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Industrial fans — Performance testing using standardized airways

Ventilateurs industriels — Essais aérauliques sur circuits normalisés

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Foreword

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

Draft International Standards adopted by the technical committees are circulated to the member bodies for voting. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

International Standard ISO 5801 was prepared by Technical Committee ISO/TC 117, *Industrial fans*, Subcommittee SC 1, *Fan performance testing using standardized airways*.

Annexes A, B and C form an integral part of this International Standard. Annexes D and E are for information only.

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.

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

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

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

3.7 isentropic exponent, κ:

For an ideal gas and an isentropic process,

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

$\kappa = 1{,}4$ 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 ($m \cdot s^{-1}$).

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

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

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.

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

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

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

where

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

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_a: 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:

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

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_ρ: Fluid density at the fan inlet divided by the mean fluid density in the fan.

$$k_\rho = \frac{2\rho_1}{\rho_1 + \rho_2}$$

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.

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

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.

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

3.66 friction loss coefficient, $\left(\zeta_{x-y}\right)_y$: 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

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

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 tappings	mm
b	Width of the rectangular section of a duct	m
C	Discharge coefficient	—
c	Velocity of sound $c = \sqrt{\kappa R_w \Theta_x}$	$m \cdot s^{-1}$
c_p	Specific heat at constant pressure	$J \cdot kg^{-1} \cdot K^{-1}$
c_V	Specific heat at constant volume	$J \cdot kg^{-1} \cdot K^{-1}$
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 $\dfrac{4 \times \text{area of cross-section}}{\text{perimeter of cross-section}}$	m
D_x	Internal diameter of a circular conduit in the x plane	m
D_r	Outside diameter of the impeller	m
F_{Mx}	Mach factor for correction of dynamic pressure at section x	—
g	Gravitational acceleration	$m \cdot s^{-2}$
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 $\left(i_{kx} = \dfrac{V_{mx}^2}{2y}\right)$	—
k_c	Resulting coefficient used in the conversion of test results	—
k_ρ	Fan density ratio $\left(k_\rho = \dfrac{2\rho_1}{\rho_1 + \rho_2}\right)$	—
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	—
$Ma_{x\,ref}$	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	$r \cdot s^{-1}$
N	Rotational speed of impeller	$r \cdot min^{-1}$
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	$kg \cdot s^{-1}$
q_V	Volume flowrate	$m^3 \cdot s^{-1}$
q_{Vsg1}	Volume flowrate at stagnation conditions upstream of the fan inlet corresponding to standard conditions of use	$m^3 \cdot s^{-1}$
q_{Vx}	Volume flowrate at section x	$m^3 \cdot s^{-1}$
r	Pressure ratio	—
r_d	Pressure ratio for a flowmeter $r_d = p_{do} / p_u$	—
$r_{\Delta p}$	$\dfrac{\Delta p}{p_{do}}$ for a flowmeter	—
R	Gas constant of dry air or gas	$J \cdot kg^{-1} \cdot K^{-1}$
R_w	Gas constant of humid air or gas	$J \cdot kg^{-1} \cdot K^{-1}$
Re_{Dx}	Reynolds number at section x	—
t_a	Ambient temperature	°C
t_b	Barometer temperature	°C
t_d	Dry-bulb thermometer temperature	°C
t_w	Wet-bulb thermometer temperature	°C
t_x	Static temperature at section x	°C
t_{sgx}	Stagnation temperature at section x	°C
u	Peripheral velocity of impeller, or tip speed	$m \cdot s^{-1}$
u_x	Relative uncertainty of x	%
U_x	Absolute uncertainty of x	same as x
v	Velocity of gas at a point	$m \cdot s^{-1}$
v_{mx}	Average velocity of the gas at section x	$m \cdot s^{-1}$
y	Fan work per unit mass	$J \cdot kg^{-1}$
y_s	Fan static work per unit mass	$J \cdot kg^{-1}$
Z	Compressibility factor in equation of state $Z = \dfrac{p}{\rho R_w \Theta_x}$; 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 $\eta_a = \dfrac{P_u}{P_a}$	—
η_e	Overall efficiency $\eta_e = \dfrac{P_u}{P_e}$	—
η_o	Fan motor shaft efficiency $\eta_o = \dfrac{P_u}{P_o}$	—
η_r	Fan efficiency $\eta_r = \dfrac{P_u}{P_r}$	—
η_{sr}	Fan static efficiency $\eta_{sr} = \dfrac{P_{us}}{P_r}$	—
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	$kg \cdot m^{-3}$
ρ_x	Mean density of gas at section x	$kg \cdot m^{-3}$
ρ_m	Mean density of gas in the fan	$kg \cdot m^{-3}$
Φ	Flow coefficient $\Phi = \dfrac{q_m}{\rho_m D_r^2 u}$	—
Ψ	Fan work per unit mass coefficient $\Psi = \dfrac{y}{u^2}$	—

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

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 tappings 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 $J \cdot kg^{-1}$ corresponding to an increase in fan pressure approximately equal to 30 000 Pa for a mean density in the fan of 1,2 $kg \cdot m^{-3}$.

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 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 ± 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 ± 0,01 m · s^{-2}) and for room temperature (within ± 5 °C).

Barometers of the aneroid or pressure transducer type may be used provided they have a calibrated accuracy of ± 200 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 ± 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 ± 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 tappings 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 tappings

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 tappings constructed in accordance with 7.3.

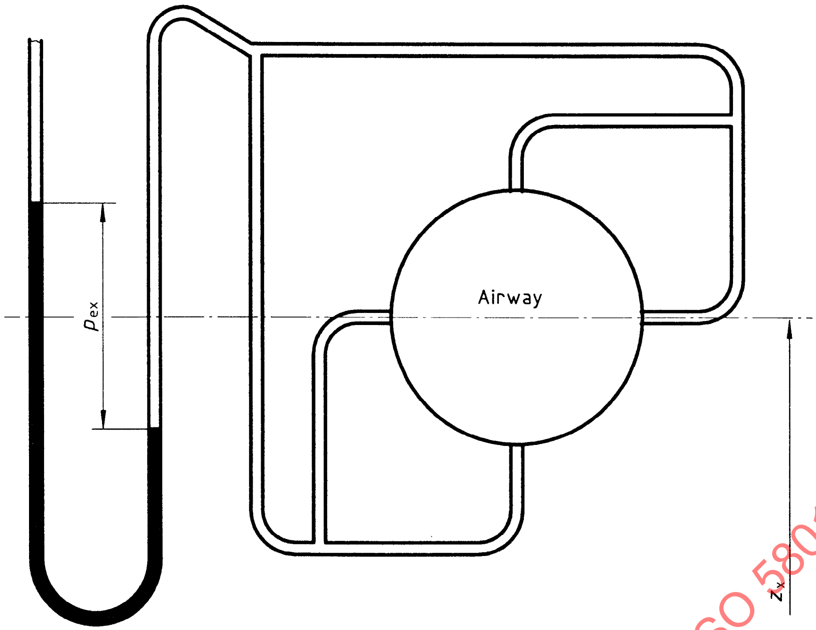


Figure 1 — Tapping connections to obtain average static pressure and altitude of manometer

7.3 Construction of tappings

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 tappings 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 tappings 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 tappings 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 tappings 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 $p_{ex} \leqslant$ 1 000 Pa or 2% for 1 000 Pa $< p_{ex} <$ 30 000 Pa, p_{ex} being the mean gauge pressure, the tappings and manometer connections should be examined for defects. If none are found, eight pressure tappings 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.

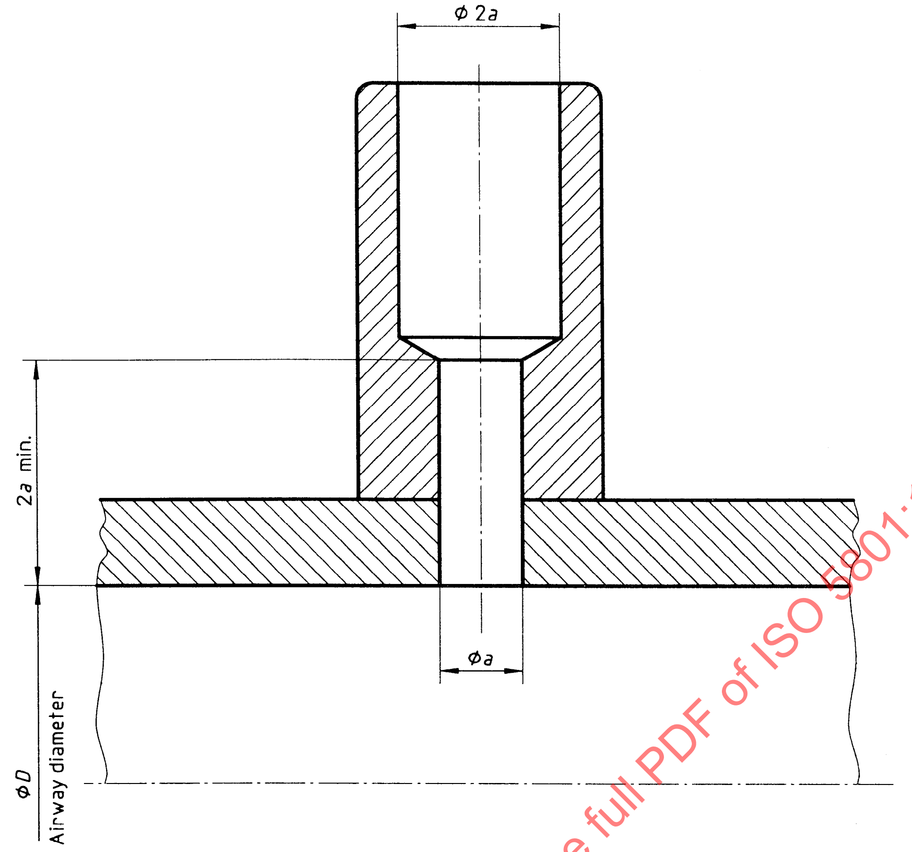


Figure 2 — Construction of wall pressure tappings

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

If the measurement is taken in a section where the air velocity is less than 25 $m \cdot s^{-1}$, 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 $m \cdot s^{-1}$ 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 $m \cdot s^{-1}$. 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 ± 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 ± 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 ± 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 ± 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 ± 0,2 %. The length of the torque arm shall be determined to an accuracy of ± 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 ± 0,2 %. The length of the torque arm shall be determined to an accuracy of ± 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 $^{+10}_{-1}$ %, 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 ± 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

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

NOTE 22 For standard air:

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

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

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

$h_u = 0{,}40$

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

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:

$$p_v = (p_{sat})_{tw} - p_a \cdot A_w (t_d - t_w)(1 + 0{,}001\,15\, t_w)$$

where

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

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

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

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

$(p_{sat})_{tw}$ 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 °C to 49,5 °C.

p_{sat} may be obtained from the following expression, between 0 °C and 30 °C:

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

or, between 0 °C and 100 °C:

$$p_{sat} = 610{,}8 + 44{,}442\, t_w + 1{,}413\,3\, t_w^2 + 0{,}027\,68\, t_w^3 + 2{,}556\,67\,e^{-4}\, t_w^4 + 2{,}891\,66\,e^{-6}\, t_w^5$$

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

$$p_v = h_u\,(p_{sat})_{td}$$

where $(p_{sat})_{td}$ 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 °C to + 100 °C to obtain the dynamic viscosity, in pascal seconds:

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

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:

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

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 ± 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.

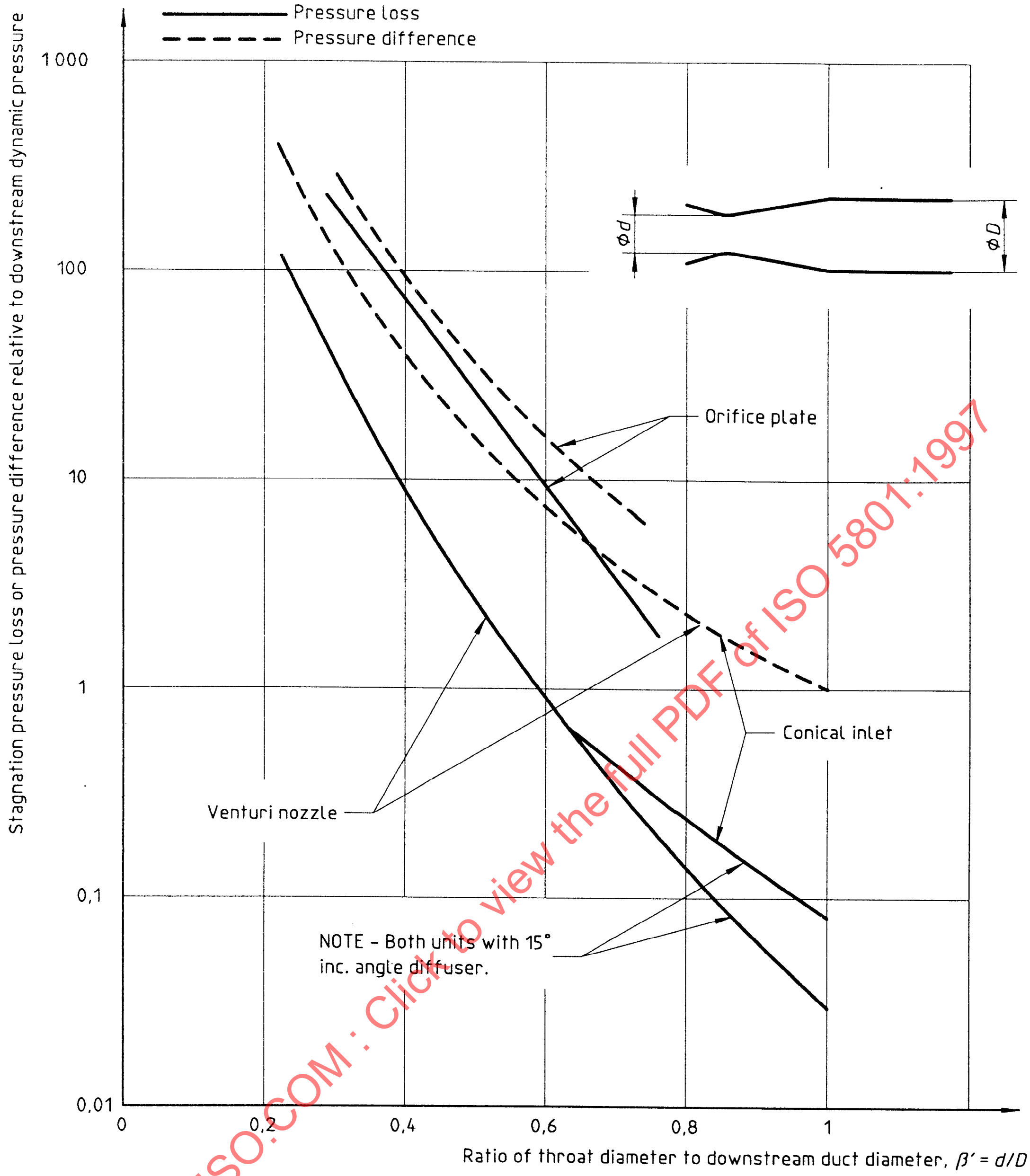


Figure 3 — Pressure loss and pressure difference of standard primary systems

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 ($m^3 \cdot s^{-1}$).

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 $J \cdot kg^{-1} \cdot K^{-1}$ 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 $m \cdot s^{-1}$ 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.

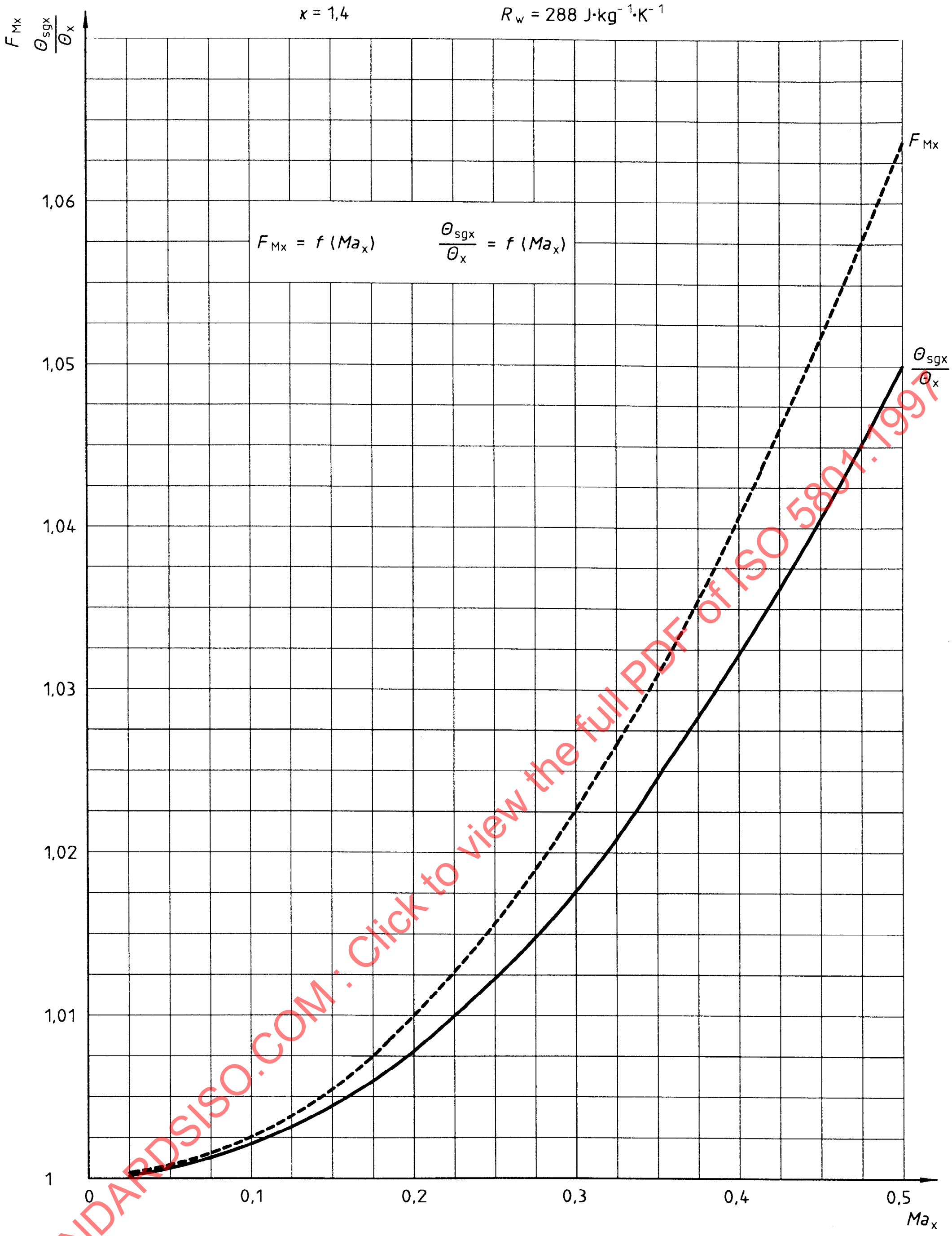


Figure 4 — Changes in F_{Mx} and the ratio Θ_{sgx}/Θ_x as functions of Ma_x

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 m · 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 $Ma_{sgx} \leqslant 0{,}45$

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

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

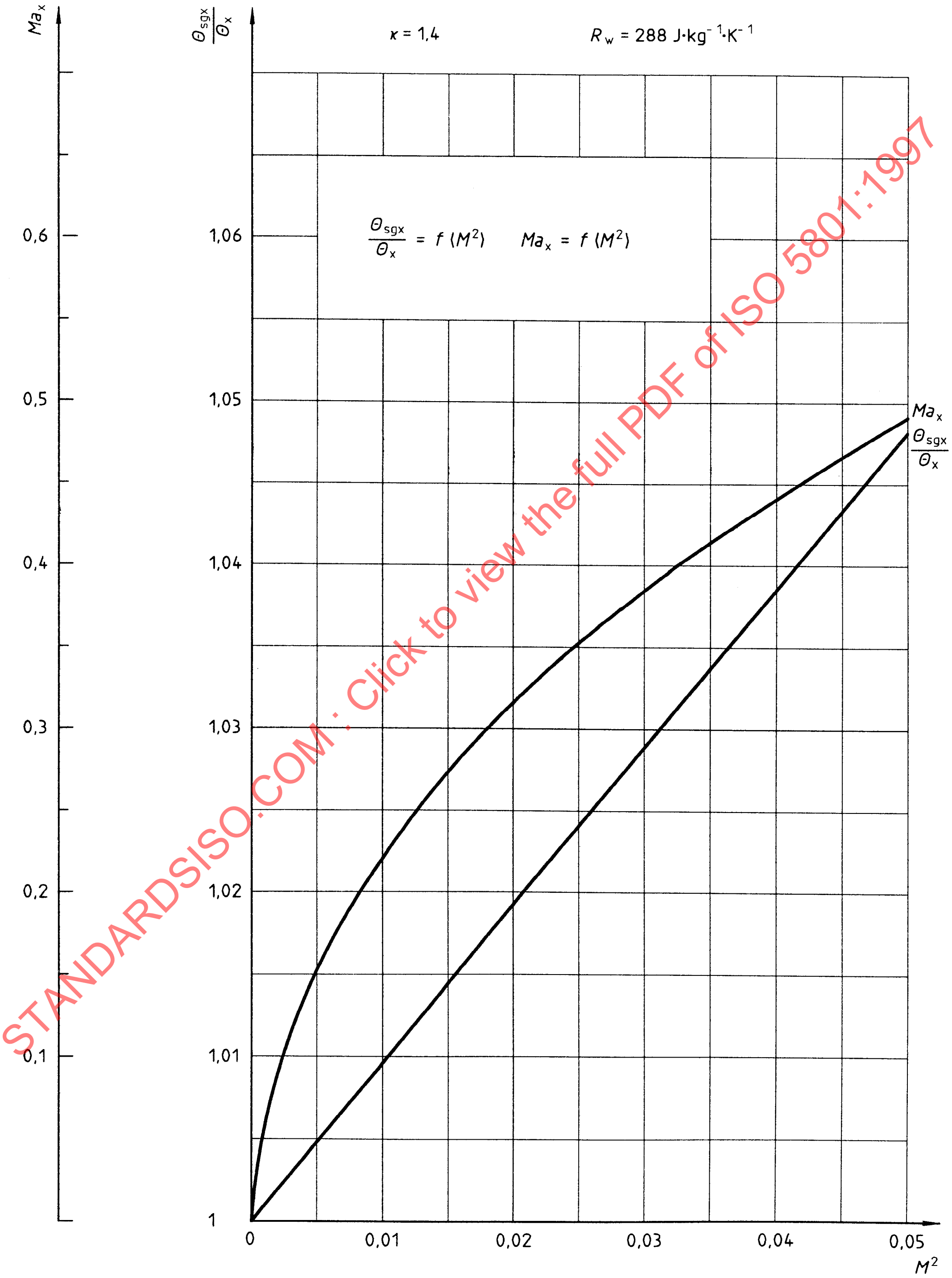


Figure 5 — Changes in Ma_x and the ratio Θ_{sgx}/Θ_x as functions of M^2

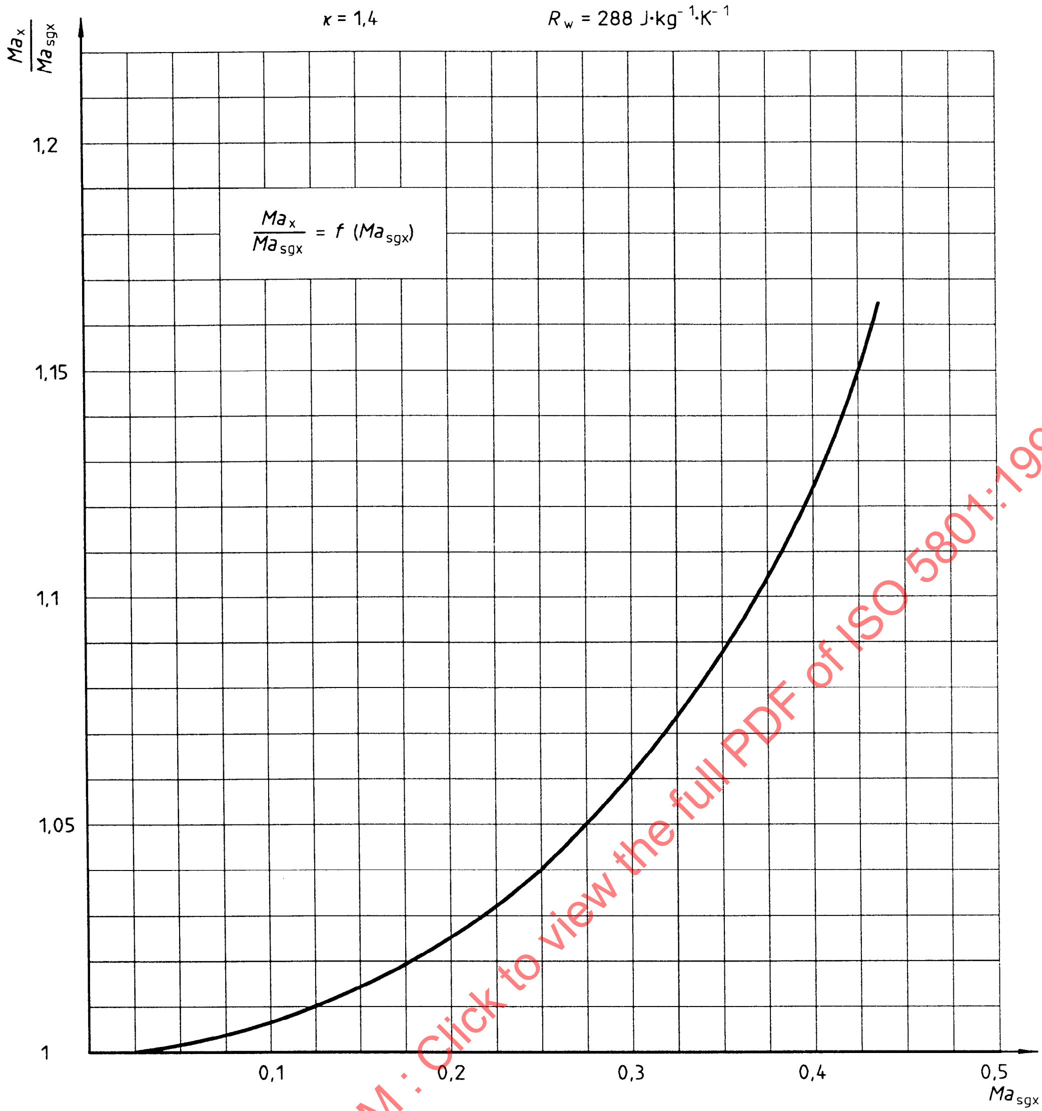


Figure 6 — Changes in the ratio Ma_x / Ma_{sgx} as a function of Ma_{sgx}

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:

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

with $\rho_x = \dfrac{p_x}{R_w \Theta_x}$

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

$$F_{Mx} = \frac{p_{sgx} - p_x}{\frac{1}{2} \rho_x v_{mx}^2} = 1 + \frac{Ma_x^2}{4} + \frac{Ma_x^4}{40} + \frac{Ma_x^6}{1\,600} + \dots$$

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.

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

where $p_{e1} \leqslant 0$

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, ρ_x is given by:

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

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

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.

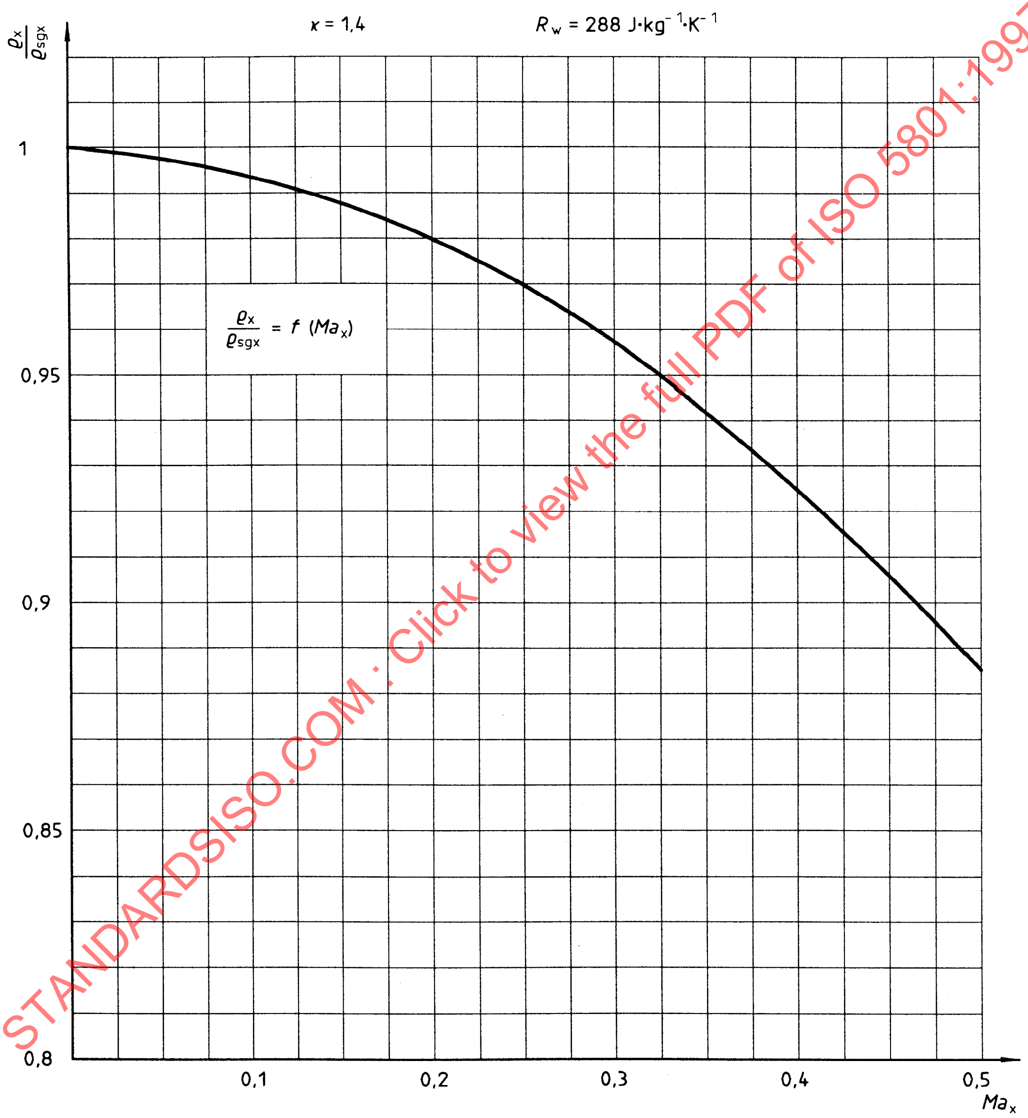


Figure 7 — Changes in the ratio ρ_x/ρ_{sgx} as a function of Ma_x

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

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

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.

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

$(\zeta_{n-x})_x \leqslant 0$ 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:

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

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.

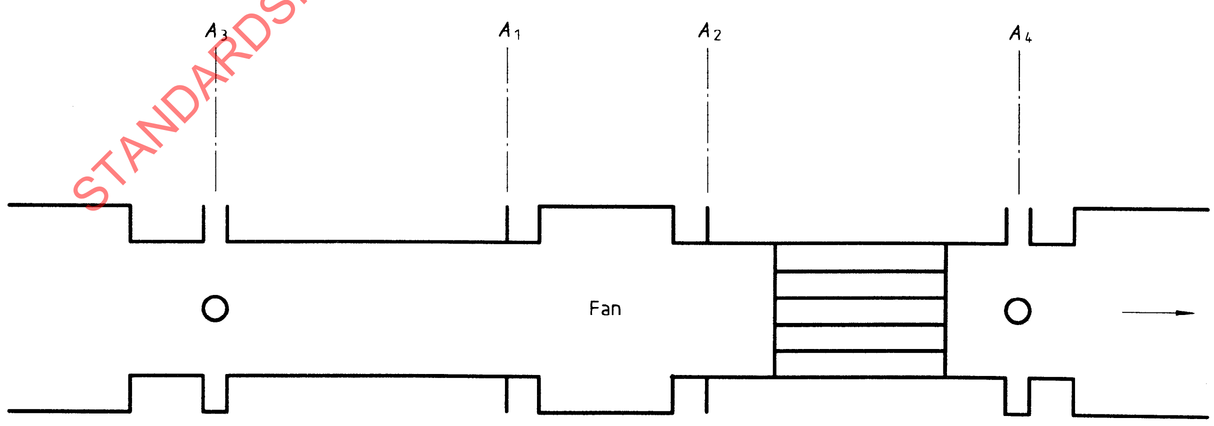


Figure 8 — Measurement planes and reference planes

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

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

and is plotted in figure 9 as a fonction 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.

