor   Make . . . . . Type . . . . No. . . . .

Transmission : Direct drive   Make . . . . . Type . . . . No. . . . .

TABLE 25 – **Specified conditions for the guarantee**

Item	Symbol	Figure	Unit
Absolute inlet pressure	p_{1c}	1,4	MPa
Inlet temperature	t_{1c}	15	°C
Relative inlet humidity	φ_{1c}	100	%
Absolute discharge pressure	p_{4c}	30,1	MPa
Shaft speed	n_c	5,000	rev/s
Cooling water thermal difference	$t_{w2} - t_{w1}$	15	K
Cooling water consumption	q_w	120	m^3/h
Water inlet temperature	t_{w1c}	$(t_1 + 10)$	°C

TABLE 26 – **Guaranteed performance**

Item	Symbol	Figure	Unit
Capacity (dry gas at inlet conditions)	q_c	324,2	l/s
Power requirement	P_c	2 150	kW
Specific energy consumption	w_c	6 624	J/l

Capacity and specific energy consumption are for this non-standard compressor guaranteed with an additional tolerance of ± 2 % to the normal test tolerances.

D.6.2 Methods and equipment used for the test

Flow measurement : With nozzle according to ISO/R 541. Overcritical expansion in the valves before and after the nozzle reduces pulsations at the measuring pipe.

Instruments : . . . .

Calibration of instruments : see appendix (not included)

