

Australian Standard™

**Industrial fans—Performance testing
using standardized airways**



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Australian Chamber of Commerce and Industry
Australian Industry Group
Australian Institute of Refrigeration Air Conditioning and Heating
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Australian Standard™

Industrial fans—Performance testing using standardized airways

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PREFACE

This Standard was prepared by the Standards Australia Committee ME-013, Industrial Fans, to supersede AS 2936—1987, *SAA Fan Test Code*. It is identical to, and reproduced from ISO 5801:1997, *Industrial fans—Performance testing using standardized airways*.

The terms ‘normative’ and ‘informative’ have been used in this Standard to define the application of the appendix to which they apply. A ‘normative’ appendix is an integral part of a Standard, whereas an ‘informative’ appendix is only for information and guidance.

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References to International Standards should be replaced by references to the following Australian Standards:

<i>Reference to International Standard or other Publication</i>		<i>Australian/New Zealand Standard</i>	
ISO		AS	
5167-1	Measurement of fluid flow by means of pressure differential devices—Part 1: Orifice plates, nozzles and Venturi tubes inserted in circular cross-section conduits running full	2360.1.1	Measurement of fluid flow in closed conduits—Pressure differential methods
5168	Measurement of fluid flow—Evaluation of uncertainties	3778.2.4	Measurement of water flow in open channels— General—Estimation of uncertainty of a flow rate measurement
IEC			
34-2	Rotating electrical machines—Part 2: Methods for determining losses and efficiency of rotating electrical machinery from tests (excluding machines for traction vehicles)	1359.102.1	Rotating electrical machines—General requirements—Methods for determining losses and efficiency—General

CONTENTS

	<i>Page</i>
1 Scope.....	1
2 Normative references.....	1
3 Definitions	2
4 Symbols and units	8
4.1 Symbols.....	8
4.2 Subscripts.....	12
5 General	12
6 Instrumentation for pressure measurement	13
6.1 Barometers.....	13
6.2 Manometers	13
6.3 Damping of manometers.....	14
6.4 Checking of manometers	14
6.5 Position of manometers	14
7 Determination of average pressure in an airway.....	14
7.1 Methods of measurement.....	14
7.2 Use of wall tapings	14
7.3 Construction of tapings	15
7.4 Position and connections.....	15
7.5 Checks for compliance	15
7.6 Use of Pitot-static tube.....	16
8 Measurement of temperature.....	16
8.1 Thermometers.....	16
8.2 Thermometer location	16
8.3 Humidity	17

	<i>Page</i>
9 Measurement of rotational speed	17
9.1 Fan shaft speed	17
9.2 Examples of acceptable methods	17
10 Determination of power input.....	18
10.1 Measurement accuracy.....	18
10.2 Fan shaft power	18
10.3 Determination of fan shaft power by electrical measurement.....	18
10.4 Impeller power	19
10.5 Transmission systems.....	19
11 Measurement of dimensions and determination of areas.....	19
11.1 Flow measurement devices	19
11.2 Tolerance on dimensions	19
11.3 Determination of cross-sectional area.....	19
12 Determination of air density, humid gas constant and viscosity .	20
12.1 Density of the air in the test enclosure, gas constant for humid air and average density in a section x	20
12.2 Determination of vapour pressure.....	20
12.3 Determination of air viscosity.....	21
13 Determination of flowrate.....	23
13.1 General	23
13.2 In-line flowmeters (standard primary devices)	23
13.3 Traverse methods	24
14 Calculation of test results	25
14.1 General	25
14.2 Units	25
14.3 Temperature.....	26
14.4 Mach number and reference conditions	27
14.5 Fan pressure.....	31
14.6 Calculation of stagnation pressure at a reference section of the fan from gauge pressure p_{ex} measured at a section x of the test duct	32
14.7 Inlet volume flowrate	34
14.8 Fan air power and efficiency	34
14.9 Simplified calculation methods.....	39
15 Rules for conversion of test results.....	43
15.1 Laws on fan similarity.....	44
15.2 Conversion rules.....	44
16 Fan characteristic curves	49
16.1 General	49

	<i>Page</i>
16.2 Methods of plotting.....	49
16.3 Characteristic curves at constant speed.....	49
16.4 Characteristic curves at inherent speed.....	49
16.5 Characteristic curves for adjustable-duty fan.....	49
16.6 Complete fan characteristic curve	50
16.7 Test for a specified duty.....	50
17 Uncertainty analysis	51
17.1 Principle.....	51
17.2 Pre-test and post-test analysis.....	52
17.3 Analysis procedure.....	52
17.4 Propagation of uncertainties	53
17.5 Reporting uncertainties	53
17.6 Maximum allowable uncertainties measurement.....	53
17.7 Maximum allowable uncertainty of results	54
18 Selection of test method.....	54
18.1 Classification	54
18.2 Installation types	54
18.3 Test report.....	56
18.4 User installations.....	56
18.5 Alternative methods.....	56
18.6 Duct simulation	56
19 Installation of fan and test airways.....	56
19.1 Inlets and outlets.....	56
19.2 Airways.....	56
19.3 Test enclosure.....	57
19.4 Matching fan and airway	57
19.5 Outlet area	57
20 Carrying out the test.....	57
20.1 Working fluid	57
20.2 Rotational speed.....	57
20.3 Steady operation	57
20.4 Ambient conditions	57
20.5 Pressure readings	58
20.6 Tests for a specified duty.....	58
20.7 Tests for a fan characteristic curve.....	58
20.8 Operating range	58
21 Determination of flowrate	58
21.1 ISO Venturi nozzle	58

	<i>Page</i>
21.2 Multiple nozzle or Venturi nozzle	58
21.3 Quadrant inlet nozzle	58
21.4 Conical inlet	58
21.5 Orifice plate	58
21.6 Pitot-static tube traverse	59
22 Determination of flowrate using ISO Venturi nozzle	59
22.1 Geometric form	59
22.2 Venturi nozzle in free-inlet condition.....	59
22.3 Nozzle performance	59
22.4 Uncertainties	64
23 Determination of flowrate using multiple nozzles or Venturi nozzle.....	65
23.1 Installation	65
23.2 Geometric form	65
23.3 Inlet zone	66
23.4 Multiple-nozzle and Venturi-nozzle characteristics	66
23.5 Uncertainty	68
24 Determination of flowrate using a quadrant inlet nozzle	68
24.1 Installation	68
24.2 Geometric form	69
24.3 Unobstructed space in front of inlet nozzle.....	69
24.4 Quadrant inlet nozzle performance	70
24.5 Uncertainty	70
25 Determination of flowrate using a conical inlet	70
25.1 Geometric form	70
25.2 Screen loading	70
25.3 Inlet zone	71
25.4 Conical inlet performance.....	71
25.5 Uncertainties	72
26 Determination of flowrate using an orifice plate	73
26.1 Installation	73
26.2 Orifice plate	73
26.3 Ducts	77
26.4 Pressure tapplings.....	77
26.5 Calculation of mass flowrate	77
26.6 Reynolds number	78
26.7 In-duct orifice with D and $D/2$ taps.....	78
26.8 In-duct orifice with corner taps.....	80
26.9 Outlet orifice with wall tapplings.....	80

	<i>Page</i>
26.10 Inlet orifice with corner taps	83
26.11 Inlet orifice with wall tapings.....	86
27 Determination of flowrate using a Pitot-static tube traverse	86
27.1 General.....	86
27.2 Pitot-static tube	86
27.3 Limits of air velocity	91
27.4 Location of measurement points	91
27.5 Determination of flowrate	92
27.6 Flowrate coefficient	92
27.7 Uncertainty of measurement	93
28 Installation categories and setups	93
28.1 Type A: free inlet and free outlet	93
28.2 Type B: free inlet and ducted outlet.....	93
28.3 Type C: ducted inlet and free outlet.....	93
28.4 Type D: ducted inlet and ducted outlet	94
28.5 Test installation type	94
29 Component parts of standardized airways	94
29.1 Symbols.....	94
29.2 Component parts	94
29.3 Flowrate measurement devices.....	97
30 Common airway segments for ducted fan installation.....	98
30.1 Common segments	98
30.2 Common segment at fan outlet.....	98
30.3 Common segment at fan inlet	101
30.4 Outlet duct simulation	103
30.5 Inlet duct simulation	103
30.6 Loss allowances for standardized airways.....	104
31 Standardized test chambers	107
31.1 Test chamber	107
31.2 Variable supply and exhaust systems	111
31.3 Standardized inlet test chambers.....	111
31.4 Standardized outlet test chambers	114
32 Standard methods with test chambers — type A installations ...	115
32.1 Types of fan setup	115
32.2 Inlet-side test chambers.....	115
32.3 Outlet-side test chambers.....	130
33 Standard methods with outlet-side test ducts — type B installations.....	138
33.1 Types of fan setup	138

33.2	Outlet-side test ducts with antiscirl — device	138
33.3	Outlet chamber test ducts without antiscirl — device....	153
34	Standard methods with inlet-side test ducts or chambers — Type C installations.....	161
34.1	Types of fan setup.....	161
34.2	Inlet-side test ducts.....	162
34.3	Inlet-side test chambers.....	175
35	Standard methods with inlet- and outlet-side test ducts — type D installations	189
35.1	Types of fan setup.....	189
35.2	Installation type B with outlet antiscirl device and inlet duct or inlet-duct simulation	193
35.3	Installation type B without outlet antiscirl device nor common segment and with inlet duct or inlet-duct simulation	201
35.4	Installation type C with outlet-duct common segment and antiscirl device and common inlet duct.....	204
35.5	Installation type C with outlet-duct simulation without antiscirl device.....	210
Annexes		
A	Fan pressure and fan installation types	220
B	Fan-powered roof exhaust ventilators.....	224
C	Direct calculations of p_{sqn} and p_{ri} at section n of the fan — Installation types B, C and D	226
D	Fan outlet elbow in case of a non-horizontal discharge axis	229
E	Bibliography	232

Introduction

This International Standard is the result of almost thirty years of discussion, comparative testing and detailed analyses by leading specialists from the fan industry and research organizations throughout the world.

It was demonstrated many years ago that the codes for fan performance testing established in different countries do not always lead to the same results.

The need for an International Standard has been evident for some time and ISO/TC 117 started its work in 1963. Important progress has been achieved over the years and although the International Standard itself was not yet published, the subsequent revisions of various national standards led to much better agreement among them.

It has now become possible to complete this International Standard by agreement on certain essential points. It must be borne in mind that the test equipment, especially for large fans, is very expensive and it was necessary to include in the present International Standard many set-ups from various national codes in order to authorize their future use. This explains the sheer volume of this document.

Essential features of the present standard are as follows:

a) Types of installation

Since the connection of a duct to a fan outlet and/or inlet modifies its performance, it has been agreed that four standard installation types should be recognized.

These are:

- Type A: free inlet and outlet;
- Type B: free inlet and ducted outlet;
- Type C: ducted inlet and free outlet;
- Type D: ducted inlet and outlet.

A fan adaptable to more than one installation type will have more than one standardized performance characteristic. The user should select the installation type closest to his application.

b) Common parts

The differences obtained by testing the same fan according to various test codes depend chiefly on the flow pattern at the fan outlet and, while often minor, can be of substantial significance. There is general agreement that it is essential that all standardized test airways to be used with fans have

portions in common adjacent to the fan inlet and/or outlet sufficient to ensure consistent determination of fan pressure.

Geometric variations of these common segments are strictly limited.

However, conventional agreement has been achieved for some particular situations:

1) For centrifugal or cross-flow fans without outlet swirling flow, it is possible to use a simplified outlet duct as described in 30.2 f) without straightener when discharging to the atmosphere or to a measuring chamber.

2) For large fans (outlet diameter exceeding 800 mm) it may be difficult to carry out the tests with standardized common airways at the outlet including a straightener. In this case, by mutual agreement between the parties concerned, the fan performance may be measured using the set-up described in 30.2 f) with a duct of length $2D$ on the outlet side. Results obtained in this way may differ to some extent from those obtained using the normal type D installation, especially if the fan produces a large swirl. Establishment of a possible value of differences, is still a subject of research.

c) Calculations

Fan pressure is defined as the difference between the stagnation pressure at the outlet of the fan and the stagnation pressure at the inlet of the fan. The compressibility of air must be taken into account when high accuracy is required. However, simplified methods may be used when the reference Mach number does not exceed 0,15.

A method for calculating the stagnation pressure and the fluid or static pressure in a reference section of the fan, which stemmed from the work of the ad hoc group of Subcommittee 1 of ISO/TC 117, is given in annex C.

Three methods are proposed for calculation of the fan power output and efficiency. All three methods give very similar results (difference of a few parts per thousand for pressure ratios equal to 1,3).

d) Flowrate measurement

Determination of flowrate has been completely separated from the determination of fan pressure. A number of standardized methods may be used.

AUSTRALIAN STANDARD

Industrial fans — Performance testing using standardized airways

1 Scope

This International Standard deals with the determination of the performance of industrial fans of all types except those designed solely for air circulation e.g. ceiling fans and table fans.

Estimates of uncertainty of measurement are provided and rules for the conversion, within specified limits, of test results for changes in speed, gas handled and, in the case of model tests, size are given.

2 Normative references

The following standards contain provisions which, through reference in this text, constitute provisions of this International Standard. At the time of publication, the editions indicated were valid. All standards are subject to revision, and parties to agreements based on this International Standard are encouraged to investigate the possibility of applying the most recent editions of the standards indicated below. Members of IEC and ISO maintain registers of currently valid International Standards.

ISO 3966:1977, *Measurement of fluid flow in closed conduits — Velocity area method using Pitot static tubes*.

ISO 5167-1:1991, *Measurement of fluid flow by means of pressure differential devices — Part 1: Orifice plates, nozzles and Venturi tubes inserted in circular cross-section conduits running full*.

ISO 5168:—1), *Measurement of fluid flow — Evaluation of uncertainties*.

ISO 5221:1984, *Air distribution and air diffusion — Rules to methods of measuring air flowrate in an air handling duct*.

IEC 34-2:1972, *Rotating electrical machines — Part 2: Methods for determining losses and efficiency of rotating electrical machinery from tests (excluding machines for traction vehicles)*.

IEC 51-2:1984, *Direct acting indicating analogue electrical-measuring instruments and their accessories — Part 2: Special requirements for ammeters and voltmeters*.

IEC 51-3:1984, *Direct acting indicating analogue electrical-measuring instruments and their accessories — Part 3: Special requirements for wattmeters and varmeters*.

IEC 51-4:1984, *Direct acting indicating analogue electrical-measuring instruments and their accessories — Part 4: Special requirements for frequency meters*.

1) To be published. (Revision of ISO 5168:1978)

3 Definitions

For the purposes of this International Standard, the definitions given in ISO 5168 and the following definitions apply.

NOTE 1 All the symbols used in this International Standard are listed with their units in clause 4.

3.1 area of the conduit section, A_x : Area of the conduit at section x .

3.2 fan inlet area, A_1 : Surface plane bounded by the upstream extremity of the air-moving device.

The inlet area is, by convention, taken as the gross area in the inlet plane inside the casing.

3.3 fan outlet area, A_2 : Surface plane bounded by the downstream extremity of the air-moving device.

Fan outlet area is, by convention, taken as the gross area in the outlet plane inside the casing.