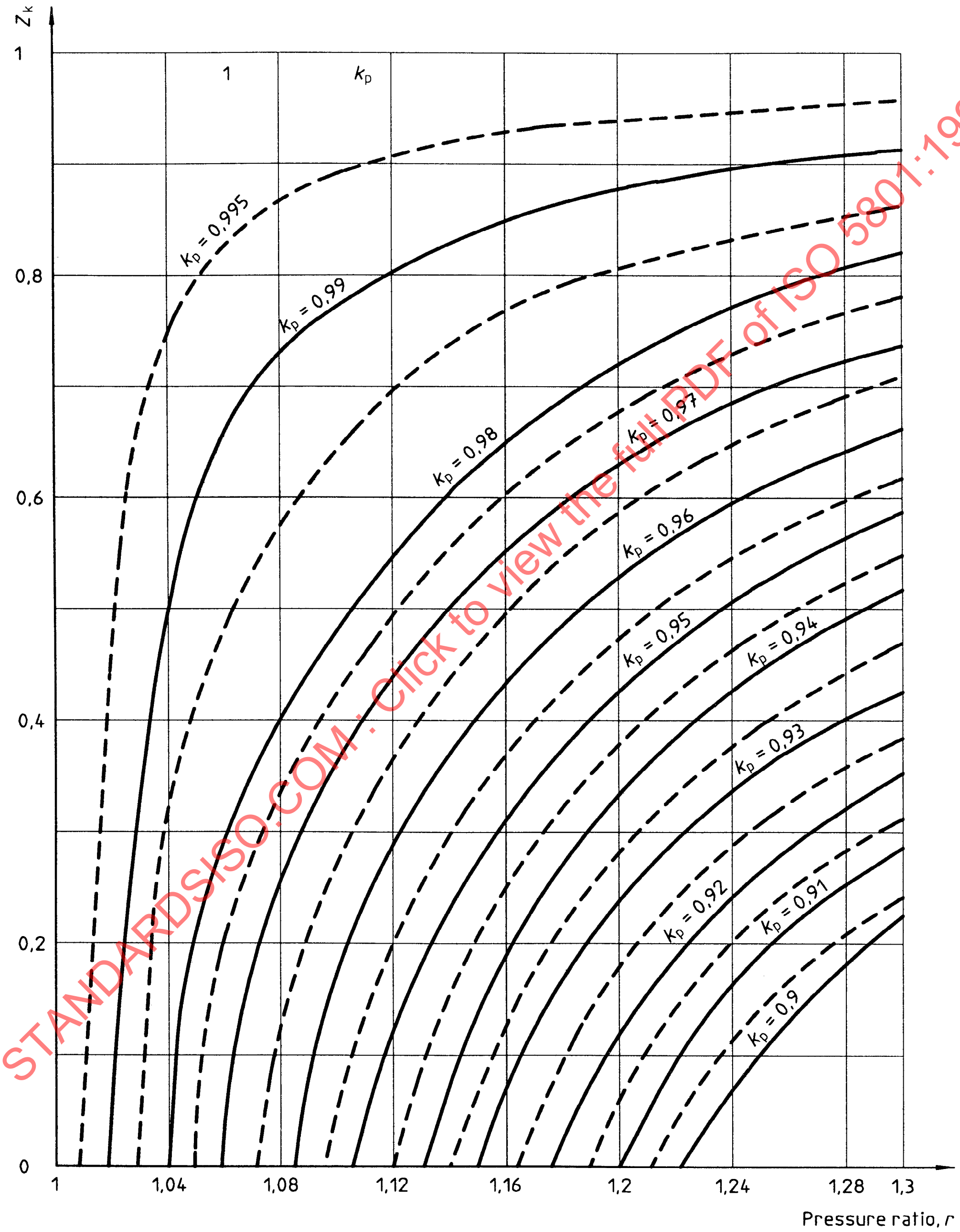


Figure 9 — Chart for determination of the compressibility coefficient, k_p

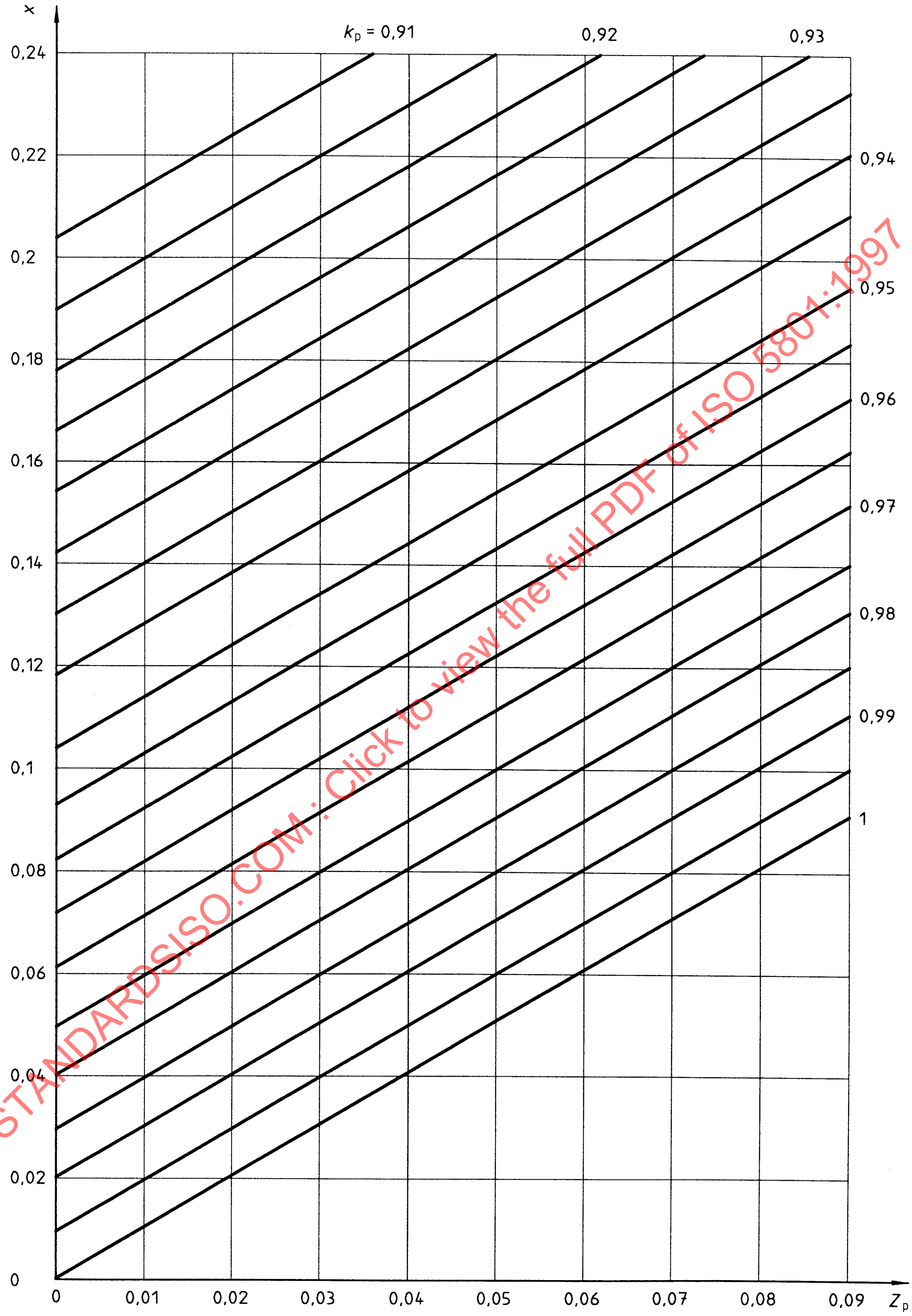


Figure 10 — Chart for the determination of compressibility coefficient, k_p

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}).

14.8.3.3 Determination of the fan static work per unit mass y_s from the fan static air power P_{us}

The fan static work per unit mass y_s is determined using the following expression:

$$y_s = \frac{P_{us}}{q_m} = \frac{P_{us}}{q_{Vsg1}\rho_{sg1}}$$

14.8.4 Determination of the kinetic index at the fan inlet, i_{k1} or at the fan outlet i_{k2}

The kinetic index i_{kx} is given by the following expressions:

— at the fan inlet:

$$i_{k1} = \frac{v_{m1}^2}{2y}$$

— at the fan outlet:

$$i_{k2} = \frac{v_{m2}^2}{2y}$$

14.9 Simplified calculation methods

When the reference Mach number Ma_{2ref} and/or the fan pressure do not exceed certain values, simplified calculation methods may be used.

14.9.1 Reference Mach number Ma_{2ref} less than 0,15 but fan pressure p_F greater than 2 000 Pa

In this case:

— the Mach factor F_{Mx} may be taken as 1;

— the stagnation temperature Θ_{sgx} and the static or fluid temperature Θ_x may be taken as equal and measured.

14.9.1.1 Determination of mass flowrate

The stagnation and static temperatures may be considered as equal and the temperature upstream of the flowmeter may be measured.

The determination of flowrate does not need a trial and error procedure for the calculation of the density upstream of the flowmeter as described in clauses 22 to 27.

$$\Theta_u = t_u + 273{,}15 = \Theta_{sgu}$$

$$p_u = p_{eu} + p_a$$

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

However, the Reynolds number correction on the flow coefficient of the flowmeter α should be applied after a first determination of the mass flowrate and of the corresponding Reynolds number.

14.9.1.2 Determination of the stagnation pressure at section x, p_{sgx}

In accordance with the assumptions described in 14.9.1 we have

— for an outlet duct:

$$\Theta_2 = \Theta_4 = \Theta_{sg2} = \Theta_{sg4} = \Theta_{sg1} + \frac{P_r \text{ or } P_e}{q_m c_p}$$

(the temperature Θ_{sg4} can be measured),

— for an inlet duct:

$$\Theta_1 = \Theta_{sg1} = \Theta_3 = \Theta_{sg3}$$

The absolute pressure in the section of measurement is given by the expression

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

and

$$p_{sgx} = p_x + \frac{1}{2}\rho_x v_{mx}^2$$

or

$$p_{sgx} = p_x + \frac{1}{2\rho_x}\left(\frac{q_m}{A_x}\right)^2$$

where

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

The gauge stagnation pressure, p_{esgx}, is given by the following expression:

$$p_{esgx} = p_{ex} + \frac{1}{2}\rho_x v_{mx}^2$$

$$= p_{ex} + \frac{1}{2\rho_x}\left(\frac{q_m}{A_x}\right)^2$$

14.9.1.3 Determination of the stagnation pressure at a reference section of the fan from the fluid pressure measured at section x, p_{ex}

Assuming

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

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

Given that

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

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

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

the stagnation pressure at section n is given by

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

or

$$p_{sgn} = p_x + \frac{1}{2\rho_x}\left(\frac{q_m}{A_x}\right)^2\left[1 + (\zeta_{n-x})_x\right]$$

where

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

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

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

NOTES

32 The gauge pressure p_{ex} is negative for an inlet test duct or an inlet test chamber.

33 The gauge stagnation pressure at section n is given by the following expression:

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

$$= p_{ex} + \frac{1}{2\rho_x}\left(\frac{q_m}{A_x}\right)^2\left[1+\left(\zeta_{n-x}\right)_x\right]$$

14.9.1.4 Determination of the static pressure at a reference section of the fan

In accordance with 14.9.1.2: $\Theta_n = \Theta_{sgn} = \Theta_x = \Theta_{sgx}$

$$p_n = p_{sgn} - \rho_n \frac{v_{mn}^2}{2} = p_{sgn} - \frac{1}{2\rho_n}\left(\frac{q_m}{A_n}\right)^2$$

where

$$\rho_n = \frac{p_n}{R_w \Theta_n}$$

but p_n is unknown.

Assuming that for a first approximation:

$$\left(\rho_n\right)_1 = \rho_{sgn} = \frac{p_{sgn}}{R_w \Theta_{sgn}} = \frac{p_{sgn}}{R_w \Theta_n}$$

$$\left(p_n\right)_1 = p_{sgn} - \frac{1}{2\left(\rho_n\right)_1}\left(\frac{q_m}{A_n}\right)^2$$

and

$$\rho_n = \frac{\left(p_n\right)_1}{R_w \Theta_n}$$

$$p_n = p_{sgn} - \frac{1}{2\rho_n}\left(\frac{q_m}{A_n}\right)^2$$

$$p_{en} = p_{esgn} - \frac{1}{2\rho_n}\left(\frac{q_m}{A_n}\right)^2$$

Two or three iterations are sufficient for an accuracy of 10^{-3} on p_{en}; p_n may be obtained by the following equation:

$$p_n = \frac{1}{2}\left[p_{sgn} + \sqrt{p_{sgn}^2 - 2\left(\frac{q_m}{A_n}\right)^2 R_w \Theta_{sgn}}\right]$$

$$\rho_n = \frac{p_n}{R_w \Theta_n}$$

14.9.1.5 Calculation of fan pressure

The fan pressure p_F and fan static pressure p_{sF} are given by the following expressions:

$$p_F = p_{sg2} - p_{sg1} = p_{esg2} - p_{esg1}$$

$$p_{sF} = p_2 - p_{sg1} = p_{e2} - p_{esg1}$$

14.9.1.6 Determination of fan air power, P_u

The fan air power, P_u, the fan static air power, P_{us}, and the fan work per unit mass y and y_s are calculated in accordance with 14.8.1, 14.8.2, 14.8.3.

The various efficiencies are calculated from P_u or P_{us} and the various types of power supplied to the fan in accordance with 14.8.1.

14.9.2 Reference Mach number Ma_{2ref} less than 0,15 and fan pressure p_F less than 2 000 Pa

In this case,

— the Mach factor F_{Mx} may be taken as 1;

— the static and stagnation inlet temperatures and the static and stagnation outlet temperatures may be taken as equal, and, in the absence of an auxiliary fan upstream of the test fan, equal to the ambient temperature:

$$\Theta_1 = \Theta_{sg1} = \Theta_2 = \Theta_{sg2} = \Theta_3 = \Theta_{sg3} = \Theta_u = \Theta_a = t_a + 273{,}15$$

— the air flow through the fan and the test airway is considered as incompressible;

— in the presence of an auxiliary fan, the airflow is considered as incompressible between sections 3 and 4 (see figure 8).

14.9.2.1 Determination of mass flowrate

In accordance with 14.9.1.1

$$p_u = p_{eu} + p_a$$

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

However, the Reynolds number correction on the flow coefficient of the flowmeter α should be applied after a first determination of the mass flowrate and the corresponding Reynolds number.

14.9.2.2 Determination of the stagnation pressure at section x, p_{sgx}

According to 14.9.1, 14.9.1.2 and 14.9.2

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

$$p_{sgx} = p_x + \frac{1}{2}\rho_1 v_{mx}^2 = p_x + \frac{1}{2\rho_1}\left(\frac{q_m}{A_x}\right)^2$$

or

$$p_{esgx} = p_{ex} + \frac{1}{2\rho_1}\left(\frac{q_m}{A_x}\right)^2$$

where, in the absence of an auxiliary fan upstream of the test fan,

$$\rho_1 = \frac{p_a}{R_w \Theta_{sg1}} = \frac{p_a}{R_w \Theta_a} = \rho_a$$

With an auxiliary fan upstream of the test fan,

$$\rho_1 = \rho_2 = \rho_3 = \rho_4 = \frac{p_3}{R_w \Theta_3}$$

14.9.2.3 Determination of the stagnation pressure at a reference section of the fan from the gauge pressure measured at section x, p_{ex}

In accordance with 14.9.1.2 and 14.9.2

$$p_{sgn} = p_x + \frac{1}{2}\rho_1 v_{mx}^2 \left[1 + \left(\zeta_{n-x}\right)_x\right]$$

$$= p_x + \frac{1}{2\rho_1}\left(\frac{q_m}{A_x}\right)^2 \left[1 + \left(\zeta_{n-x}\right)_x\right]$$

The gauge stagnation pressure at section n is given by the following expression:

$$p_{esgn} = p_{ex} + \frac{1}{2\rho_1}\left(\frac{q_m}{A_x}\right)^2 \left[1 + \left(\zeta_{n-x}\right)_x\right]$$

14.9.2.4 Determination of the static pressure at a reference section of the fan

In accordance with 14.9.2 and 14.9.2.2

$$p_n = p_{sgn} - \frac{1}{2}\rho_1 v_{mn}^2$$

$$p_n = p_{sgn} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_x}\right)^2 \left(\frac{A_x}{A_n}\right)^2 = p_{sgn} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_n}\right)^2$$

which may also be written

$$p_{en} = p_{esgn} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_x}\right)^2 \left(\frac{A_x}{A_n}\right)^2 = p_{esgn} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_n}\right)^2$$

14.9.2.5 Fan pressure

The fan pressure p_F and the fan static pressure p_{sF} are given by the following expressions:

$$p_F = p_{sg2} - p_{sg1} = p_{esg2} - p_{esg1}$$

$$p_{sF} = p_2 - p_{sg1} = p_{e2} - p_{esg1}$$

14.9.2.6 Determination of fan air power, P_u

The fan air power P_u and the fan static air power P_{us} are calculated by the following expressions:

$$P_u = q_{Vsg1}\, p_F$$

$$P_{us} = q_{Vsg1}\, p_{sF}$$

The various efficiencies are calculated from P_u or P_{us} and the various types of power supplied in accordance with 14.8.1.

15 Rules for conversion of test results

The test results can only be compared directly with the guaranteed values if, during the acceptance tests, the measurements of the performance of the fan are taken under the conditions specified.

In most tests completed on fans, it is not possible to exactly reproduce and maintain the operating and/or driving conditions on the test airway as specified in the operating conditions.

Only the results converted to these operating conditions may be compared with the specified values.

For very large fans, model tests may be conducted in standardized airways when a full-scale test is impracticable owing to the limitations on power supply or dimensions of standardized test airways.

15.1 Laws on fan similarity

15.1.1 Similarity

Two fans which have similar flow conditions will have similar performance characteristics. The degree of similarity of the performance characteristics will depend on the degree of similarity of both the fans and of the flows through the fans.

15.1.1.1 Geometrical similarity

Complete geometrical similarity requires that the ratios of all corresponding dimensions for both fans be equal.

This includes ratios of thickness, clearances and roughness as well as the other linear dimensions for the flow passages.

All corresponding angles shall be equal.

15.1.1.2 Reynolds number similarity

Reynolds number similarity is necessary in order to keep relative thicknesses of boundary layer, velocity profiles and friction losses equal.

$$Re_u = \frac{uD_\mathrm{r}\rho_\mathrm{sg1}}{\mu} = \frac{uD_\mathrm{r}p_\mathrm{sg1}}{\mu R_\mathrm{w}\Theta_\mathrm{sg1}}$$

When the peripheral Reynolds number increases, the friction losses decrease.

Therefore efficiency and possibly performance may increase.

A difference in efficiency of 0,04 (4%) may be obtained for a Reynolds numbers ratio equal to 20.

15.1.1.3 Mach number and similarity of velocity triangles

In order to keep velocity triangles equal, variations of pressure, velocity and temperature through the fan must also be the same.

For peripheral Mach numbers higher than 0,15 important differences may arise if the Mach number is not kept equal for test and specified conditions.

For fans, the peripheral Mach number is given by.

$$Ma_u = \frac{u}{\sqrt{\kappa R_\mathrm{w}\Theta_\mathrm{sg1}}}$$

When this Mach number increases, the peripheral Reynolds number increases, as does the fan pressure.

When the fan pressure increases, ρ_m increases, while k_p and the ratio $\rho_\mathrm{sg1}/\rho_\mathrm{msg}$ both decrease. The velocity triangle similarity is no longer respected and losses increase.

This is why, when the Mach number increases, fan performance and efficiency first improve and then tend to deteriorate.

This effect depends on fan type, impeller design and position of the operating point on the characteristic curve of the fan.

As the compressibility coefficient k_p defined in 14.8.2.1 and 14.8.2.2 is close to $\rho_\mathrm{sg1}/\rho_\mathrm{msg}$ it can be used to represent the density variation through the fan and to characterize the similarity of the velocity triangles.

NOTE 34 There are never shock waves in fans: $Ma < 0{,}7$.

15.2 Conversion rules

The subscript Te is applied to the test measurements and test results, and the subscript Gu to the operating conditions and performance guaranteed by contract.

Figure 11 shows the permissible variations of the ratio

$$\frac{Re_{u\mathrm{Te}}}{Re_{u\mathrm{Gu}}}$$

as a function $Re_{u\mathrm{Gu}}$, and figure 12 gives an indication of the variations of the ratio $n_{\mathrm{Gu}}/n_{\mathrm{Te}}$ as a function of k_{pGu} and Δk_{p},

where

$$\Delta k_{\mathrm{p}} = k_{\mathrm{pGu}} - k_{\mathrm{pTe}}$$

15.2.1 Conversion rules for compressible flow

There is insufficient evidence to establish universal rules for the conversion of fan performance from a test to a specified condition involving a change in the compressibility coefficient k_{p} of more than ± 0,01 and which may be as great as 0,06.

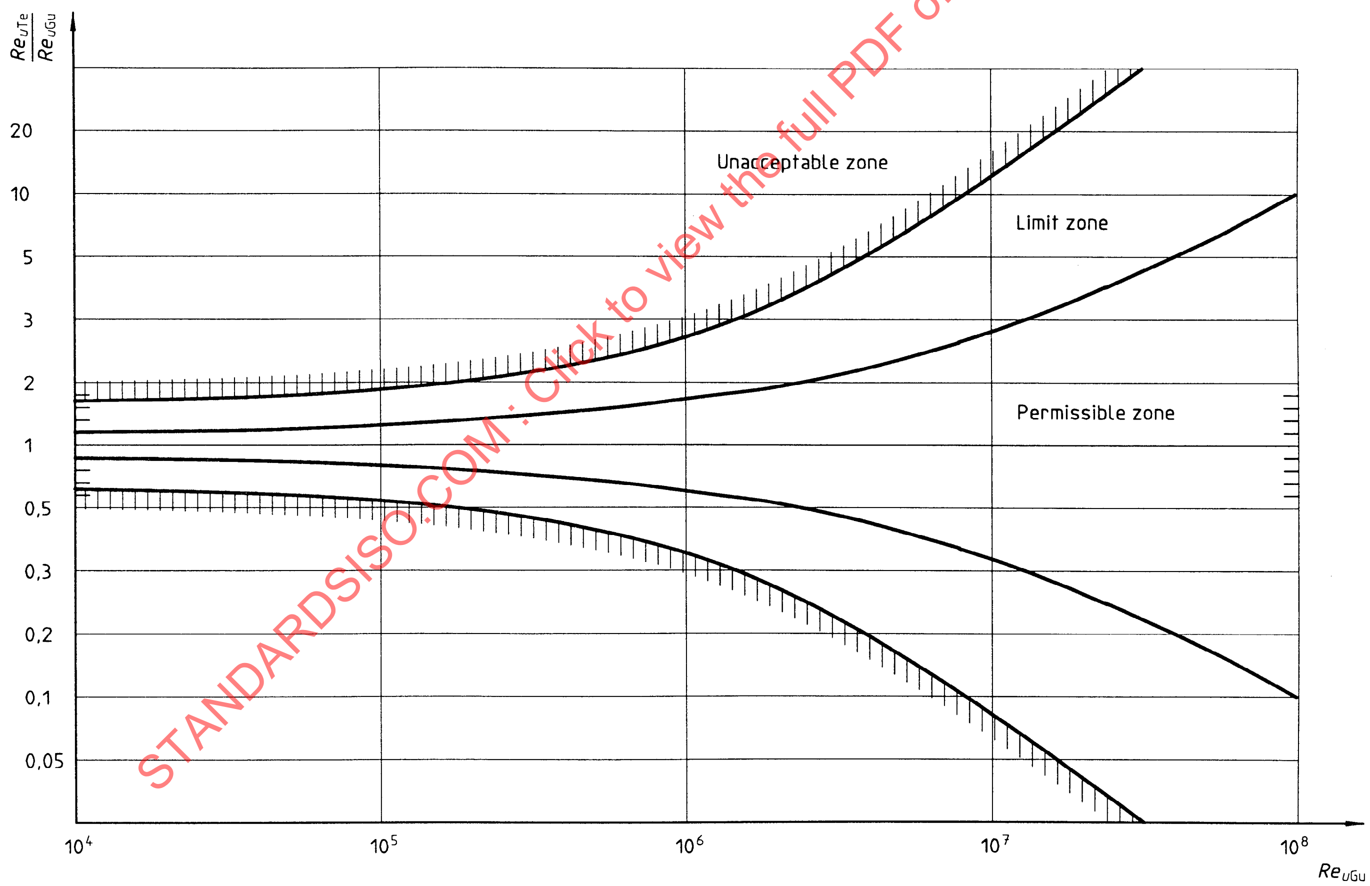


Figure 11 — Permissible variations of $Re_{u\mathrm{Te}}/Re_{u\mathrm{Gu}}$ as function of $Re_{u\mathrm{Gu}}$

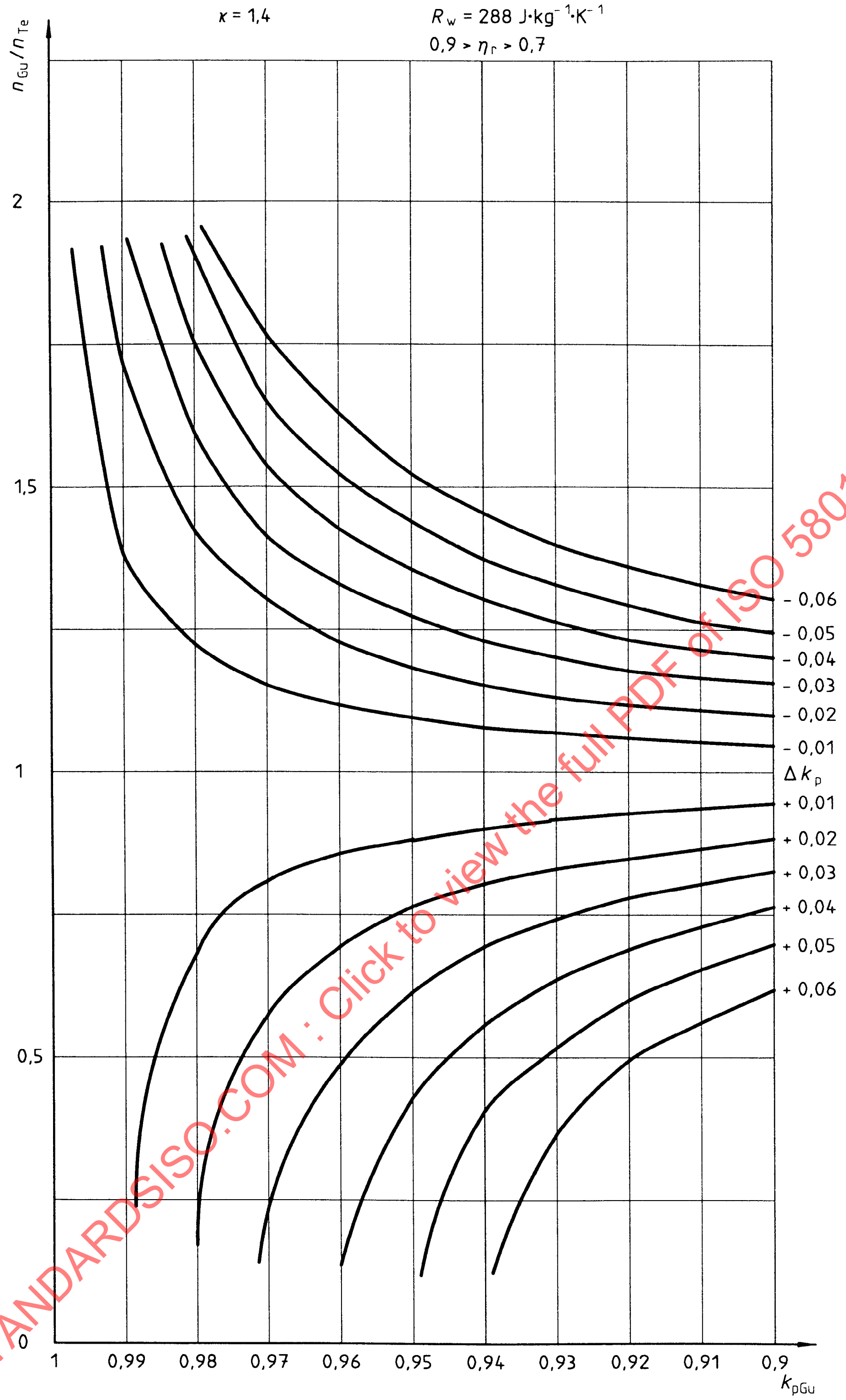


Figure 12 — Variation of n_{Gu}/ n_{Te} as a function of k_{pGu} and Δk_p

15.2.1.1 Conversion rules for a change of more than ± 0,01 in the compressibility coefficient, k_p

These conversion rules can be represented by the following expressions, in which q is an exponent which may vary from one design to another, values from 0 to – 0,5 having been demonstrated.

A type-test is recommended (which may be at model scale) to determine the range of pressure ratio r and the range of fan characteristic on either side of the best efficiency point, over which q may be regarded as constant without unduly increasing the uncertainty of performance prediction.

An agreement between purchaser and manufacturer is needed to apply these conversion rules.

The compressibility coefficient k_{pGu} after conversion may be found from the following approximate formula, which is correct within a few parts per 1 000:

$$\frac{1-k_{pGu}}{1-k_{pTe}} = \left(\frac{n_{Gu}D_{rGu}}{n_{Te}D_{rTe}}\right)^2 \left(\frac{R_{wTe}\Theta_{sg1Te}}{R_{wGu}\Theta_{sg1Gu}}\right) \frac{\kappa_{Te}}{\kappa_{Gu}} \left[\frac{1-\kappa_{Gu}(1-\eta)}{1-\kappa_{Te}(1-\eta)}\right] = k^2$$

where η is η_r or η_{sr}.

The fan performance after conversion may then be found from the following expressions:

$$\frac{q_{Vsg1Gu}}{q_{Vsg1Te}} = \frac{n_{Gu}}{n_{Te}} \left(\frac{D_{rGu}}{D_{rTe}}\right)^3 \left(\frac{k_{pGu}}{k_{pTe}}\right)^q$$

$$\frac{p_{FGu}}{p_{FTe}} = \left(\frac{n_{Gu}}{n_{Te}}\right)^2 \left(\frac{D_{rGu}}{D_{rTe}}\right)^2 \left(\frac{\rho_{sg1Gu}}{\rho_{sg1Te}}\right) \left(\frac{k_{pGu}}{k_{pTe}}\right)^{-1}$$

$$\frac{p_{sFGu}}{p_{sFTe}} = \left(\frac{n_{Gu}}{n_{Te}}\right)^2 \left(\frac{D_{rGu}}{D_{rTe}}\right)^2 \left(\frac{\rho_{sg1Gu}}{\rho_{sg1Te}}\right) \left(\frac{k_{pGu}}{k_{pTe}}\right)^{-1}$$

$$\frac{P_{rGu}}{P_{rTe}} = \left(\frac{n_{Gu}}{n_{Te}}\right)^3 \left(\frac{D_{rGu}}{D_{rTe}}\right)^5 \left(\frac{\rho_{sg1Gu}}{\rho_{sg1Te}}\right) \left(\frac{k_{pGu}}{k_{pTe}}\right)^q$$

The Reynolds number Re_u shall be within the limits of figure 11.

These expressions are established in the case of change in:

— rotational speed, N, or rotational frequency, n;

— impeller diameter, D_r;

— gas: R_w, κ;

— inlet temperature, Θ_{sg1} and density, ρ_{sg1}.

NOTE 35 Simplifications may be introduced as functions of the parameters which may be regarded as constant.

15.2.1.2 Conversion rules for a change of less than ± 0,01 in compressibility coefficient k_p

In the limits of the peripheral Reynolds number allowed according to figure 11, the following expressions may be applied.