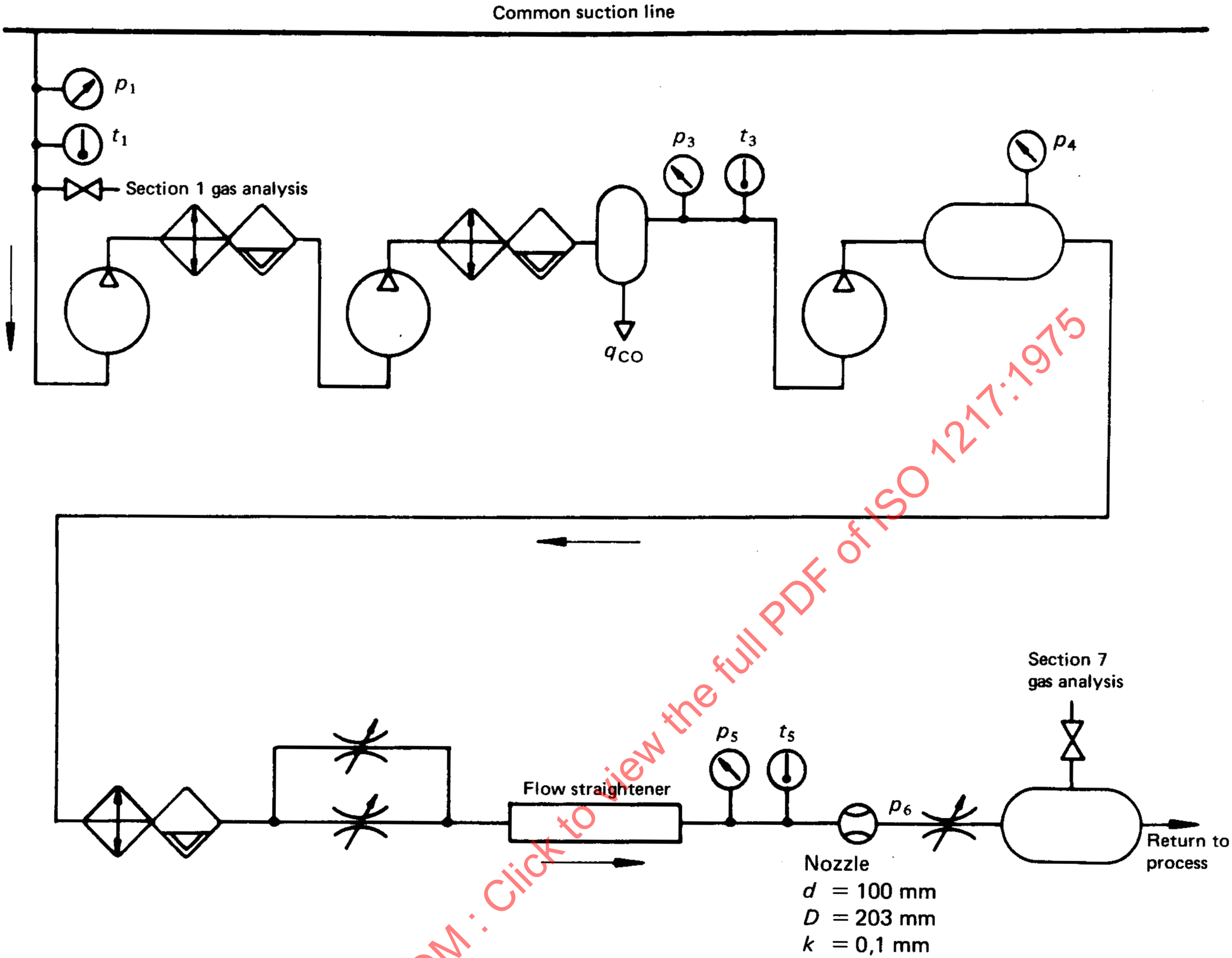


FIGURE 8 – **Test layout**

The condensate collected during the test was measured in the intercoolers, the aftercooler and the gas receivers. The aftercooler was not subject to test.

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

Motor efficiency
(determined by separate test) : Compressor full load 0,945 ± 0,006.

D.6.3 TABLE 27 – Average of test readings

Item		Symbol	Unit	Test figure	See page
1 Date					
2 Test number				1	
3 Number of readings				12	
4 Duration of test			min	180	
5 Compressor load			%	100	
6 Site temperature		t_o	°C	18,2	
7 Barometric pressure		p_B	MPa	0,105 0	
8 Section 1					
Gas analysis by volume	H_2		%	69,6	
	N_2		%	23,5	
	CO		%	6,4 ± 0,3	
	CO_2		%	0,3	
	Ar		%	0,2	
9 Section 7					
Gas analysis by volume	H_2		%	74,2	
	N_2		%	25,1	
	CO		%	0,2	
	CO_2		%	0,3	
	Ar		%	0,2	
Gas constant		R_5	J/(kg·K)	945	62
10 Inlet temperature	1st stage	t_1	°C	17,5 ± 1,4	
11 Absolute inlet pressure	1st stage	p_1	MPa	1,365	
12 Absolute discharge pressure	2nd stage	p_2	MPa	12,2 ± 0,1	
13 Absolute inlet pressure	3rd stage	p_3	MPa	12,1 ± 0,05	
14 Inlet temperature	3rd stage	t_3	°C	30	
15 Discharge gauge pressure	3rd stage	p_4	MPa	31,0	
16 Absolute pressure before nozzle		p_5	MPa	2,275	
17 Differential pressure over the nozzle		$p_5 - p_6$	Pa	2 657 ± 34	
18 Temperature before the nozzle		t_5	°C	52 ± 0,6	
19 Shaft speed		n_m	rev/s	4,967	
20 Power consumption		P	kW	2 162	
21 Water consumption		q_w	m^3/h	107,3	
22 Cooling water inlet temperature		t_{w1}	°C	25,5	
23 Cooling water outlet temperature		t_{w2}	°C	44,5	
24 Lubricating oil gauge pressure			MPa	0,12	
25 Oil sump temperature			°C	47	
26 Gas temperature after 1st stage			°C	149	
27 Gas temperature after 1st stage intercooler			°C	31,3	
28 Gas temperature after 2nd stage			°C	138	
29 Gas temperature after 2nd stage intercooler			°C	30,5	
30 Gas temperature after 3rd stage			°C	128	
31 Condensate in intercoolers			kg/h	13,6	

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

Capacity calculation :

From the flow rate q_2 (at the conditions of the standard inlet point) through the first and second stage, carbon monoxide will be extracted. The flow rate q_5 through the third stage and the nozzle (at the conditions of the standard inlet point) is calculated according to the following formula (see ISO/R 541) :

$$q_5 = 10^3\ \alpha\ \epsilon\ d^2\ \frac{\pi}{4}\left[\frac{2\ (p_5 - p_6)}{\rho_5}\right]^{1/2}\ \frac{\rho_5\ T_1}{p_1\ T_5}\ \text{l/s}$$

R_5 is calculated from the gas analysis at section 7.

$$\rho_5 = \frac{p_5}{R_5\ T_5}$$

The capacity q_{1m} of the compressor is according to 3.21.

$$q_{1m} = q_5 + q_{CO} + q_v$$

The volume q_v of vapour at the inlet point is here neglected as the guaranteed value q_1 refers to the flow rate of dry gas and the inlet humidity at the test is equal to the inlet humidity at guarantee conditions.