3.4 temperature, t : Air or fluid temperature measured by a temperature sensor.

3.5 absolute temperature, Θ : Thermodynamic temperature.

$$\Theta = t + 273,15$$

NOTE 2 In this document, Θ represents the absolute temperature and t the temperature in degrees Celsius.

3.6 specific gas constant, R :

For an ideal dry gas, the equation of state is written

$$\frac{p}{\rho} = R\Theta$$

$$R = 287 \text{ J} \cdot \text{kg}^{-1} \cdot \text{K}^{-1} \text{ for dry air.}$$

3.7 isentropic exponent, κ :

For an ideal gas and an isentropic process,

$$\frac{p}{\rho^\kappa} = \text{constant}$$

$$\kappa = 1,4 \text{ for atmospheric air.}$$

3.8 specific heat capacity at constant pressure, c_p :

For an ideal gas,

$$c_p = \frac{\kappa}{\kappa - 1} R$$

3.9 specific heat capacity at constant volume, c_V :

For an ideal gas,

$$c_V = \frac{1}{\kappa - 1} R$$

3.10 compressibility factor, Z :

For an ideal gas, $Z = 1$

For a real gas,

$$Z = \frac{p}{\rho R \Theta}$$

and Z is a function of the ratios p/p_c and Θ/Θ_c

where

p_c is the critical pressure of the gas;

Θ_c is the critical temperature of the gas.

3.11 stagnation temperature at a point, Θ_{sg} : Absolute temperature which exists at an isentropic stagnation point for ideal gas flow without addition of energy or heat.

The stagnation temperature is constant along an airway and, for an inlet duct, is equal to the absolute ambient temperature in the test enclosure.

3.12 static or fluid temperature at a point, Θ : Absolute temperature registered by a thermal sensor moving at the fluid velocity.

For a real gas flow,

$$\Theta = \Theta_{sg} - \frac{v^2}{2c_p}$$

where v = fluid velocity at a point, in metres per second ($\text{m} \cdot \text{s}^{-1}$).

In a duct, when the velocity increases the static temperature decreases.

3.13 dry-bulb temperature, t_d : Air temperature measured by a dry temperature sensor in the test enclosure, near the fan inlet or airway inlet.

3.14 wet-bulb temperature, t_w : Air temperature measured by a temperature sensor covered by a water-moistened wick and exposed to air in motion.

When properly measured, it is a close approximation of the temperature of adiabatic saturation.

3.15 stagnation temperature at a section x , Θ_{sgx} : Mean value, over time, of the stagnation temperature averaged over the area of the specified airway cross-section.

3.16 static or fluid temperature at a section x , Θ_x : Mean value, over time, of the static or fluid temperature averaged over the area of the specified airway cross-section.

3.17 absolute pressure at a point; absolute pressure, p : Pressure, measured with respect to absolute zero pressure, which is exerted at a point at rest relative to the air around it.

3.18 atmospheric pressure, p_a : Absolute pressure of the free atmosphere at the mean altitude of the fan.

3.19 gauge pressure, p_e : Value of the pressure when the datum pressure is the atmospheric pressure at the point of measurement.

It may be negative or positive.

$$p_e = p - p_a$$

3.20 absolute stagnation pressure at a point, p_{sg} : Absolute pressure which would be measured at a point in a flowing gas if it were brought to rest via an isentropic process:

$$p_{sg} = p \left(1 + \frac{\kappa - 1}{2} Ma^2 \right)^{\frac{\kappa}{\kappa - 1}}$$

where Ma is the Mach number at this point (see 3.23).

3.21 Mach factor, F_M : Correction factor applied to the dynamic pressure at a point, given by the expression

$$F_M = \frac{p_{sg} - p}{p_d}$$

The Mach factor may be calculated by

$$F_M = 1 + \frac{Ma^2}{4} + \frac{(2 - \kappa) Ma^4}{24} + \frac{(2 - \kappa)(3 - 2\kappa) Ma^6}{192} + \dots$$

3.22 dynamic pressure at a point, p_d : Pressure calculated from the velocity v and the density ρ of the air at the point.

$$p_d = \rho \frac{v^2}{2}$$

3.23 Mach number at a point, Ma : Ratio of the gas velocity at a point to the velocity of sound.

$$Ma = \frac{v}{\sqrt{\kappa R_w \Theta}} = \frac{v}{c}$$

where

c is the velocity of sound,

$$c = \sqrt{\kappa R_w \Theta}$$

R_w is the gas constant of humid gas.

3.24 gauge stagnation pressure at a point, p_{esg} : Difference between the absolute stagnation pressure p_{sg} and the atmospheric pressure p_a .

$$p_{esg} = p_{sg} - p_a$$

3.25 mass flowrate, q_m : Mean value, over time, of the mass of air which passes through the specified airway cross-section per unit of time.

NOTE 3 The mass flow will be the same at all cross-sections within the fan airway system excepting leakage.

3.26 average gauge pressure at a section x , p_{ex} : Mean value, over time, of the gauge pressure averaged over the area of the specified airway cross-section.

3.27 average absolute pressure at a section x , p_x : Mean value, over time, of the absolute pressure averaged over the area of the specified airway cross-section.

$$p_x = p_{ex} + p_a$$

3.28 average density at a section x , ρ_x : Fluid density calculated from the absolute pressure p_x and the static temperature Θ_x .

$$\rho_x = \frac{p_x}{R_w \Theta_x}$$

where R_w is the gas constant of humid gas.

3.29 volume flowrate at a section x , q_{vx} : Mass flowrate at the specified airway cross-section divided by the corresponding mean value, over time, of the average density at that section.

$$q_{vx} = \frac{q_m}{\rho_x}$$

3.30 average velocity at a section x , v_{mx} : Volume flowrate at the specified airway cross-section divided by the cross-sectional area A_x .

$$v_{mx} = \frac{q_{Vx}}{A_x}$$

NOTE 4 This is the mean value, over time, of the average component of the gas velocity normal to that section.

3.31 conventional dynamic pressure at a section x , p_{dx} : Dynamic pressure calculated from the average velocity and the average density at the specified airway cross-section.

$$p_{dx} = \rho_x \frac{v_{mx}^2}{2} = \frac{1}{2\rho_x} \left(\frac{q_m}{A_x} \right)^2$$

NOTE 5 The conventional dynamic pressure will be less than the average of the dynamic pressures across the section.

3.32 Mach number at a section x , Ma_x : Average gas velocity divided by the velocity of sound at the specified airway cross-section.

$$Ma_x = v_{mx} / \sqrt{\kappa R_w \Theta_x}$$

3.33 average stagnation pressure at a section x , p_{sgx} : Sum of the conventional dynamic pressure p_{dx} corrected by the Mach factor coefficient F_{Mx} at the section and the average absolute pressure p_x :

$$p_{sgx} = p_x + p_{dx} F_{Mx}$$

NOTE 6 The average stagnation pressure may be calculated by the expression

$$p_{sgx} = p_x \left(1 + \frac{\kappa - 1}{2} Ma_x^2 \right)^{\frac{\kappa}{\kappa - 1}}$$

3.34 gauge stagnation pressure at a section x , p_{esgx} : Difference between the average stagnation pressure, p_{sgx} , at a section and the atmospheric pressure, p_a .

$$p_{esgx} = p_{sgx} - p_a$$

3.35 inlet stagnation temperature, Θ_{sg1} : Absolute temperature in the test enclosure near the fan inlet at a section where the gas velocity is less than 25 m·s⁻¹.

In this case the stagnation temperature may be considered as equal to the ambient temperature Θ_a .

$$\Theta_{sg1} = \Theta_a = t_a + 273,15$$

3.36 stagnation density, ρ_{sg1} : Density calculated from the inlet stagnation pressure p_{sg1} and the inlet stagnation temperature Θ_{sg1} :

$$\rho_{sg1} = \frac{p_{sg1}}{R_w \Theta_{sg1}}$$

3.37 inlet stagnation volume flowrate, q_{Vsg1} : Mass flowrate divided by the inlet stagnation density.

$$q_{Vsg1} = \frac{q_m}{\rho_{sg1}}$$

3.38 fan pressure, p_F : Difference between the stagnation pressure at the fan outlet and the stagnation pressure at the fan inlet.

$$p_F = p_{sg2} - p_{sg1}$$

When the Mach number is less than 0,15,

$$p_F = p_{tF} = p_{t2} - p_{t1}$$

NOTE 7 Fan pressure should be referred to the installation type A, B, C or D.

3.39 fan dynamic pressure, p_{d2} : Average dynamic pressure at the fan outlet calculated from the mass flowrate, the average gas density at the outlet and the fan outlet area.

$$p_{d2} = \rho_2 \frac{v_{m2}^2}{2} = \frac{1}{2\rho_2} \left(\frac{q_m}{A_2} \right)^2$$

3.40 fan static pressure, p_{sF} : Conventional quantity defined as the fan pressure minus the fan dynamic pressure corrected by the Mach factor.

$$p_{sF} = p_{sg2} - p_{d2} \cdot F_{M2} - p_{sg1} = p_2 - p_{sg1}$$

NOTE 8 Fan static pressure should be referred to the installation type A, B, C or D.

3.41 mean density, ρ_m : Arithmetic mean value of inlet and outlet densities.

$$\rho_m = \frac{\rho_1 + \rho_2}{2}$$

3.42 mean stagnation density, ρ_{msg} : Arithmetic mean value of inlet and outlet stagnation densities.

$$\rho_{msg} = \frac{\rho_{sg1} + \rho_{sg2}}{2}$$

3.43 fan work per unit mass, y : Increase in mechanical energy per unit mass of fluid passing through the fan.

$$y = \frac{p_2 - p_1}{\rho_m} + \alpha_{A2} \frac{v_{m2}^2}{2} - \alpha_{A1} \frac{v_{m1}^2}{2}$$

y may be calculated as in 3.47, i.e.

$$y = \frac{P_u}{q_m}$$

The value obtained differs by only a few parts per thousand of the value given by the above expression:

NOTE 9 y should be referred to the installation type A, B, C or D.

3.44 fan static work per unit mass, y_s :

$$y_s = \frac{p_2 - p_1}{\rho_m} - \alpha_{A1} \frac{v_{m1}^2}{2}$$

NOTE 10 y_s should be referred to the installation type A, B, C or D.

3.45 fan pressure ratio, r : Ratio of the average absolute stagnation pressure at the outlet section of a fan to that at its inlet section.

$$r = p_{sg2} / p_{sg1}$$

3.46 compressibility coefficient, k_p : Ratio of the mechanical work done by the fan on the air to the work that would be done on an incompressible fluid with the same mass flow, inlet density and pressure ratio.

The work done is derived from the impeller power on the assumption of polytropic compression with no heat transfer through the fan casing.

k_p is given by the expression

$$k_p = \frac{Z_k \log_{10} r}{\log_{10} [1 + Z_k (r - 1)]}$$

where

$$Z_k = \frac{\kappa - 1}{\kappa} \cdot \frac{\rho_{sg1} P_r}{q_m p_r}$$

NOTE 11 k_p and ρ_{sg1}/ρ_{msg} differ by less than 2×10^{-3} .

3.47 fan air power, P_U : Conventional output power which is the product of the mass flow by the fan work per unit mass, or the product of the inlet volume flow, the compressibility coefficient k_p and the fan pressure.

$$P_U = q_m y = q_{Vsg1} \cdot p_F \cdot k_p$$

NOTE 12 P_U should be referred to the installation type A, B, C or D.

3.48 fan static air power, P_{US} : Conventional output power which is the product of the mass flowrate q_m by the fan static work per unit mass or the product of the inlet volume flowrate, the compressibility coefficient k_{ps} and the fan static pressure p_{sF} ; k_{ps} is calculated using $r = p_2/p_{sg1}$.

$$P_{US} = q_m y_s = q_{Vsg1} \cdot k_{ps} \cdot p_{sF}$$

NOTE 13 P_{US} should be referred to the installation type A, B, C or D.

3.49 impeller power, P_r : Mechanical power supplied to the fan impeller.

3.50 fan shaft power, P_g : Mechanical power supplied to the fan shaft.

3.51 motor output power, P_o : Shaft power output of the motor or other prime mover.

3.52 motor input power, P_e : Electrical power supplied at the terminals of an electric motor drive.

3.53 rotational speed of the impeller, N : Number of revolutions of the fan impeller per minute.

3.54 rotational frequency of the impeller, n : Number of revolutions of the fan impeller per unit time.

3.55 tip speed of the impeller, u : Peripheral speed of the impeller blade tips.

3.56 peripheral Mach number, Ma_u : Dimensionless parameter equal to the ratio of tip speed to the velocity of sound in the gas at the stagnation conditions of fan inlet:

$$Ma_u = u / \sqrt{\kappa R_w \Theta_{sg1}}$$

3.57 fan impeller efficiency, η_r : Fan air power divided by the impeller power P_r .

$$\frac{P_U}{P_r}$$

NOTE 14 η_r shall be referred to the installation type A, B, C or D.

3.58 fan impeller static efficiency, η_{sr} : Fan static power divided by the impeller power.

NOTE 15 η_{rs} should be referred to the installation type A, B, C or D.

3.59 fan shaft efficiency, η_a : Fan air power divided by the fan shaft power.

NOTES

16 Fan shaft power includes bearing losses, whilst fan impeller power does not.

17 η_a should be referred to the installation type A, B, C or D.

3.60 fan motor shaft efficiency, η_o : Fan air power P_U divided by the motor output power P_o .

NOTE 18 η_o should be referred to the installation type A, B, C or D.

3.61 overall efficiency, η_e : Fan air power divided by the motor input power for the fan and motor combination.

NOTE 19 η_e should be referred to the fan type A, B, C or D.

3.62 ratio of inlet density to mean density, k_p : Fluid density at the fan inlet divided by the mean fluid density in the fan.

$$k_p = \frac{2\rho_1}{\rho_1 + \rho_2}$$

3.63 kinetic energy factor at a section x , α_{Ax} : Dimensionless coefficient equal to the time-averaged flux of kinetic energy through the considered area A_x divided by the kinetic energy corresponding to the mean air velocity through this area.

$$\alpha_{Ax} = \frac{\iint_{A_x} (\rho v_n^2) dA_x}{q_m v_{mx}^2}$$

where

v is the local absolute velocity, in metres per second;

v_n is the local velocity normal to the cross-section.

NOTE 20 By convention $\alpha_{A1} = 1$ and $\alpha_{A2} = 1$.

3.64 kinetic index at a section x , i_{kx} : Dimensionless coefficient equal to the ratio of the kinetic energy per unit mass at the section x and the fan work per unit mass.

$$i_{kx} = \frac{v_{mx}^2}{2y}$$

3.65 Reynolds number at a section x , Re_{Dx} : Dimensionless parameter which defines the state of development of a flow and is used as a scaling parameter. It is the product of the local velocity, the local density and a relevant scale velocity, the local density and a relevant scale length (duct diameter, blade chord) divided by the dynamic viscosity.

$$Re_{Dx} = \frac{v_{mx} D_x}{\nu_x} = \frac{4q_m}{\pi \mu l_x}$$

3.66 friction loss coefficient, $(\zeta_{x-y})_y$: Dimensionless coefficient for friction losses between sections x and y of a duct, calculated for the velocity and density at section y .