The compressibility coefficient for guaranteed conditions k_{pGu} may be estimated from the formula given in 15.2.1.1.

$$\frac{1-k_{pGu}}{1-k_{pTe}} = \left(\frac{n_{Gu}D_{rGu}}{n_{Te}D_{rTe}}\right)^2 \left(\frac{R_{wTe}\Theta_{sg1Te}}{R_{wGu}\Theta_{sg1Gu}}\right) \frac{\kappa_{Te}}{\kappa_{Gu}} \left(\frac{1-\kappa_{Gu}(1-\eta)}{1-\kappa_{Te}(1-\eta)}\right)$$

The fan performance after conversion may then be found using the following expressions:

$$\frac{q_{Vsg1Gu}}{q_{Vsg1Te}} = \frac{n_{Gu}}{n_{Te}} \left(\frac{D_{rGu}}{D_{rTe}}\right)^3 \left(\frac{k_{pGu}}{k_{pTe}}\right)^q$$

$$\frac{p_{FGu}}{p_{FTe}} = \left(\frac{n_{Gu}}{n_{Te}}\right)^2 \left(\frac{D_{rGu}}{D_{rTe}}\right)^2 \left(\frac{\rho_{sg1Gu}}{\rho_{sg1Te}}\right) \left(\frac{k_{pGu}}{k_{pTe}}\right)^{-1}$$

$$\frac{P_{sFGu}}{P_{sFTe}} = \left(\frac{n_{Gu}}{n_{Te}}\right)^2 \left(\frac{D_{rGu}}{D_{rTe}}\right)^2 \left(\frac{\rho_{sg1Gu}}{\rho_{sg1Te}}\right) \left(\frac{k_{pGu}}{k_{pTe}}\right)^{-1}$$

$$\frac{P_{rGu}}{P_{rTe}} = \left(\frac{n_{Gu}}{n_{Te}}\right)^3 \left(\frac{D_{rGu}}{D_{rTe}}\right)^5 \left(\frac{\rho_{sg1Gu}}{\rho_{sg1Te}}\right) \left(\frac{k_{pGu}}{k_{pTe}}\right)^{q}$$

where η is η_r or η_{sr}, and q is an index which may vary from one design to another, values from 0 to – 0,5 having been demonstrated.

NOTE 36 Simplifications may be introduced as functions of the parameters which may be regarded as constant.

15.2.2 Simplified conversion rules for incompressible flow

When the fan pressure for test and guaranteed conditions is less than 2 000 Pa, k_p is close to 1 and the following simplified expressions may be used for the calculation of converted performance.

$$\frac{q_{Vsg1Gu}}{q_{Vsg1Te}} = \left(\frac{n_{Gu}}{n_{Te}}\right) \left(\frac{D_{rGu}}{D_{rTe}}\right)^3$$

$$\frac{p_{FGu}}{p_{FTe}} = \left(\frac{n_{Gu}}{n_{Te}}\right)^2 \left(\frac{D_{rGu}}{D_{rTe}}\right)^2 \left(\frac{\rho_{sg1Gu}}{\rho_{sg1Te}}\right) = \frac{p_{sFGu}}{p_{sFTe}}$$

$$\frac{P_{rGu}}{P_{rTe}} = \left(\frac{n_{Gu}}{n_{Te}}\right)^3 \left(\frac{D_{rGu}}{D_{rTe}}\right)^5 \left(\frac{\rho_{sg1Gu}}{\rho_{sg1Te}}\right)$$

15.2.3 Shaft power and impeller power

The measured and specified input powers will usually be the fan shaft power P_{aTe} and P_{aGu}.

It may be necessary to estimate the bearing losses P_{bTe} at n_{Te} and P_{bGu} at n_{Gu} and to use the relations

$$P_{rTe} = P_{aTe} - P_{bTe}$$

and

$$P_{aGu} = P_{rGu} + P_{bGu}$$

in order to carry out the conversion specified in 15.2.

However, the error incurred by assuming

$$\frac{P_{rGu}}{P_{rTe}}$$

as equal to

$$\frac{P_{aGu}}{P_{aTe}}$$

will not exceed the following, in percent,

$$\frac{200\left(n_{Gu} - n_{Te}\right) P_b}{n_{Te} P_a}$$

which is often negligible.

16 Fan characteristic curves

16.1 General

This clause deals with the graphical representation of the test results on a single fan.

Charts representing the performance of a series of fans over a range of speed and size by means of dimensionless coefficients or otherwise are outside the scope of this International Standard.

16.2 Methods of plotting

The actual test results, or the results after conversion according to the rules given in clause 15, shall be plotted as a series of test points against inlet volume flow. Smooth curves should be drawn through these points, with broken-line sections joining any discontinuities where stable results are not obtainable.

The results of conversion according to the rules given in clause 15 may be used, provided those changes which are outside the conversion limits given in 15.2.1 are clearly indicated on the plotted curves.

For fans for which the fan pressure is more than 2 000 Pa, indications of the fan outlet density shall be plotted using the ratio

$$\frac{\rho_2}{\rho_{sg1}}$$

or

$$k_\rho = \frac{\rho_1}{\rho_m}$$

16.3 Characteristic curves at constant speed

Fan characteristic curves at constant rotational speed are obtained from results converted in accordance with the rules given in clause 15 to a constant stated, rotational speed N_{Gu}, to a constant stated density ρ_{sg1Gu}, which should, unless otherwise agreed, be 1,2 kg · m^{-3}, and to a stated absolute inlet stagnation pressure p_{sg1Gu}.

The fan pressure p_F and the fan static pressure p_{sF} or either one of them together with the fan dynamic pressure corrected for the Mach number effect: $p_{d2}F_{M2}$ shall be plotted against the inlet volume flowrate q_{Vsg1}. The fan efficiency η_r and/or the fan static efficiency η_{sr} or their shaft power equivalents may also be plotted.

An example is given in figure 13.

16.4 Characteristic curves at inherent speed

Characteristic curves at inherent speed may be used if so desired for a unit consisting of the fan and its driving means.

The driving means should be operated under fixed and stated conditions, e.g. at the rated voltage and frequency for an electric motor. The rotational speed should also be indicated on the fan performance characteristic curve plotted against the inlet volume flowrate. Conversion to another air density is permissible within the Reynolds number criteria given in 15.2 provided the rotational speed is corrected with respect to motor input power by use of performance data on the driving means.

16.5 Characteristic curves for adjustable-duty fan

Adjustable-duty fan characteristic curves are required for fans having means for altering their performance, such as variable-pitch blades or variable inlet guide vanes. A family of constant speed characteristic curves at 1,2 kg · m^{-3} inlet density is recommended, selected at suitable steps of adjustment over the whole available range of volume flowrates. Efficiencies may be shown by means of smooth contours drawn through points of equal efficiency on the fan pressure characteristic curve. An example is given in figure 14.

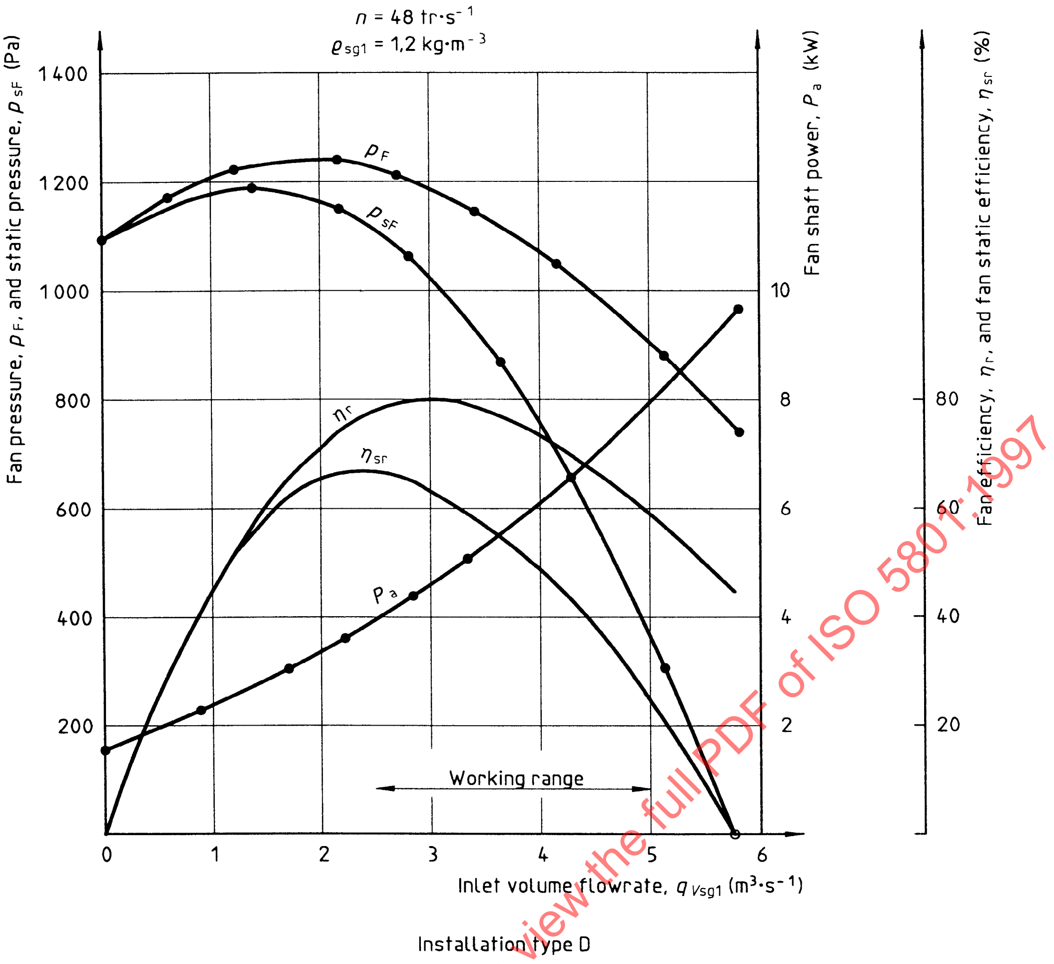


Figure 13 — Example of a set of complete, constant-speed, fan characteristic curves

16.6 Complete fan characteristic curve

A complete fan characteristic curve extends from zero fan static pressure to zero inlet volume flowrate.

Only part of this curve is normally used however, and it is recommended that the supplier should state the range of inlet volume flowrates for which the fan is suitable.

The plotted fan characteristic curve may then be limited to this normal operating range. Outside the normal operating range of inlet volume flowrates, the uncertainty of measurement is liable to increase and unsatisfactory flow patterns may develop at inlet or outlet.

16.7 Test for a specified duty

Tests for a specified duty should comprise not less than three test points determining a short part of the fan characteristic curve, including both the specified inlet volume flowrate and the specified fan stagnation or static pressure.

A system resistance line should also be drawn, passing through the specified duty point, and such that the stagnation or static pressure varies with the square of the inlet volume flowrate (see figure 15).

The actual operating point of the fan will be at the intersection of the fan characteristic curve and the system resistance line.

NOTE 37 Deviations or tolerances should be determined in accordance with the planned ISO standard concerning fan tolerances.

17 Uncertainty analysis

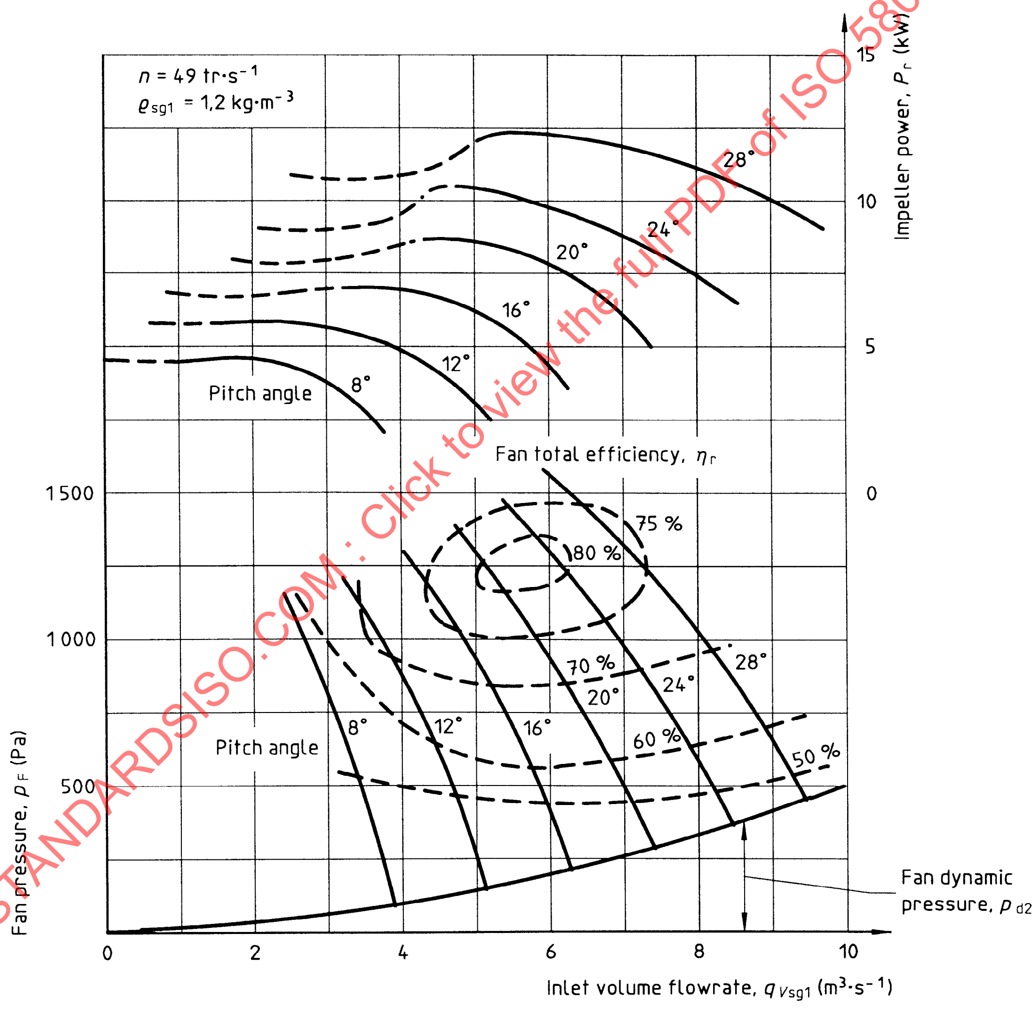
17.1 Principle

It is an accepted principle that all measurements have a margin of error. It is also clear that any results, such as fan flowrate and fan pressure calculated from measured data, will also contain errors, due not only to the errors in the data, but also to approximations or errors in the calculation procedure.

Accordingly, the quality of a measurement or a result is a function of the associated error. Uncertainty analysis provides a means of quantifying the errors with various levels of coverage. The quality of any fan test is best evaluated by performing an uncertainty analysis.

ISO 5168 includes an excellent discussion of uncertainty analysis that can be applied to all aspects of fan testing, not just fluid flow measurements. The concepts contained in ISO 5168 provide the basis for the following.

In this International Standard, 95 % coverage is required.



Installation type D

Figure 14 — Example of characteristic curves for an adjustable-duty fan

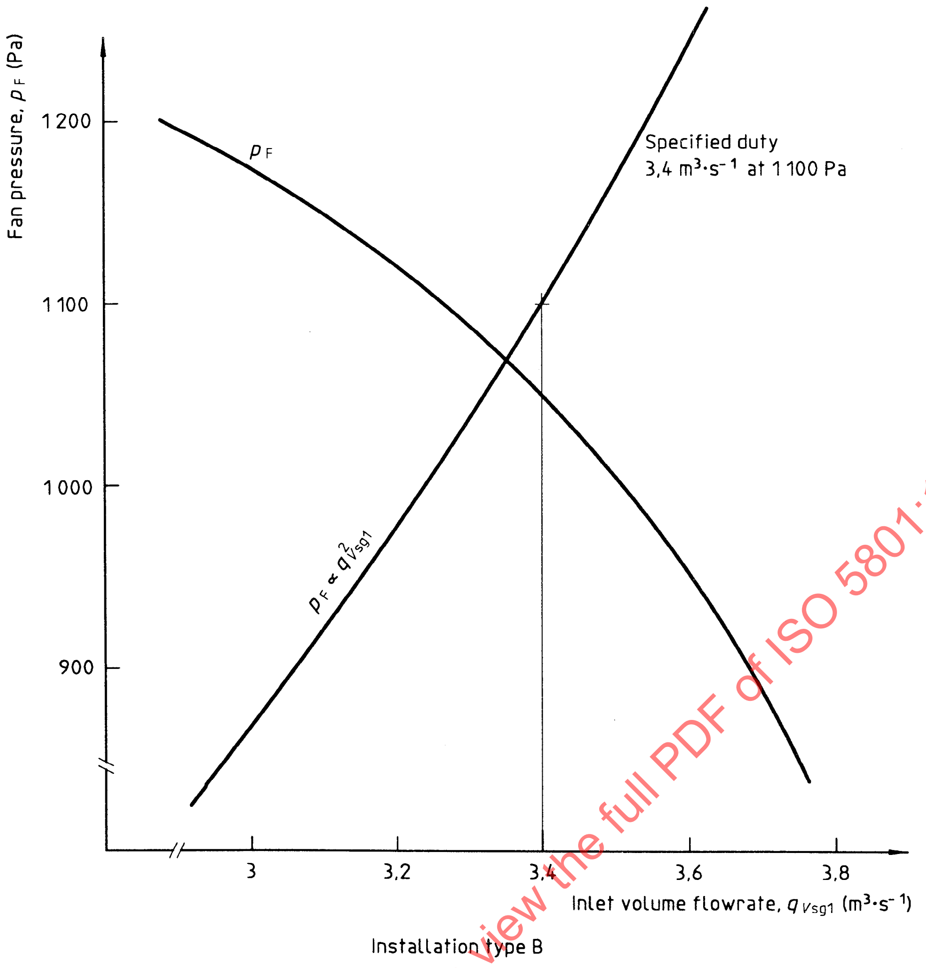


Figure 15 — Example of test for a specified duty

17.2 Pre-test and post-test analysis

A pre-test uncertainty analysis is recommended to identify potential measurement problems and to permit design of the most cost-effective test. A post-test uncertainty analysis is required to establish the quality of the test. This analysis will also show which measurements were associated with the largest errors.

17.3 Analysis procedure

A rigorous uncertainty analysis for a fan test requires significant effort as well as detailed information concerning the instruments, calibrations, calculations and other factors. There are at least five (and perhaps as many as 15) parameters that can be considered the results of a fan test. Each result is dependent on one or more measurements. Each measurement can have five or more components of uncertainty. All of these components should be considered in an uncertainty analysis.

The procedure outlined in ISO 5168 includes the following steps:

a) list all possible sources of error;

b) calculate or estimate, as appropriate, elementary errors for each source;

c) for each measurement, combine separately the element bias limits and the element precision indices by the root-sum-square method;

d) for each parameter, propagate separately measurement bias limits and measurement precision indices, either by using sensitivity factors or by regression;

e) calculate the uncertainty for each parameter;

f) establish the uncertainty interval for each parameter.

NOTE 38 In addition to measurement errors, there may be errors associated with extracting data from tables or charts, or from using formulas.

17.4 Propagation of uncertainties

ISO 5168 explains how to combine the uncertainties due to calibration errors, data acquisition errors, data reduction errors, errors of method and personal errors into an uncertainty of a measurement. It also details how to propagate various measurement and other uncertainties into an uncertainty of a result. It is important to maintain a separate accounting of precision indices and bias limits, even though they may be combined in the ultimate calculation.

17.5 Reporting uncertainties

The test report should state the following for each parameter of interest:

a) the test value[2)] of the parameter;

b) the precision index and associated degrees of freedom, ν;

c) the bias limit;

d) the uncertainty based on a 95 % confidence level.

EXAMPLE

a) $R = q_V = 5\ \text{m}^3 \cdot \text{s}^{-1}$

b) $s = 0{,}05\ \text{m}^3 \cdot \text{s}^{-1}$ $\nu = 5$

c) $B = 0{,}025\ \text{m}^3 \cdot \text{s}^{-1}$

d) $U = \sqrt{B^2 + (t_{95}s)^2}$ (U corresponds to U_{RSS} in ISO 5168)

$$U = \sqrt{0{,}025^2 + (2{,}57 \times 0{,}05)^2} = 0{,}131\ \text{m}^3 \cdot \text{s}^{-1}$$

then $u = \dfrac{U}{R} = \dfrac{0{,}131}{5}$

17.6 Maximum allowable uncertainties measurement

This International Standard lists certain requirements for measuring instruments. These include the accuracy and legibility of the instrument itself and, in some cases, similar information about the working standard which must be used to calibrate the instrument before and after the test. None of this information is given in terms of precision index and bias limit, nor is coverage stated. However, values may be assumed to be for uncertainty at 95 % confidence level. The same assumption is usually justified when interpreting technical data supplied by an instrument manufacturer.

Table 2 contains a summary of maximum allowable relative uncertainties for each of the parameters measured, either directly or indirectly, during a fan test. The instrument (or combination of instruments) used to determine the parameter value must be sufficiently accurate so that when the various error estimates are combined, the resulting uncertainty will not exceed the value given in table 2.

2) The best estimate of a parameter is the test value. This estimate can be improved by repeating the test and using the average result.

17.7 Maximum allowable uncertainty of results

The different parameters comprising the results of a fan test are listed in table 3. Also listed is the maximum allowable relative uncertainty for each result, if the test is to qualify as a test conducted under this International Standard. Better quality (lower uncertainty) results might be attainable by using instruments with proven uncertainties lower than those required to satisfy the preceding subclause.

The uncertainties in table 3 are based on 95 % confidence level. Precision indices and bias limits are not separately stated. Nevertheless, any test conducted in accordance with this International Standard should include an uncertainty analysis. The precision indices and bias limits should be listed separately in such an analysis.

18 Selection of test method

18.1 Classification

The fan to be tested shall be classified according to one of the four types specified in 18.2. The supplier should state the type or types of installation for which the fan is intended, and the user should select from the types available the one which is closest to his application.

18.2 Installation types

The four types of installation are as follows:

— type A: free inlet, free outlet;

— type B: free inlet, ducted outlet;

— type C: ducted inlet, free outlet;

— type D: ducted inlet, ducted outlet.

In the above classification, the terms shall be taken to have the following meanings:

Free inlet or outlet signifies that the air enters or leaves the fan directly from or to the unobstructed free atmosphere. Ducted inlet or outlet signifies that the air enters or leaves the fan through a duct directly connected to the fan inlet or outlet respectively.

Table 2 — Maximum allowable uncertainties of measurement of individual parameters

Parameter	Symbol	Relative uncertainty of measurement	Remarks	Clause or subclause
Atmospheric pressure	p_a	$u_{pa} = \pm 0{,}2$ %	corrected for temperature and altitude	6.1
Ambient temperature	Θ_a	$u_{\Theta a} = \pm 0{,}2$ %	measured near fan inlet or inlet duct, or in a chamber where the velocity is less than 25 m·s^{-1} (0,5 °C)	8.1
Humidity	h_u	$u_{hu} = \pm 0{,}2$ %	uncertainty in air density due to an uncertainty of ± 2 °C in $(t_d - t_w)$ for t_d = 30 °C	8.3
Gauge pressure	p_e	$u_{pe} = \pm 1{,}4$ %	static pressure greater than 150 Pa: combining 1 % manometer and 1 % reading fluctuation uncertainty may be reduced to 1 % or less for high-pressure fans as a function of fluctuations.	6.2 6.3
Differential pressure	Δp	$u_{\Delta p} = \pm 1{,}4$ %	as for gauge pressure	6.2 6.3
Rotational speed of impeller	N	$u_N = \pm 0{,}5$ %	may be reduced to 0,2 % by use of electrical scanning	9
Rotational frequency of impeller	n	$u_n = \pm 0{,}5$ %	as for rotational speed	9
Power input	P_r	$u_{Pr} = \pm 2$ %	measured by torquemeter or two-wattmeter method uncertainty according to class of wattmeter and transformer	10
Area of a nozzle throat	A_d	$u_{Ad} = \pm 0{,}2$ %	u_d = 0,1 %	
Area of a duct	A_x	$u_{Ax} = \pm 0{,}5$ %	u_D = 0,1 %	11
Mass flowrate	q_m	u_{qm}		22 to 27

Table 3 — Maximum allowable uncertainty for the results

Parameter	Symbol	Relative uncertainty	Remarks
Ambient density	ρ_a	$u_{\rho a} = \pm 0{,}4$ %	$\sqrt{u_{\Theta a}^2 + u_{hu}^2 + u_{pa}^2}$
Fan temperature rise	$\Delta\Theta$	$u_{\Delta\Theta} = \pm 2{,}8$ %	$\sqrt{u_{pr}^2 + u_{qm}^2}$
Outlet stagnation temperature	Θ_{sg2}	$u_{\Theta sg2} = \pm 0{,}4$ %	$\dfrac{u_{\Delta\Theta}\Delta\Theta}{\Theta_{sg2}}$
Outlet stagnation density	ρ_{sg2}	$u_{\rho sg2} = \pm 0{,}7$ %	u_{p2}
Dynamic pressure	p_{d2}	$u_{pd2} = \pm 4$ %	$\sqrt{4u_{qm}^2 + 4u_A^2 + u_{\rho 2}^2}$
Fan pressure	p_F	$u_{pF} = \pm 1{,}4$ %	$= u_{pe}$
Fan air power	P_u	$u_{Pu} = \pm 2{,}5$ %	$\sqrt{u_{qm}^2 + u_{pF}^2}$
Fan efficiency	η_r	$u_{\eta r} = \pm 3{,}2$ %	$\sqrt{u_{Pu}^2 + u_{Pr}^2}$
Fan flowrate	q_m or q_V	u_{qm} or $u_{qV} = \pm 2$ %	see individual clauses for various flow measurement methods

18.3 Test report

All references to fan performance stated to be in accordance with this International Standard shall also state the installation type to which they refer. This is because a fan adaptable for use in all four installation types will have differing performance characteristics for each installation, the extent of the difference depending on the fan type and design.

In reporting a test, the method selected from clauses 32 to 35 shall also be stated, but this is not necessary for catalogue data or contracts of sale since the alternative methods permissible within each installation type may be expected to give results falling within the uncertainty of measurement.

18.4 User installations

In selecting a type of installation to match his system, the user should note that a system connected to the fan through a length of duct equal to one diameter is usually sufficient (see 30.3) to establish ducted inlet performance provided bends, sudden expansions or other upstream sources of flow separation are not too close by.

On the outlet side a duct length of two or three diameters is required to establish ducted outlet performance.

Rectangular-to-round transition has little effect provided there is no change in cross-sectional area. A change in performance may be expected when the cross-sectional area is increased through a diffuser fitted to the fan outlet, both for free outlet and ducted outlet systems.

18.5 Alternative methods

For any one installation type, the alternative methods available differ only in the method of flowrate measurement and control. The relative merits of nozzle, orifice and traverse methods of flowrate measurement are discussed in clause 13. Other methods complying fully with the requirements of International Standards or other well-known standards may also be employed.

The alternative standardized airways and the required measurements and calculations are described in clauses 32, 33, 34 and 35 and in the accompanying figures.

18.6 Duct simulation

To limit the number of standardized airways required in a test laboratory, those designed for free inlet or outlet tests may be adapted to ducted inlet or outlet tests by the addition of the inlet and outlet duct simulation sections described in clause 30.

Standardized airways designed for type A installation tests may be adapted to provide tests for type B, C or D installations. It follows that the inlet side or outlet side test chambers described in clause 31, which will also cover a wide range of fan sizes, are well suited to the needs of a permanent, universal, test installation.

Standardized airways for type B or C installation tests may be adapted to provide tests for type D installations.

19 Installation of fan and test airways

19.1 Inlets and outlets

The fan shall be tested as supplied without additions except for the test airways, and without removal of any component part which might affect the flow unless otherwise agreed before the test.

It is nevertheless permissible, subject to prior agreement between supplier and purchaser, to determine the combined performance of the fan and a transition airway such as an inlet box or outlet diffuser which is not supplied with the fan. Such an addition shall be fully specified with the test report, and its inlet or outlet shall be regarded as the fan inlet or outlet for the purposes of test.

19.2 Airways

All test airways should be straight and of circular cross-section, except where otherwise specified.

Joints between airway sections should be in good alignment and free from internal protrusions, and leakage should be negligible compared with the mass flow of the fan under test. Where provision is made for the insertion and manipulation of measuring instruments, special care should be taken to minimize leakage and obstruction of the airway

19.3 Test enclosure

The assembly of the fan with its test airways should be so situated that, when the fan is not operating, there is no draught in the vicinity of the inlet or outlet of the assembly of speed greater than 1 $m \cdot s^{-1}$. Care should be taken to avoid the presence of any obstruction which might significantly modify the air flow at inlet or outlet. In particular, no wall or other major obstruction should be closer than two airway diameters from the inlet or outlet of the airways or the test fan. Greater unobstructed space at the inlet and outlet of flow measurement devices is specified in clauses 22 to 26. The test enclosure shall be large enough to permit free return from outlet to inlet.

19.4 Matching fan and airway

For the purposes of confirming compliance with the limitations on test duct dimensions, the fan inlet and/or outlet areas shall be taken as the gross area at the inlet or outlet flange without deduction for motors, fairings or other obstructions. Where motors, fairings or other obstructions extend beyond an inlet or outlet flange at which the performance for ducted installation is to be determined, the casing should be extended by a duct of the same size and shape as the inlet or outlet and of sufficient length to cover the obstruction. The test airway dimensions should be measured from the plane through the outermost extension of the obstruction as if this were the plane of the inlet or outlet flange.

19.5 Outlet area

For the purpose of determining the fan dynamic pressure, the fan outlet area shall be taken as the gross area at the outlet flange or the outlet opening in the casing without deduction for motors, fairings or other obstructions.

Some free-outlet fans without casings have no well-defined outlet area. A nominal area may then be defined and stated, e.g. the area within the ring of a propeller wall fan or the circumferential outlet area of an open-running centrifugal impeller. The corresponding fan dynamic pressure and fan pressure will also be nominal and should be so described.

20 Carrying out the test

20.1 Working fluid

The working fluid for tests with standardized airways shall be atmospheric air, and the pressure and temperature should be within the normal atmospheric range, either at fan outlet or at fan inlet.

20.2 Rotational speed

20.2.1 For constant speed characteristics, the fan should preferably be operated at a speed close to that specified. Where the speed is substantiallly different, or where the fan is intended for use with a gas other than air, or at a different density, the provisions of clause 15 shall be applied.

20.2.2 In the case of inherent speed characteristics, as defined in clause 16, the fan motor shall be operated at steady supply conditions within the range permissible for the motor or prime mover.

20.3 Steady operation

Before taking measurements for any point on the fan flowrate curve, the fan shall be run until steady operation is achieved within a band of speed fluctuation not exceeding 1 %.

Readings of speed and power input shall be taken at each point on the fan characteristic curve. If they are fluctuating, sufficient readings should be taken to obtain, by averaging, a value which is compatible with the accuracy of measurement given in clauses 9 and 10.

20.4 Ambient conditions

Readings of atmospheric pressure, dry-bulb temperature and wet-bulb temperature shall be taken within the test enclosure (except as permitted by the recommendations of 6.1) during the series of observations required to determine the fan characteristic curves. If the ambient conditions are varying, sufficient readings should be taken to obtain for each test point on the characteristic curve by averaging, a value which is compatible with the accuracy of measurement given in clauses 6 and 8.

20.5 Pressure readings

Pressure in the test airways should be observed over a period of not less than 1 min for each point on the fan characteristic curve. Rapid fluctuations should be damped at the manometer and if the readings still show random variations, a sufficient number of observations should be recorded to ensure that a time-average is obtained within the accuracy limits given in 6.3.

20.6 Tests for a specified duty

Tests for a specified duty should comprise not less than three test points determining a short portion of the fan characteristic curve including the specified flowrate.

20.7 Tests for a fan characteristic curve

Test for determining fan characteristic curves should comprise a sufficient number of test points to permit the characteristic curve to be plotted over the normal operating range. Closely spaced points will be necessary where there is evidence of sharp changes in the shape of the characteristic curve.

20.8 Operating range

Test points outside the normal operating range may be recorded, and the complete fan characteristic curve plotted, for information only. Tests made outside the normal operating range will not necessarily have the accuracy expected for tests made within the normal range.

21 Determination of flowrate

Six methods of flowrate measurement are listed in 21.1 to 21.6 and described in clauses 22 to 27.

21.1 ISO Venturi nozzle (see ISO 5167-1)

— Inlet Venturi nozzle

— In-duct Venturi nozzle

— Outlet Venturi nozzle

21.2 Multiple nozzle or Venturi nozzle

— Multiple nozzle in test chamber

— Inlet Venturi nozzle

— In-duct Venturi nozzle

— Outlet Venturi nozzle

21.3 Quadrant inlet nozzle

21.4 Conical inlet

21.5 Orifice plate

— Inlet orifice plate

— In-duct orifice plate (see ISO 5167-1)

— Outlet orifice plate

— Orifice plate in chamber

21.6 Pilot-static tube traverse (see ISO 3966 and ISO 5221)

22 Determination of flowrate using ISO Venturi nozzle

22.1 Geometric form

The nozzle dimensions are shown in figure 16 and are in accordance with ISO 5167-1:1991, subclauses 9.1 and 10.2. The nozzle profile shall be axially symmetrical, the curved portion being checked by template. The throat shall be cylindrical, with no diameter differing from the corresponding mean diameter by more than 0,0005d, as measured in any plane at right angles to the axis of the nozzle. The roughness expressed by the arithmetical deviation of the profile surface (R_a) shall not exceed $10^{-4}d$ ($R_a \leqslant 10^{-4}d$). The throat, the two circular arcs and the flat face shall meet tangentially without perceptible steps. The nozzle axis and the axis of the airway shall be coincident.