According to the gas analysis : (see table 27)

stage I and II : $q_{CO\ II} = q_2 \times 0{,}064$

stage III : $q_{CO\ III} = q_5 \times 0{,}002$

$q_{CO} = q_{CO\ II} - q_{CO\ III} = (q_2 \times 0{,}064) - (q_5 \times 0{,}002)$

Further

$$q_{CO} = q_2 - q_5$$

giving

$$\frac{q_2}{q_5} = \frac{1 - 0{,}002}{1 - 0{,}064} = 1 + \frac{0{,}064 - 0{,}002}{1 - 0{,}064} = 1 + \frac{0{,}062}{0{,}936} = 1{,}066\ 2$$

Finally

$$q_{1m} = 10^3\ \alpha\ \epsilon\ d^2\ \frac{\pi}{4}\left[\frac{2\ (p_5 - p_6)\ R_5\ p_5}{T_5}\right]^{1/2} \times \frac{T_1}{p_1} \times \frac{q_2}{q_5}\ \text{l/s}$$

For results see item 33 below.

To enable a first determination of α and ϵ, which factors depend on the Reynold's number, an estimate of the gas flow must be made. For this purpose the last formula above is used. The same formula is further used for the error calculation.

D.6.4 TABLE 28 – **Calculated figures**

Item	Symbol	Unit	Complete unit	1st + 2nd stage	3rd stage	See page
32 Guaranteed capacity	q_{1c}	l/s	324,2	324,2	305,6	
33 Capacity at actual test condition	q_{1m}	l/s	334,7	334,7	313,9	64
34 Correction for shaft speed	K_1	–	1,007	1,007	1,007	64
35 Correction for polytropic exponent and pressure ratio	K_2	–	1,000	1,000	1,000	64
36 Correction for coolant temperature	K_3	–	0,999	0,999	0,999	64
37 Corrected measured capacity	q_{corr}	l/s	336,7	336,7	315,8	
38 Shaft input	P_m	kW	2 162	1 535	627	
39 Specific energy consumption	w_m	J/l	6 473	4 586	1 998	
40 Correction for inlet pressure, pressure ratio, isentropic exponent	K_4	–		1,018	0,958	
41 Correction for coolant temperature	K_5	–		1,000	1,000	
42 Correction for humidity	K_6	–		1,000	1,000	
43 Correction for shaft speed	K_7	–		1,000	1,000	
44 Correction for CO	K_8	–		1,000	1,000	
45 Corrected specific energy consumption	w_{corr}	J/l		4 669	1 912	
46 Corrected power consumption	P_{corr}	kW	2 164	1 572	604	
47 Corrected specific energy consumption	w_{corr}	J/l	6 462			

D.6.5 Tolerance calculation

The general rules for tolerance calculation are given in 10.1.1 to 10.1.6.

Tolerance for corrected capacity

Base for the calculation is the formula above for q_{1m}.

Tolerances for

α	(according to ISO/R 541, 6.4.2)	± 0,686 %
ϵ	(according to ISO/R 541, 6.4.2)	± 0,005 %
T_5	± 0,6/325	± 0,185 %
$p_5 - p_6$	± 34/2 652	± 1,282 %
T_1	± 1,4/290,5	± 0,482 %
CO	± 0,064 × 0,05	± 0,320 %

The errors in n, d, D, p_1, p_5 and R_5 are negligible.

The tolerance caused by the correction method is estimated to be 20 % of the correction.

$$\text{i.e.} \quad \pm\ 0,2\ \frac{1\ 212 - 1\ 205}{1\ 205} = \pm\ 0,12\ \%$$

Manufacturing tolerance . . . . ± 2,00 %

$$\tau_{q\ \mathrm{corr}} = \pm \left\{ [0{,}686^2 + 0{,}005^2 + 0{,}5^2\ (1{,}282^2 + 0{,}185^2) + 0{,}482^2]^{1/2} + 0{,}320 + 0{,}12 + 2{,}00 \right\}$$

$$= \pm\ (1{,}50 + 2{,}00) = \pm\ 3{,}50\ \%$$

Tolerance for specific energy consumption

Tolerance for motor efficiency (known from a separate test) . . . . ± 0,64 %

Tolerance for wattmeter reading . . . . ± 0,30 %

The principles for calculating of tolerance τ_{lm} for the two-wattmeter instrument errors are given in 10.3.2 to 10.3.5.