For incompressible flow

$$\Delta p_{xy} = \frac{1}{2} \rho_y v_{my}^2 (\zeta_{x-y})_y$$

4 Symbols and units

4.1 Symbols

For the purposes of this International Standard, the following symbols and units apply.

Symbol	Represented quantity	SI unit
A_x	Area of the conduit at section x	m ²
a	Hole diameter of wall pressure tapings	mm
b	Width of the rectangular section of a duct	m
C	Discharge coefficient	—
c	Velocity of sound $c = \sqrt{\kappa R_w \Theta_x}$	m · s ⁻¹
c_p	Specific heat at constant pressure	J · kg ⁻¹ · K ⁻¹
c_v	Specific heat at constant volume	J · kg ⁻¹ · K ⁻¹
d	Diameter of orifice or nozzle throat	m
d_i	Diameter of stagnation pressure hole in Pitot-static tube	mm
D	Internal diameter of a circular conduit upstream of an in-line flowmeter	m
D_h	Hydraulic diameter of a rectangular section of a duct $\frac{4 \times \text{area of cross-section}}{\text{perimeter of cross-section}}$	m
D_x	Internal diameter of a circular conduit in the x plane	m
D_f	Outside diameter of the impeller	m
F_{Mx}	Mach factor for correction of dynamic pressure at section x	—
g	Gravitational acceleration	m · s ⁻²
h	Height of the rectangular section of a duct	m
h_u	Relative humidity p_v/p_{sat}	—
i_{kx}	Kinetic index at section x $\left(i_{kx} = \frac{V_{mx}^2}{2y} \right)$	—
k_c	Resulting coefficient used in the conversion of test results	—
k_p	Fan density ratio $\left(k_p = \frac{2p_1}{p_1 + p_2} \right)$	—
k_p	Compressibility coefficient for the calculation of fan air power P_u	—
k_{ps}	Compressibility coefficient for the calculation of fan static air power	—
Ma	Mach number	—
Ma_x	Mach number at section x	—
$Ma_{x \text{ ref}}$	Reference Mach number at section x at inlet stagnation conditions	—
Ma_u	Peripheral Mach number of impeller	—
m	Area ratio of an orifice plate $(d/D)^2$	—
n	Rotational frequency of impeller	r · s ⁻¹
N	Rotational speed of impeller	r · min ⁻¹
p	Absolute pressure of the fluid	Pa
p_a	Atmospheric pressure at the mean altitude of the fan	Pa
p_e	Gauge pressure $(p_e = p - p_a)$	Pa
p_{sg}	Absolute stagnation pressure	Pa
p_{esg}	Gauge stagnation pressure at a point	Pa
p_{esgx}	Gauge stagnation pressure at section x	Pa
p_d	Dynamic pressure at a point	Pa
p_x	Mean absolute pressure in space and time of the fluid at section x	Pa
p_{ex}	Mean gauge pressure in space and time at section x	Pa
p_{sgx}	Mean absolute stagnation pressure in space and time at section x	Pa

Symbol	Represented quantity	SI unit
p_{dx}	Conventional dynamic pressure at section x	Pa
p_{sat}	Saturation vapour pressure	Pa
p_v	Partial pressure of water vapour	Pa
p_F	Fan pressure ($p_F = p_{sg2} - p_{sg1}$)	Pa
p_{sF}	Fan static pressure ($p_{sF} = p_2 - p_{sg1}$)	Pa
p_{d2}	Fan dynamic pressure	Pa
p_u	Mean absolute pressure upstream of an in-line flowmeter	Pa
p_{do}	Mean absolute pressure downstream of an in-line flowmeter	Pa
P_a	Mechanical power supplied to the fan shaft	W
P_e	Motor input power	W
P_o	Power available at the output shaft of the drive	W
P_r	Mechanical power supplied to the impeller of the fan	W
P_u	Fan air power	W
P_{us}	Fan static power	W
q_m	Mass flowrate	$\text{kg} \cdot \text{s}^{-1}$
q_v	Volume flowrate	$\text{m}^3 \cdot \text{s}^{-1}$
q_{vsg1}	Volume flowrate at stagnation conditions upstream of the fan inlet corresponding to standard conditions of use	$\text{m}^3 \cdot \text{s}^{-1}$
q_{vx}	Volume flowrate at section x	$\text{m}^3 \cdot \text{s}^{-1}$
r	Pressure ratio	—
r_d	Pressure ratio for a flowmeter $r_d = p_{do}/p_u$	—
$r_{\Delta p}$	$\frac{\Delta p}{p_{do}}$ for a flowmeter	—
R	Gas constant of dry air or gas	$\text{J} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}$
R_w	Gas constant of humid air or gas	$\text{J} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}$
Re_{Dx}	Reynolds number at section x	—
t_a	Ambient temperature	$^{\circ}\text{C}$
t_b	Barometer temperature	$^{\circ}\text{C}$
t_d	Dry-bulb thermometer temperature	$^{\circ}\text{C}$
t_w	Wet-bulb thermometer temperature	$^{\circ}\text{C}$
t_x	Static temperature at section x	$^{\circ}\text{C}$
t_{sgx}	Stagnation temperature at section x	$^{\circ}\text{C}$
u	Peripheral velocity of impeller, or tip speed	$\text{m} \cdot \text{s}^{-1}$
u_x	Relative uncertainty of x	%
U_x	Absolute uncertainty of x	same as x
v	Velocity of gas at a point	$\text{m} \cdot \text{s}^{-1}$
v_{mx}	Average velocity of the gas at section x	$\text{m} \cdot \text{s}^{-1}$
y	Fan work per unit mass	$\text{J} \cdot \text{kg}^{-1}$
y_s	Fan static work per unit mass	$\text{J} \cdot \text{kg}^{-1}$
Z	Compressibility factor in equation of state	—
	$Z = \frac{p}{\rho R_w \theta_x}$	
	$Z = 1$ for an ideal gas	
Z_k	Coefficient used for the calculation of the compressibility factor k_p (first method)	—
Z_p	Coefficient used for the calculation of the compressibility factor k_p (second method)	—

Symbol	Represented quantity	SI unit
z_x	Mean altitude of section x	m
α	Flowrate coefficient of an in-line flowmeter	—
α_{Ax}	Coefficient of kinetic energy of flow in the section x of area A_x : α_{Ax} is assumed equal to 1	—
β	Ratio of the internal diameter of an orifice or nozzle to the upstream diameter of the duct d/D	—
β'	Ratio of the internal diameter of an orifice or nozzle to the downstream diameter of the duct	—
Δp	Differential pressure	Pa
Δz_b	Difference in altitude between the barometer and the mean altitude of the fan	m
ε	Expansibility factor	—
$(\zeta_{x-y})_y$	Conventional energy loss coefficient between sections x and y calculated for section y	—
η	Efficiency	—
η_s	Static efficiency	—
η_a	Fan shaft efficiency	—
	$\eta_a = \frac{P_u}{P_a}$	
η_e	Overall efficiency	—
	$\eta_e = \frac{P_u}{P_e}$	
η_o	Fan motor shaft efficiency	—
	$\eta_o = \frac{P_u}{P_o}$	
η_r	Fan efficiency	—
	$\eta_r = \frac{P_u}{P_r}$	
η_{sr}	Fan static efficiency	—
	$\eta_{sr} = \frac{P_{us}}{P_r}$	
θ_{sgx}	Stagnation temperature at section x	K
θ_x	Fluid temperature at section x	K
θ_a	Ambient temperature	K
θ_u	Temperature upstream of an in-line flowmeter	K
κ	Isentropic exponent $\kappa = c_p/c_v$ for an ideal gas	—
λ	Fan power coefficient	—
Λ	Specific friction loss coefficient for a length of one diameter of a straight duct	—
μ	Dynamic viscosity	Pa·s
ρ	Density of gas	kg·m ⁻³
ρ_x	Mean density of gas at section x	kg·m ⁻³
ρ_m	Mean density of gas in the fan	kg·m ⁻³
Φ	Flow coefficient	—
	$\Phi = \frac{q_m}{\rho_m D_f^2 u}$	
Ψ	Fan work per unit mass coefficient	—
	$\Psi = \frac{y}{u^2}$	

ω	Angular velocity	$\text{rad} \cdot \text{s}^{-1}$
ν	Kinematic viscosity	$\text{m}^2 \cdot \text{s}^{-1}$

4.2 Subscripts

1	Test fan inlet
2	Test fan outlet
3	Pressure measurement section in an inlet-side airway
4	Pressure measurement section in an outlet-side airway
5	Throat or downstream tapping for Δp for an inlet-side measurement
6	Upstream tapping for Δp and p_u for an outlet-side measurement
7	Upstream tapping for Δp and p_u for an inlet-side measurement
8	Throat or downstream tapping for Δp for an outlet-side measurement
a	Ambient atmosphere in the test enclosure
b	Barometer
c	Centrepoint of the test section
do	Downstream of a flow-measurement device
F	Fan
Gu	Guaranteed relative to the characteristics specified in the contract
n	Reference plane of the fan; $n = 1$ for inlet, $n = 2$ for outlet
s	Static conditions
sat	Saturation conditions
sg	Stagnation conditions
Te	Tested relative to the characteristics specified in the contract
u	Reference air conditions upstream of a flow-measurement device
x-y	Airway length from plane x to plane y

5 General

The upper limit of fan work per unit mass is $25\,000 \text{ J} \cdot \text{kg}^{-1}$ corresponding to an increase in fan pressure approximately equal to $30\,000 \text{ Pa}$ for a mean density in the fan of $1,2 \text{ kg} \cdot \text{m}^{-3}$.

The working fluid for test with standardized airways shall be atmospheric air, and the pressure and temperature should be within the normal atmospheric range.

There are four types of installation:

- type A: free inlet, free outlet,
- type B: free inlet, ducted outlet,
- type C: ducted inlet, free outlet,
- type D: ducted inlet, ducted outlet,

to which correspond four performance characteristics.

Fan performance cannot be considered as invariable. The performance curve of fan pressure versus flowrate may be modified by the upstream fluid flow, e.g. if the velocity profile is distorted or if there is swirl.

Although the downstream flow generally cannot act on the flow through the impeller, the losses in the downstream duct may be modified by the fluid flow at the fan outlet.

Methods of measurement and calculation for the flowrates, fan pressures and fan efficiencies are specified in clauses 14 to 27 and annex A. They are established in the case of compressible flow, taking into account Mach number effect and density variation. However, simplified methods are given for reference Mach numbers less than 0,15 and/or fan pressures less than $2\,000 \text{ Pa}$.

It is agreed that for the purposes of this International Standard, calculations are made using absolute pressures and temperatures but equivalent expressions using gauge pressures are provided.

It is conventionally agreed that:

— for fan installation types C and D, a common airway section should be provided upstream of the fan inlet to simulate a long, straight inlet duct;

— for fan installation categories B and D, a common airway section (incorporating a standardized flow straightener: an eight-radial-vane straightener, or honeycomb straightener) adjacent to the fan outlet should be provided upstream of the outlet pressure measurement section to simulate a long, straight outlet duct.

When the test installation is intended to simulate an on-site installation corresponding to type C but with a short duct discharging to the atmosphere, the test fan should be equipped with a duct having the same shape as the fan outlet and a length of two equivalent diameters.

For large fans of installation type D (800 mm diameter or larger) it may be difficult to carry out the tests with standardized common airways at the outlet side including straighteners. In this case, by mutual agreement between the parties concerned, the fan performance may be measured using the set-up described in 30.2 f) with a duct of length $2D$ on the outlet side. Results obtained in this way may differ to some extent from those obtained by using common airways on both the inlet and outlet side, especially if the fan produces a large swirl.

By convention the kinetic energy factors α_{A1} , α_{A2} at fan inlet and fan outlet are considered equal to one.

6 Instruments for pressure measurement

6.1 Barometers

The atmospheric pressure in the test enclosure shall be determined at the mean altitude between the centre of fan inlet and outlet sections with an uncertainty not exceeding $\pm 0,2\%$. Barometers of the direct-reading mercury column type should be read to the nearest 100 Pa (1 millibar) or to the nearest 1 mm of mercury. They should be calibrated and corrections applied to the readings for any difference in mercury density from standard, any change in length of the graduated scale due to temperature and for the local value of g .

Correction may be unnecessary if the scale is preset for the regional value of g (within $\pm 0,01 \text{ m} \cdot \text{s}^{-2}$) and for room temperature (within $\pm 5^\circ\text{C}$).

Barometers of the aneroid or pressure transducer type may be used provided they have a calibrated accuracy of $\pm 200 \text{ Pa}$ and the calibration is checked at the time of test.

The barometer should be located in the test enclosure at the mean altitude between fan inlet and fan outlet. A correction $\rho_a g(z_b - z_m)$ in Pa should be added for any difference in altitude exceeding 10 m,

where

z_b is the altitude at barometer reservoir or at barometer transducer;

z_m is the mean altitude between fan inlet and fan outlet;

g is the local value of acceleration due to gravity;

ρ_a is the ambient air density.

6.2 Manometers

Manometers for the measurement of pressure difference shall have an uncertainty under conditions of steady pressure, and after applying any calibration corrections (including that for any temperature difference from calibration temperature and for g value), not exceeding $\pm 1\%$ of the significant pressure or 1,5 Pa, whichever is greater.

The significant pressure should be taken as the fan stagnation pressure at rated duty or the pressure difference when measuring rated volume flow according to the manometer function. Rated duty will normally be near the point of best efficiency on the fan characteristic curve.

The manometers will normally be of the liquid column type, vertical or inclined, but pressure transducers with indicating or recording instrumentation are acceptable, subject to the same accuracy and calibration requirements.

Calibration should be carried out at a series of steady pressures, in both rising and falling sequences to check for any difference.

The reference instrument should be a precision manometer or micromanometer capable of being read to an accuracy of $\pm 0,25$ % or 0,5 Pa, whichever is greater.

6.3 Damping of manometers

Rapid fluctuations of manometer readings should be limited by damping so that it is possible to estimate the average reading within ± 1 % of the significant pressure. The damping may be in the air connections leading to the manometer or in the liquid circuit of the instrument. It should be linear, and of a type which ensures equal resistance to movement in either direction. The damping should not be so heavy that it prevents the proper indication of slower changes. If these occur a sufficient number of readings should be taken to determine an average within ± 1 % of the significant pressure.

6.4 Checking of manometers

Liquid column manometers should be checked in their test location to confirm their calibration near the significant pressure. Inclined tube instruments should be frequently checked for level and rechecked for calibration if disturbed. The zero reading of all manometers shall be checked before and after each series of readings without disturbing the instrument.

6.5 Position of manometers

The altitude of zero level of manometers or of pressure transducers should be the mean altitude of the section for pressure measurement (see figure 1).

7 Determination of average pressure in an airway

7.1 Methods of measurement

A differential manometer complying with the specifications of 6.2 to 6.5 shall be used with one side connected either to wall tapplings or to the pressure connections of a set of Pitot-static tubes in the plane of pressure measurement.

To determine the average static pressure in this plane, the other side of the manometer shall be open to the atmospheric pressure in the test enclosure.

To determine the pressure difference between planes of pressure measurement on opposite sides of the fan, either or both sides of the manometer may be connected between sets of four tapping connections arranged as recommended in 7.4.