22.1.1 The throat diameter, d, is the arithmetic mean of four measurements, to within an accuracy of 0,001d, taken at angular spacings of about 45°, in the plane of the throat pressure tappings.

22.1.2 The pressure tappings shall conform to the dimensions shown in figure 16, the requirements specified in clause 7 and ISO 5167-1:1991 subclauses 9.1.5 and 10.2.3.

22.1.3 The pressure difference Δp across the nozzle shall be measured in accordance with the requirements of clause 6 and 13.2.3.

22.1.4 Except where otherwise specified, the included angle of the divergent section may lie in the range $0° < \theta < 30°$. An included angle of 15° is recommended since the loss is then at a minimum. Except in the case of some type C installations (see clause 34), the Venturi nozzle may be truncated. The divergent section may then be shortened by about 35 % of its full length without greatly modifying the pressure loss in the device. The divergent or cylindrical connection piece shall be of length not less than 3d.

22.1.5 The installation conditions and the length of the upstream and downstream ducts shall be in accordance with ISO 5167-1, clause 7. The diameter of the upstream duct shall be cylindrical, with no diameter differing from the corresponding mean diameter by more than 0,003D [see figure 17 a)].

22.2 Venturi nozzle in free-inlet condition

22.2.1 Within the inlet zone defined in figure 17 b), there shall be no external obstruction to the free movement of air entering the inlet and the velocity of any cross-currents should not exceed 5 % of the nozzle throat velocity.

22.2.2 Steps shall be taken to ensure that the pressure registering at the high-pressure limb of the differential pressure-reading manometer is the ambient pressure in the inlet zone.

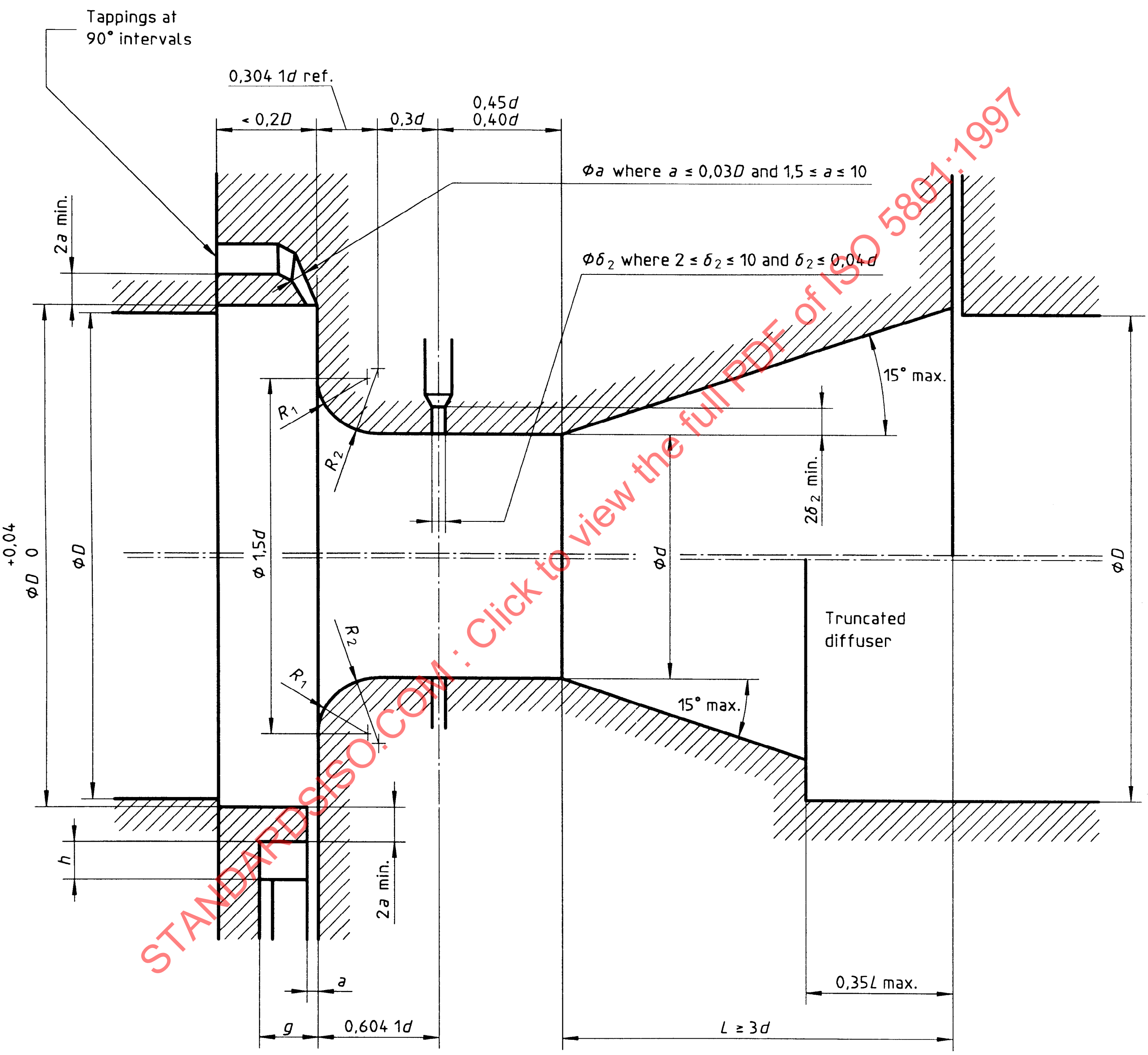
22.2.3 Screen loading in accordance with the requirements of 25.2 and figure 17 c) is permissible with the free-inlet Venturi nozzle installation, but there will be an increase in the uncertainty of the flowrate coefficient α, and the pressure ratio r_d should not be less than 0,90 if the uncertainty of flowrate determination is not to exceed ± 2 %.

22.2.4 Screen loading, in accordance with the requirements of 22.4.3 and figure 17 b), located not less than 4d downstream of the inlet face of the nozzle may be used without increasing the uncertainty of the flowrate coefficient α.

22.3 Nozzle performance

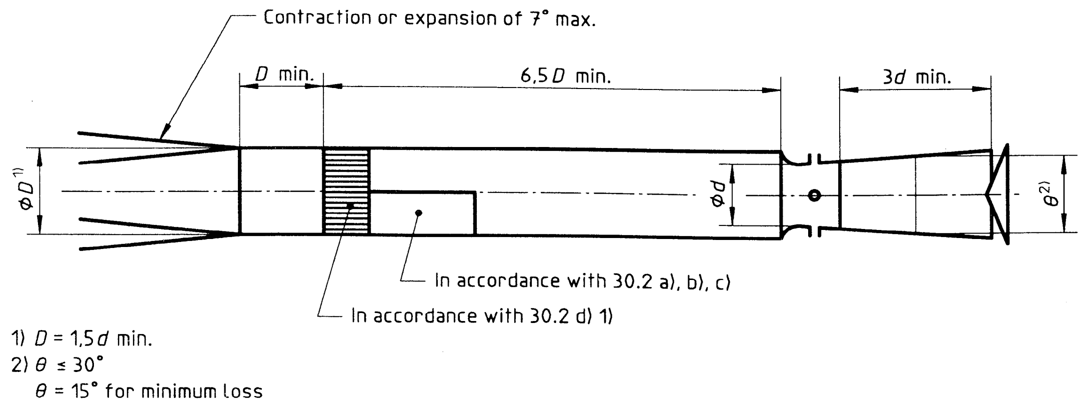
22.3.1 Diameter ratio

In this International Standard the Venturi nozzle may be used in conditions ranging from a free inlet, i.e. $\beta = 0$, to a maximum diameter ratio $\beta = 0{,}67$ for an in-duct Venturi nozzle.

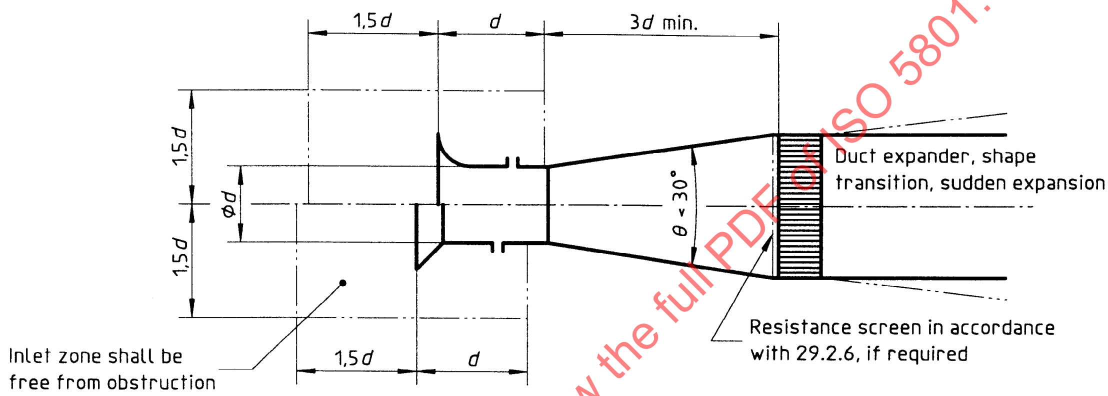


$R_1 = 0{,}2d$
$R_2 = d/3$

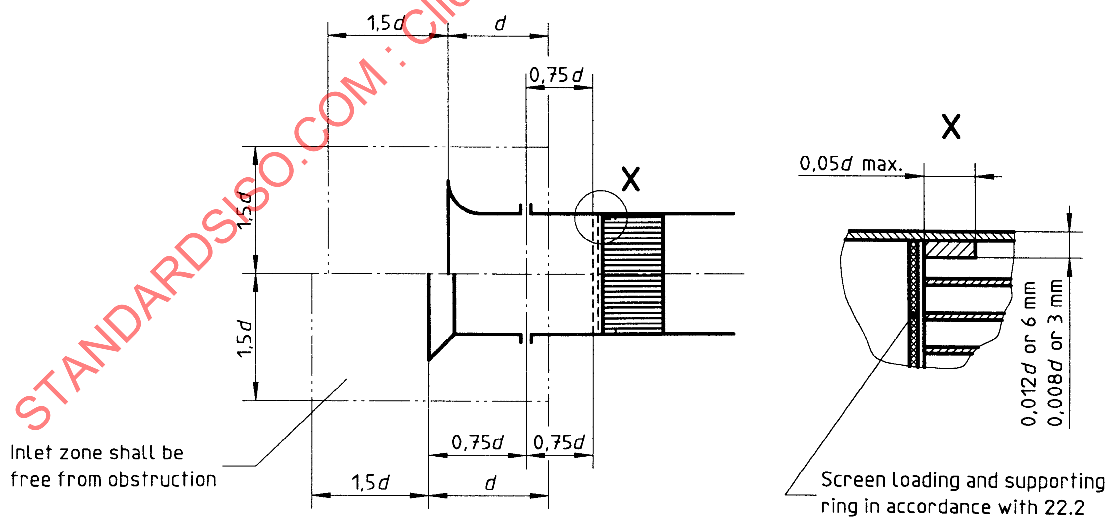
Figure 16 — Geometry of ISO Venturi nozzle



a) In-duct Venturi nozzle



b) Free-inlet nozzle or conical inlet



c) Free-inlet Venturi nozzle or conical inlet with adjustable screen loading

Figure 17 — Flow-metering installations

22.3.2 Flowrate coefficient

The flowrate coefficient α is independent of Reynolds number within the following constraints:

— diameter ratio: $\beta = d/D$ where $0 \leqslant \beta \leqslant 0{,}67$

— throat diameter: $d \geqslant 0{,}05$ m

— throat Reynolds number:

$$Re_d = \frac{4q_m}{\pi\mu d} \geqslant 10^5$$

where

μ is the dynamic viscosity of air upstream of the nozzle;

D is the diameter of the upstream airway;

d is the nozzle throat diameter.

Under these conditions it has been shown that the relationship between α and β is described by the following empirical expression:

$$\alpha = \frac{0{,}985\,8 - 0{,}196\beta^{4,5}}{\left(1-\beta^4\right)^{0,5}}$$

22.3.3 Expansibility factor

The expansibility factor ε is related to the pressure ratio r_d, where

$$r_d = \frac{p_{do}}{p_u} = 1 - \frac{\Delta p}{p_u}$$

(Δp is assumed to be positive) by the following expression which is valid when $r_d > 0{,}75$:

$$\varepsilon = \left[\frac{\kappa r_d^{2/\kappa}}{\kappa - 1}\;\frac{1-\beta^4}{1-\beta^4 r_d^{2/\kappa}}\;\frac{1-r_d^{\frac{\kappa-1}{\kappa}}}{1-r_d}\right]^{0,5}$$

22.3.4 Compound coefficient

The compound coefficient $\alpha\varepsilon$ is shown plotted against the ratio β in figure 18 for representative values of the pressure ratio r_d. The mass flowrate is derived from the following expression:

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

where ρ_u is the density upstream of the nozzle.

22.3.5 Calculation of the density upstream of the nozzle

22.3.5.1 Venturi nozzle in free-inlet condition

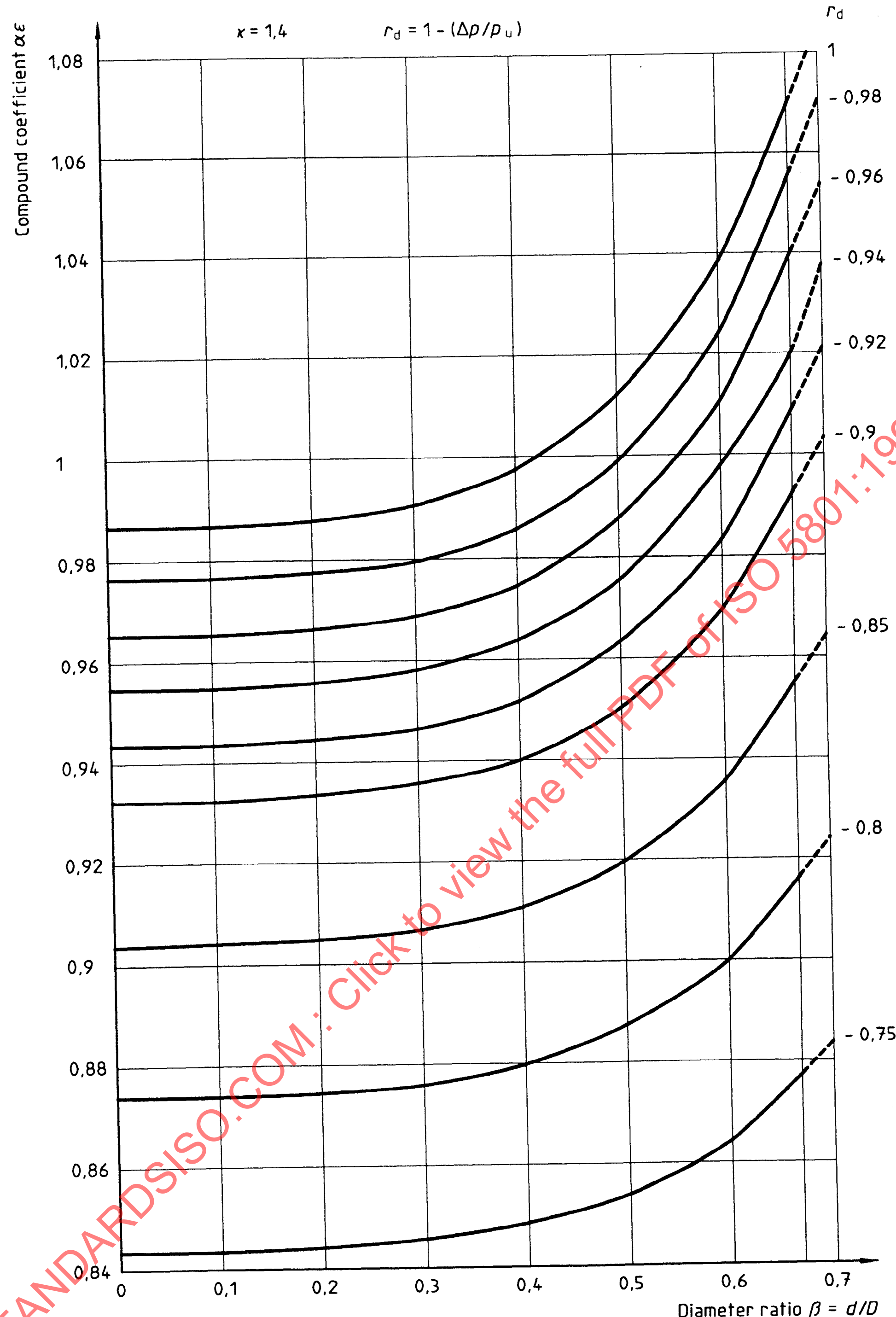
The density upstream of the nozzle is given by the following expression:

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

where

$p_u = p_a$

$\Theta_u = \Theta_a = t_a + 273{,}15$



NOTE — For a free-entry Venturi nozzle, β should be taken as 0.

Figure 18 — Compound flow coefficients of ISO Venturi nozzles

22.3.5.2 Venturi nozzle in in-duct conditions

The density upstream the nozzle is given by

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

where

p_u is the upstream pressure;

Θ_u is the temperature upstream the Venturi nozzle.

22.3.5.2.1 The temperature t_u is measured and the upstream Mach number is less than 0,15. The absolute temperature

$$\Theta_u = t_u + 273{,}15$$

is greater than the true static temperature and less than the stagnation temperature but the difference between these two values is less than 5×10^{-3}.

Calculations are carried out with Θ_u.

22.3.5.2.2 The temperature t_u is not measured and the upstream Mach number is greater than 0,15.

The stagnation temperature upstream of the flowmeter is given by the following expressions:

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

for an inlet duct without auxiliary fan;

$$\Theta_{sgu} = \Theta_a + \frac{P_{rx} \text{ or } P_{ex}}{q_m c_p}$$

for an auxiliary fan upstream of an inlet duct where P_{rx} is the shaft power of the auxiliary fan or P_{ex} is the input power of the motor inside the fan housing

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

for an outlet duct.

For all cases, the static temperature Θ_u may be calculated by the following expression:

$$\Theta_u = \Theta_{sgu} - \frac{q_m^2}{2A_u^2 \rho_u^2 c_p}$$

For a first approximation, $\Theta_u = \Theta_{sgu}$ and a first value of q_m : q_{m1} is obtained. This value allows the calculation of Θ_u, the flowmeter Reynolds number Re_D or Re_d and the mass flowrate q_m.

NOTE 39 For $\beta = 0{,}67$ and

$$\frac{p_{do}}{p_u} = 0{,}75,$$

the Mach number $Ma_u = 0{,}235$ and

$$\frac{\Theta_{sgu}}{\Theta_u} = 1{,}011\,0$$

$q_m\,(\Theta_{sgu})$ is $5{,}5 \times 10^{-3}$ less than the true value of q_m.

22.4 Uncertainties

22.4.1 The uncertainty u_α, in percent, of the flowrate coefficient α of a Venturi nozzle with ducted inlet as shown in figure 17 a) is as follows:

$$u_\alpha = \pm\left[1{,}2 + 1{,}5\left(\frac{d}{D}\right)^4\right]$$

22.4.2 The uncertainty u_α of the flow coefficient α of a Venturi nozzle with free inlet as shown in figure 17 b) is ± 1,2 %.

22.4.3 When screen loading is used as shown in figure 17 c) for a free-inlet Venturi nozzle with the screen less than $4d$ from the inlet face of the nozzle, an arithmetic addition of 0,5 % should be made to the uncertainty of the flowrate coefficient. u_α will then be equal to ± 1,7 %.

22.4.4 The uncertainty u_ε of the expansibility factor ε is given by the following expression:

$$u_\varepsilon = \pm\left(4 + 100\beta^8\right)\frac{\Delta p}{p_u}$$

22.4.5 If the uncertainties of measurement of pressure, density and diameter are equal to the maxima specified in clauses 7 to 11 the overall uncertainty u_{qm} or u_{qV} of determination of the mass flowrate q_m or the volume flowrate q_V will not exceed the values in table 4 provided that the pressure ratio r_d is not less than the value indicated in table 4.

Table 4 — Uncertainties of flowrate determination

Installation of Venturi nozzle	**Duct inlet** [figure 17 a)]	**Free inlet** [figure 17 b)]	**Free inlet with screen** [figure 17 c)]
Pressure ratio $r_d = 1 - \Delta p/p_u$	0,75	0,75	0,90
Uncertainty of flow coefficient u_α	± 1,5 %	± 1,2 %	± 1,7 %
Uncertainty of flowrate u_{qm} or u_{qV}	± 2,0 %	± 1,75 %	± 2,0 %

23 Determination of flowrate using multiple nozzles or Venturi nozzle

23.1 Installation

For tests in standardized airways, multiple nozzles shall be used within inlet or outlet chambers. The nozzles may be of varying sizes but shall be symmetrically positioned relative to the axis of the chamber, as to both size and radius.

Nozzles with integral throat taps shall be used for ducted installations. Upstream pressure taps shall be located as shown in the figure for the appropriate installation. Downstream taps are integral throat taps and shall be located as shown in figure 19.

23.2 Geometric form

23.2.1 Multiple-nozzle dimensions and tolerances are shown in figure 19.

The profile shall be axially symmetrical and the outlet edge shall be square, sharp and free from burrs, nicks or roundings. The axes of the nozzle(s) and of the chamber in which they are installed shall be parallel. Nozzle throat length L shall be either $0{,}6d \pm 0{,}005d$ (recommended) or $0{,}5d \pm 0{,}005d$.

23.2.2 Nozzles shall have an elliptical form as shown in figure 19, but two or three radii approximations that do not differ at any point, in the normal direction, by more than $0{,}015d$ from the elliptical form may also be used.

23.2.3 The nozzle throat diameter d shall be measured to an accuracy of $0{,}001d$ at the minor axis of the ellipse and the nozzle exit. Four measurements shall be taken at angular spacings of 45° and shall be within $\pm 0{,}002d$ of the mean.

At the entrance to the throat, the mean diameter may be $0{,}002d$ greater, but no less than the mean diameter at the nozzle exit.

23.2.4 The nozzle interior surface shall be faired smooth so that a straightedge may be rocked over the surface without clicking and the surface waviness shall not be greater than $0{,}001d$ peak-to-peak.

23.2.5 Where nozzles are used in a chamber, either of the types shown in figure 19 shall be used.

Where a nozzle discharges directly to a duct or a diffuser, a nozzle with throat taps shall be used, the nozzle outlet shall be flanged, and the four pressure taps shall be connected to a piezometer ring.

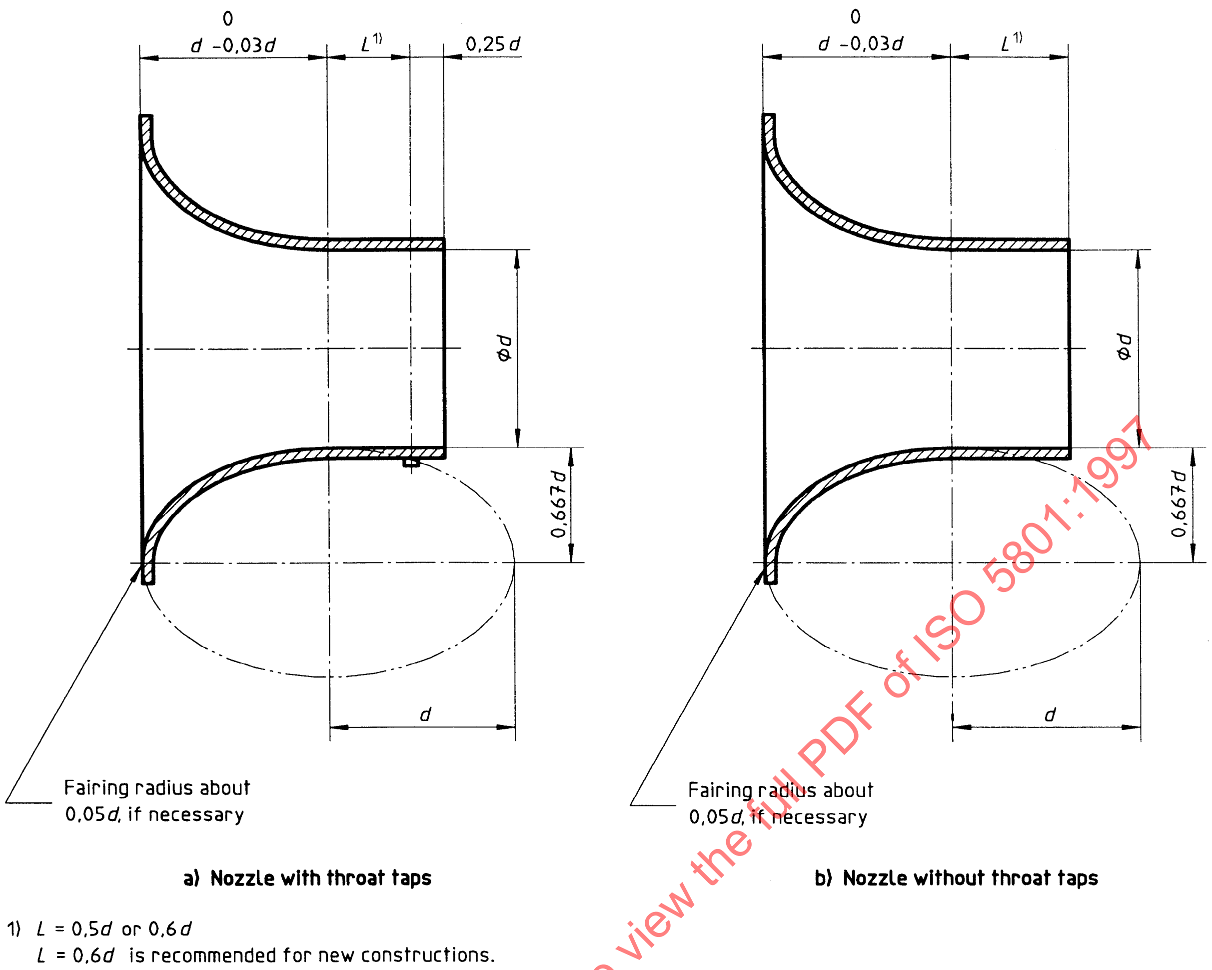


Figure 19 — Nozzle geometry

23.2.6 Nozzle throat taps shall consist of four static pressure taps 90° apart connected to a piezometer ring.

23.2.7 When a duct is connected to the nozzle, the ratio of the throat diameter to the diameter of the inlet duct shall not exceed 0,525. Ducts connected to the upstream side of a nozzle shall be straight and have uniform circular cross-sections. They shall have a length of between 6,5 and 6,75 times their diameter when used to provide a measuring section, or between 9,5 and 9,75 times their diameter when used as an outlet duct.

23.3 Inlet zone

Multiple nozzles shall be positioned such that the centreline of each nozzle is not less than 1,5d from the chamber wall. The minimum distance between the centres of any two nozzles in simultaneous use shall be 3d where d is the diameter of the large nozzle.

23.4 Multiple-nozzle and Venturi-nozzle characteristics

23.4.1 A multiple nozzle installation manufactured in accordance with the requirements in 23.3 may be used uncalibrated for pressure ratios $r_d > 0{,}9$ (i.e. $\Delta p < 10$ kPa).

23.4.2 The nozzle flowrate coefficient α is obtained from table 5 or may be calculated from the following expressions:

$$\alpha = \left[0{,}998\,6 - \frac{7{,}006}{\sqrt{Re_d}} + \frac{134{,}6}{Re_d}\right]\left[\frac{1}{\sqrt{1-\alpha_{Au}\beta^4}}\right] = \frac{C}{\sqrt{1-\alpha_{Au}\beta^4}}$$

for $L/d = 0{,}6$

Table 5 — Flowrate coefficients for nozzles used in a chamber

Nozzle flowrate coefficient α	Reynolds number, Re_d L/d = 0,5	Reynolds number, Re_d L/d = 0,6	Nozzle flowrate coefficient α	Reynolds number, Re_d L/d = 0,5	Reynolds number, Re_d L/d = 0,6
0,950	12 961	14 720	0,973	57 519	63 948
			0,974	62 766	69 736
0,951	13 657	15 491			
0,952	14 401	16 314	0,975	68 713	76 295
0,953	15 196	17 195	0,976	75 488	83 765
0,954	16 047	18 137	0,977	83 249	92 320
			0,978	92 195	102 180
0,955	16 961	19 148	0,979	102 576	113 620
0,956	17 942	20 234			
0,957	18 998	21 402	0,980	114 715	126 992
0,958	20 136	22 661	0,981	129 024	142 753
0,959	21 365	24 021	0,982	146 048	161 500
			0,983	166 513	184 032
0,960	22 695	25 492	0,984	191 401	211 428
0,961	24 137	27 086			
0,962	25 703	28 817	0,985	222 073	245 182
0,963	27 407	30 701	0,986	260 450	287 409
0,964	29 268	32 758	0,987	309 324	341 172
			0,988	372 865	411 057
0,965	31 303	35 006	0,989	457 538	504 164
0,966	33 535	37 472			
0,967	35 989	40 184	0,990	573 788	631 966
0,968	38 697	43 174	0,991	739 389	813 986
0,969	41 693	46 482	0,992	986 593	1 085 643
			0,993	1 378 954	1 516 727
0,970	45 018	50 153	0,994	2 056 291	2 260 760
0,971	48 723	54 242	0,995	3 377 887	3 712 194
0,972	52 866	58 815			

or

$$\alpha = \left[0{,}998\,6 - \frac{6{,}688}{\sqrt{Re_d}} + \frac{131{,}5}{Re_d}\right]\left[\frac{1}{\sqrt{1-\alpha_{Au}\beta^4}}\right] = \frac{C}{\sqrt{1-\alpha_{Au}\beta^4}}$$

for L/d = 0,5

where

Re_d is the Reynolds number based on the exit diameter, which may be estimated by the following expression:

$$Re_d = 0{,}95\varepsilon d \frac{\sqrt{2\rho_u \Delta p}}{(17{,}1 + 0{,}048 t_u)} \times 10^6$$

αA_u is the kinetic energy coefficient upstream of the nozzle, equal to 1,043 for an in-duct nozzle and 1 for a nozzle and a multiple nozzle in chamber or a free-inlet nozzle;

$\beta = d/D$ (which may be taken as 0 for a chamber) ($\beta \leqslant$ 0,525 for an in-duct nozzle);

C is the nozzle discharge coefficient.

23.4.3 The expansibility factor is obtained from table 6 or may be calculated from:

$$\varepsilon = \left[\frac{\kappa r_d^{2/\kappa} \left(1 - r_d^{\frac{\kappa - 1}{\kappa}} \right)}{(\kappa - 1)\,(1 - r_d)} \right]^{0,5} \left[\frac{1 - \beta^4}{1 - r_d^{2/\kappa} \beta^4} \right]^{0,5}$$

$$r_d = \frac{p_u - \Delta p}{p_u} = 1 - \frac{\Delta p}{p_u}$$

This expression may be replaced by the following:

$$\varepsilon = \left[\frac{\kappa r_d^{2/k}}{\kappa - 1} \; \frac{1 - \beta^4}{1 - \beta^4 r_d^{2/\kappa}} \; \frac{1 - r_d^{\frac{\kappa - 1}{\kappa}}}{1 - r_d} \right]^{0,5}$$

Table 6 — Expansibility factors for nozzles used in a chamber

Static pressure ratio r_d	Ratio of diameters, β					
	0	**0,20**	**0,25**	**0,30**	**0,40**	**0,50**
	Expansibility factor, ε					
1,00	1,000 00	1,000 00	1,000 00	1,000 00	1,000 00	1,000 00
0,98	0,989 23	0,989 21	0,989 17	0,989 11	0,988 86	0,988 29
0,96	0,978 34	0,978 29	0,978 23	0,978 11	0,977 61	0,976 50
0,94	0,967 32	0,967 26	0,967 16	0,966 99	0,966 25	0,964 61
0,92	0,956 19	0,956 10	0,955 98	0,955 75	0,954 78	0,952 63
0,90	0,944 92	0,944 81	0,944 66	0,944 38	0,943 19	0,940 55

23.4.4 The mass flowrate is given by the following expression:

$$q_m = \varepsilon \sum_{i=1}^{n} \left(\alpha_i d_i^2 \right) \frac{\pi}{4} \sqrt{2 \rho_u \Delta p}$$

for a multiple nozzle,

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

for a Venturi nozzle,

where

$\sum_{i=1}^{n} \left(\alpha_i d_i^2 \right)$ is the sum of the square of the various open nozzle diameters multiplied by their respective flowrate coefficients;

ρ_u is the upstream density according to 22.3.5.