The class of instruments used were the following :

	Class	Angle error, arc minutes
voltage transformer	$f_u = \pm 0{,}1\ \%$	$\vartheta_u = 5$
current transformer	$f_i = \pm 0{,}1\ \%$	$\vartheta_i = 5$
wattmeters	$f_w = \pm 0{,}2\ \%$	–

The power factor $\cos \varphi \approx 1$, $\mathrm{tg}\ \varphi \approx 0$, $\alpha_1 + \alpha_2 = 1{,}2$

$$\tau_{\mathrm{lm}} = \pm \left[0{,}1^2 + 0{,}1^2 + \left(\frac{2 \times 0{,}2}{1{,}2} \right)^2 \right]^{1/2} = \pm\ 0{,}36\ \%$$

An additional tolerance caused by the correction methods is estimated to be 20 % of the correction.

$$\pm\ 0{,}2 \times \frac{2\ 179 - 2\ 162}{2\ 162} = \pm\ 0{,}16\ \%$$

Manufacturing tolerance . . . . ± 2,0 %

$$\tau_P = \pm\ [(0{,}64^2 + 0{,}30^2 + 0{,}36^2)^{1/2} + 0{,}16 + 2{,}00] = \pm\ (0{,}95 + 2{,}00) = \pm\ 2{,}95\ \%$$

The tolerance on the specific energy consumption

$$\tau_w = \pm\ [(1{,}50^2 + 0{,}64^2 + 0{,}30^2 + 0{,}36^2)^{1/2} + 0{,}16 + 2{,}00] = \pm\ (1{,}86 + 2{,}00) = \pm\ 3{,}86\ \%$$

D.6.6 Test data for compressor type . . . . working at 298 rev/min and an absolute discharge pressure of 31,1 MPa (311 bar) and an absolute inlet pressure of 1,365 MPa (13,65 bar).

Capacity : 336,7 (± 3,5 %) l/s

Power requirement 2 164 (± 3,0 %) kW

Specific energy consumption : 6 462 (± 3,9 %) J/l (= 1,795 kWh/m^3)

Conclusion : The specified performance has been met.

Date : 19 . . - . . - .

. . . .
Test engineer

. . . .
Purchaser's representative

Further participants :

. . . .

. . . .

D.6.7 Detailed calculations

TABLE 29 — **Gas constant** R_5 (see page 58)

Gas component	Molecular mass M_i	Volume fraction r_i	$M_i r_i$
H_2	2,016	0,742	1,496
N_2	28,016	0,251	7,032
CO	28,011	0,002	0,056
CO_2	44,011	0,003	0,132
Ar	39,944	0,002	0,080
	Sum :	1,000	8,796

Equivalent molecular mass $M_F = \Sigma M_i r_i = 8{,}796$ kg/kmol

The universal gas constant $R = 8\,314$ J/(kmol·K)

Fictive (equivalent) gas constant $R_7 = R_5 = \dfrac{8\,314}{8{,}796} = 945$ J/(kg·K)

Reynolds number, αε

According to Zipperer[1)]

$$\nu_{20} = \frac{\nu_m \times 100}{O_2 + CO + CH_4 + N_2 + 2\,(CO_2 + C_nH_m) + H_2/7}$$

where O_2, CO etc. is the volumetric composition in percent. With $\nu_m = 15{,}56 \times 10^{-6}\ m^2/s$

Substituting oxygen with argon we obtain

$$\nu_{20} = \frac{15{,}56 \times 10^{-6} \times 100}{0{,}2 + 0{,}2 + 0 + 25{,}1 + (2 \times 0{,}3) + 74{,}2/7} = \frac{100}{36{,}7} \times 15{,}56 \times 10^{-6}$$

$$\nu_{20} = 42{,}4 \times 10^{-6}\ m^2/s$$

In section 5

$$\nu_5 = 1{,}83 \times 10^{-6}\ m^2/s$$

Assume $q_5 = 305{,}6$ l/s which gives an actual flow in the measuring pipe of $q = 0{,}19\ m^3/s$