7.2 Use of wall tapplings

At each of the sections for pressure measurement in the standardized airways specified in clauses 21 to 27 and 32 to 35 the average static pressure shall be taken to be the average of the static pressures at four wall tapplings constructed in accordance with 7.3.

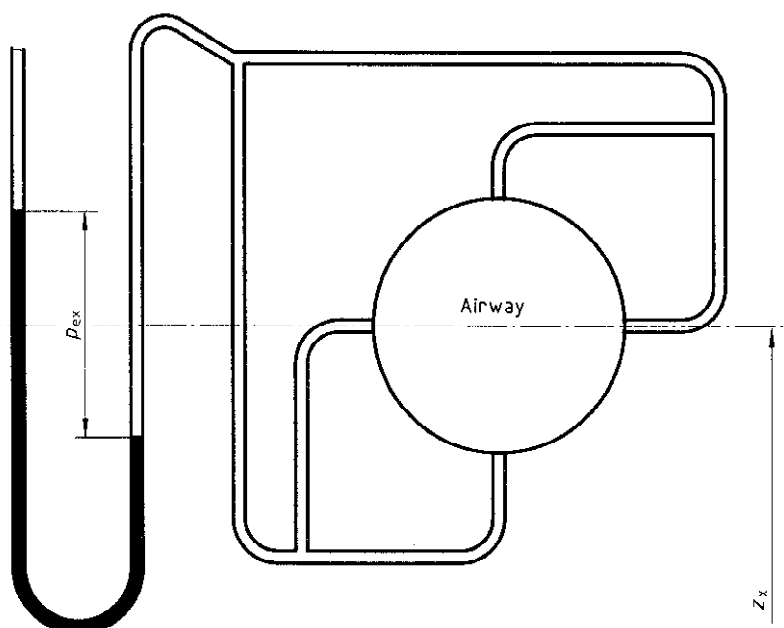


Figure 1 — Tapping connections to obtain average static pressure and altitude of manometer

7.3 Construction of tapplings

Each tapping takes the form of a hole through the wall of the airway conforming to the dimensional limits shown in figure 2. Additional limits are specified in clauses 22 to 26 for the tapplings used in flow measurement devices. It is essential that the hole be carefully produced so that the bore is normal to and flush with the inside surface of the airway, and that all internal protrusions are removed. Rounding of the edge of the hole up to a maximum of $0,1a$ is permissible.

The bore diameter a shall be not less than 1,5 mm, not greater than 5 mm and not greater than $0,1 D$.

Special care is required when the velocity in the airway is comparable with that at the fan inlet and outlet. In these cases the tapping should be situated in a section of the airway that is free from joints or other irregularities for a distance of $1D$ upstream and $D/2$ downstream, D being the airway diameter. In very large airways it may not be practicable to meet this condition. In such cases the Pitot-static tube method described in 7.6 may be used.

7.4 Position and connections

In the case of a cylindrical airway the four tapplings should be equally spaced around the circumference. In the case of a rectangular airway they should be at the centres of the four sides. Four similar tapplings may be connected to a single manometer. They should be connected as shown in figure 1.

7.5 Checks for compliance

Care shall be taken to ensure that all tubing and connections are free from blockage and leakage, and are empty of liquid. Before beginning any series of observations, the pressure at the four side tapplings should be individually measured at a flowrate approaching the maximum of the series. If any one of the four readings lies outside a range equal to 5 % for $p_{ex} \leq 1\,000$ Pa or 2 % for $1\,000 \text{ Pa} < p_{ex} < 30\,000$ Pa, p_{ex} being the mean gauge pressure, the tapplings and manometer connections should be examined for defects. If none are found, eight pressure tapplings should be used.

NOTE 21 By the mean gauge pressure is meant the pressure across the nozzle or orifice at rated flow in the case of flow measurement, or the rated fan pressure in the case of pressure measurement.

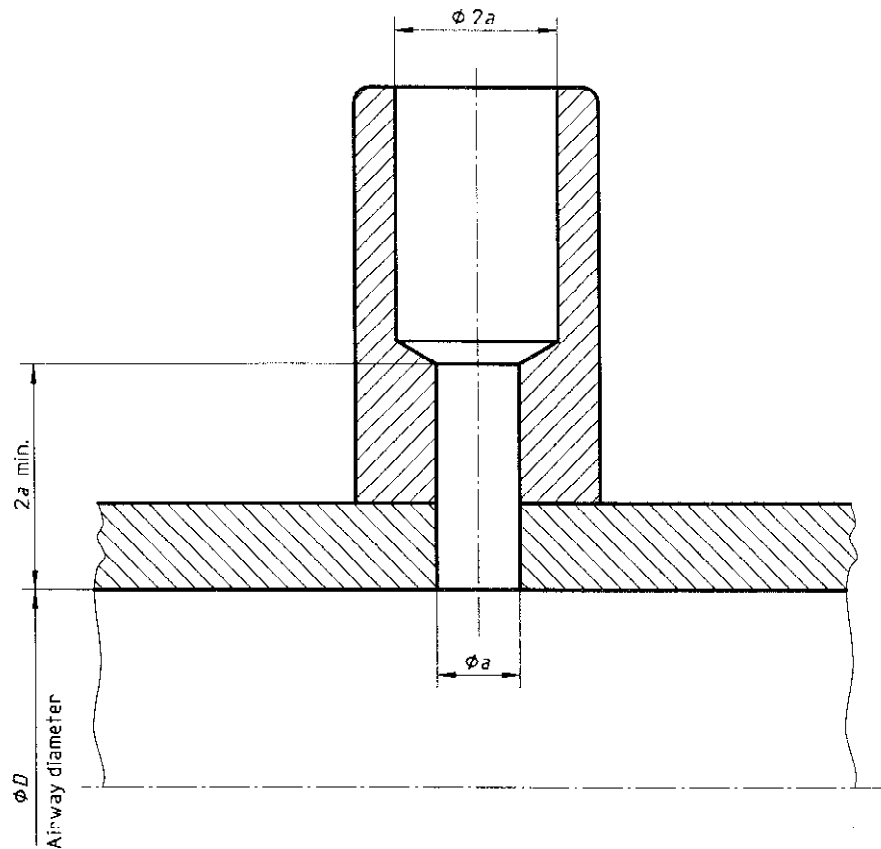


Figure 2 — Construction of wall pressure tapplings

7.6 Use of Pitot-static tube

At the appropriate pressure measurement plane in a circular airway, a minimum of four points should be selected, equally and symmetrically spaced around the axis at approximately one-eighth of the airway diameter from the wall or, in the case of a rectangular airway, one-eighth of the duct width from the centre of each wall. Under steady flow conditions, a static pressure reading should be taken at each point and the average calculated.

Alternatively, if desired, the static pressure connections of four separate Pitot-static tubes may be connected together to give a single average reading in the manner described in 7.4 and figure 1.

8 Measurement of temperature

8.1 Thermometers

Instruments for the measurement of temperature shall have an accuracy of $\pm 0,5$ °C after the application of any calibration correction.

8.2 Thermometer location

When a probe is put inside an airway to take temperature measurements, the measurement accuracy is a function of the fluid velocity.

The measured temperature, which is neither the stagnation temperature nor the static temperature, is a value lying between them and usually a bit closer to the stagnation value.

If the air velocity is equal to $25 \text{ m} \cdot \text{s}^{-1}$, the difference between stagnation and static temperatures is $0,31$ °C; at $35 \text{ m} \cdot \text{s}^{-1}$, the same difference is $0,61$ °C (for a static temperature of $293,15 \text{ K}$).

If the measurement is taken in a section where the air velocity is less than $25 \text{ m} \cdot \text{s}^{-1}$, the measured temperature is assumed equal to both stagnation and static temperatures.

It is therefore recommended that measurement of the stagnation temperature be made upstream of the fan inlet or of the test airway, either in a section where the air velocity lies between zero and $25 \text{ m} \cdot \text{s}^{-1}$ or in the inlet chamber.

In order to measure the mean stagnation temperature, one or several probes shall then be put in the appropriate section, located on a vertical diameter at different altitudes symmetrically situated from the diameter centre. Probes shall be shielded against radiation from heated surfaces.

If it is not possible to meet these requirements, probes can be placed inside an airway on a horizontal diameter, at least 100 mm from the wall or one-third of the airway diameter, whichever is less.

8.3 Humidity

The dry-bulb and wet-bulb temperatures in the test enclosure should be measured at a point where they can record the condition of the air entering the test airway. The instruments should be shielded against radiation from heated surfaces.

The wet-bulb thermometer should be located in an air stream of velocity at least $3 \text{ m} \cdot \text{s}^{-1}$. The sleeving should be clean, in good contact with the bulb, and kept wetted with pure water.

Relative humidity may be measured directly provided the apparatus used has an accuracy of $\pm 2 \%$.

9 Measurement of rotational speed

9.1 Fan shaft speed

The fan shaft speed shall be measured at regular intervals throughout the period of test for each test point, so as to ensure the determination of average rotational speed during each such period with an uncertainty not exceeding $\pm 0,5 \%$.

No device used should significantly affect the rotational speed of the fan under test or its performance.

9.2 Examples of acceptable methods

9.2.1 Digital counter measuring revolutions for a given time interval

The number of impulses counted shall be not less than 1 000 during the measured time interval. The timing device shall be actuated automatically by the starting and stopping of the counter, and shall not be in error by more than 0,25 % of the time needed to count the total number of impulses.

9.2.2 Revolution counter

The revolution counter shall be free from slip and programmed for a period of not less than 60 s per reading.

9.2.3 Direct-readout mechanical or electrical tachometer

These devices shall be free from slip and calibrated before and after use. The smallest division on the scale of such an instrument should represent not more than 0,25 % of the measured rotational speed.

9.2.4 Stroboscope methods

Stroboscopes shall be calibrated against a rotating standard before and after use, unless fed by or checked against a source whose frequency is known or measured within $\pm 0,25 \%$.

9.2.5 Frequency meter

When the fan is direct-driven by a synchronous or induction motor, the supply frequency can be measured and in the latter case, also the slip frequency. The frequency meter shall have an uncertainty of not more than 0,5 % (i.e. accuracy class index of 0,5 in accordance with IEC 51-4).

Alternatively, a digital instrument of lower class index, i.e. smaller uncertainty, is permissible. The device used for indicating slip frequency shall be used in such a manner as to permit direct counting with an uncertainty not exceeding $\pm 0,25$ % of the shaft speed

10 Determination of power input

10.1 Measurement accuracy

The power input to the fan over the specified performance range shall be determined by a method, including the averaging of a sufficient number of readings at each test point, which achieves a result with an uncertainty not exceeding ± 2 %.

10.2 Fan shaft power

When the power to be determined is the input to the fan shaft, acceptable methods include the following.

10.2.1 Reaction dynamometer

The torque is measured by means of a cradle or torque-table type dynamometer. The weights shall have certified accuracies of $\pm 0,2$ %. The length of the torque arm shall be determined to an accuracy of $\pm 0,2$ %.

The zero-torque equilibrium (tare) shall be checked before and after each test. The difference shall be within 0,5 % of the maximum value measured during the test.

10.2.2 Torsion meter

The torque is measured by means of a torsion meter having an uncertainty which should not be more than 2,0 % of the torque to be measured. For the calibration, the weights shall have certified accuracies of $\pm 0,2$ %. The length of the torque arm shall be determined to an accuracy of $\pm 0,2$ %.

The zero-torque equilibrium (tare) and the span of the readout system shall be checked before and after each test. In each case, the difference shall be within 0,5 % of the maximum value measured during the test.

10.3 Determination of fan shaft power by electrical measurement

10.3.1 Summation of losses

The power output of an electric motor for direct drive is deduced from its electrical power input by the summation of losses method specified in IEC 34-2. For this purpose, measurements of voltage, current, speed and, in the case of a.c. motors, power input and slip of induction motors shall be made for each test point and the no-load losses of the motor when uncoupled from the fan shall be measured.

10.3.2 Calibrated motor

The power output of an electric motor for direct drive is determined from an efficiency calibration acceptable to both manufacturer and purchaser. The motor should be run on charge for a time sufficient to ensure that it is running at its normal working temperature. The electrical supply should be within the statutory limits, i.e.:

voltage: ± 6 %

frequency: ± 1 %

10.3.3 Electrical instruments

The electrical power input to the motor during the fan tests described in 10.3.1 or 10.3.2 shall be measured by one of the following methods:

- a) for a.c. motors, by the two-wattmeter method or by an integrating wattmeter;
- b) for d.c. motors, by measurement of the input voltage and current.

For standardized airway tests, the instruments used for these measurements shall be of class index 0,5 in accordance with IEC 51-2 and IEC 51-3 to which calibration corrections are applied or, alternatively, of class index 0,2 for which calibration corrections are unnecessary.

10.4 Impeller power

To determine the power input to the fan impeller hub it is necessary, unless the impeller is mounted directly on the motor shaft, to deduct from the fan shaft power an allowance for bearing losses and for the losses in any flexible coupling. This may be determined by running a further test at the same speed with the impeller removed from the shaft and measuring the torque losses due to bearing friction. If considered necessary, the fan impeller may be substituted by an equivalent mass (having negligible aerodynamic loss) to provide similar bearing loadings.

10.5 Transmission systems

For tests with standardized airways, the interposition of a transmission system between the fan and the point of power measurement should be avoided unless it is of a type in which the transmission losses under the specified working conditions can be reliably determined, or the specified power input is required to include those losses.

11 Measurement of dimensions and determination of areas

11.1 Flow measurement devices

The dimensions of nozzles, orifices and airways used for flow measurement shall conform to the tolerances specified in the appropriate subclauses covering their use.

11.2 Tolerance on dimensions

11.2.1 Specified airway component lengths shall be measured after manufacture and shall conform to the requirements of the test method within a tolerance of $+10_{-1}^{\circ}$ %, except where otherwise stated.

11.2.2 Specified airway component diameters shall be measured after manufacture and shall conform to the requirements of the test method within a tolerance of ± 1 % except where otherwise stated.

11.3 Determination of cross-sectional area

11.3.1 Dimensional measurements

Sufficient dimensional measurements shall be taken across the reference planes of airways to determine cross-sectional areas within $\pm 0,5$ % in standardized airways and other well-defined regular sections.

11.3.2 Circular sections

For circular sections, the mean diameter of the section is taken as being equal to the arithmetic mean of the measured values on at least three diameters of the measuring section. The diameters shall be so positioned that they are at equal angles within the cross-section.

If the difference in linear measurement between two adjacent diameters is more than 1 %, the number of measured diameters shall be doubled. The area of the circular section shall be calculated as follows:

$$\pi \frac{D^2}{4}$$

where D is the arithmetic mean of the measured diameters.

11.3.3 Rectangular sections

The width and height of a rectangular section shall be measured along five equidistant lines parallel to the width and height. If the difference between two adjacent widths or heights is more than 2 %, then the number of measurements in that direction shall be doubled.

The average width of the section shall be taken as the arithmetic mean of all the widths measured and the average height of the section shall be taken as the arithmetic mean of all the heights measured. The cross-sectional area of the section shall be taken as being the average width multiplied by the average height.