23.5 Uncertainty

23.5.1 The uncertainty in the discharge coefficient C is ± 1,2 % for $Re_d \geqslant 1{,}2 \times 10^4$.

24 Determination of flowrate using a quadrant inlet nozzle

24.1 Installation

The inlet nozzle shall only be used in open (free) inlet conditions and shall conform to figure 20.

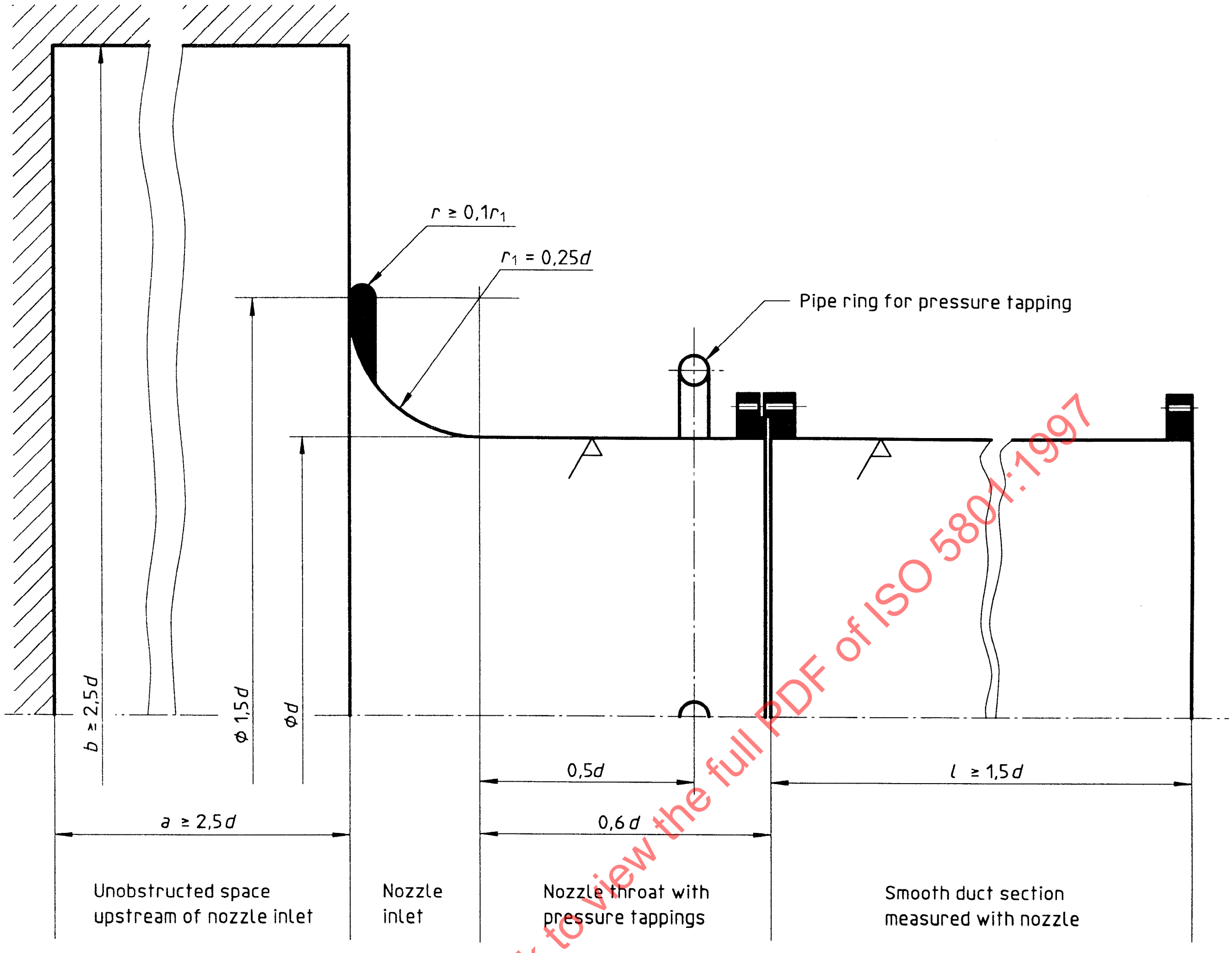


Figure 20 — Quadrant inlet geometry

24.2 Geometric form

24.2.1 The inlet quadrant measuring nozzle shall correspond to a standard nozzle with an extended cylindrical duct section that is hollowed out to a smooth finish of defined accuracy. This ensures that the flow in the element directly connected to the duct does not influence the measurement conditions at the nozzle inlet. If it is necessary to make any enlargements to the cross-section of the test airway, these shall be made only in the connection to the cylindrical duct of the inlet measuring nozzle. The nozzle and extended cylindrical duct section are shown in figure 20.

24.2.2 The nozzle throat diameter d shall be measured to an accuracy of $0{,}001d$.

Four measurements shall be taken at angular spacings of 45° and shall be within $\pm\,0{,}002d$ of the mean value.

Pressure tappings shall be in accordance with the dimensions shown in figure 20 and the requirements specified in clause 7.

The surface of the nozzle, the measuring duct and the extended duct shall be faired smooth.

24.3 Unobstructed space in front of the inlet nozzle

The minimum dimensions of the unobstructed rectangular space around the midpoint of the inlet depend on the size of the nozzle diameter d.

$$a = b \geqslant 2{,}5d$$

where

a is the distance between the midpoint of the inlet and upstream plane, see figure 17 b) and c) and figure 20.

b is the side of the rectangular space.

24.4 Quadrant inlet nozzle performance

The mass flowrate is given by

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

where

$$\alpha = 1 - 0{,}004 \sqrt{\frac{10^6}{Re_d}}$$

for $10^5 \leqslant Re_d \leqslant 10^7$ and $d \geqslant 50$ mm;

ε is the expansibility factor

$$\varepsilon = 1 - 0{,}55 \frac{\Delta p}{p_a}$$

for $\kappa = 1{,}4$ and $\Delta p \leqslant 2\,000$ Pa;

$\rho_u = \rho_a$ is the upstream density calculated in accordance with 22.3.5.

24.5 Uncertainty

The uncertainty in the flowrate coefficient is $\pm\, 0{,}003$ for $\alpha \leqslant 1$ with $Re_d \geqslant 10^5$.

25 Determination of flowrate using a conical inlet

The conical inlet shall only be used in open (free) inlet conditions.

25.1 Geometric form

25.1.1 The conical inlet dimensions and tolerances are given in figure 21. The profile shall be axially symmetric, the junctions between the cone and the face and between the cone and the cylindrical throat each having a sharp edge, free from ridges and projections. The axis of the inlet and that of the airway shall be coincident.

25.1.2 The throat diameter, d, is the arithmetic mean of four measurements, to within an accuracy of $0{,}001d$, taken at angular spacings of about 45° in the plane of the throat pressure tappings.

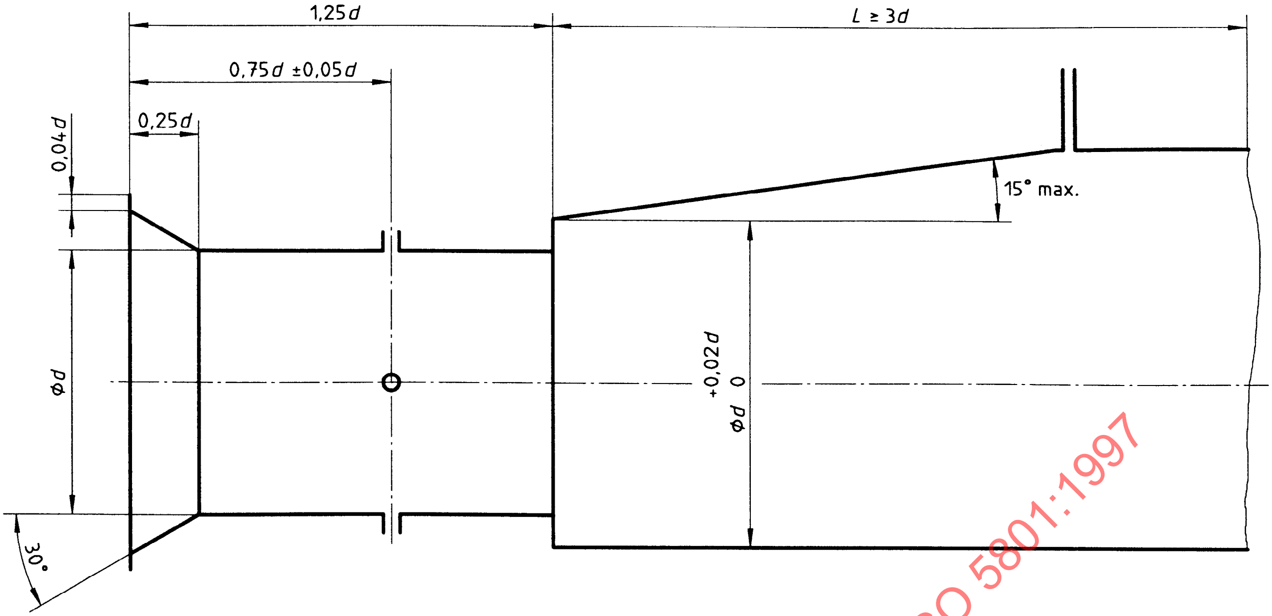
25.1.3 The pressure tappings shall conform to the requirements of clause 7.

25.1.4 The pressure difference Δp shall be measured in accordance with the requirements in 13.2.3.

25.1.5 Except where otherwise specified, the included angle of the divergent section may lie anywhere in the range $\theta \leqslant 30°$. The divergent or cylindrical connection piece shall be not less than $3d$ long.

25.2 Screen loading

25.2.1 Adjustable screen loading in accordance with figure 17 c) is permissible with the conical inlet, but the uncertainty of the flowrate coefficient α is increased (see 25.5.3).



NOTE — Four wall tappings as specified in clause 7.

Figure 21 — Geometry of conical inlet

25.2.2 Screens, antiswirl devices and their supports may be installed in the connection piece, but they shall not be allowed to encroach upon the nozzle throat.

25.2.3 Supports for screens shall have the minimal frontal area consistent with strength and stiffness for their purpose. For example, no single transverse member should present a blockage greater that 2 %. The supports shall ensure that the screens are not allowed to bow in the middle.

NOTE 40 An antiswirl device makes an excellent screen support, see figure 17 c).

Screens shall be accurately cut and a supporting ring with a radial thickness of 0,012d or 6 mm max. and 0,008d or 3 mm min. and a length of 0,05d max. shall be fitted, or other means adopted of eliminating leakage at the wall.

25.3 Inlet zone

25.3.1 Within the inlet zone defined in figure 17 c), there shall be no external obstruction to the free movement of the air entering the inlet, and the velocity of any cross-currents should not exceed 5 % of the nozzle throat velocity.

25.3.2 Steps shall be taken to ensure that the pressure registering at the high-pressure limb of the differential pressure-reading manometer is the ambient pressure in the inlet zone.

25.4 Conical inlet performance

25.4.1 A conical inlet manufactured in accordance with the above requirements may be used uncalibrated for pressure ratios $r_d \geqslant 0{,}96$, i.e. $\Delta p < 4\ 000$ Pa.

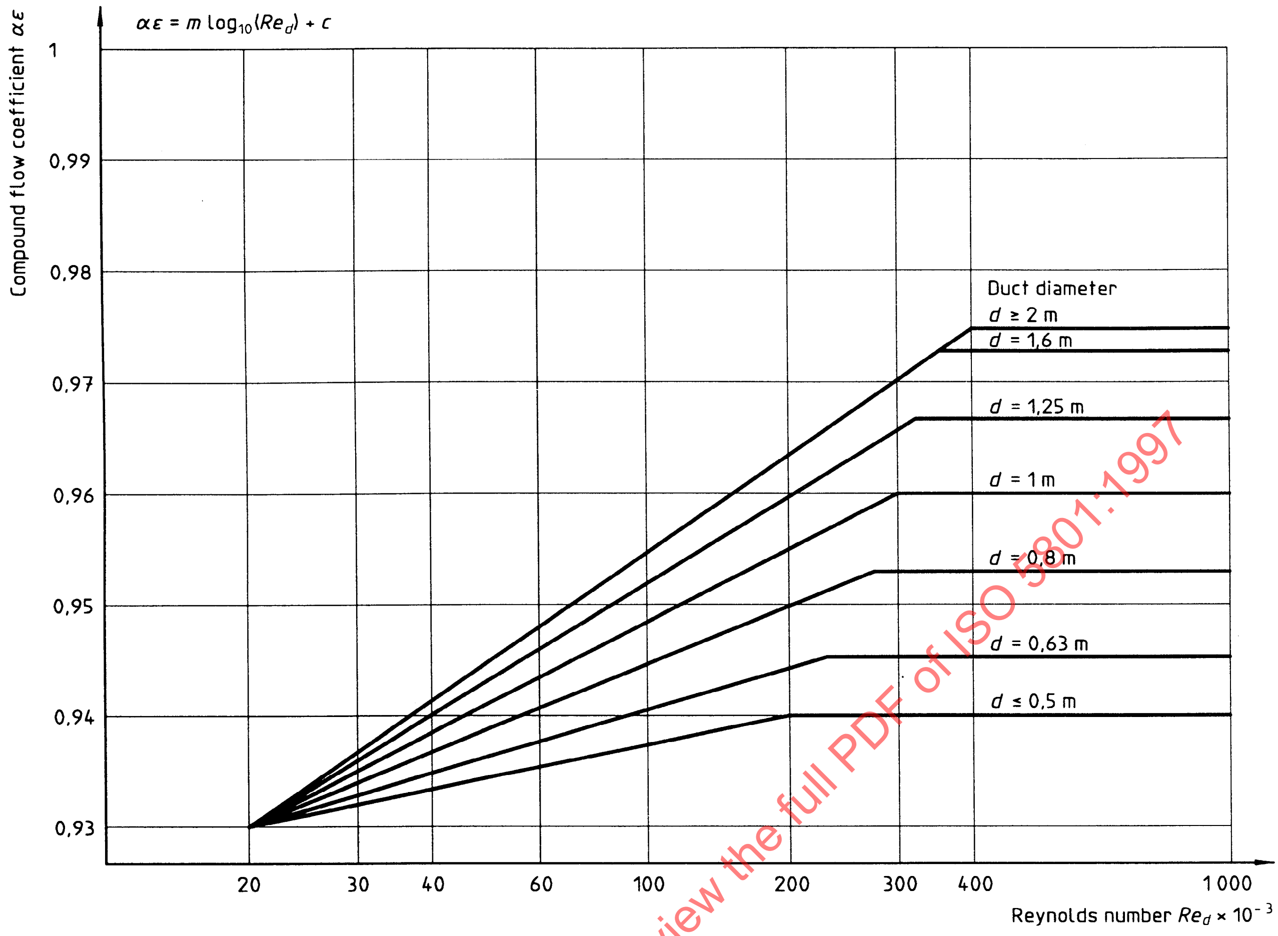
25.4.2 The compound coefficient $\alpha\varepsilon$ is dependent on the Reynolds number Re_d and is plotted in figure 22.

Conical inlets shall not be used when $Re_d < 20\ 000$.

25.4.3 The mass flowrate is given by the following expression:

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

where ρ_u is the upstream density calculated in accordance with 22.3.5.



For $d \leqslant 0{,}5$ m: $m = 0{,}011\ 07$, $c = 0{,}882\ 4$, $\alpha\varepsilon$ max. = 0,940
For 0,5 m $< d \leqslant 2$ m: $m = 0{,}009\ 63 + 0{,}047\ 83d + 0{,}055\ 33d^2$,
$c = 0{,}971\ 5 - 0{,}205\ 8d + 0{,}055\ 33d^2$,
$\alpha\varepsilon$ max. = $0{,}913\ 1 + 0{,}062\ 3d - 0{,}0156\ 7d^2$.
For $d \geqslant 2$ m: $m = 0{,}345\ 9$, $c = 0{,}781\ 2$, $\alpha\varepsilon$ max. = 0,975

Figure 22 — Compound flowrate coefficient $\alpha\varepsilon$ of conical inlets

25.5 Uncertainties

25.5.1 The uncertainty in the compound coefficient $\alpha\varepsilon$ and that in the flow coefficient α are the same. The basic uncertainty, applicable when $Re_d \geqslant 3 \times 10^5$ and when no screen loading is allowed in the connection piece is ± 1,5 %. To this shall be arithmetically added (if applicable) the next additional uncertainty associated with low Re_d and screen loading.

25.5.2 The additional uncertainty (in percent) due to low Re_d (i.e. $2 \times 10^4 < Re_d < 3 \times 10^5$) is as follows:

$$\pm\left(\frac{2\times 10^4}{Re_d} - \frac{1}{15}\right)$$

25.5.3 The additional uncertainty due to the presence of a uniform screen complying with 25.2 is 0,5 % and shall be added arithmetically.

25.5.4 These uncertainties may be reduced if a calibrated value of $\alpha\varepsilon$ is used in place of the value given in 25.4.2. The calibration may be carried out using a Pitot-static traverse in accordance with the requirements of ISO 3966 or by means of a primary device with an uncertainty of flowrate coefficient not exceeding 1,0 %. The overall uncertainty of mass or volume flowrate measurement with screen loading in accordance with figure 17 c) may then be taken as ± 2 %.

26 Determination of flowrate using an orifice plate

26.1 Installation

For tests in standardized airways, a common design of orifice plate may be used at the inlet to a test duct (inlet orifice), at the outlet from a test duct (outlet orifice) or between upstream and downstream ducts of the same diameter (in-duct orifice in accordance with ISO 5167-1). The ducts shall conform to the requirements of the relevant test method.

26.2 Orifice plate

26.2.1 The orifice plate and the associated pressure tappings shall conform to the dimensions shown in figure 23, to the additional requirements of this clause and to ISO 5167-1:1991, clauses 7 and 8.

Two alternative types of tapping are available, the piezometer ring being generally the more convenient for small ducts and the wall tapping for larger sizes, although neither usage is exclusive.

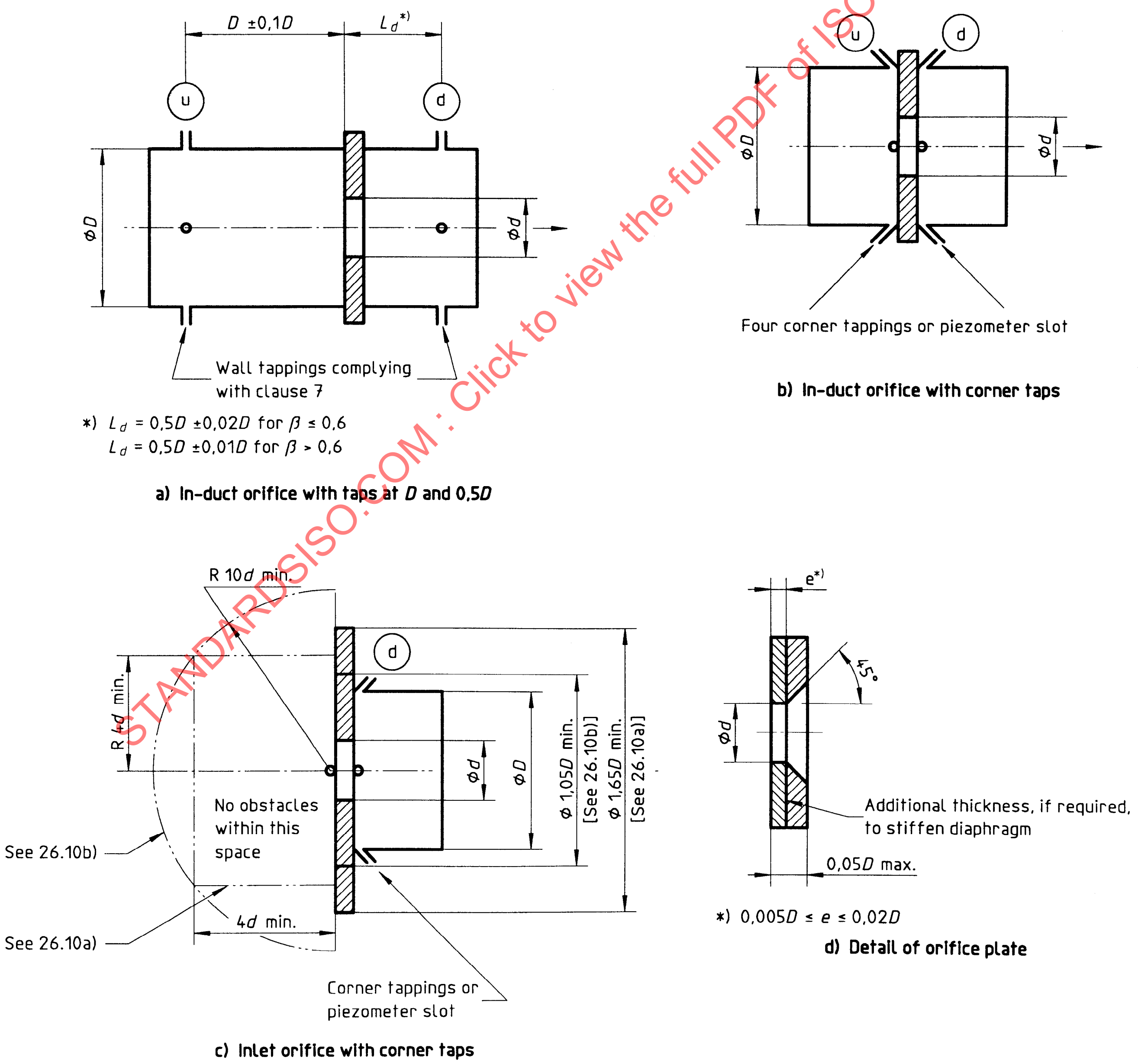
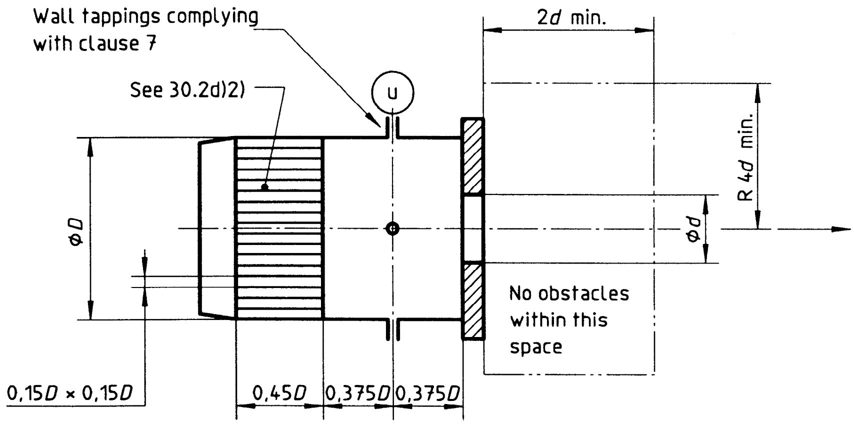
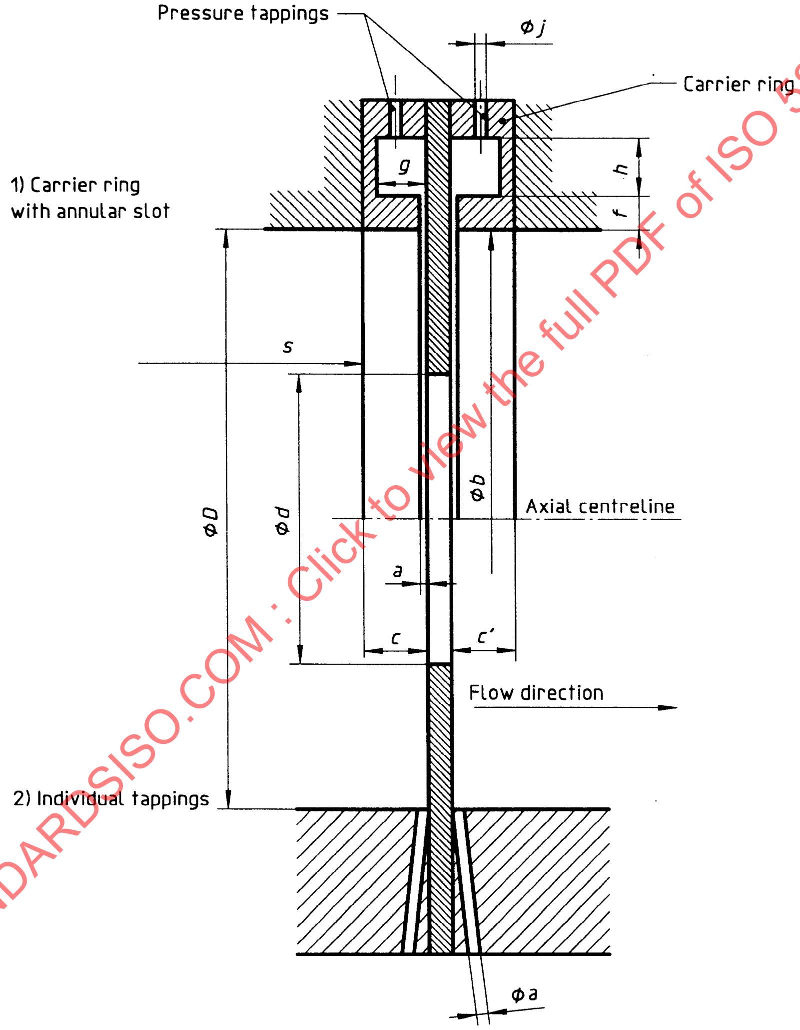


Figure 23 — Orifice plates and assemblies



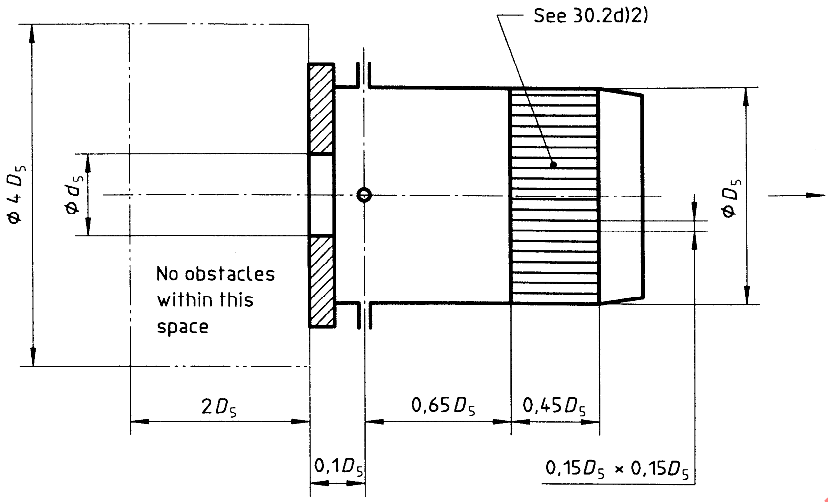
e) Outlet orifice with wall tappings



f : Thickness of the slot
c : Length of the upstream ring
c': Length of the downstream ring
b : Diameter of the carrier ring
a : Width of the annular slot or diameter of single tapping
s : Distance from upstream step to carrier ring
j : Diameter of pressure tapping
$4 < j < 10$
$gh \geq \pi a D$

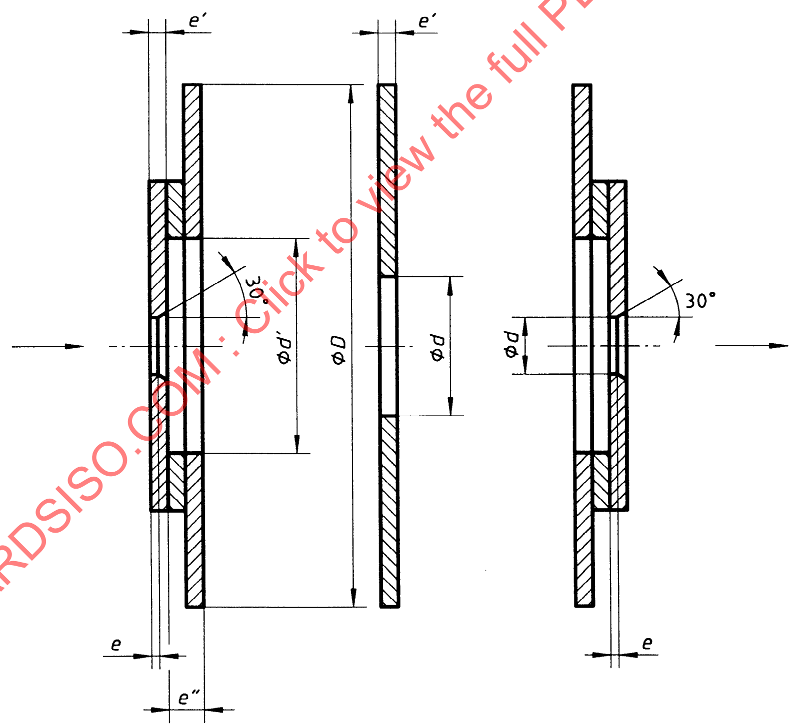
f) Corner tappings

Figure 23 — Orifice plates and assemblies (*continued*)



NOTE – Orifice plate fastening:
- by flange: internal diameter $\geq D_5$
  thickness $\leq 0{,}01D_5$
- by clamp: total thickness $\leq 0{,}15D_5$
  radial obstruction $\leq 0{,}15D_5$ above D_5

g) Inlet orifice with wall tappings



Inlet orifice

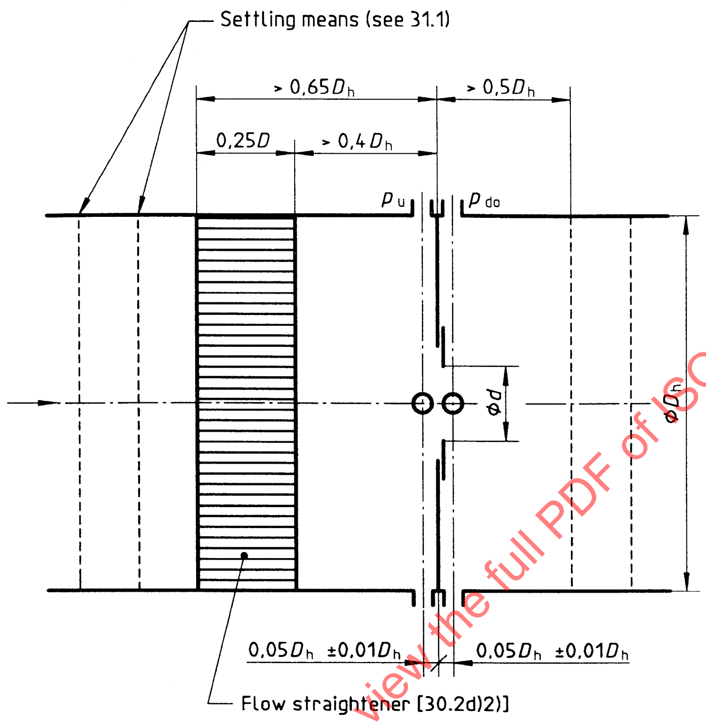
Outlet orifice

$e' \geq 0{,}003D$
$e \leq 0{,}01d$

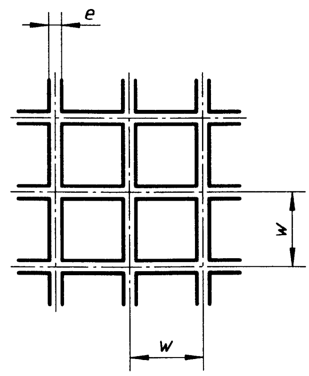
NOTES
1 Chamfer for $e' \geq 0{,}01d$
2 Where the orifice plate is bolted to a supplementary plate:
$d' \geq 1{,}25d + 4e'$

h) Detail of inlet or outlet orifice plates for wall tappings, see 26.9, 26.9.1 and 26.11

Figure 23 — Orifice plates and assemblies *(continued)*



i) Orifice plate in test chamber (inlet-side or outlet-side), see 26.9.1, 31.4 and 31.3



w = 0,075D_h ± 0,005D_h
e < 0,005D_h

j) Detail of flow straightener

Figure 23 — Orifice plates and assemblies *(end)*

26.2.2 The orifice plate should be constructed from material which will not corrode in service, and it should be protected from damage when handling and cleaning. It is particularly important that the edges of the orifice should not be burred or rounded, or sustain other damage visible to the naked eye.

The upstream edge of the orifice shall be sharp and shall not reflect light. Any edge radius should not exceed $0{,}0004d$. These conditions may be met by machining the orifice plate, fine-boring the orifice, and then finishing the upstream face by a very fine radial cut from the centre outwards.