Pipe diameter $D = 203$ mm

Thus $A = 324 \times 10^{-4}\ m^2$

$$V_5 = \frac{q}{A} = \frac{0{,}19}{324} \times 10^4 = 5{,}86\ m/s$$

$$Re = \frac{V_5\, D}{\nu} = \frac{5{,}86 \times 0{,}203}{1{,}83} \times 10^{-6} = 6{,}5 \times 10^5$$

Nozzle diameter $d = 100$ mm

$$\frac{d}{D} = \frac{100}{203} = \beta = 0{,}492\,6 \qquad \beta^2 = 0{,}242\,7 \qquad \beta^4 = 0{,}058\,9$$

According to ISO/R 541 table 10, $\beta^4 = 0{,}058\,9$ and $Re = 6{,}5 \times 10^5$

gives $\alpha'' = 1{,}006$

$$\alpha = \alpha''\, r_{Re}$$

Here $Re > 3{,}2 \times 10^5$ giving $r_{Re} = r_o$

According to ISO/R 541 table 11, $D/k = \frac{203}{0{,}1}$ and $\beta^2 = 0{,}242\,7$ gives $r_{Re} = 1{,}000$

Thus $\alpha = 1{,}006 \times 1{,}000 = 1{,}006$

1) Zipperer L & Müller G : Beitrag zur Bestimmung und Berechnung der Zähigkeit von Gasgemischen. G.W.J Bd 75 (1932) p. 623.

According to ISO/R 541, 7.1.7.3, $k = 1,4$; $\beta^4 = 0,058\,9$ and $x = \dfrac{0,026\,3}{22,75} = 1,156 \times 10^{-3}$

gives $\epsilon = 0,996$

Calculation of q_{1m} (see page 59)

$$q_{1m} = 10^3 \times 1,006 \times 0,996 \times 0,1^2 \frac{\pi}{4}\left(\frac{2 \times 2\,652 \times 945 \times 22,75 \times 10^5}{325,2}\right)^{1/2} \times \frac{290,7}{13,65 \times 10^5} \times \left(1 + \frac{0,064 - 0,002}{1 - 0,064}\right)$$

$$q_{1m} = 334,7 \text{ l/s}$$

D.6.8 Flow corrections

According to 9.2.1 to 9.2.3

K_1 : $K_1 = \dfrac{n_c}{n_m} = \dfrac{5,000}{4,967} = 1,007$

K_2 : $K_2 = 1,000$

K_3 : $t_{w1c} = t_1 + 10,0 = 17,5 + 10,0 = 27,5\ °C$

$t_{w1m} = 25,5\ °C$

$t_{w1c} - t_{w1m} = 2,0\ °C$

According to prototype tests : 10 K lower cooling water temperature gives, at constant gas inlet temperature, a 0,5 % higher capacity.

$K_3 = 0,999$

$K_1\ K_2\ K_3 = 1,006$

$q_{corr} = 1,006 \times 334,7 = 336,7$ l/s

D.6.9 Specific energy consumption corrections

To enable the calculation of K_4 to be made, the power needed for the different stages has to be calculated. The total power is divided over the different stages in proportion to the isentropic power requirement.

Pressure ratios r_{I}, r_{II} and r_{III}

$$r_I = r_{II} = \left(\frac{p_2}{p_1}\right)^{1/2} = \left(\frac{12,2}{1,365}\right)^{1/2} = 2,99$$

$$r_{III} = \frac{p_4}{p_3} = \frac{31,1}{12,1} = 2,57$$

Power

$$P_{is}\ \text{I + II} = 2\, q_2\, p_1 \frac{\kappa}{\kappa - 1} \left(r_{\text{I}}^{(\kappa - 1)/\kappa} - 1\right) = 1\ 177 \text{ kW}$$

$$P_{is}\ \text{III} = q_5\, p_1 \frac{\kappa}{\kappa - 1} \left(r_{\text{III}}^{(\kappa - 1)/\kappa} - 1\right) = 481 \text{ kW}$$

$$P_{is}\ \text{tot} = 1\ 658 \text{ kW}$$

These theoretical isentropic power figures are used to apportion the measured shaft input (2 162 kW) over the two stage groups as follows :

$$\frac{1\ 177}{1\ 658} \times 2\ 162 = 1\ 535$$

$$\frac{481}{1\ 658} \times 2\ 162 = 627$$

For the first two stages K_4 shall be calculated according to 9.4.1 b) with $p_{2c} = 12{,}3$ MPa.