12 Determination of air density, humid gas constant and viscosity

12.1 Density of air in the test enclosure, gas constant for humid air and average density in a section x

The density of the ambient air in the test enclosure is given by the following expression:

$$\rho_a = \frac{p_a - 0,378 p_v}{287 \Theta_a}$$

where

Θ_a is the absolute ambient temperature, in kelvins.

$$\Theta_a = t_a + 273,15$$

where $t_a = t_d$ (dry-bulb temperature, in degrees Celsius) (see 14.3);

p_v is the partial water vapour pressure in the air, in pascals;

287 is the gas constant for dry air, R , in joules per kilogram kelvin;

$$0,378 = \frac{R_v - R}{R_v}$$

with $R_v = 461$ which is the gas constant of water vapour.

The gas constant of humid air, R_w , is then given by

$$R_w = \frac{p_a}{\rho_a \Theta_a} = \frac{287}{1 - 0,378 \frac{p_v}{p_a}}$$

NOTE 22 For standard air:

$$\rho_a = 1,2 \text{ kg} \cdot \text{m}^{-3}$$

$$\Theta_a = 293,15 \text{ K}$$

$$p_a = 101\,325 \text{ Pa}$$

$$h_v = 0,40$$

$$R_w = 288 \text{ J} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}$$

The average density of the air in an airway section x may be obtained from the following expression:

$$\rho_x = \frac{p_x}{R_w \Theta_x}$$

12.2 Determination of vapour pressure

The partial vapour pressure p_v , is obtained from the following expression when the air humidity is measured by means of a psychrometer at the fan inlet:

$$p_v = (p_{\text{sat}})_{t_w} - p_a \cdot A_w (t_d - t_w) (1 + 0,00115 t_w)$$

where

t_d is the dry-bulb temperature, in degrees Celsius;

t_w is the wet-bulb temperature, in degrees Celsius;

$A_w = 6,6 \times 10^{-4} \text{ } ^\circ\text{C}^{-1}$ when t_w is between $0 \text{ } ^\circ\text{C}$ and $150 \text{ } ^\circ\text{C}$;

$A_w = 5,94 \times 10^{-4} \text{ } ^\circ\text{C}^{-1}$ when t_w is less than $0 \text{ } ^\circ\text{C}$;

$(p_{\text{sat}})_{t_w}$ is the pressure of saturated vapour at the wet-bulb temperature t_w .

Table 1 lists values of saturated vapour pressure (p_{sat}) over the temperature range $-4 \text{ } ^\circ\text{C}$ to $49,5 \text{ } ^\circ\text{C}$.

p_{sat} may be obtained from the following expression, between $0 \text{ } ^\circ\text{C}$ and $30 \text{ } ^\circ\text{C}$:

$$p_{\text{sat}} = \exp \left(\frac{17,438 t_w}{239,78 + t_w} + 6,4147 \right)$$

or, between $0 \text{ } ^\circ\text{C}$ and $100 \text{ } ^\circ\text{C}$:

$$p_{\text{sat}} = 610,8 + 44,442 t_w + 1413,3 t_w^2 + 0,02768 t_w^3 + 2,55667 e^{-4} t_w^4 + 2,89166 e^{-6} t_w^5$$

The air relative humidity, h_u , can also be directly measured in order to obtain

$$p_v = h_u (p_{\text{sat}})_{t_d}$$

where $(p_{\text{sat}})_{t_d}$ is the saturation vapour pressure at the dry-bulb temperature t_d calculated using the above formula with t_d instead of t_w .

12.3 Determination of air viscosity

The following formula can be used in the range $-20 \text{ } ^\circ\text{C}$ to $+100 \text{ } ^\circ\text{C}$ to obtain the dynamic viscosity, in pascal seconds:

$$\mu = (17,1 + 0,048 t_x) \times 10^{-6}$$

The kinematic viscosity ν is given by the following expression:

$$\nu = \frac{\mu}{\rho}$$

Table 1 — Saturation vapour pressure p_{sat} of water as a function of wet-bulb temperature t_w

Wet-bulb temperature t_w °C	Saturation vapour pressure p_{sat} of water (above water) hPa									
	0,0	0,1	0,2	0,3	0,4	0,5	0,6	0,7	0,8	0,9
– 4	4,55	4,51	4,48	4,44	4,41	4,37	4,35	4,31	4,28	4,24
– 3	4,89	4,87	4,83	4,79	4,76	4,72	4,68	4,65	4,61	4,59
– 2	5,28	5,24	5,20	5,16	5,12	5,08	5,04	5,01	4,97	4,93
– 1	5,68	5,64	5,60	5,56	5,52	5,47	5,44	5,39	5,36	5,32
– 0	6,11	6,07	6,03	5,97	5,93	5,89	5,84	5,80	5,76	5,72
0	6,11	6,16	6,19	6,24	6,29	6,33	6,37	6,43	6,47	6,52
1	6,56	6,61	6,67	6,71	6,76	6,80	6,85	6,91	6,96	7,00
2	7,05	7,11	7,16	7,21	7,25	7,31	7,36	7,41	7,47	7,52
3	7,57	7,63	7,68	7,73	7,79	7,85	7,91	7,96	8,01	8,08
4	8,13	8,19	8,24	8,31	8,36	8,43	8,48	8,53	8,60	8,65
5	8,72	8,79	8,84	8,91	8,96	9,03	9,09	9,16	9,21	9,28
6	9,35	9,41	9,48	9,53	9,61	9,68	9,75	9,81	9,88	9,95
7	10,01	10,08	10,15	10,23	10,29	10,36	10,43	10,51	10,57	10,65
8	10,72	10,80	10,87	10,95	11,01	11,09	11,17	11,24	11,32	11,40
9	11,48	11,55	11,63	11,71	11,79	11,87	11,95	12,03	12,11	12,19
10	12,27	12,36	12,44	12,52	12,61	12,69	12,77	12,87	12,95	13,04
11	13,12	13,21	13,29	13,39	13,47	13,56	13,65	13,75	13,84	13,93
12	14,01	14,11	14,20	14,29	14,39	14,48	14,59	14,68	14,77	14,87
13	14,97	15,07	15,17	15,27	15,36	15,47	15,57	15,67	15,77	15,88
14	15,97	16,08	16,19	16,29	16,40	16,51	16,61	16,72	16,83	16,93
15	17,04	17,16	17,27	17,37	17,49	17,60	17,72	17,83	17,96	18,05
16	18,17	18,29	18,41	18,52	18,64	18,76	18,88	19,00	19,12	19,25
17	19,37	19,49	19,61	19,73	19,87	19,99	20,12	20,24	20,37	20,51
18	20,63	20,76	20,89	21,03	21,16	21,29	21,43	21,56	21,69	21,83
19	21,96	22,11	22,24	22,39	22,52	22,67	22,80	22,95	23,09	23,23
20	23,37	23,52	23,67	23,81	23,96	24,11	24,25	24,41	24,56	24,71
21	24,87	25,01	25,17	25,32	25,48	25,64	25,80	25,95	26,11	26,27
22	26,43	26,60	26,76	26,92	27,08	27,25	27,41	27,59	27,75	27,92
23	28,09	28,25	28,43	28,60	28,77	28,95	29,12	29,31	29,48	29,65
24	29,84	30,01	30,19	30,37	30,56	30,75	30,92	31,11	31,29	31,48
25	31,68	31,87	32,05	32,24	32,44	32,63	32,83	33,01	33,21	33,41
26	33,61	33,81	34,01	34,21	34,41	34,61	34,83	35,03	35,24	35,44
27	35,65	35,87	36,08	36,28	36,49	36,71	36,93	37,15	37,36	37,57
28	37,80	38,03	38,24	38,47	38,69	38,92	39,15	39,37	39,60	39,83
29	40,05	40,29	40,52	40,76	41,00	41,23	41,47	41,71	41,95	42,19
30	42,43	42,68	42,92	43,17	43,41	43,67	43,92	44,17	44,43	44,68
31	44,93	45,19	45,44	45,71	45,96	46,23	46,49	46,75	47,01	47,28
32	47,56	47,83	48,09	48,37	48,64	48,92	49,19	49,47	49,75	50,03
33	50,31	50,60	50,88	51,16	51,45	51,73	52,03	52,32	52,61	52,91
34	53,20	53,51	53,80	54,11	54,40	54,71	55,01	55,32	55,63	55,93

Table 1 — (end)

Wet-bulb temperature t_w °C	Saturation vapour pressure p_{sat} of water (above water) hPa									
	0,0	0,1	0,2	0,3	0,4	0,5	0,6	0,7	0,8	0,9
35	56,24	56,55	56,87	57,17	57,49	57,81	58,13	58,45	58,77	59,11
36	59,43	59,76	60,08	60,41	60,75	61,08	61,41	61,75	62,08	62,43
37	62,77	63,11	63,45	63,80	64,15	64,49	64,85	65,20	65,56	65,91
38	66,27	66,63	66,99	67,35	67,72	68,08	68,45	68,83	69,19	69,56
39	69,95	70,32	70,69	71,07	71,45	71,84	72,23	72,61	73,00	73,39
40	73,79	74,17	74,57	74,97	75,37	75,77	76,17	76,59	76,99	77,40
41	77,81	78,23	78,64	79,05	79,47	79,89	80,32	80,73	81,16	81,59
42	82,03	82,45	82,89	83,32	83,76	84,20	84,64	85,08	85,53	85,97
43	86,43	86,88	87,33	87,79	88,25	88,71	89,17	89,64	90,11	90,57
44	91,04	91,52	91,99	92,47	92,95	93,43	93,91	94,40	94,88	95,37
45	95,87	96,36	96,85	97,35	97,85	98,36	98,85	99,36	99,88	100,39
46	100,89	101,41	101,93	102,45	102,97	103,51	104,04	104,57	105,09	105,63
47	106,17	106,71	107,25	107,79	108,33	108,89	109,44	109,99	110,55	111,11
48	111,67	112,23	112,80	113,37	113,93	114,51	115,08	115,65	116,24	116,83
49	117,41	118,00	118,59	119,17	119,79	120,37	120,99	121,57	122,19	122,80

13 Determination of flowrate

13.1 General

The measurement of flowrate may be carried out in accordance with ISO 5167-1 and 3966, and any flowrate measurement obtained in this way will conform to the requirements of this International Standard.

This International Standard specifies different flow-metering methods which are appropriate for fan testing purposes, and in each case the associated uncertainty of measurement is given.

The flow shall be effectively swirl-free. Provisions to ensure that this condition is met are included in the methods of test.

Two basic flow-metering methods are permissible under these relaxed conditions: i.e. the use of an in-line flowmeter or a traversing method.

13.2 In-line flowmeters (standard primary devices)

13.2.1 The flowmeters which may be used are the Venturi nozzle, the orifice plate, the conical inlet and the inlet nozzle.

The first two may be used at the inlet to or outlet from an airway as well as between two sections of an airway.

The conical inlet and the inlet nozzle may only be used at the inlet to an airway, drawing air from free space.

The requirements for these instruments and for the simplified installations in which they may be used are given in clauses 22 to 26 and in ISO 5167-1.

13.2.2 The general expression for the mass flowrate through an in-line differential pressure flowmeter is as follows:

$$q_m = \frac{\alpha \epsilon \pi d^2}{4} \sqrt{2 \rho_a \Delta p}$$

where

q_m is the mass flowrate, in kilograms per second;

- d is the throat diameter, in metres;
- ρ_u is the upstream density, in kilograms per cubic metre;
- Δp is the pressure difference, in pascals;
- α is the flow coefficient;
- ε is the expansibility factor.

$$\rho_u = \frac{p_u}{R_w \Theta_u}$$

Normally, Θ_u should be the fluid temperature upstream of the flowmeter.

When the flowmeter is at the inlet side of the fan to be tested

$$\Theta_u = \Theta_{sgu} - \frac{q_m^2}{2A_u^2 \rho_u^2 c_p} + \frac{P_{rx} \text{ or } P_{ex}}{q_m c_p}$$

where

- P_{rx} or P_{ex} is the power provided by any auxiliary fan;
- A_u is the area of the duct upstream the flowmeter;
- $A_u = \infty$ for an inlet orifice or an inlet nozzle.

When the flowmeter is at the outlet side of the fan to be tested

$$\Theta_u = \Theta_{sg1} + \frac{P_r \text{ or } P_e}{q_m c_p} - \frac{q_m^2}{2A_u^2 \rho_u^2 c_p}$$

The value of q_m is obtained by an iterative procedure.

For a given device, ε is a function of the pressure ratio and α is a function of the Reynolds number. Values for these coefficients are given in clauses 22 to 26, tables 5 and 6, and figures 18, 22 and 24 to 28.

13.2.3 The pressure difference across an in-line flowmeter shall be measured with an uncertainty not exceeding $\pm 1,4$ % of the observed value.

13.2.4 The values for the uncertainties of the flow coefficient associated with each flow metering element are given in clauses 22 to 26. It shall always be possible to reduce the uncertainties associated with any in-line flowmeter installation different from those defined in ISO 5167-1 by calibrating the installation against an improved or calibrated standard device in accordance with ISO 5167-1.

13.2.5 In order to facilitate the selection of type and size of flowmeter, the losses associated with each type are given in figure 3. Approximate values for the pressure difference (expressed as a multiple of the dynamic pressure in the down-stream airway) which will be registered across each device are also shown.

13.2.6 The Venturi nozzle has a relatively low pressure loss and a lower sensitivity to disturbances in the approaching airflow. The orifice plate, in particular, incurs higher pressure losses and an auxiliary booster fan is required if the fan characteristic is to be extended to maximum volume flow. For tests at one or more preset points on a fan characteristic, an orifice plate can, simultaneously with the flow measurement, control the pressure drop, and this can be a useful property.

13.2.7 In-line flowmeters are normally used for tests in a laboratory; they may however, be used for site tests, provided the installation meets the appropriate requirements specified in clauses 22 to 26.

13.3 Traverse methods

The local velocity should be measured at a number of positions across a duct and the individual velocity values combined, using an integration technique, to yield an estimate of the mean velocity in the duct. Measurement of the cross-sectional area of the duct in the traverse plane then allows calculation of the flowrate (see clause 11 and 27).

In standardized airways, a Pitot-static tube conforming to the requirements of ISO 3966 [see figure 29 a), b), c) and d)] shall be used.