26.2.3 The orifice shall be cylindrical within $\pm 0{,}000\,5d$, its diameter being measured to the nearest $0{,}001d$. After assembly, the orifice shall be coaxial with the upstream duct within $\pm 1°$ and $\pm (0{,}005D)/(0{,}1 + 2{,}3\beta^4)$.

26.2.4 The upstream face of the orifice plate shall be flat within 1 mm per 100 mm and its roughness, R_a, should not exceed $10^{-4}d$. Any gasket sealing the plate and the duct flange shall not project internally.

26.3 Ducts

For in-duct orifices in accordance with ISO 5167-1, the upstream duct diameter D shall be determined, to the nearest $0{,}003D$, as the average of 12 measurements at about 45° in three cross-sections equally distributed between the upstream tapping and the section at $0{,}5D$ upstream. It is sufficient for the downstream side duct to be nominally cylindrical and of diameter $D \pm 0{,}03D$.

The length of the upstream and downstream ducts is given in clause 7 of ISO 5167-1:1991.

A flow straightener as shown in figure 55 or 57 shall be fitted in the upstream duct. The length of the upstream and downstream ducts and the installation conditions are given in clauses 7 and 8 of ISO 5167-1:1991.

26.4 Pressure tappings

26.4.1 Wall tappings shall be four in number, in accordance with clause 7, and in the locations shown in figure 23. The axis of each tap should intersect the duct axis at right angles, except in the case of corner taps which may be inclined at an angle sufficient to enable them to be satisfactorily drilled and connected.

The dimensions of the wall tapping holes shall conform to the dimensions shown in figure 2, except that, in the case of corner taps, no part of the hole should extend more than $0{,}03D$ for $\beta \leqslant 0{,}65$ or $0{,}02D$ for $\beta > 0{,}65$ from the orifice plate at the point where the hole breaks through the carrier ring. Any gasket shall be included in this dimension.

26.4.2 Piezometer rings take the form of a continuous circumferential slot of axial width a to the orifice plate and radial depth $2a$ minimum. Width a shall not exceed $0{,}03D$ for $\beta \leqslant 0{,}65$ or $0{,}02D$ for $\beta > 0{,}65$. Provided the cross-sectional area of the circumferential recess is at least equal to the minimum indicated in figure 23 f), a single tapping can be used for connection to the manometer.

The bore of the piezometer carrier ring shall not protrude within the bore of duct of diameter D at any point, a diametral allowance of $0{,}04D$ being made to accommodate out-of-roundness in the duct.

26.5 Calculation of mass flowrate

$$q_m = \alpha\, \varepsilon\, \pi \frac{d^2}{4} \sqrt{2\rho_u \Delta p}$$

where ρ_u is in accordance with 22.3.5.

The definitions and limitations on the quantities on the right-hand side of the equation differ slightly according to the orifice installation adopted, and are therefore considered separately for each case. The following limits apply for an in-duct orifice (ISO 5167-1):

— The duct diameter D shall be not less than 50 mm and not more than 1 000 mm for D and $D/2$ taps and corner taps.

— The orifice diameter d shall not be less than 12,5 mm (see ISO 5167-1:1991, subclause 8.1.7).

— The flowrate coefficient α depends on the orifice diameter ratio $\beta = d/D$ and on the duct Reynolds number Re_D (see 26.6). The ranges of β and Re_D are limited for each installation. In some cases the Re_D limits are expressed in terms of limiting pressures and velocities in standard air, for simplicity.

— The expansibility factor, ε, is given in clauses 26.7 to 26.11 and figure 26.

26.6 Reynolds number

The Reynolds numbers required for calculating orifice flowrate are defined as follows:

$$Re_D = \frac{D v_D}{\nu} = \frac{4 q_m}{\pi D \mu} = \frac{\alpha \varepsilon d^2}{\nu D} \sqrt{\frac{2\Delta p}{\rho_u}} = \frac{\alpha \varepsilon \beta d}{\nu} \sqrt{\frac{2\Delta p}{\rho_u}}$$

$$Re_d \frac{d v_d}{\nu} = \frac{4 q_m}{\pi d \mu} = \frac{\alpha \varepsilon d}{\nu} \sqrt{\frac{2\Delta p}{\rho_u}}$$

where μ is calculated in accordance with 12.3.

The kinematic viscosity ν is given by the following expression:

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

Strictly speaking, the derivation of Re from a test value of Δp requires an iterative calculation since α and q_m are not known. Only a rough approximation of Re_D is needed, however, and it may be considered sufficient to calculate Re_D or Re_d from the first approximation of q_m.

For inlet orifice it may be sufficient to take the dynamic viscosity μ as its value for standard air: $\mu = 18 \times 10^{-6}$ Pa · s.

In this case,

$$Re_D = \frac{71 q_m}{D} \times 10^3$$

or

$$Re_d = \frac{71 q_m}{d} \times 10^3$$

where D and d are expressed in metres and q_m is expressed in kilograms per second.

26.7 In-duct orifice with D and $D/2$ taps [see figure 23 a) and ISO 5167-1]

The following conditions shall apply:

$$\Delta p = p_u - p_{do} = p_{eu} - p_{edo}$$

$$p_{do}/p_u \geqslant 0{,}75$$

ρ_u is the air density at the upstream tapping (see 22.3.5.2).

$\beta = d/D$, and shall be not less than 0,2 nor greater than 0,75.

The flowrate coefficient α is given by the Stolz formula:

$$\alpha = \left(1-\beta^4\right)^{-0{,}5} \Big[0{,}595\,9 + 0{,}031\,2\beta^{2{,}1} - 0{,}184\beta^8 + 0{,}002\,9\beta^{2{,}5}\left(\frac{10^6}{Re_D}\right)^{0{,}75} + 0{,}039\beta^4\left(1-\beta^4\right)^{-1} - 0{,}015\,8\beta^3\Big]$$

and is shown in figure 24.

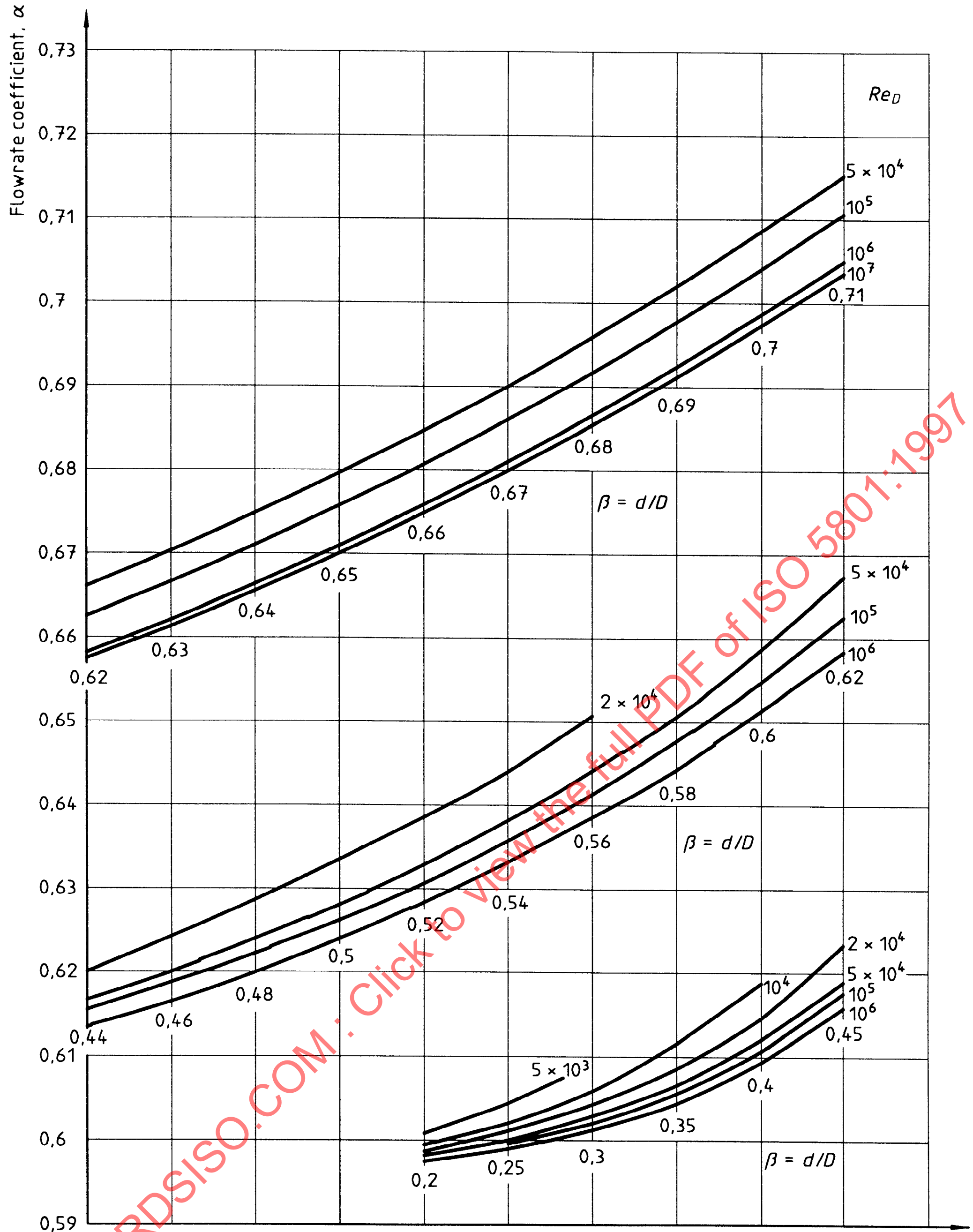


Figure 24 — Flowrate coefficient, α, of in-duct orifice with taps at D and D/2
(see 26.7)

The expansibility factor ε is given by

$$\varepsilon = 1 - \left(0{,}41 + 0{,}35\beta^4\right)\frac{\Delta p}{\kappa p_u}$$

and is shown in figure 26.

The uncertainty with which α is known is 0,6 % for $Re_D \geqslant 1\,260\beta^2 D$ (D in millimetres) for $\beta \leqslant 0{,}6$ or β % for $0{,}6 < \beta \leqslant 0{,}75$ provided the straight lengths of the ducts are in accordance with ISO 5167-1:1991, 7.2. An additional uncertainty of 0,5 % shall be arithmetically added when these lengths are divided by 2.

The uncertainty on ε, in percent, is

$$4\frac{\Delta p}{p_u}$$

26.8 In-duct orifice with corner taps [see figure 23 b) and ISO 5167-1]

The following conditions shall apply:

$\Delta p = p_u - p_{do} = p_{eu} - p_{edo}$

$p_{do}/p_u \geqslant 0{,}75$

ρ_u is the air density at the upstream tapping (see 22.3.5.2).

$\beta = d/D$, and shall be not less than 0,2 nor greater than 0,75.

The flowrate coefficient α is given by the Stolz formula:

$$\alpha = \left(1 - \beta^4\right)^{-0{,}5}\left[0{,}595\,9 + 0{,}031\,2\beta^{2{,}1} - 0{,}184\beta^8 + 0{,}002\,9\beta^{2{,}5}\left(\frac{10^6}{Re_D}\right)^{0{,}75}\right]$$

and plotted in figure 25.

The expansibility factor ε is given by:

$$\varepsilon = 1 - \left(0{,}41 + 0{,}35\beta^4\right)\frac{\Delta p}{\kappa p_u}$$

and plotted in figure 26.

The uncertainty with which α is known has the same value as for taps at D and $D/2$ (see 26.7) provided

$Re_D \geqslant 5\,000$ for $0{,}2 \leqslant \beta \leqslant 0{,}45$

$Re_D \geqslant 10\,000$ for $0{,}45 < \beta \leqslant 0{,}75$

26.9 Outlet orifice with wall tappings [see figure 23 e) and h)]

The following conditions shall apply:

$\Delta p = p_u - p_a = p_{eu} = p_{e6}$

where p_a is the ambient atmospheric pressure.

ρ_u = air density at the upstream tapping determined in accordance with 22.3.5.

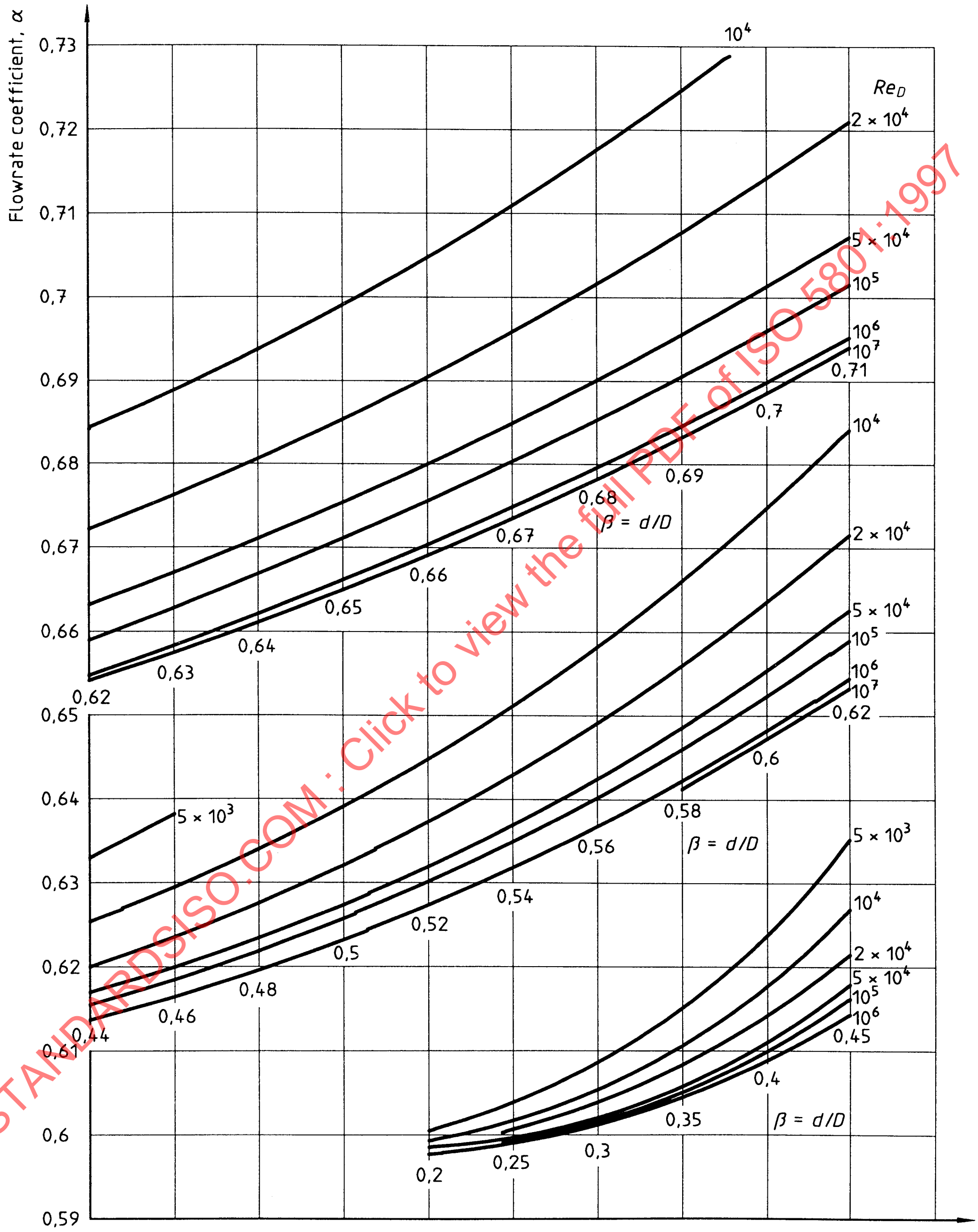


Figure 25 — Flowrate coefficient, α, of in-duct or outlet orifice with corner taps
(see 26.8)

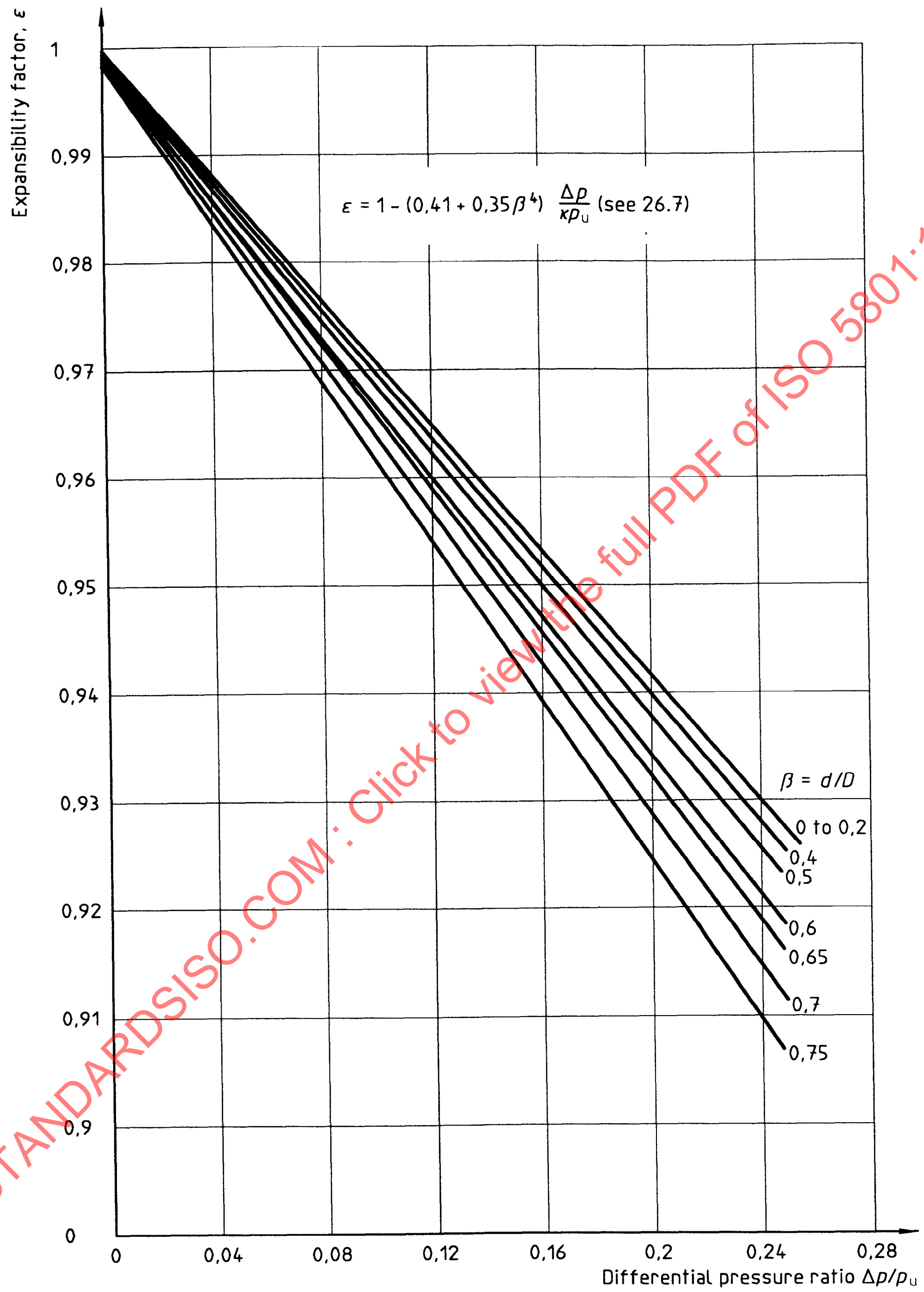


Figure 26 — Expansibility factor, ε, for orifice plates in atmospheric air (see 26.7, 26.8 and 26.10)

$\beta = d/D$ shall not exceed 0,5 (or 0,7 with additional uncertainty).

$\alpha\varepsilon$ is given by the following expression and plotted in figure 27 as function of

$$\frac{p_{e6}}{p_a} = \frac{p_{eu}}{p_a} = r_{\Delta p} = \frac{\Delta p}{p_u - \Delta p}$$

$$\alpha\varepsilon = A\left[1 - r_{\Delta p}\left(B - Cr_{\Delta p}\right)\right]$$

where A, B and C are respectively equal to

$A = 0{,}599\,3 + 0{,}159\,9\beta^2 - 0{,}915\,6\beta^4 + 6{,}567\,5\beta^6 - 9{,}142\,9\beta^8$ for $\beta \leqslant 0{,}5$

$A = 0{,}6\,(2{,}04)\beta^{3{,}2}$ for $\beta > 0{,}5$

$B = 0{,}249 + 0{,}070\,1\beta^2 + 0{,}243\beta^4 + 0{,}113\beta^6$

$C = 0{,}075\,7 + 0{,}058\beta^2 + 0{,}22\beta^4 + 0{,}25\beta^6$

The uncertainty with which $\alpha\varepsilon$ is known may be taken as $\pm 0{,}5$ % provided β is not greater than 0,5 and the Reynolds number referred to the orifice diameter d is not less than 10^5. The latter condition requires that, for normal atmospheric conditions Δp is not less than $(2\,000/d)^2$, where d is expressed in millimetres.

26.9.1 Orifice plate with wall tappings in the test chamber [see figure 23 h) and i)]

The following conditions shall apply:

$$\Delta p = p_{eu} - p_{edo} = p_u - p_{do}$$

The temperature t_u is measured in the test chamber.

$$\Theta_u = \Theta_{sgu} = t_u + 273{,}15$$

$\beta = d/D_h$ shall not exceed 0,25.

$\alpha\varepsilon$ is determined in accordance with 26.9.

The other remarks of 26.9 shall apply.

26.10 Inlet orifice with corner taps [see figure 23 c)]

Two types of setup may be used:

— The external diameter of the orifice plate is more than $1{,}65D$, D being the diameter of the down-stream duct [see figure 23 c)].

— The external diameter of the orifice plate is more than $1{,}05D$.

In the two cases the following conditions shall apply:

$$\Delta p = p_a - p_{do} = p_{e5}$$

where p_a is the ambient atmospheric pressure.

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

a) External diameter of the orifice plate more than $1{,}65D$.

$\beta' = d/D$ is the ratio of the orifice diameter to the downstream duct diameter.

β' shall not be greater than 0,84. There is no lower limit except for the diameter d min. specified in 26.5.

The flow coefficient, α, is given in figure 28 as a function of β' only.

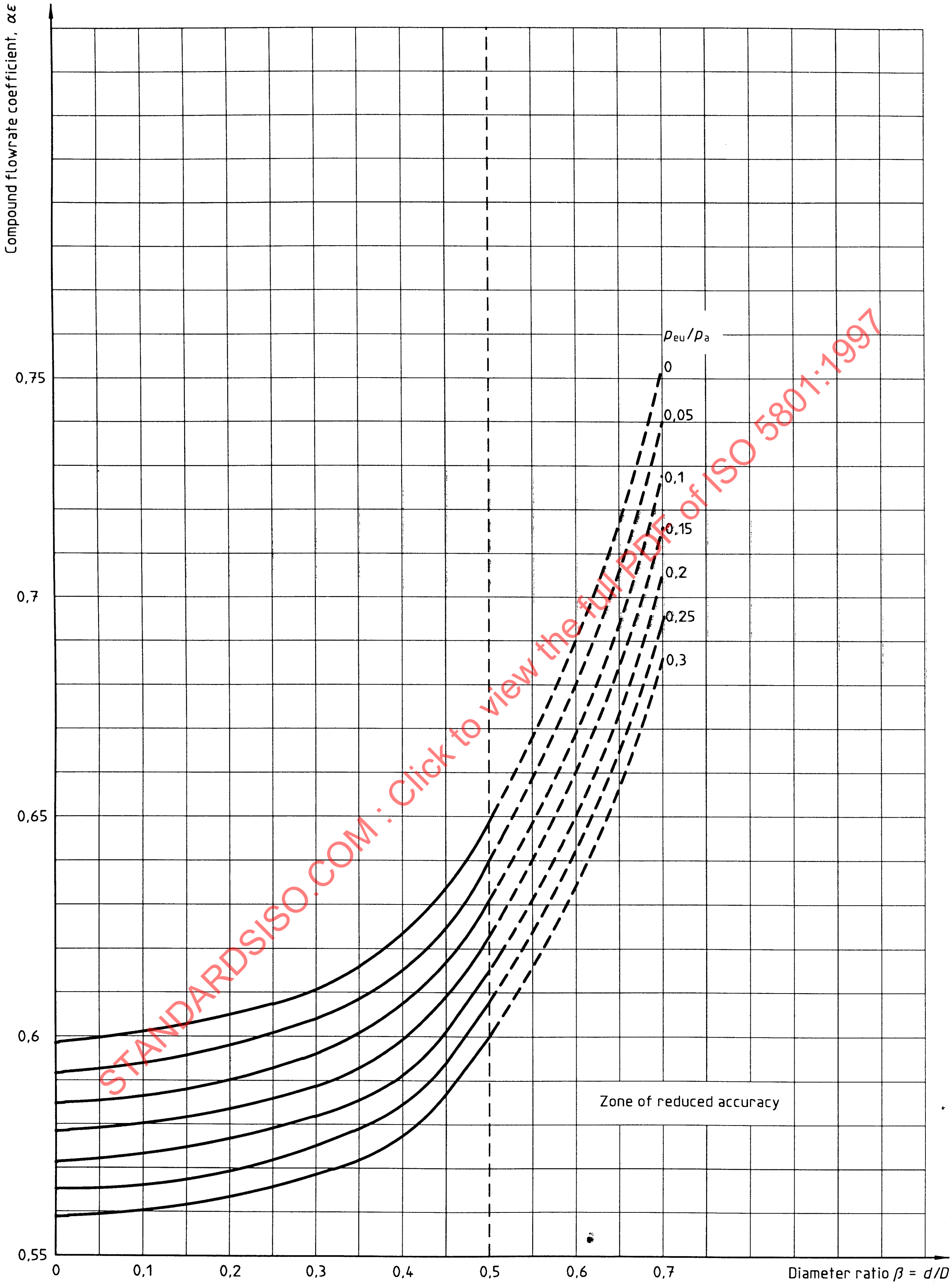


Figure 27 — Compound flowrate coefficient, $\alpha\varepsilon$, of outlet orifices with wall taps (see 26.9)

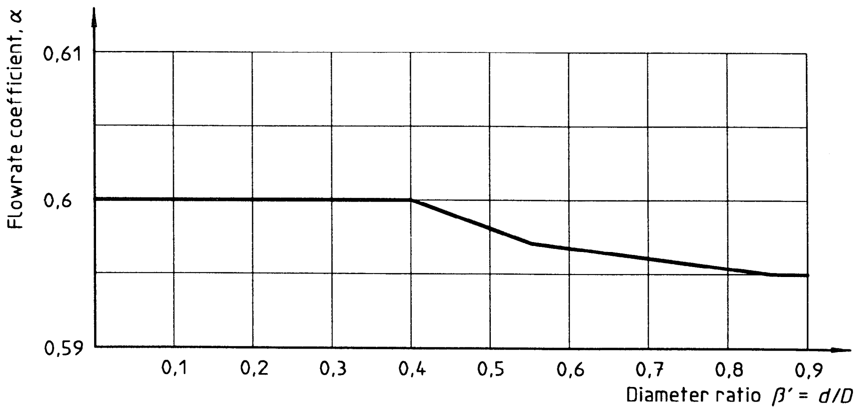


Figure 28 — Flowrate coefficient, α, of inlet orifices with corner taps [see 26.10 a)]

The expansibility factor, ε, may be determined by

$$\varepsilon = 1 - \left(0{,}41 + 0{,}35\beta^4\right)\frac{\Delta p}{\kappa p_u} = 1 - 0{,}41\frac{\Delta p}{\kappa p_u}$$

valid for

$$\frac{\Delta p}{p_u} \leqslant 0{,}25$$

and is plotted in figure 26.

The uncertainty with which α is known may be taken as ± 1,5 % provided Re_d is not less than 5×10^4. This condition requires that, for normal atmospheric conditions, Δp is not less than

$$\left[\frac{1\,000}{d}\right]^2$$

where d is expressed in millimetres.

The uncertainty, in percent, with which ε is known may be taken as

$$\pm 4\frac{\Delta p}{p_u} = \pm 4\frac{\Delta p}{p_a}$$

b) The external diameter of the orifice plate is more than 1,05D

$\beta' = d/D$ shall be not less than 0,20 and not greater than 0,75.

α is not a function of $\beta' = d/D$; α is constant and equal to 0,6.

The expansibility factor, ε, may be determined by

$$\varepsilon = 1 - 0{,}41\frac{\Delta p}{\kappa p_u}$$

valid for

$$\frac{\Delta p}{p_u} \leqslant 0{,}25$$

and is plotted in figure 26.

The uncertainty with which α is known is ± 1,5 % provided

$Re_D \geqslant 5\,000$ for $0{,}2 \leqslant \beta' \leqslant 0{,}45$

$Re_D \geqslant 10\,000$ for $0{,}45 \leqslant \beta' \leqslant 0{,}75$

The uncertainty, in percent, with which ε is known is

$$\pm 8 \frac{\Delta p}{p_u} = \pm 8 \frac{\Delta p}{p_a}$$

26.11 Inlet orifice with wall tappings [see figure 23 g) and h)]

The following conditions shall apply:

$$\Delta p = p_a - p_{do} = p_{e5}$$

where p_a is the ambient atmospheric pressure.

$$\rho_u = \rho_a$$

where ρ_a the density of the ambient atmosphere.

$\beta' = d/D$ is in this case the orifice ratio to the downstream duct.

β' shall not be greater than 0,7. There is no lower limit except for the minimum d specified in 26.5.

$$\alpha = 0{,}598$$

$$\varepsilon = 1 - r_{\Delta p}\left(0{,}249 - 0{,}075\,7 r_{\Delta p}\right)$$

$$r_{\Delta p} = p_{e5}/p_5 = \Delta p/\left(p_a - \Delta p\right)$$

The uncertainty with which α is known may be taken as $\pm 1{,}0$ % provided that $Re_D \geqslant 5 \times 10^4$ and $r_{\Delta p} = \Delta p/(p_u - \Delta p) \leqslant 0{,}3$.

27 Determination of flowrate using a Pitot-static tube traverse

27.1 General

For standardized airway tests, only traverses using Pitot-static tube in cylindrical ducts are recognized. The locations of the traverse planes will be those shown in figures 70 e), 72 d) and e, 74 f) and g), 75 b) and 76 g), and the working fluid will normally be atmospheric air.

Measurements may be made and corrections applied in accordance with ISO 3966, but for the purposes of this International Standard, it is possible to measure uncorrected velocities at the points specified, average the results and apply a single correction factor α given in 27.6 as a function of Reynolds number to determine the average velocity at the section with an uncertainty of ± 2 %.

27.2 Pitot-static tube

The instrument shall conform to the requirements of ISO 3966. The external diameter of the tube d shall not exceed one forty-eighth of the diameter D of the airway. The diameter of the stagnation pressure hole shall not be less than 1 mm.