$$\text{Stage I + II :} \quad \frac{1{,}4}{1{,}365} \times \frac{\log \dfrac{12{,}3}{1{,}4}}{\log \dfrac{12{,}2}{1{,}365}} = \frac{1{,}4}{1{,}365} \times \frac{\log 8{,}786}{\log 8{,}938} = 1{,}018$$

For the third stage K_4 shall be calculated according to 9.4.1 a) with $p_{3c} = 12{,}2$ MPa.

$$\text{Stage III :} \quad \frac{12{,}2}{12{,}1} \times \frac{\left(\dfrac{30{,}1}{12{,}2}\right)^{0{,}286} - 1}{\left(\dfrac{31{,}1}{12{,}1}\right)^{0{,}286} - 1} = \frac{12{,}2}{12{,}1} \times \frac{2{,}467^{0{,}286} - 1}{2{,}570^{0{,}286} - 1} = 0{,}959$$

and the final calculation will be

$$w_{m\ \text{I + II}} = \frac{1\ 535 \times 10^3}{336{,}7} = 4\ 559 \text{ J/l}$$

$$w_{m\ \text{III}} = \frac{627 \times 10^3}{315{,}8} = 1\ 985 \text{ J/l}$$

$$w_{corr\ \text{I + II}} = 1{,}018 \times 4\ 559 = 4\ 641 \text{ J/l}$$

$$w_{corr\ \text{III}} = 0{,}958 \times 1\ 985 = 1\ 902 \text{ J/l}$$

$$P_{corr\ \text{I + II}} = 4\ 641 \times 336{,}7 \times 10^{-3} = 1\ 563 \text{ kW}$$

$$P_{corr\ \text{III}} = 1\ 902 \times 315{,}8 \times 10^{-3} = 601 \text{ kW}$$

$$P_{corr} = 2\ 164 \text{ kW}$$

The following correction factors are neglected :

K_5 Correction factor for cooling water temperature

K_6 Correction factor for humidity

K_7 Correction factor for speed

K_8 Correction factor for carbon monoxide

D.7 TEST EXAMPLE No. 7

Type of gas : natural gas

Type of compressor : reciprocating, integral with gas engine

Number of stages : 2

Cooling : water

Capacity : 2 778 l/s

Inlet abs. pressure : 0,1 MPa (= 1 bar)

Discharge abs. pressure : 0,9 MPa (= 9 bar)

Certificate of Compressor Test according to International Standard ISO 1217

D.7.1 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 (Lloyds etc.) . . . . Certificate (for pressure vessels) . . . .

Short description of compressor . . . .

Medium to be compressed : Gas (for analysis : see "Fuel")

Primemover : Four stroke gas engine Make . . . . . . Type . . . . . . No. . . . . . .

Transmission : Direct drive Make . . . . . . Type . . . . . . No. . . . . . .

Fuel : Natural gas with the following gas analysis (by volume) :

CH_4 : 0,889 5 N_2 : 0,041 0 C_2H_6 : 0,029 2

CO_2 : 0,028 1 C_3H_8 : 0,006 9 C_4H_{10} : 0,003 4

C_5H_{12} : 0,001 6 C_6H_{14} : 0,000 3

TABLE 30 — **Specified conditions for the guarantee**

Item	Symbol	Unit	Figure	
Compressor loads	%	–	100	50
Absolute inlet pressure	p_{1c}	MPa	0,1	0,1
Inlet temperature	t_{1c}	°C	–	–
Inlet humidity	φ_c	–	none	none
Absolute discharge pressure	p_{2c}	MPa	0,8	0,8
Shaft speed	n_c	rev/min	300	150
Absolute pressure after 1st stage		MPa	0,4	0,4
Absolute pressure after 2nd stage		MPa	0,9	0,9
Maximum temperature after 1st stage		°C	180	180
Maximum temperature after 2nd stage		°C	180	180