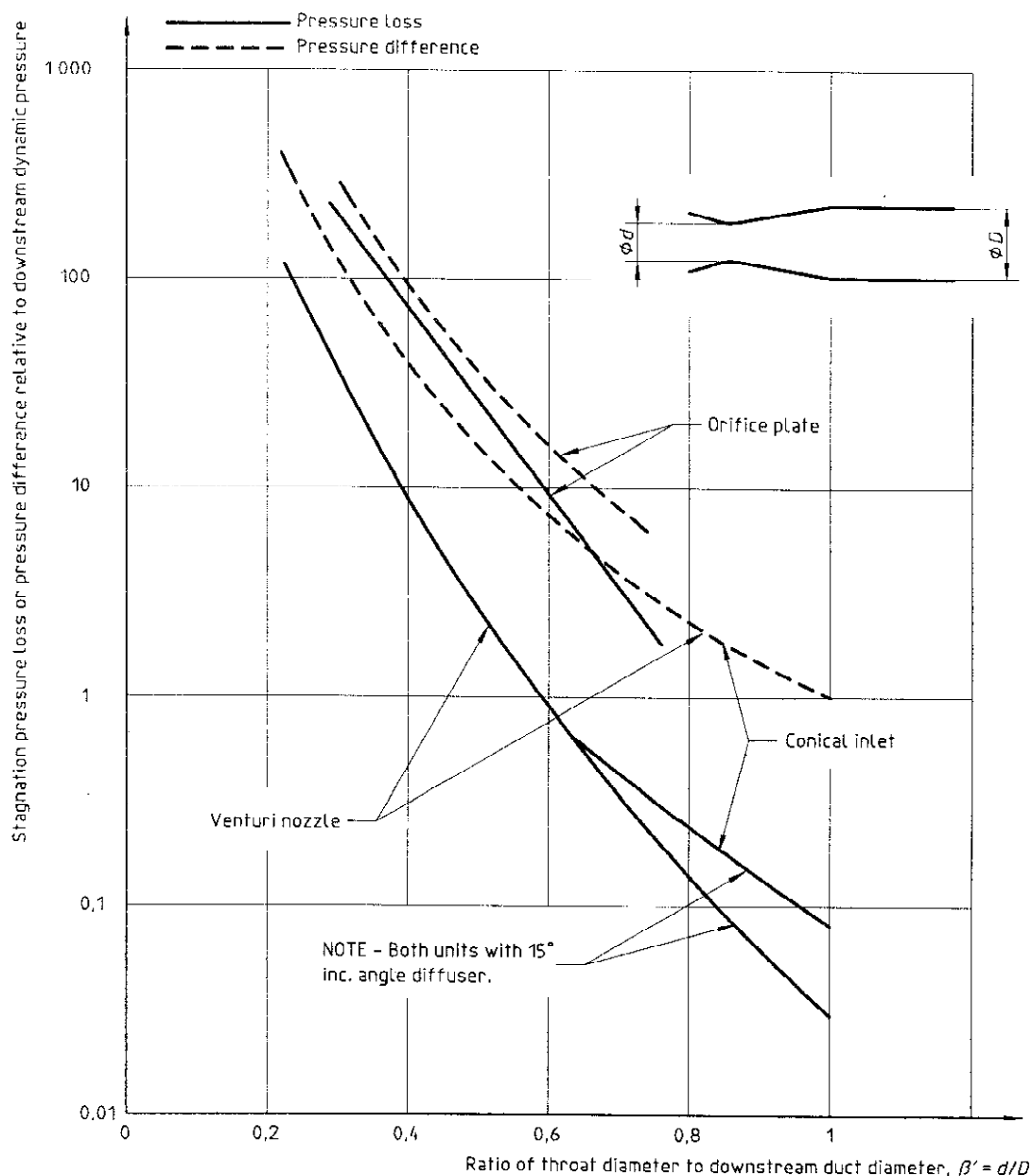


Figure 3 — Pressure loss and pressure difference of standard primary systems

14 Calculation of test results

14.1 General

Specific instructions for the calculation of fan performance from measurements at a single test point are given in clauses 18 to 35 depending on the test method used.

The method of calculation in the general case of compressible fluid flow is explained in this clause. The use of simplified methods and their limitations are given in 14.9.

14.2 Units

The units throughout these calculations shall be the SI units given in clause 4. The results will then also be in these units, viz. pressure in pascals (Pa); power in watts (W); volume flow in cubic metres per second ($\text{m}^3 \cdot \text{s}^{-1}$).

14.3 Temperature

14.3.1 In this International Standard, the mean temperature measured at section x is assumed to be the stagnation value Θ_{sgx} rather than the fluid temperature or static temperature Θ_x , which is slightly lower at high velocities.

The static temperature Θ_x is determined in accordance with 14.4.3 and used in the fluid state equation to calculate the density.

14.3.2 The behaviour of air in the test airways for the provisions of this International Standard is considered as adiabatic, because the air is taken from the atmosphere and because of the absence of heat or mechanical energy increase, except in the test fan. Consequently, the stagnation temperature Θ_{sgx} in all sections upstream of the fan tested shall be considered constant and equal to the ambient temperature at the test site, Θ_a :

$$\Theta_{sg1} = \Theta_{sg3} = \Theta_a$$

except when an auxiliary fan is used upstream of a test chamber or test airway.

14.3.3 The stagnation temperature at the fan outlet Θ_{sg2} and in the downstream airways is equal to the stagnation temperature at the fan inlet, increased by the temperature rise through the fan which is dependent upon the impeller power P_r , the mass flow q_m and the heat capacity of air at constant pressure c_p .

$$\Theta_{sg2} = \Theta_{sg1} + \frac{P_r \text{ or } P_e}{q_m c_p} = \Theta_{sg4}$$

NOTES

23 In the above expression, c_p can be taken as $1\,008\text{ J} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}$ as a first approximation for air.

24 P_r should be replaced by the electric input power P_e when the motor is wholly immersed in the airstream.

14.3.4 When the above conditions do not apply, for instance if the impeller power is not measured, the stagnation temperature shall be measured by a measuring device (e.g. thermometer) inserted into the airway in accordance with 8.2 at a point where the velocity does not exceed $35\text{ m} \cdot \text{s}^{-1}$ provided this is reasonably close to the relevant section.

14.3.5 The fluid temperature at section x , Θ_x , is less than the stagnation measured or derived temperature at that section. It is expressed in terms of the Mach number Ma_x and of the stagnation temperature Θ_{sgx} as:

$$\frac{\Theta_{sgx}}{\Theta_x} = 1 + \frac{\kappa - 1}{2} Ma_x^2$$

The ratio

$$\frac{\Theta_{sgx}}{\Theta_x}$$

is plotted in figure 4 as a function of Ma_x .

Because Ma_x is usually unknown, Θ_x shall be calculated from:

- the mass flowrate, q_m ,
- the stagnation temperature, Θ_{sgx} ,
- the section area, A_x ,
- the pressure p_x or the stagnation pressure at section x , p_{sgx} ,

in accordance with 14.4.3.

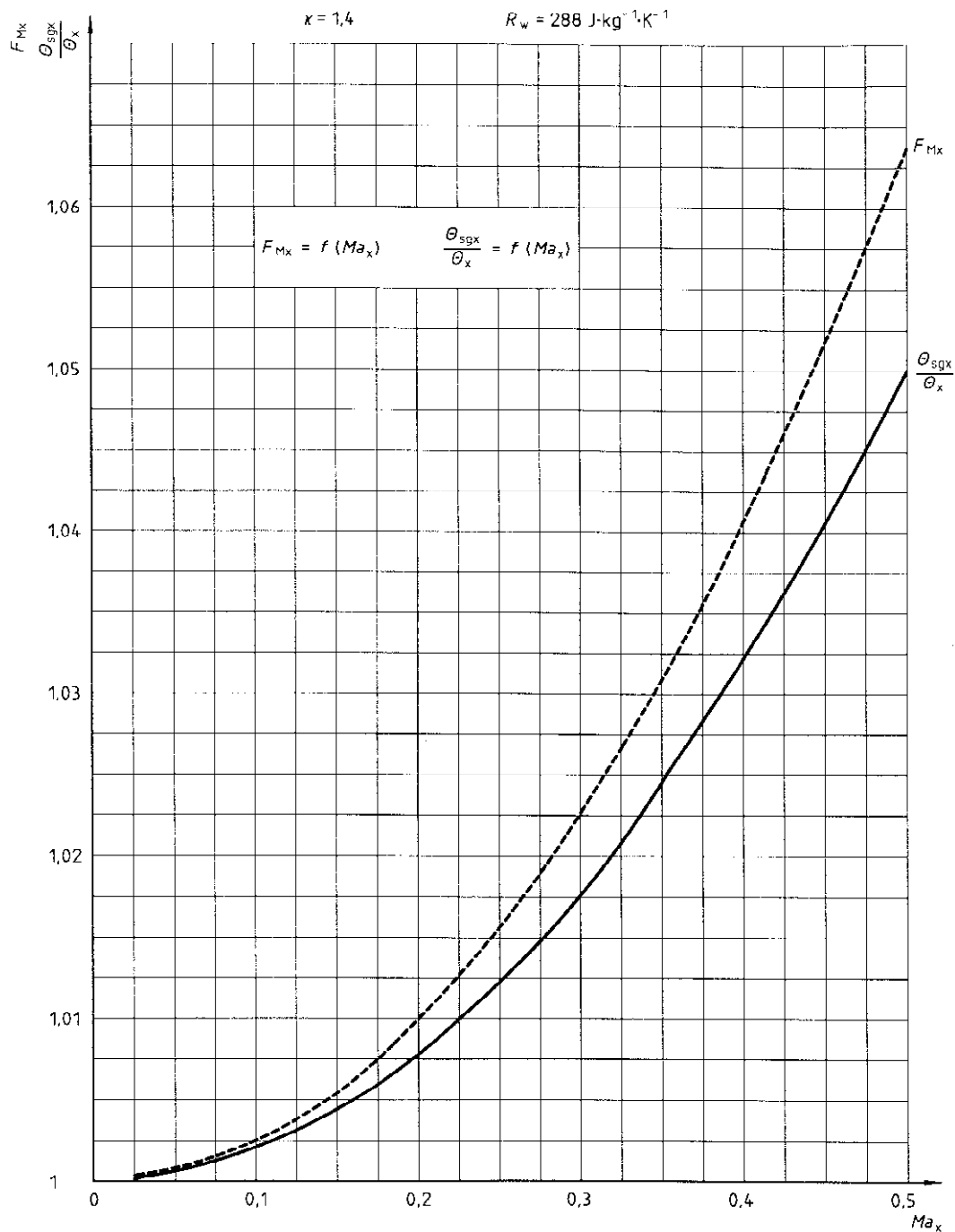


Figure 4 — Changes in F_{Mx} and the ratio θ_{sgx}/θ_x as functions of Ma_x

14.4 Mach number and reference conditions

14.4.1 General

When carrying out low-pressure fan tests using standardized airways, it is usually agreed that the air velocity is sufficiently low so that its influence on parameters such as gas pressure, temperature and density can be neglected. For high- or medium-pressure fans, a distinction shall be made between the stagnation and the static values, of pressure, temperature and density, unless the reference Mach number is less than 0,15, corresponding to a velocity of standard air of $51,5 \text{ m} \cdot \text{s}^{-1}$.

The Mach number of 0,15 is considered as the limit above which this distinction shall be made.

14.4.2 Reference Mach number

In order to obtain a rapid evaluation of the limit above which compressibility phenomena due to air velocity shall be taken into account, the reference Mach number Ma_{2ref} is defined as:

$$Ma_{2ref} = \frac{v_{m2}}{c_{ref}} = \frac{q_m}{A_2 \rho_a \sqrt{\kappa R_w \Theta_{sga}}} = \frac{q_m}{A_2 \rho_{sg1} \sqrt{\kappa R_w \Theta_{sg1}}}$$

It is assumed that the air reference conditions are those in the test enclosure. The reference Mach number limit above which a distinction between the stagnation and static values of temperature, pressure and density shall be made is regarded as equal to 0,15.

14.4.3 Mach number at a section x, Ma_x

It is defined as the mean velocity at the section x, v_{mx} , divided by the velocity of sound c_x at the same section, i.e.

$$Ma_x = \frac{v_{mx}}{c_x} = \frac{q_m}{A_x \rho_x \sqrt{\kappa R_w \Theta_x}}$$

where

$$\rho_x = \frac{p_x}{R_w \Theta_x}$$

$$v_{mx} = \frac{q_m}{A_x \rho_x}$$

14.4.3.1 Calculation of Ma_x and Θ_x when p_x and Θ_{sgx} are known

Assuming that

$$M^2 = \left(\frac{q_m}{A_x} \right)^2 \cdot \frac{\kappa - 1}{2\kappa} \cdot \frac{R_w \Theta_{sgx}}{p_x^2}$$

and

$$\frac{\Theta_{sgx}}{\Theta_x} = \frac{1 + \sqrt{1 + 4M^2}}{2}$$

$$Ma_x = \sqrt{\left(\frac{\Theta_{sgx}}{\Theta_x} - 1 \right) \frac{2}{\kappa - 1}}$$

$\frac{\Theta_{sgx}}{\Theta_x}$ and Ma_x are plotted as functions of M^2 in figure 5.

14.4.3.2 Calculations of Ma_x and Θ_x when p_{sgx} and Θ_{sgx} are known

Assuming that

$$Ma_{sgx}^2 = \frac{q_m^2}{A_x^2 \rho_{sgx}^2 \kappa R_w \Theta_{sgx}} = \frac{q_m^2}{A_x^2 \kappa p_{sgx} \rho_{sgx}}$$

the Mach number Ma_x is given by:

$$Ma_x = Ma_{sgx} \sqrt{\left(1 + 1,217 Ma_{sgx}^2 + 1,369 Ma_{sgx}^4 + 10 Ma_{sgx}^6 \right)}$$

For $\kappa = 1,4$ and $Ma_{sgx} \leq 0,45$

$$\frac{\theta_{sgx}}{\theta_x} = 1 + \frac{\kappa - 1}{2} Ma_x^2$$

Figure 6 shows $\frac{Ma_x}{Ma_{sgx}}$ plotted against Ma_{sgx} .