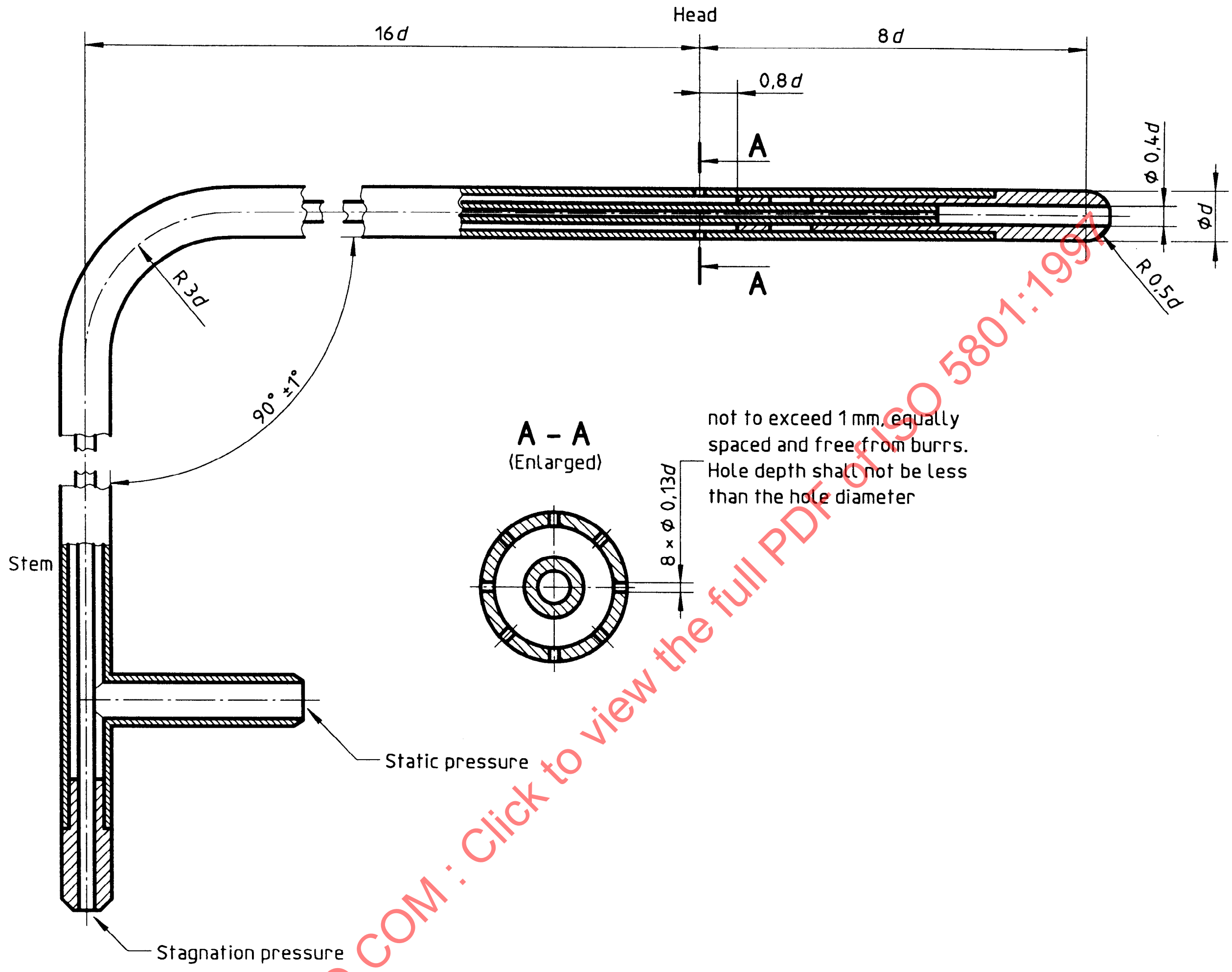
Four types of Pitot-static tube may be used:

— AMCA Type, see figure 29 a);

— NPL modified ellipsoidal-nosed, see figure 29 b);

— CETIAT Type, see figure 29 c);

— DLR Type, see figure 29 d).



NOTES

1 Head shall be free from nicks and burrs.

2 All dimensions shall be within ±2 %.

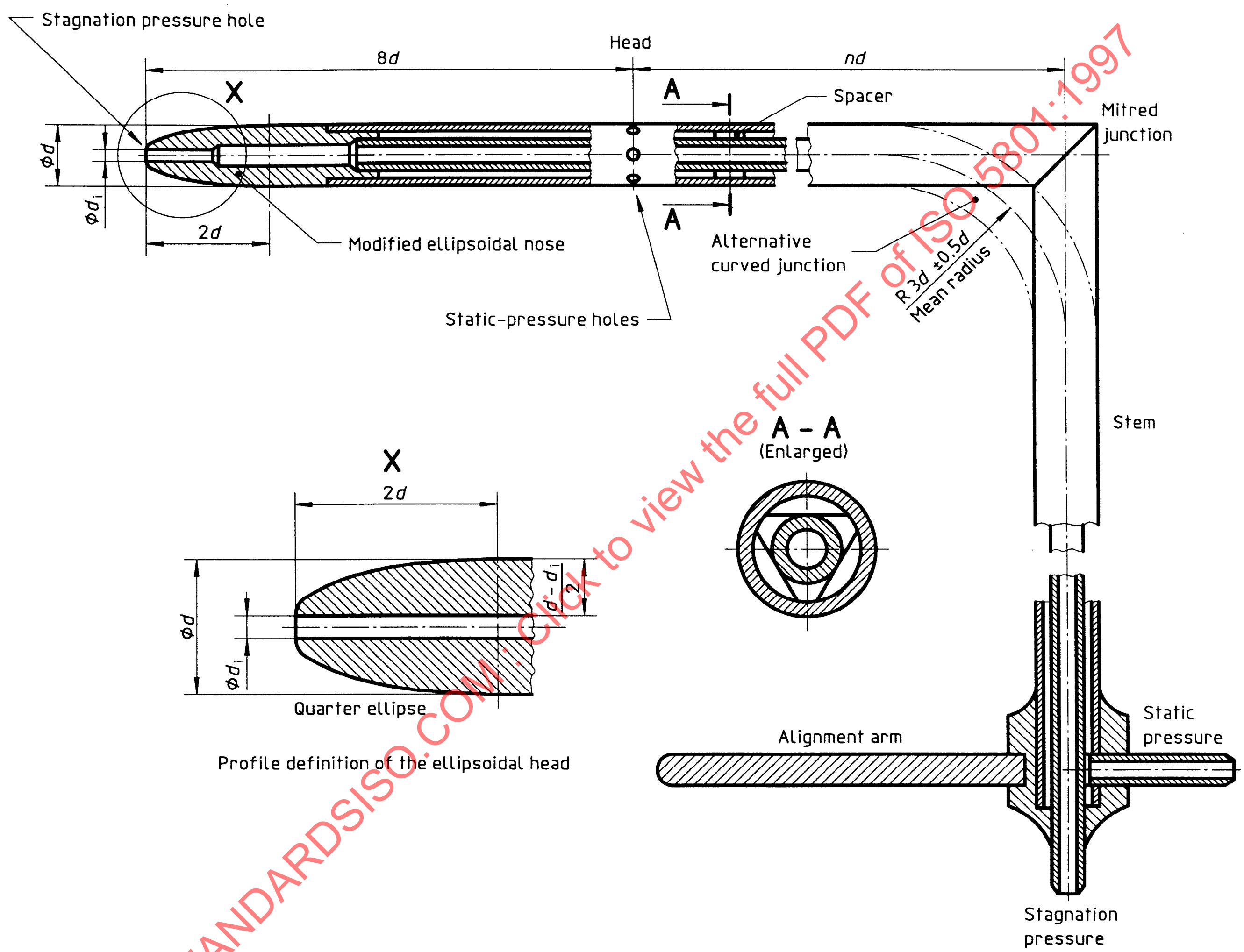
3 Surface roughness shall be 0,8 µm or better.

4 The static orifices may not exceed 1 mm in diameter.

5 The minimum Pitot tube stem diameter allowed by this International Standard is 2,5 mm. In no case shall the stem diameter exceed 1/30 of the test duct diameter.

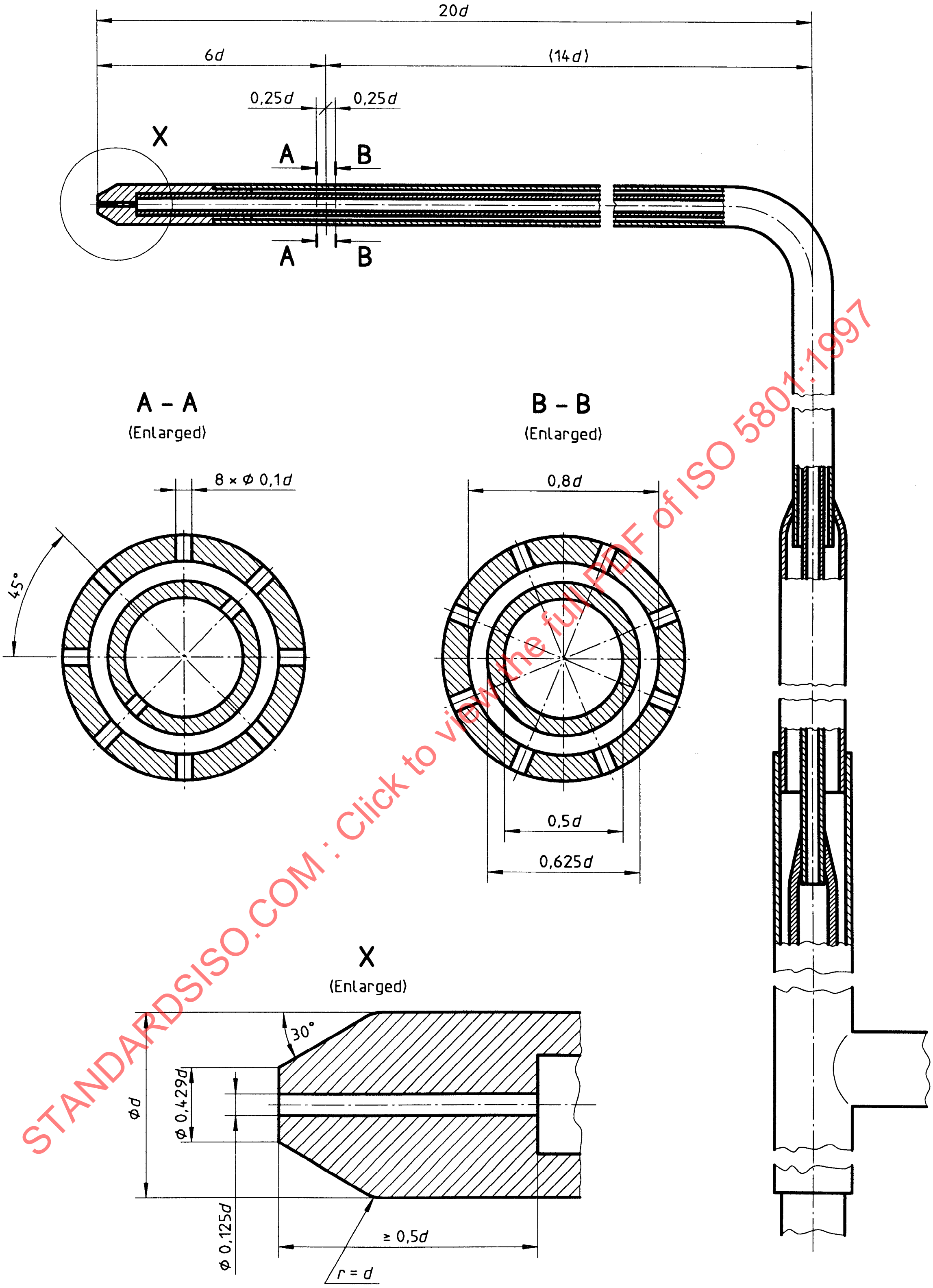
a) AMCA type

Figure 29 — Types of Pitot-static tubes



b) NPL type with modified ellipsoidal nose

Figure 29 — Types of Pitot-static tubes *(continued)*

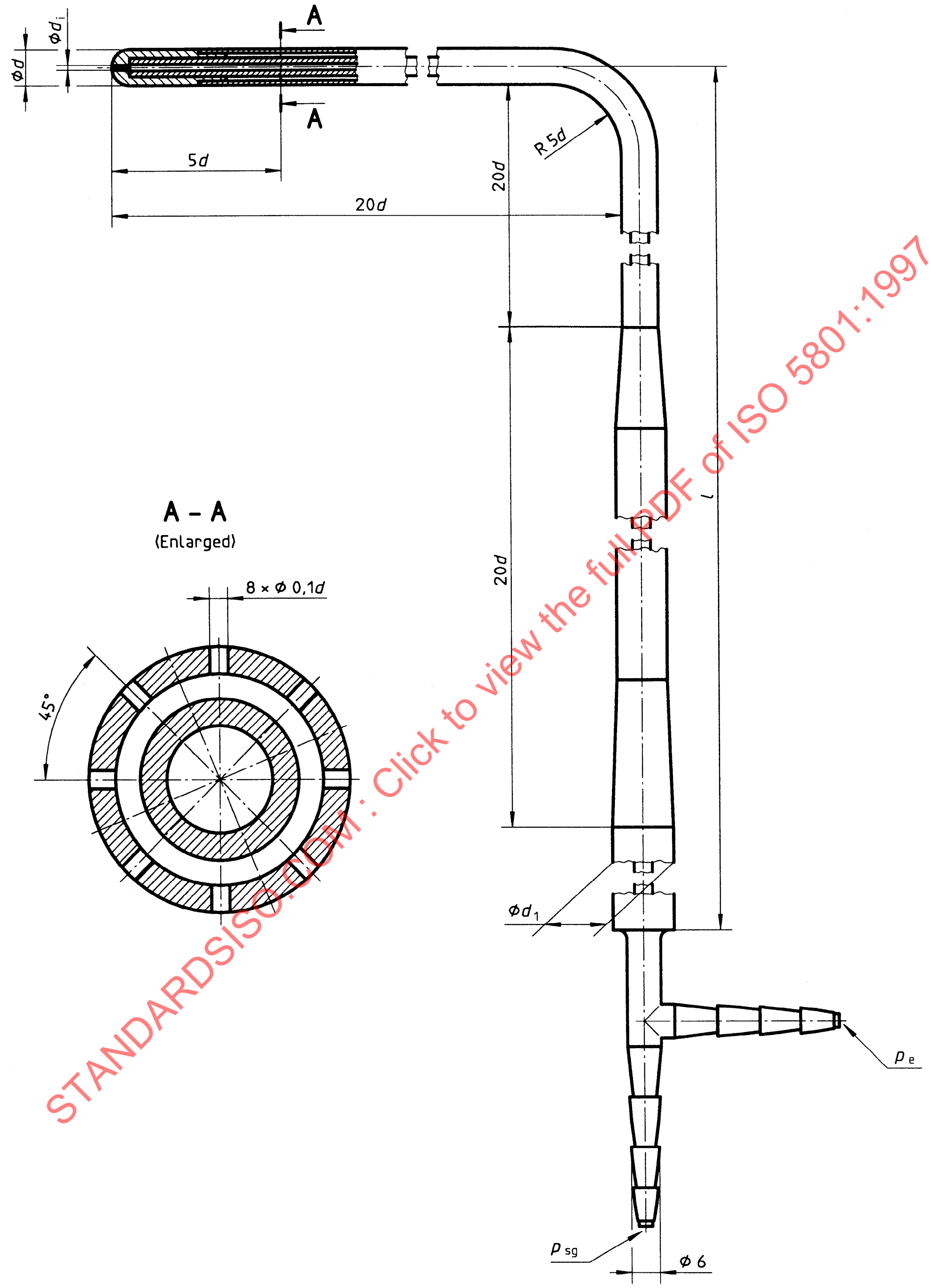


NOTE – Static-pressured taps may be limited to those indicated on section A-A, in which case section A-A shall be placed at 6d from the tube tip.

c) CETIAT type

Figure 29 — Types of Pitot-static tubes (*continued*)

Dimensions in millimetres



d) DLR type

Figure 29 — Types of Pitot-static tubes (*end*)

27.3 Limits of air velocity

The Mach number of the flow past the tube should not exceed 0,25 (85 $m \cdot s^{-1}$ atmospheric air).

The Reynolds number referred to the diameter of the stagnation pressure tapping d_i, in metres, should exceed 200. This means that, for tests with atmospheric air, the velocity, in metres per second, should not be less than:

$$v = 3/d_i$$

27.4 Location of measurement points

The centre of the nose of the Pitot-static tube shall be located successively at not less than 24 measurement points spaced along three symmetrically disposed diameters of the airway, as shown in figure 30.

The head of the Pitot-static tube shall be aligned parallel with the airway axis within ± 2°.

The distance of the measurement points (when 8 per diameter) from one inside wall of the airway shall lie within the limits given below, except that the minimum positional tolerance shall be ± 1 mm.

$0{,}021D \pm 0{,}000\,6D$

$0{,}117D \pm 0{,}003\,5D$

$0{,}184D \pm 0{,}005D$

$0{,}345D \pm 0{,}005D$

$0{,}655D \pm 0{,}005D$

$0{,}816D \pm 0{,}005D$

$0{,}883D \pm 0{,}003\,5D$

$0{,}979D \pm 0{,}000\,6D$

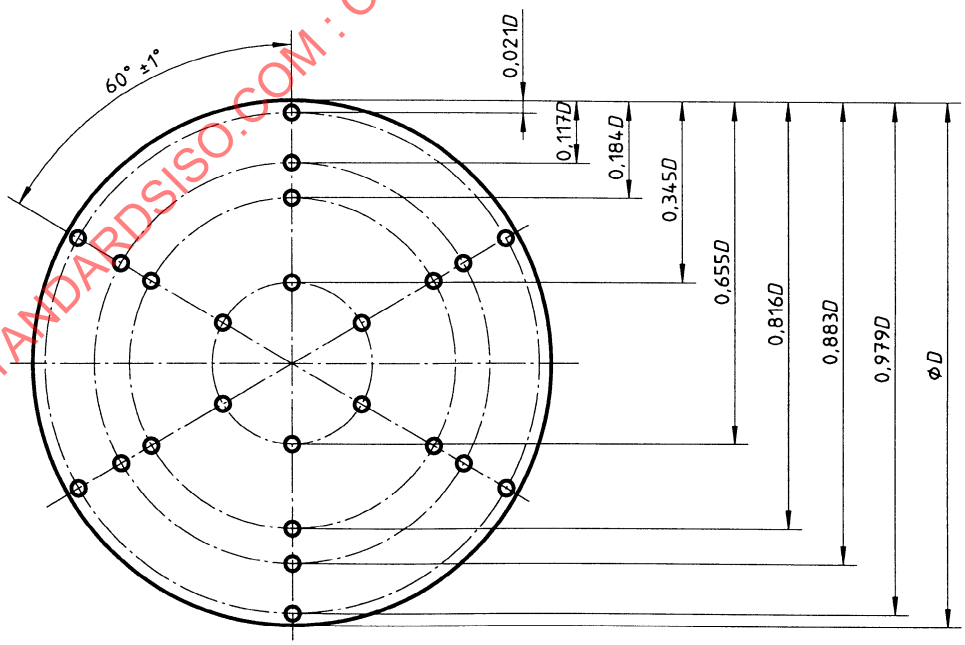


Figure 30 — Positions for traverse measurements in standardized airways

27.5 Determination of flowrate

At each measurement point the differential pressure Δp_j across the Pitot-static tube shall be measured.

The mean differential pressure at the section, Δp_m, is the square of the average of the square roots of the n individual differential pressures, Δp_j, given by the following expression:

$$\Delta p_m = \left[\frac{1}{n} \sum_{j=1}^{n} \Delta p_j^{0,5} \right]^2$$

$$\Delta p_m = \left[\frac{1}{n} \left(\sqrt{\Delta p_1} + \sqrt{\Delta p_2} + \ldots + \sqrt{\Delta p_n} \right) \right]^2$$

The average air density ρ_x at the section of flow measurement x shall be determined from the average static pressure:

$$p_{ex} = \frac{1}{n} \left(p_{ex1} + p_{ex2} + \ldots + p_{exn} \right)$$

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

and the static temperature Θ_x, given by the following expression:

$$\Theta_x = \Theta_{sgx} \left[\frac{p_x}{p_x + \Delta p_m} \right]^{\frac{\kappa - 1}{\kappa}}$$

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

The mass flowrate q_m is given by:

$$q_m = \alpha \varepsilon \pi \frac{D_x^2}{4} \sqrt{2 \rho_x \Delta p_m}$$

where

$$\varepsilon = \left[1 - \frac{1}{2\kappa} \frac{\Delta p_m}{p_x} + \frac{\kappa + 1}{6\kappa^2} \left(\frac{\Delta p_m}{p_x} \right)^2 \right]^{0,5}$$

is the expansibility factor. α is the correction factor or flowrate coefficient given in 27.6.

27.6 Flowrate coefficient

The flowrate coefficient α has been derived by applying each of the correction factors specified in ISO 3966 at an average value of the variables appropriate to tests with atmospheric air complying with this clause. The coefficient α is dependent on Reynolds number which is derived from the diameter D_x and average velocity v_{mx} at the section as shown below.

$$Re_{Dx} = \frac{\rho_x v_{mx} D_x}{\mu} = \frac{4 q_m}{\pi D_x \mu} \approx 71 \times 10^3 \frac{q_m}{D_x}$$

for atmospheric air and with SI units.

Re_{Dx}	3×10^4	10^5	3×10^5	10^6	3×10^6
α	0,986	0,988	0,990	0,991	0,992

27.7 Uncertainty of measurement

The use of an average value for α involves disregarding systematic errors which may reach ± 0,8 % of volume flowrate or mass flowrate. Random uncertainties of measurement total ± 1,1 %. Therefore the uncertainty of flowrate measurement may be taken as ± 2 %.

This estimate assumes that the uncertainty of manometer calibration is ± 1 %. Sensitive manometers are necessary to meet this requirement at moderately low air velocities, and the manometer calibration required for air with a density of 1,2 kg · m^{-3} is shown below:

± 1,5 Pa	± 1 Pa	± 0,5 Pa	± 0,25 Pa
16 m · s^{-1}	13 m · s^{-1}	9 m · s^{-1}	6 m · s^{-1}

28 Types of installation and setups

There are four types of site installation which can be used for fans:

Type A: free inlet and free outlet;

Type B: free inlet and ducted outlet;

Type C: ducted inlet and free outlet;

Type D: ducted inlet and ducted outlet.

The test installation shall reproduce as much as possible these working conditions, therefore four types of test set-up have been defined.

28.1 Type A: free inlet and free outlet

In order to qualify for installation type A, the fan must be tested without any auxiliary device added for the tests, for instance inlet bell or outlet duct, but the auxiliaries supplied with the fan, i.e. protection grid, inlet bell, etc., shall be fitted.

An inlet or outlet chamber is used in this case as defined in clauses 31.3 and 31.4.

28.2 Type B: free inlet and ducted outlet

In order to qualify for installation type B, an outlet duct with straightener shall be used, which be of the short duct variety when there is no swirl at the fan outlet.

The fan shall be tested without any auxiliary device added to the fan inlet, except those supplied with the fan.

Normally the outlet pressure is measured in the outlet duct after an antiswirl device. Duct and antiswirl device form a common segment at the fan outlet (see 30.2).

When an outlet chamber is used, and when there is no swirl flow at the fan outlet, particularly for centrifugal fans, a short duct [see 30.2 f)] may be used between fan and chamber.

28.3 Type C: ducted inlet and free outlet

In order to qualify for installation type C, an inlet duct simulation shall be used and no outlet duct or auxiliary device shall be used, except those supplied with the fan (protection grid, diffuser, etc.).

When the inlet pressure is measured in the inlet duct, a common segment at the fan inlet is used (see 30.3).

An inlet test chamber may be used (see 31.3). If the fan at the outlet side is connected to a short duct, this will considerably influence its performance, even if this duct is very short, for instance 0,5D, because practically the entire flow resistance is at the inlet side.

Therefore such a duct should also be included in the test airway, if the in-site fan has a short outlet side duct.

The length of the duct employed during tests should be mentioned in the test report.

The fan performance is calculated as for other type C fan tests.

28.4 Type D: ducted inlet and ducted outlet

In order to qualify for installation type D, an inlet duct simulation shall be used and an outlet duct shall be used.

Normally inlet and outlet ducts shall be of the common-segment type, as specified in 30.2 and 30.3 respectively.

When an inlet or outlet chamber is used, the outlet duct may be of the short variety described in 30.2 f) when there is no swirl at the fan outlet.

For large fans (800 mm diameter and larger) it may be difficult to carry out the tests with the standardized common-segment airways on the outlet side, including straighteners. In this case, by mutual agreement between the parties concerned, the fan performance may be determined using the method described in 30.2 f) and 30.4 with a duct of length $2D_h$ 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. It is still a subject of research to determine which method gives the most representative values.

In this case, the outlet static pressure is not measured in the outlet duct but considered as equal to the atmospheric pressure.

28.5 Test installation type

To identify the performance, the symbols of the characteristics influenced by the installation type, shall have an additional letter indicating the test installation type.

p_{FA}, p_{FB}, p_{FC} or p_{FD}

p_{sFA}, p_{sFB}, p_{sFC} or p_{sFD}

η_{rA}, η_{rB}, η_{rC} or η_{rD}

29 Component parts of standardized airways

29.1 Symbols

The following graphical symbols are used to specify the component parts of the standardized airways defined in clauses 30, 32, 33, 34 and 35. Each component shall comply with the relevant description specified in 29.2.

29.2 Component parts

29.2.1 Figure 31 symbolizes a cylindrical duct of internal diameter, D_x measured at plane x, in accordance with clause 11. Joints between airway sections should be as few as possible, and accurately aligned without internal protrusions. Leakage shall be negligible compared with the flowrate under test, particular care being taken where a Pitot tube or thermometer is inserted for measurement.

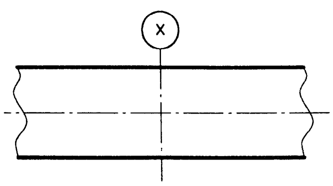


Figure 31 — Symbol for cylindrical duct of internal diameter D_x

29.2.2 Figure 32 symbolizes a group of wall tappings for the measurement of average static pressure, p_{ex}, at section x. The wall tappings and their connections to the manometer unit shall conform to clause 7.

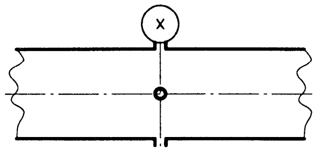


Figure 32 — Symbol for a group of wall tappings

29.2.3 Figure 33 symbolizes the fan under test to be installed and operated in accordance with clauses 18 and 19.

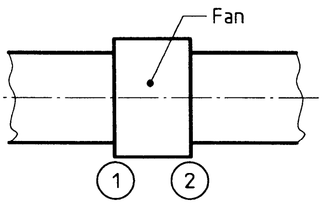


Figure 33 — Symbol for fan under test

29.2.4 Figure 34 symbolizes an auxiliary fan designed to overcome the flow resistance of the test airways. Flowrate control may be secured by speed control, pitch control, damper or otherwise, provided the flow remains steady at any control setting.

An antiswirl device shall be interposed between the auxiliary fan and any test airway to which it is connected.

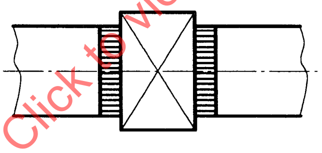


Figure 34 — Symbol for auxiliary fan

29.2.5 Figure 35 symbolizes a flowrate control device at the inlet or outlet of a test airway. The device should not introduce swirl or unsymmetrical flow about the duct axis. An auxiliary fan may form part of the device.

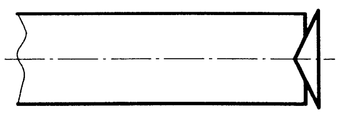


Figure 35 — Symbol for inlet or outlet flowrate control device

29.2.6 Figure 36 symbolizes an in-duct flowrate control device formed from wiremesh or perforated metal screens. The holes should be uniform, and uniformly spaced not more than $0{,}05D$ apart. Screens shall be accurately cut and a supporting ring with a radial thickness $0{,}012d$ or 6 mm max. and length between $0{,}008d$ or 3 mm and $0{,}05d$ max. shall be fitted or other means adopted for eliminating leakage at the wall.

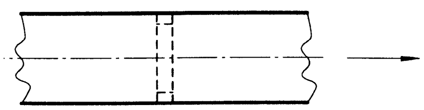


Figure 36 — Symbol for in-duct flowrate control device

29.2.7 Figure 37 symbolizes a standardized flow straightener designed to dissipate any swirl energy at the fan outlet.

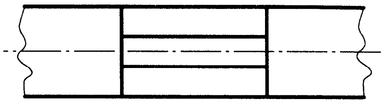


Figure 37 — Symbol for standardized flow straightener

29.2.8 Figure 38 symbolizes an antiswirl device designed to prevent the growth of swirl in a normally axial flow or a flow straightener designed to dissipate any swirl energy.

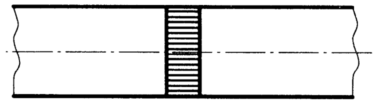


Figure 38 — Symbol for antiswirl device or flow straightener

29.2.9 Figure 39 symbolizes a thermometer inserted in a test chamber or duct to determine the average temperature applying to plane x. It should be located so as to avoid significant disturbance of the flow over wall tappings or into a flowrate measurement device.

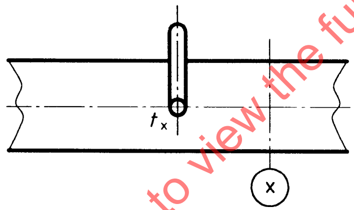


Figure 39 — Symbol for thermometer inserted in test chamber

29.2.10 Figure 40 symbolizes a conical transition section joining two cylindrical ducts or a circular fan outlet and a test duct, defined in clause 30, by included angles or by area ratio and length.

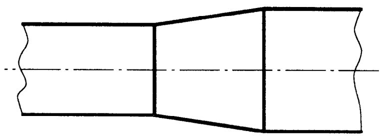


Figure 40 — Symbol for conical transition section joining cylindrical ducts

29.2.11 Figure 41 symbolizes a transition section joining a cylindrical test airway to a rectangular fan outlet or airway of dimensions $b \cdot h$, defined in clause 30 by area ratio and length and formed from sheet material in single curvature illustrated in figure 58.

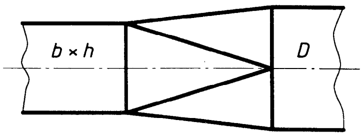


Figure 41 — Symbol for transition section joining cylindrical duct with rectangular outlet duct

29.2.12 Figure 42 symbolizes a manometer complying with the requirements of 6.2 for the measurement of the static pressure at a section or a differential pressure.

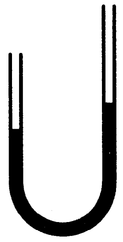


Figure 42 — Manometer

29.3 Flowrate measurement devices

Each device shown below shall comply with the relevant clause for dimensions, dimensional tolerances, and methods of measurement.

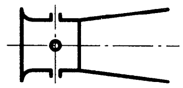


Figure 43 — Inlet ISO Venturi-nozzle (see clause 22)

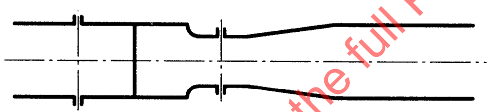


Figure 44 — In-duct ISO Venturi-nozzle (see clauses 22 and 23)

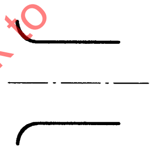


Figure 45 — Nozzle used in a test chamber (see clause 23)

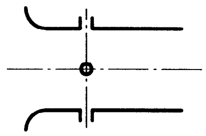


Figure 46 — Quadrant inlet nozzle (see clause 24)

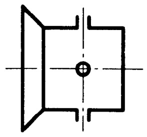


Figure 47 — Conical inlet nozzle (see clause 25)

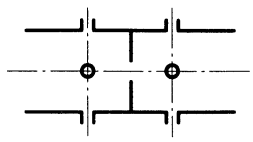


Figure 48 — In-duct orifice with taps at D and $D/2$ (see clause 26)

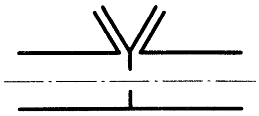


Figure 49 — In-duct orifice with corner taps (see clause 26)

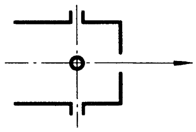


Figure 50 — Outlet orifice with wall taps (see clause 26)

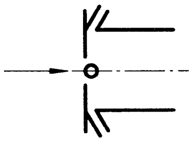


Figure 51 — Inlet orifice with corner taps (see clause 26)

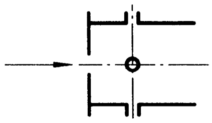


Figure 52 — Inlet orifice with wall taps (see clause 26)

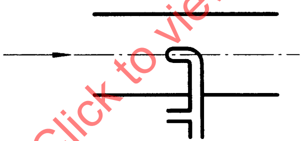


Figure 53 — Pitot-static traverse for flowrate determination (see clause 27)

30 Common airway segments for ducted fan installations

30.1 Common segments

Standardized airways for type B, C or D ducted fan installations incorporate common segments adjacent to the fan inlet and/or outlet which are described in this clause.

Pressure measurements are made at the outer ends of these common segments and geometric variations are strictly limited so that the fan pressures determined will be consistent from one installation types to another.

30.2 Common segment at fan outlet

This comprises the section of an outlet-side test airway adjacent to the fan. It incorporates a standardized flow straightener in accordance with 29.2.7 and figure 54 in the central cylindrical section, together with a set of wall tappings in accordance with clause 7. A transition section may be used to accommodate a difference of area and/or shape within the limits indicated in b) and c) below.

Figures 54, 55 and 56 show the recommended devices, figure 57 is an alternative device, while figure 59 concerns a special case.

a) Circular fan outlet when $D_4 = D_2$ (see figure 54).

The star straightener consists of eight radial vanes equally spaced and located 22,5° from the radial planes through the wall tappings in plane 4.

The vane thickness shall not exceed $0{,}007D_4$.

The length of the straightener is $2D_4 \pm 1$ %.