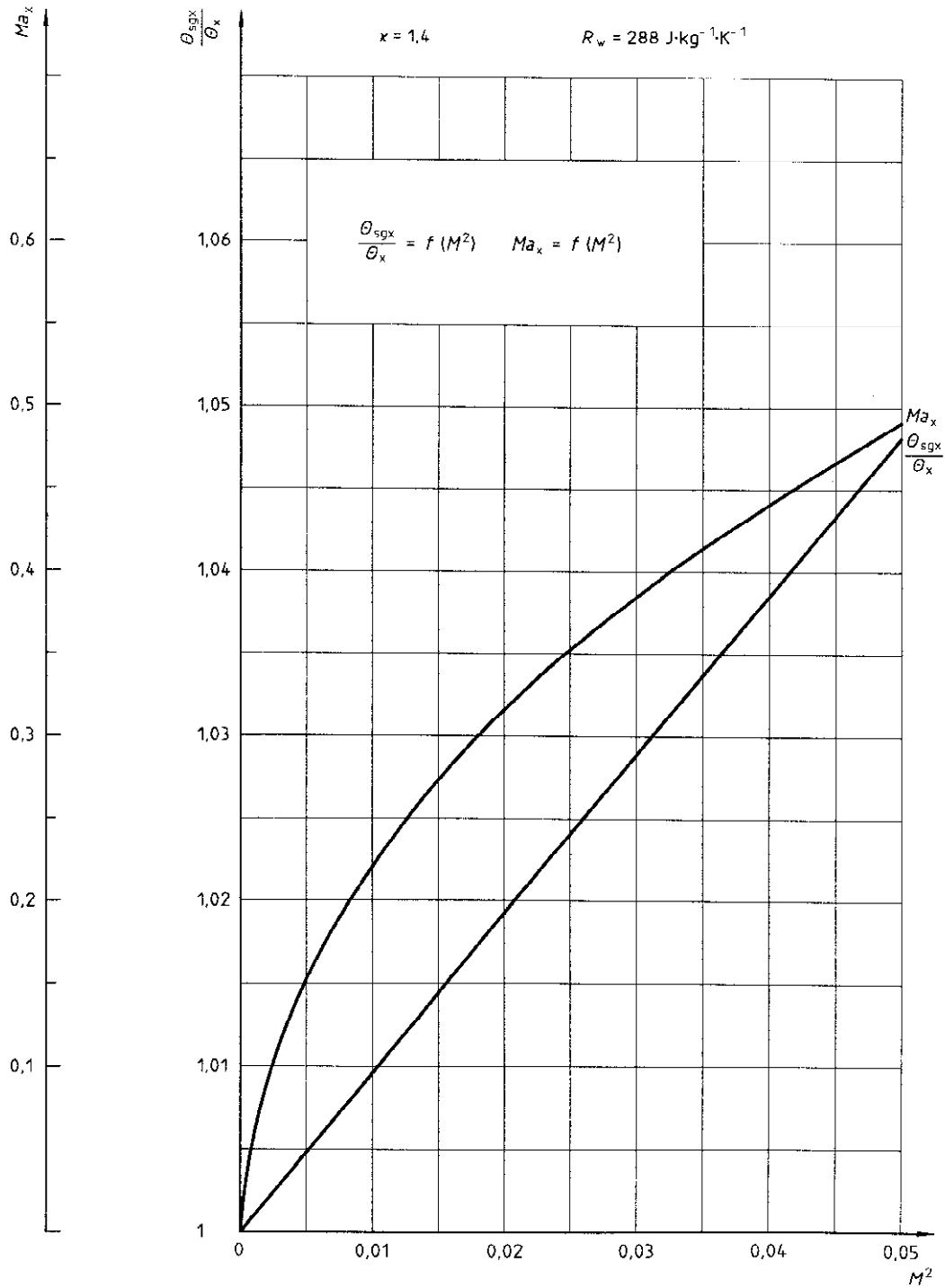


Figure 5 — Changes in Ma_x and the ratio θ_{sgx}/θ_x as functions of M^2

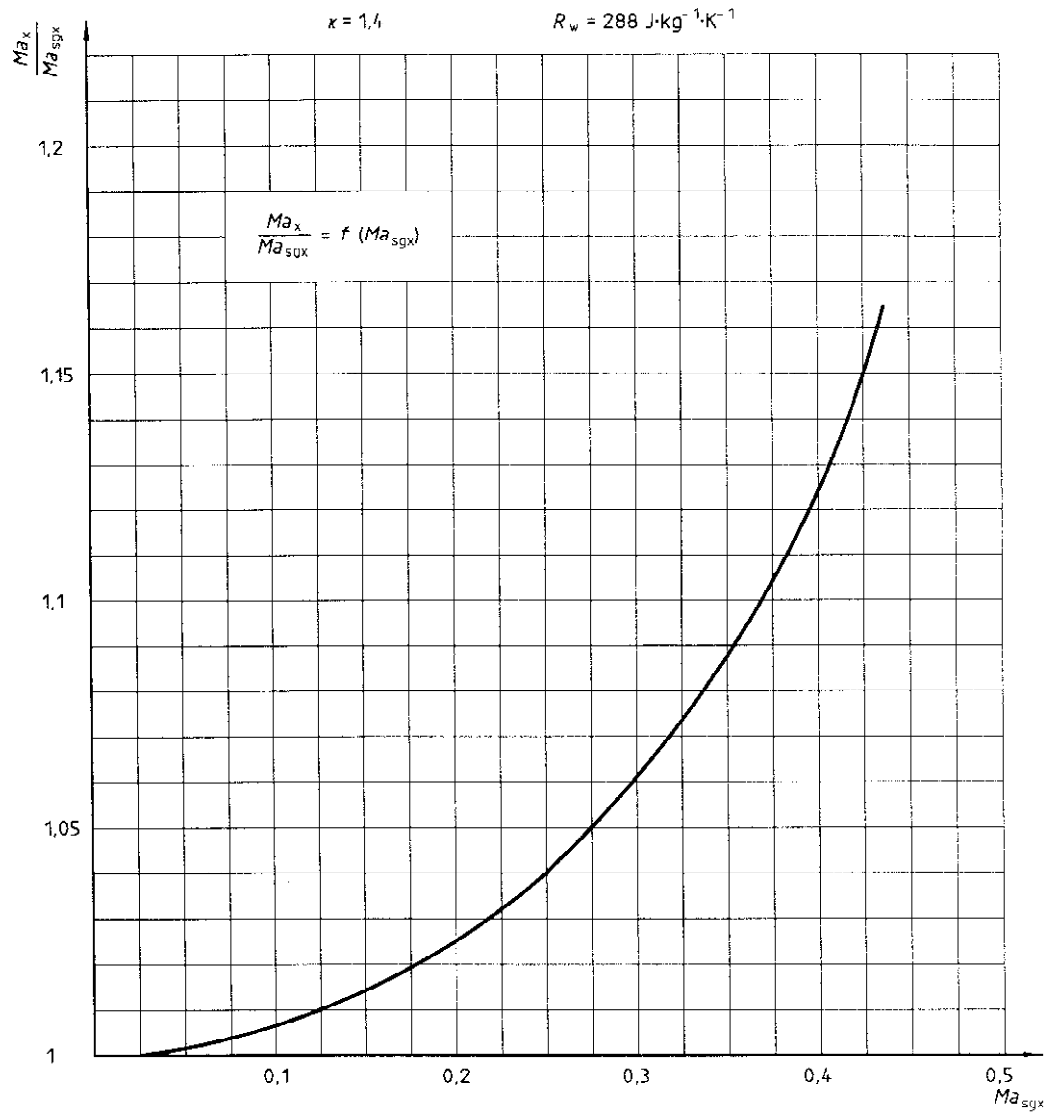


Figure 6 — Changes in the ratio Ma_x/Ma_{sgx} as a function of Ma_{sgx}

14.4.4 Calculation of the density, ρ_x at a section x and mean velocity v_{mx}

If the Mach number Ma_x is calculated in accordance with 13.4.3.1 or 13.4.3.2, the ratio:

$$\frac{\theta_{sgx}}{\theta_x}$$

is given by the following expression:

$$\frac{\theta_{sgx}}{\theta_x} = 1 + \frac{\kappa - 1}{2} Ma_x^2$$

and

$$\frac{p_{sgx}}{p_x} = \left(\frac{\theta_{sgx}}{\theta_x} \right)^{\frac{\kappa}{\kappa - 1}}$$

and

$$\frac{\rho_{sgx}}{\rho_x} = \left(\frac{\theta_{sgx}}{\theta_x} \right)^{\frac{1}{\kappa - 1}}$$

The mean velocity in the section x may be determined by the following expression:

$$v_{mx} = \frac{q_m}{A_x \rho_x}$$

where

$$\rho_x = \frac{p_x}{R_w \Theta_x} = \rho_{sgx} \left(\frac{\Theta_{sgx}}{\Theta_x} \right)^{\frac{-1}{\kappa-1}} = \frac{p_{sgx}}{R_w \Theta_{sgx}} \left[\frac{\Theta_{sgx}}{\Theta_x} \right]^{\frac{-1}{\kappa-1}}$$

14.5 Fan pressure

14.5.1 The fan pressure p_F is, by international agreement, defined as the difference between the stagnation pressure at the outlet of the fan and the stagnation pressure at the inlet of the fan, i.e.

$$p_F = p_{sg2} - p_{sg1}$$

The stagnation pressure p_{sgx} in any duct or chamber section x (with an area A_x) is given by

$$p_{sgx} = p_x + p_{dx} F_{Mx}$$

where the conventional dynamic pressure p_{dx} at section x is defined by:

$$\frac{1}{2} \rho_x v_{mx}^2 = \frac{1}{2 \rho_x} \left(\frac{q_m}{A_x} \right)^2$$

$$\text{with } \rho_x = \frac{p_x}{R_w \Theta_x}$$

The Mach factor F_{Mx} for pressure correction is given as a function of Ma_x by the expression

$$F_{Mx} = \frac{p_{sgx} - p_x}{\frac{1}{2} \rho_x v_{mx}^2} = 1 + \frac{Ma_x^2}{4} + \frac{Ma_x^4}{40} + \frac{Ma_x^6}{1600} + \dots$$

for $\kappa = 1,4$ (see 3.21)

and is plotted in figure 4 as a function of Ma_x .

NOTES

25 The difference between the gauge stagnation pressure p_{esgx} at section x of the test airway and the total pressure p_{tx} used in earlier standards is very small at low velocities when the Mach number Ma_x is less than 0,15 ($= 0,006 p_{dx}$).

26 The fan pressure may be also defined as the difference between the gauge stagnation pressure at the outlet of the fan and the gauge stagnation pressure at the inlet of the fan.

$$p_F = p_{esg2} - p_{esg1} = p_{e2} + p_{d2} F_{M2} - (p_{e1} + p_{d1} F_{M1})$$

where $p_{e1} \approx 0$

14.5.2 The fan static pressure p_{sF} is, by international agreement, defined as the difference between the static pressure at the outlet of the fan and the stagnation pressure at the inlet of the fan.

$$p_{sF} = p_2 - p_{sg1}$$

When p_{sgx} , Θ_{sgx} , q_m and A_x are known for a section x , p_x is calculated by the following method.

After the determination of Ma_x in accordance with 14.4.3.2, p_x is given by:

$$\rho_x = \frac{p_{sgx}}{\left(1 + \frac{\kappa-1}{2} Ma_x^2 \right)^{\frac{1}{\kappa-1}}} = \rho_{sgx} \frac{p_x}{p_{sgx}}$$

$$\frac{\rho_x}{\rho_{sgx}} = \left(1 + \frac{\kappa - 1}{2} Ma_x^2\right)^{\frac{-1}{\kappa - 1}}$$

is shown on figure 7 as a function of Ma_x

and

$$p_x = p_{sgx} - p_{dx} F_{Mx} = p_{sgx} - \frac{1}{2\rho_x} \left(\frac{q_m}{A_x}\right)^2 F_{Mx}$$

with F_{Mx} being determined in accordance with 14.5.1 and figure 4.

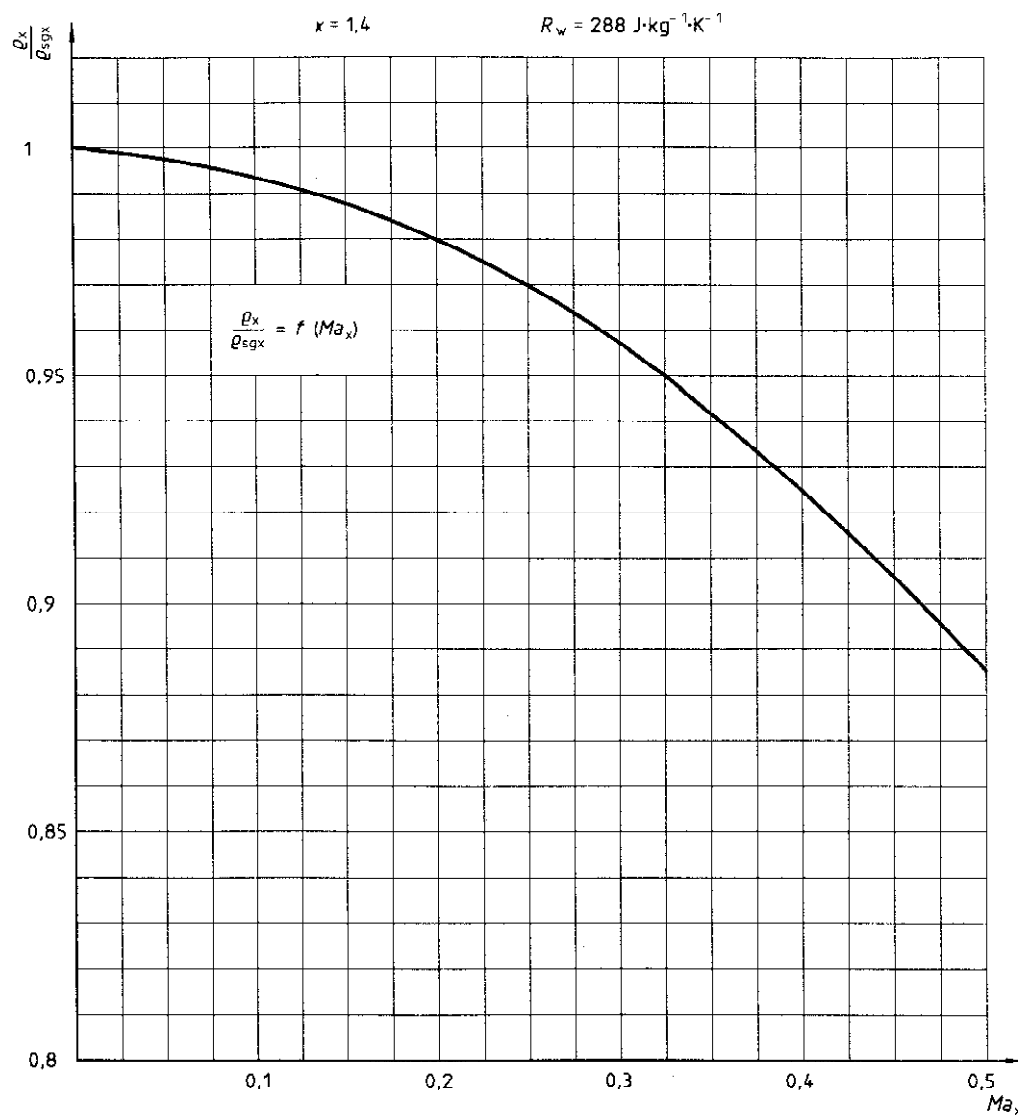


Figure 7 — Changes in the ratio ρ_x/ρ_{sgx} as a function of Ma_x

14.6 Calculation of stagnation pressure at a reference section of the fan from gauge pressure, p_{ex} , measured at a section x of the test duct

Assuming

A_n the area of the fan inlet or outlet section ($n = 1$ for inlet, $n = 2$ for outlet), and

A_x the area of the measuring section of the test duct ($x = 3$ for inlet, $x = 4$ for outlet). (See figure 8.)

The absolute pressure at section x is given by:

$$p_x = p_{ex} + p_a$$

and, in accordance with 14.3.2, 14.3.3, 14.3.4,

$$\Theta_{sgx} = \Theta_{sgn}$$

Ma_x and Θ_x are calculated in accordance with 14.3.5, 14.4.3.1

$$\rho_x = \frac{p_x}{R_w \Theta_x}$$

$$v_{mx} = \frac{q_m}{A_x \rho_x}$$

The stagnation pressure at reference section n is given by

$$p_{sgn} = p_x + \frac{1}{2} \rho_x v_{mx}^2 F_{Mx} [1 + (\zeta_{n-x})_x]$$

where

$(\zeta_{n-x})_x$ is the energy loss coefficient between section n and section x calculated for section x in accordance with 30.6.

$(\zeta_{n-x})_x \geq 0$ for an outlet test duct

$(\zeta_{n-x})_x \leq 0$ for an inlet test duct

NOTES

27 p_{ex} is negative for an inlet test duct or an inlet chamber.

28 It may be written:

$$p_{esgn} = p_{ex} + \frac{1}{2} \rho_x v_{mx}^2 F_{Mx} [1 + (\zeta_{n-x})_x]$$

14.6.1 Calculation of the fluid pressure at a reference section of the fan

The fluid or static pressure in a reference section of the fan, p_n , is calculated in accordance with 14.5.2 from p_{sgn} , Θ_{sgn} and A_n .

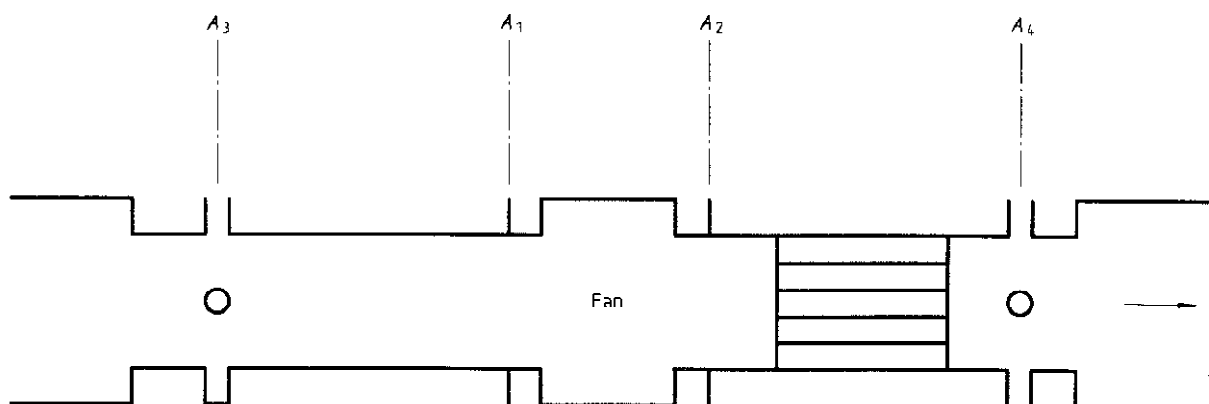


Figure 8 — Measurement planes and reference planes

14.7 Inlet volume flowrate

The methods of flow measurement in this International Standard lead to a determination of the mass flowrate q_m . In the absence of leakage, q_m will be constant throughout the airway system.

The inlet volume flowrate can be expressed as the volume flowrate under inlet stagnation conditions, i.e.

$$q_{Vsg1} = \frac{q_m}{\rho_{sg1}}$$

where

$$\rho_{sg1} = \frac{p_{sg1}}{R_w \Theta_{sg1}}$$

14.8 Fan air power and efficiency

Three methods are proposed:

- the first is derived from the concept of work per unit mass;
- the two others use the concepts of volume flowrate and pressure with a correction factor to take into account the influence of fluid compressibility.

These three methods give the same results within a few parts per thousand for a pressure ratio equal to 1,3.

14.8.1 Calculation of fan air power and efficiency from fan work per unit mass

One has

$$\begin{aligned} y &= \frac{p_2 - p_1}{\rho_m} + \frac{v_{m2}^2}{2} - \frac{v_{m1}^2}{2} \\ &= \frac{p_2 - p_1}{\rho_m} + \frac{1}{2} \left(\frac{q_m}{\rho_2 A_2} \right)^2 - \frac{1}{2} \left(\frac{q_m}{\rho_1 A_1} \right)^2 \end{aligned}$$

where

$$\rho_m = \frac{\rho_1 + \rho_2}{2}$$

and

$$\rho_1 = \frac{p_1}{R_w \Theta_1}$$

$$\rho_2 = \frac{p_2}{R_w \Theta_2}$$

ρ_1 and ρ_2 being calculated in accordance with 14.5.2.

The fan air power P_u is equal to the product $q_m y$.

The various efficiencies are calculated from P_u and the various types of power supplied to the fan, i.e.

impeller power, P_r ,

shaft power, P_a ,

motor output power, P_o ,

motor input power, P_e ;

$$\eta_r = \frac{P_u}{P_r}$$

$$\eta_a = \frac{P_u}{P_a}$$

$$\eta_o = \frac{P_u}{P_o}$$

$$\eta_e = \frac{P_u}{P_e}$$

14.8.2 Calculation of fan air power and efficiency from fan volume flowrate and fan pressure

One has:

$$P_u = q_{Vsg1} p_F k_p$$

where

q_{Vsg1} is the volume flowrate at inlet stagnation conditions;

p_F is the fan pressure, $p_{sg2} - p_{sg1}$;

k_p is the correction factor for compressibility effect.

The various efficiencies are calculated from the various types of power supplied in the same way as in 14.8.1.

Two methods for the calculation of the coefficient k_p are proposed. They give exactly the same results.

NOTE 29 The fan air power calculated by this method is always less than that calculated in accordance with 14.8.1 ($\approx 2 \times 10^{-3}$ to 3×10^{-3}).

14.8.2.1 Calculation of compressibility coefficient, k_p

The pressure ratio r is calculated as

$$r = 1 + \frac{p_F}{p_{sg1}}$$

where

p_F is the fan pressure according to 14.5.1;

p_{sg1} is the stagnation pressure at the fan inlet.

Assuming that

$$Z_k = \frac{\kappa - 1}{\kappa} \frac{\rho_{sg1} P_r}{q_m p_F} = \frac{\kappa - 1}{\kappa} \frac{P_r}{q_{Vsg1} p_F}$$

k_p is given by

$$k_p = \frac{Z_k \log_{10} r}{\log_{10} [1 + Z_k (r - 1)]}$$

and is plotted in figure 9 as a function of the pressure ratio r and of the coefficient Z_k .

NOTE 30 k_p and ρ_{sg1}/ρ_{msg} differ by less than 2×10^{-3} , where $\rho_{msg} = \frac{\rho_{sg1} + \rho_{sg2}}{2}$

The compressibility coefficient k_p may be also determined using the following expression:

$$k_p = \frac{\ln(1+x)}{x} \frac{Z_p}{\ln(1+Z_p)}$$

where

$$x = \frac{P_F}{P_{sg1}} = r - 1$$

and

$$Z_p = \frac{\kappa - 1}{\kappa} \frac{P_r}{q_{Vsg1} P_{sg1}}$$

k_p is plotted in figure 10 as a function of x and Z_p .

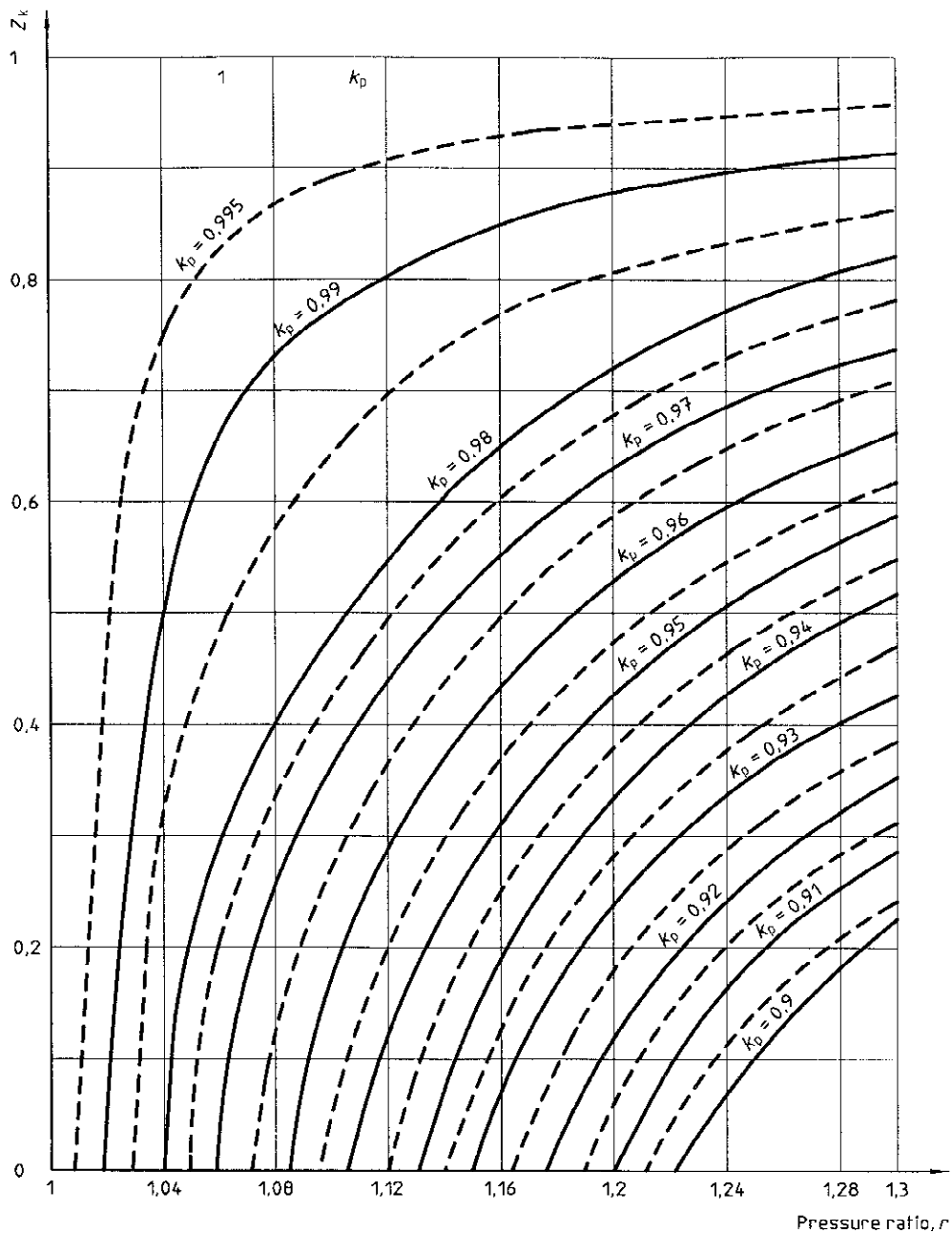


Figure 9 — Chart for determination of the compressibility coefficient, k_p

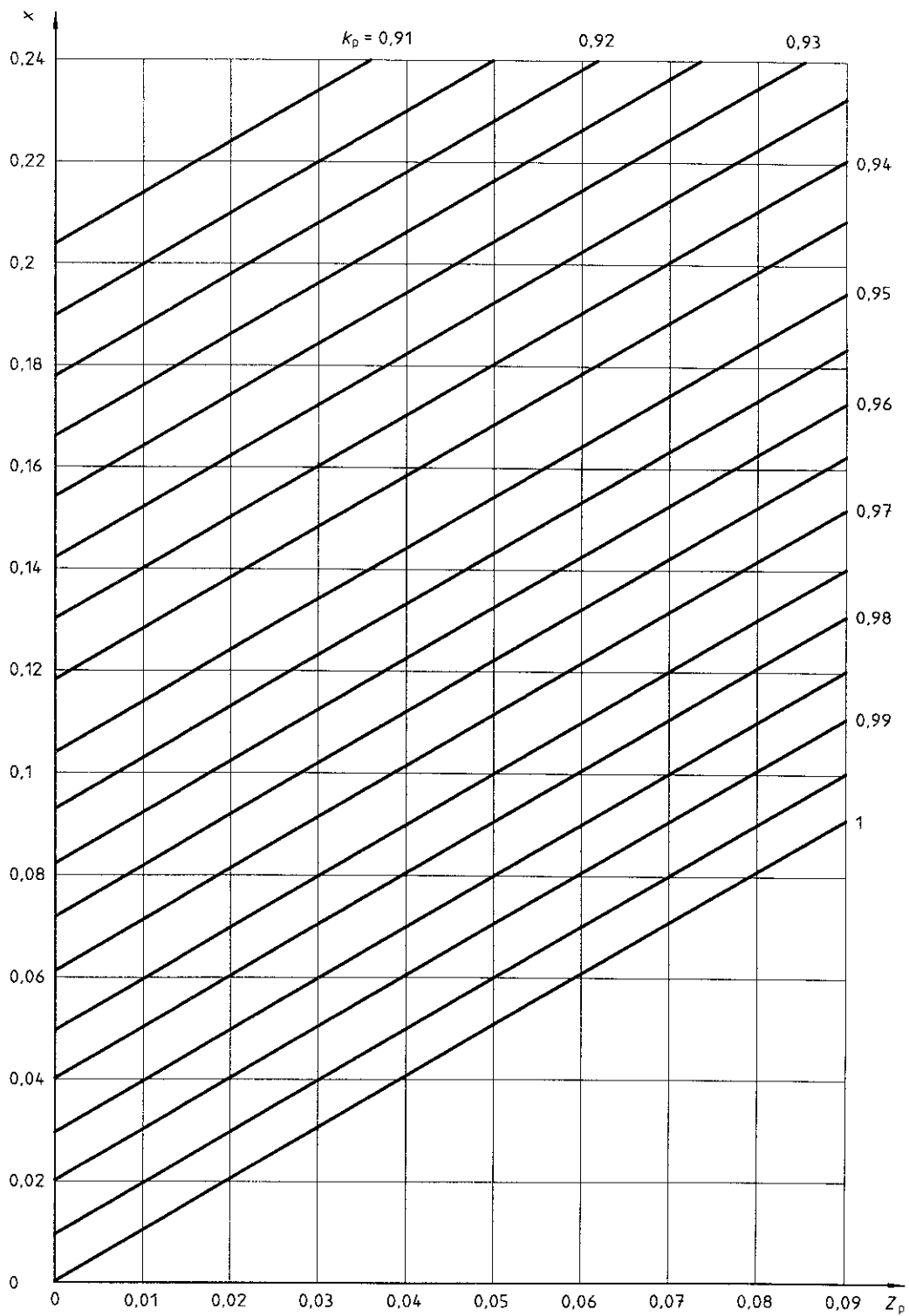


Figure 10 — Chart for the determination of compressibility coefficient, k_p

14.8.2.2 Determination of the fan work per unit mass from the fan air power, P_u

The fan work per unit mass y may be determined using the following expression:

$$y = \frac{P_u}{q_m} = \frac{P_u}{q_{Vsg1} \rho_{sg1}}$$

where

$$P_u = q_{Vsg1} p_F k_p$$

in accordance with 14.8.2 and 14.8.2.1.

14.8.3 Conventional static efficiency

14.8.3.1 Calculation of fan static air power and of static efficiency from fan static work per unit mass

One has

$$y_s = \frac{p_2 - p_1}{\rho_m} - \frac{v_{m1}^2}{2}$$

where

$$\rho_m = \frac{\rho_1 + \rho_2}{2}$$

The fan static air power is equal to the product $q_m y_s$ therefore

$$P_{us} = q_m y_s$$

The various efficiencies are calculated from P_{us} in the same way as in 14.8.1.

14.8.3.2 Calculation of the fan static air power from the fan volume flowrate and fan static pressure

The fan static power is given by the following expression:

$$P_{us} = q_{Vsg1} p_{sF} k_{ps}$$

where k_{ps} is calculated in accordance with 14.8.2.1,

and

$$r = 1 + \frac{p_{sF}}{p_{sg1}}$$

$$x = \frac{p_{sF}}{p_{sg1}} = r - 1$$

and

$$Z_k = \frac{\kappa - 1}{\kappa} \frac{\rho_{sg1} P_r}{q_m p_{sF}}$$

and

$$Z_p = \frac{\kappa - 1}{\kappa} \frac{P_r}{q_{Vsg1} p_{sg1}}$$

Static efficiencies are determined from P_{us} in accordance with 14.8.1.

NOTE 31 The fan static power calculated by this method is always greater than that calculated in accordance with 14.8.3.1 ($\approx 2 \times 10^{-3}$ to 4×10^{-3}).