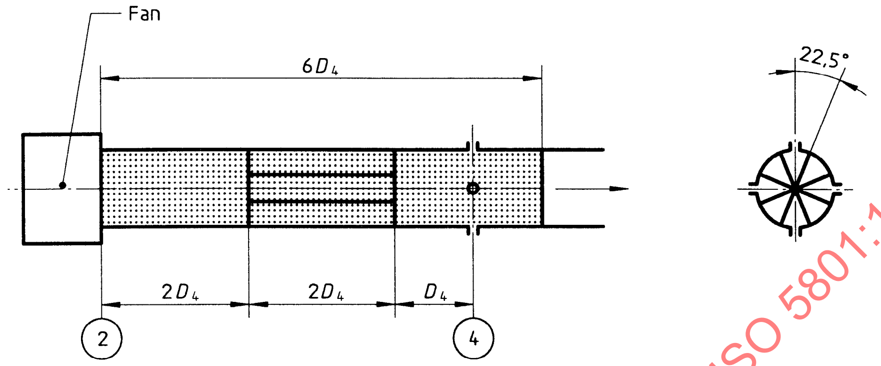


Figure 54 — Circular fan outlet for $D_2 = D_4$

b) Circular fan outlet when $D_4 \neq D_2$ (see figure 55)

$0{,}95 \leqslant (D_4/D_2)^2 \leqslant 1{,}07$

$L_{T2} = D_4$

NOTE 41 The transition section is conical, and the friction loss coefficient is that of a duct of diameter D_4 and length D_4.

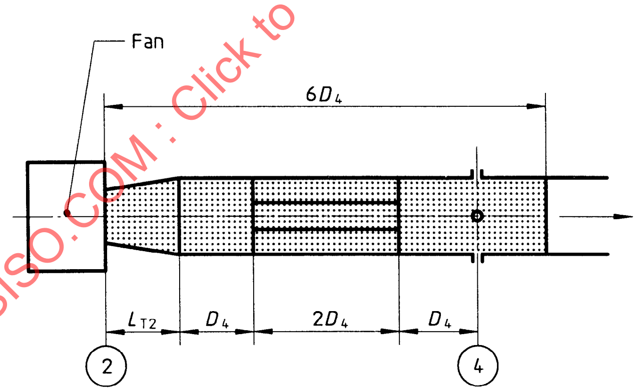


Figure 55 — Circular fan outlet for $D_2 \neq D_4$

c) Rectangular fan outlet, $b \times h$, where $b > h$ (see figure 56)

$0{,}95 \leqslant \pi D_4^2 / 4bh \leqslant 1{,}07$

$L_{T2} = 1{,}0\ D_4$ when $b \leqslant 4h/3$

$L_{T2} = 0{,}75\ (b/h)\ D_4$ when $b > 4h/3$

NOTE 42 The transition section is formed from sheet material in a single curvature.

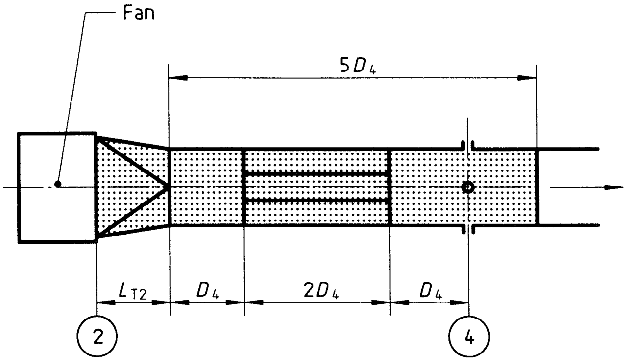


Figure 56 — Rectangular fan outlet where $b > h$

d) Circular or rectangular fan outlet where $0{,}95 \leqslant A_2/A_4 \leqslant 1{,}05$ (see figure 57)

For an outlet duct with cell straightener:

The antiswirl device consists of a nest of cells of equal cross-section (hexagonal, square, etc.) each with width w and length L. The vane thickness e shall not exceed $0{,}005D$.

1) For a normal duct straightener:

$w = 0{,}075D_4$ between axes

$L = 0{,}45D_4$

$e \leqslant 0{,}005D_4$

All dimensions shall be within $\pm\, 0{,}005D$ except e.

2) For a straightener upstream of an orifice plate with wall taps [see 26.9 and figure 23 e) and g)]:

$w = 0{,}15D_6$

$e \leqslant 0{,}003D_6$

$L = 0{,}45D_6$

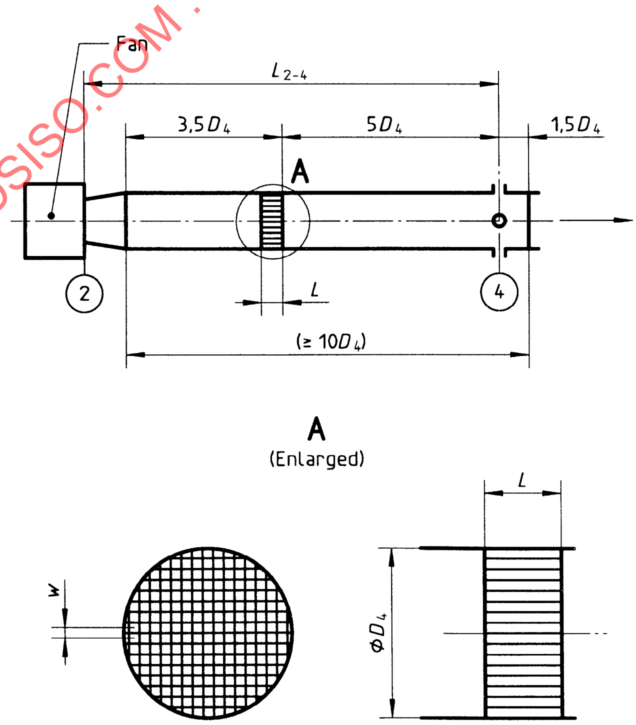


Figure 57 — Circular or rectangular fan outlet where $0{,}95 \leqslant A_2/A_4 \leqslant 1{,}05$

e) Transformation (see figure 58)

The transition section is formed from a single sheet of material as illustrated in figure 58 in accordance with 30.2 c).

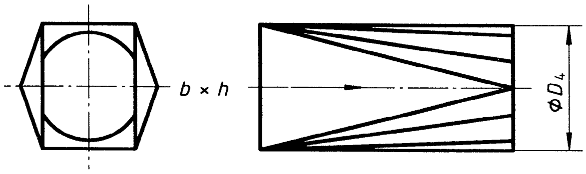


Figure 58 — Transformation

f) Special case (see figure 59).

In the particular case of tests on fans of type B or D without significant outlet swirl, such as a centrifugal volute or cross-flow fan, a simplified outlet duct may alternatively be fitted when discharging to atmosphere or a measuring chamber. This duct shall be of the same cross-section as the fan outlet and the length shall be the value determined by the expression in figure 59 a) or b).

$L \geq 2\sqrt{\frac{4 b_2 h_2}{\pi}}$

for $0{,}9 \leq h_2/h_s \leq 1$

a)

$L \geq \frac{20\, b_2 (h_2 - h_s)}{b_2 + h_2}$

for $0{,}5 \leq h_2/h_s \leq 0{,}9$

b)

Figure 59 — Special cases

The tolerance on the cross-section of the duct is ± 0,01 (1 %) of the fan outlet area.

30.3 Common segment at fan inlet

This comprises the section of the inlet side test airway adjacent to the fan and incorporates a set of wall tappings in accordance with clause 7 as shown in figure 60.

A transition section may be used to accommodate a difference in area and/or shape within the limits specified in b) and c) below.

a) A circular fan inlet when $D_3 = D_1$ (see figure 60)

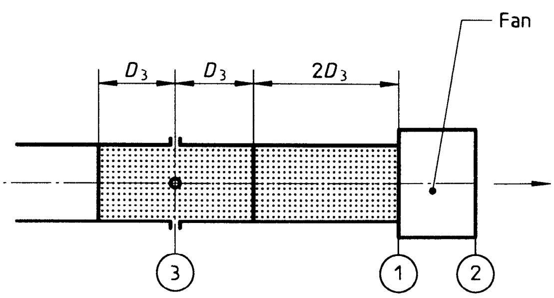


Figure 60 — Circular fan inlet for $D_3 = D_1$

b) Circular fan inlet where $0{,}975\,D_1 \leqslant D_3 \leqslant 1{,}5\,D_1$ (see figure 61)

NOTE 43 The transition section is conical, and the friction loss coefficient is that of a duct of diameter D_3 and length D_3.

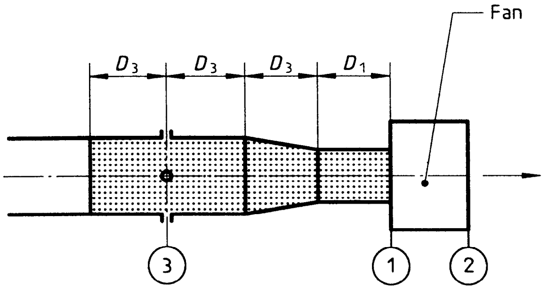


Figure 61 — Circular fan inlet for $0{,}975\,D_1 \leqslant D_3 \leqslant 1{,}5\,D_1$

c) Rectangular fan inlet, $b \times h$ (see figure 62)

The section adjacent to the fan inlet has the same rectangular cross-section, $b \times h$ as the fan inlet to which it is attached and its length L_{s1} is given below:

$$\frac{\pi D_3^2}{4} \geqslant 0{,}95bh$$

$$L_{s1} = \sqrt{\frac{4bh}{\pi}}$$

There is no upper limit on D_3 or on the aspect ratio b/h (where $b > h$), but the included angle of expansion between the short sides should not exceed 15° and the included angle of contraction between the long sides should not exceed 30°. The transition section has the form described in 30.2 e).

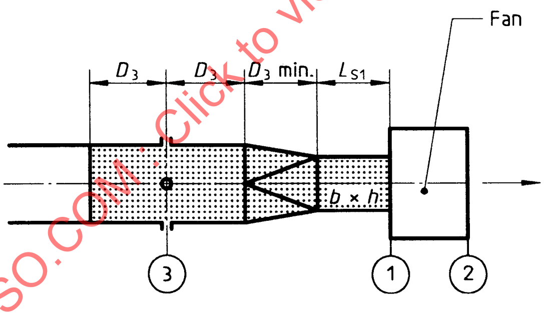


Figure 62 — Rectangular fan inlet

d) Circular or rectangular fan inlet where

$0{,}925 \leqslant A_3/A_1 \leqslant 1{,}125$

[see figure 63 and 30.2 d) 1)]

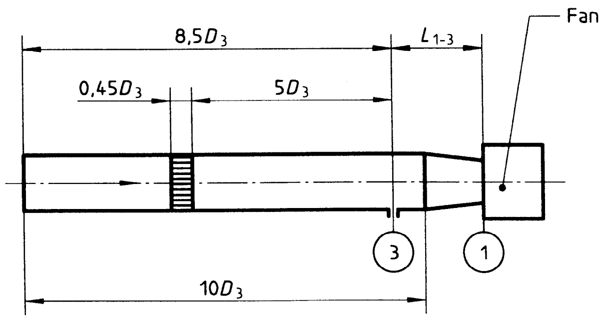


Figure 63 — Circular or rectangular fan inlet where $0{,}925 \leqslant A_3/A_1 \leqslant 1{,}125$

e) Circular or rectangular fan inlet

An inlet duct simulation section in accordance with 30.5 may be used as shown in figure 64. This method is appropriate when the test airway of diameter D_3 is sufficiently large to contain the whole of the bell-mouth entry.

$$L_{s1} = D_1$$

for a circular fan inlet subject to the conditions given in 30.5 a), and

$$L_{s1} = \sqrt{\frac{4bh}{\pi}}$$

for a rectangular fan inlet.

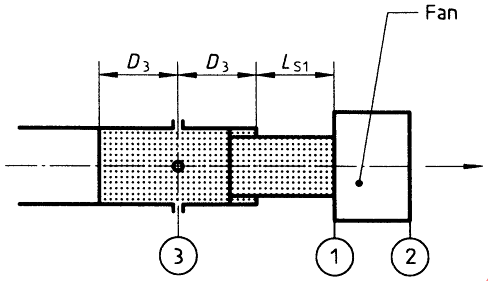


Figure 64 — Circular or rectangular fan inlet

30.4 Outlet duct simulation

A fan tested for use with free outlet but adaptable to ducted outlet may be converted for test from the former to the latter by attaching an outlet-duct simulation section to its outlet.

The outlet simulation section takes the form of the common segment defined in 30.2 a), b), c), d) 1) or f), as appropriate. The outlet of the common segment is left open to the atmosphere, but the outlet-side pressure is measured by the wall tappings in plane 4 [except for 30.2 f)].

For large fans (800 mm and larger) it may be difficult to carry out the tests with the standardized common airways on the outlet side, including straighteners.

In this case, by mutual agreement between the parties concerned, the fan performance may be measured with a duct of $2D_h$ 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 sides, especially if the fan produces a large swirl.

In this case, the static pressure p_{e4} is not measured at the wall of the outlet duct of length $2D$. This static pressure is taken equal to the atmospheric pressure.

30.5 Inlet duct simulation

A fan tested for use with free inlet but adaptable for ducted inlet may be converted from the former to the latter by attaching an inlet-duct simulation section to its inlet.

a) Circular fan inlet

The simulation section should be a cylindrical airway of the same diameter as the fan inlet to which it is attached. A bell-mouth entry should be fitted.

An inlet length equal to D_1, is the normal relationship and provides a true ducted-inlet fan characteristic for any fan over the range of normal working duty. In certain cases, however, a longer duct is needed to enable the fan to develop its full ducted-inlet pressure at or near zero-volume flow. If in such cases a complete fan characteristic curve is required, it is permissible to extend this element as required, or to use the common segment of 30.3 a) with a bell-mouth entry at its inlet end.

b) Rectangular fan inlet

The simulation section should have the same rectangular cross-section, $b \times h$, as the fan inlet to which it is attached, and its length L_{s1} given by the following relationship:

$$L_{s1} = \sqrt{\frac{4bh}{\pi}}$$

A bell-mouth entry should be fitted.

30.6 Loss allowances for standardized airways

Conventional allowances given in this subclause shall be made for airway friction in tests with standardized airways. The friction allowance factors are shown in figure 65.

These allowances depend on the Reynolds number Re_D of the flow in the test airway, and are based on fully developed flow in smooth ducts, irrespective of the actual flow pattern produced by the fan.

The allowances are calculated for the common segments described in 30.2 and 30.3 between the fan outlet or inlet and the plane of pressure measurement. The same allowances should be made when transition sections are incorporated and when an inlet-duct simulation section as described in 30.5 is used (in which case they cover the bell-mouth entry loss).

30.6.1 Loss allowances for common outlet segments described in 30.2 a), b) and c)

The coefficient of friction loss for a length of one diameter of a straight duct is given by the following expression:

$$\Lambda = 0{,}005 + 0{,}42\left(Re_{D4}\right)^{-0{,}3}$$

The conventional loss coefficient of the straightener including the external duct is given by:

$$\zeta_s = 0{,}95\left(Re_{D4}\right)^{-0{,}12}$$

and the conventional loss coefficient $(\zeta_{2-4})_4$ between the fan outlet and the measuring plane 4 is given by the following expression:

$$\left(\zeta_{2-4}\right)_4 = 3\Lambda + 0{,}95\left(Re_{D4}\right)^{-0{,}12}$$
$$= 0{,}015 + 1{,}26\left(Re_{D4}\right)^{-0{,}3} + 0{,}95\left(Re_{D4}\right)^{-0{,}12}$$

where

$$Re_{D4} = \frac{v_{m4} D_4 \rho_4}{\mu_4} \approx \frac{v_{m4} D_4}{15} \times 10^6$$

for standard air.

The loss coefficient $(\zeta_{2-4})_4$ is plotted against the Reynolds number, see figure 65 a).

The losses between planes 2 and 4 are given by the following expression:

$$\Delta p_{2-4} = \left(\zeta_{2-4}\right)_4 \frac{\rho_4 v_{m4}^2}{2} F_{M4}$$

30.6.2 Loss allowances for common outlet segments described in 30.2 d)1)

The coefficient of friction loss Λ for a duct length equal to the diameter is given by the following expression:

$$\Lambda = 0{,}14\left(Re_{Dh4}\right)^{-0{,}17}$$

and is shown plotted in figure 65 b).

The ratio of equivalent length of a cell-straightener to hydraulic diameter D_h ($D_h = D_4$ for a circular duct) is given by the following expression:

$$\frac{L_e}{D_h} = \frac{15,04}{\left[1 - 26,65\frac{e}{D_h} + 184,6\left(\frac{e}{D_h}\right)^2\right]^{1,83}}$$

The conventional loss coefficient of the common outlet segment described in 30.2 d) 1) is given by the following expression (see figure 57):

$$(\zeta_{2-4})_4 = \Lambda\left[\frac{L_{2-4}}{D_{h4}} + \frac{L_e}{D_{h4}}\right]$$

where L_{2-4} is the length of the duct between the fan outlet and the measurement section.

30.6.3 Energy loss allowances for short outlet duct described in 30.2 f)

The duct friction shall not be considered.

30.6.4 Energy loss allowances for common inlet segment described in 30.3 a), b) and c)

The coefficient of friction loss Λ is given by the following expression:

$$\Lambda = 0,005 + 0,42\,(Re_{D3})^{-0,3}$$

and

$$(\zeta_{1-3})_3 = 0,015 + 1,26\,(Re_{D3})^{-0,3}$$

where

$$Re_{D3} = \frac{v_{m3} D_3 \rho_3}{\mu_3} \approx \frac{v_{m3} D_3}{15} \times 10^6$$

in standard air.

The conventional loss coefficient

$$(\zeta_{3-1})_3 = -(\zeta_{1-3})_3$$

is always negative and is shown in figure 65 a).

The energy losses between planes 3 and 1 are given by the following expression:

$$\Delta p_{3-1} = (\zeta_{3-1})_3 \frac{\rho_3 v_{m3}^2}{2} F_{M3}$$

30.6.5 Energy loss allowances for common inlet duct with cell straightener described in 30.3 d)

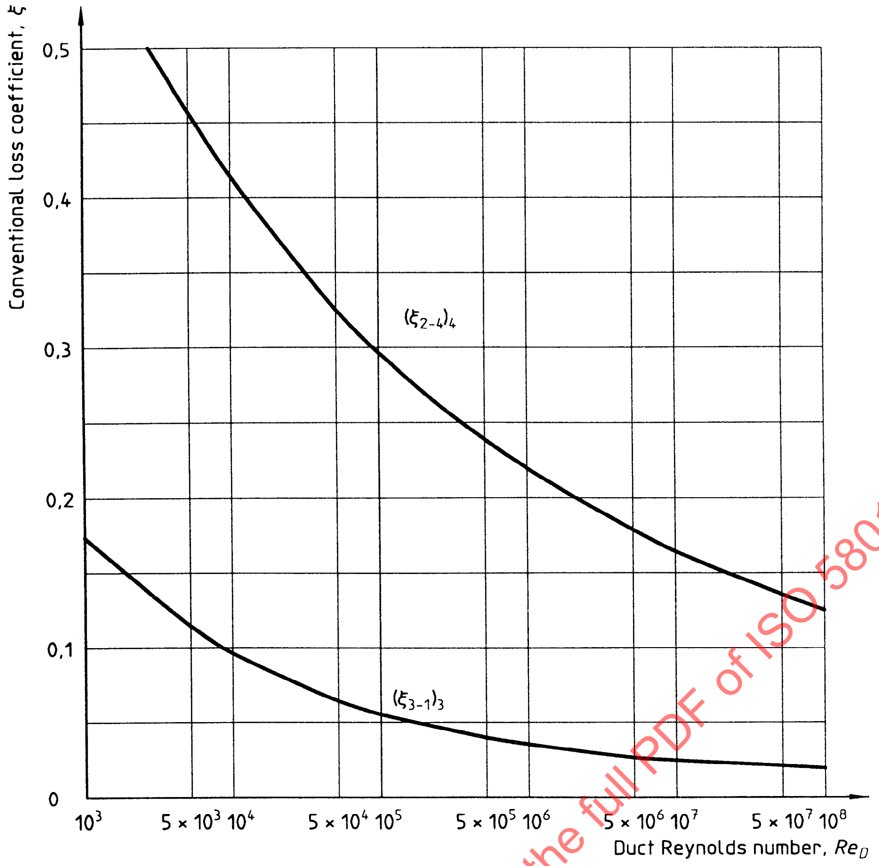
The coefficient of friction loss Λ is given by the same expression as in 30.6.2.

$$\Lambda = 0,14(Re_{Dh3})^{-0,17}$$

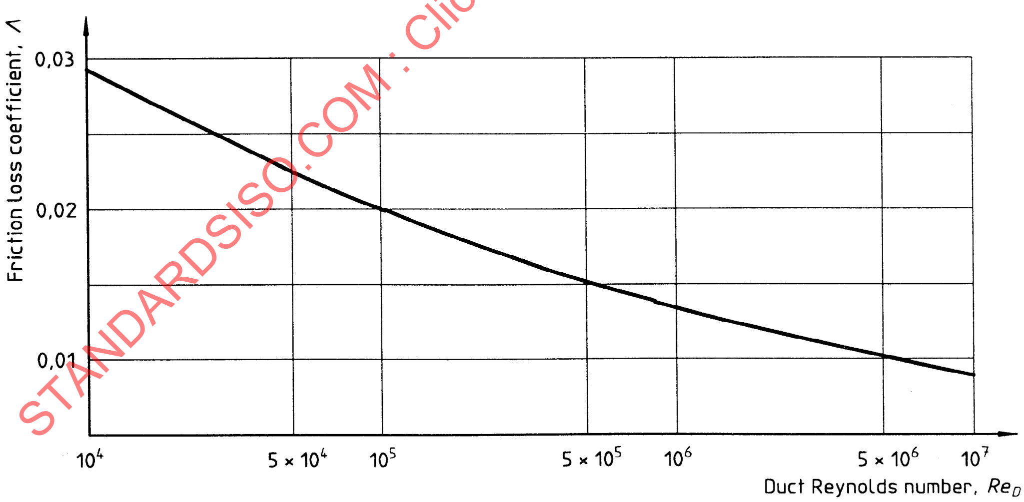
where

$$Re_{Dh3} = \frac{v_{m3} D_{h3} \rho_3}{\mu_3} \approx \frac{v_{m3} D_{h3}}{15} \times 10^6$$

in standard air.



a) Conventional loss coefficient for standardized airways (30.6.1 and 30.6.4)



b) Friction loss coefficient for ducts (30.6.2 and 30.6.5)

Figure 65 — Loss coefficients

The conventional loss coefficient between sections 1 and 3 is consequently given by the following expression:

$$(\zeta_{3-1})_3 = -\Lambda \frac{L_{1-3}}{D_{h3}}$$

It is shown in figure 65 a).

The losses in energy between planes 3 and 1 are given by the following expression:

$$\Delta p_{3-1} = \left(\zeta_{3-1}\right)_3 \frac{\rho_3 v_{m3}^2}{2} F_{M3}$$

30.6.6 Energy loss allowances for inlet duct simulation described in 30.5

There are no losses allowed for this inlet duct, unless an inlet duct corresponding to the common segments described in 30.3 a) or d) or other is required.

31 Standardized test chambers

31.1 Test chamber

A chamber may be incorporated in a laboratory set-up to provide a measuring station or to simulate the conditions the fan is expected to encounter in service, or both.

31.1.1 Dimensions

The test chamber cross-section may be circular, square or rectangular.

The length should be sufficient to accommodate any fan to be tested without infringing on the minimum distance shown in figures 66 and 67.

31.1.2 Pressure tappings

The wall tappings in the measuring planes shall be in accordance with the requirements of clause 7 and be equally spaced around a cylindrical chamber or at the centre of each of the sides of a square or rectangular chamber.

31.1.3 Flow-settling means

Flow-settling means shall be installed in chambers where indicated on the test installation plans to provide the required flow patterns.

If the measuring plane is located downstream of the settling means, the settling means is provided to ensure a substantially uniform flow ahead of the measuring plane. In this case, the maximum velocity at a distance 0,1 D_h downstream of the screen shall not exceed the average velocity by more than 25 % unless the maximum velocity is less than 2 $m \cdot s^{-1}$.

If the measuring plane is located upstream of the settling means, the purpose of the settling screen is to absorb the kinetic energy of the upstream jet, and allow its normal expansion as if in an unconfined space. This requires some backflow to supply the air to mix at the jet boundaries, but the maximum reverse velocity shall not exceed 10 % of the calculated mean jet velocity.

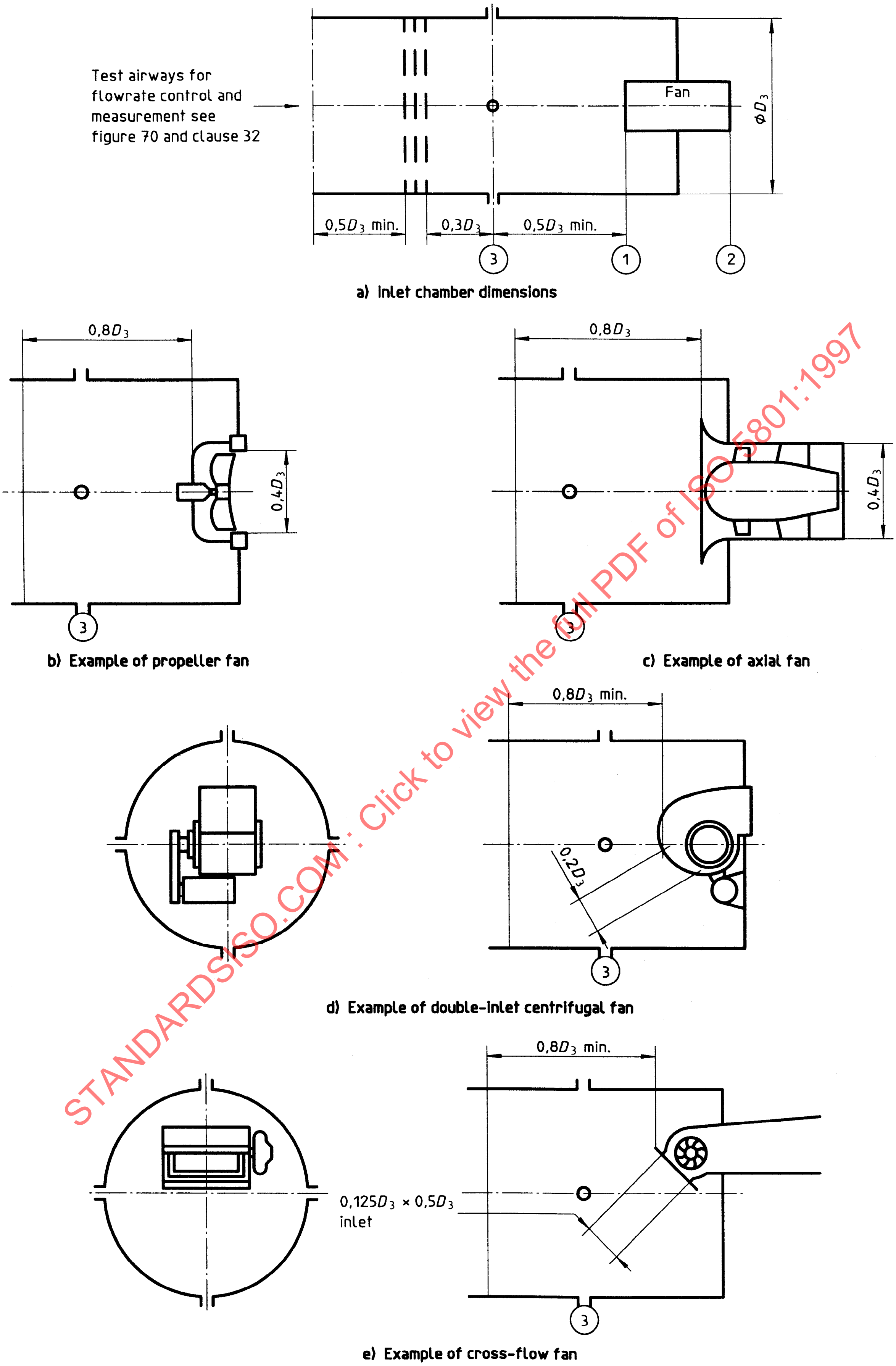
If measuring planes are located on both sides of the settling means within the chamber, the requirements for each side as outlined above shall be met.

Any combination of screens or perforated plates that will meet these requirements may be used, but in general a reasonable chamber length for the settling means is necessary to meet both requirements.

Three uniform wire-mesh or perforated-plate screens adequately supported and sealed to the chamber wall, spaced 0,1D_h apart and with 60 %, 50 % and 45 % free area successively in the direction of flow, may be expected to secure flow meeting these conditions.

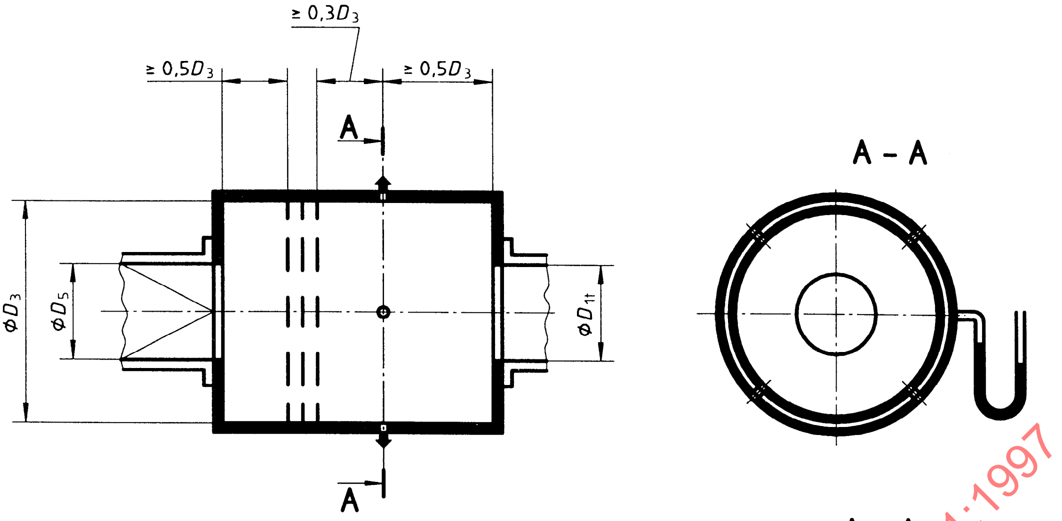
Screens shall be kept free from blocking by dirt.

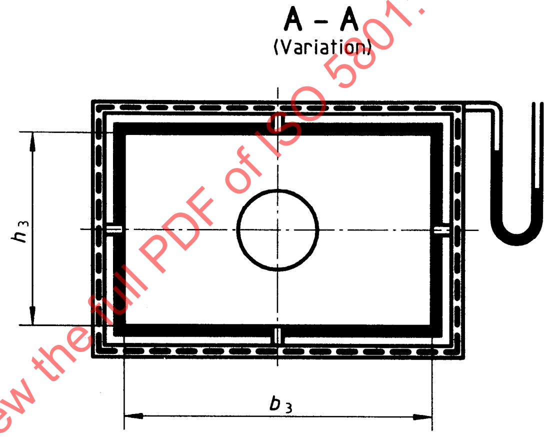
A performance check will be necessary to verify that the flow-settling means are providing the required flow patterns.



NOTE — The fans illustrated have the maximum permissible dimensions

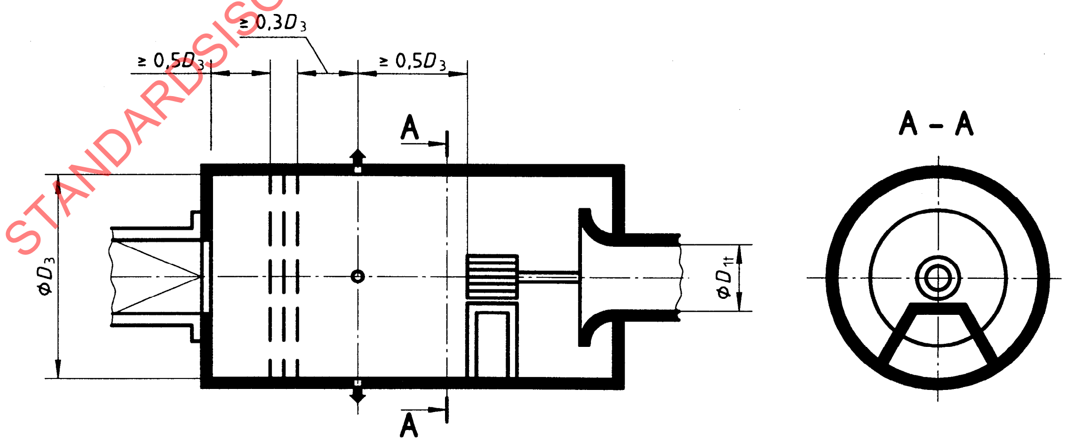
Figure 66 — Inlet-side test chamber type 1





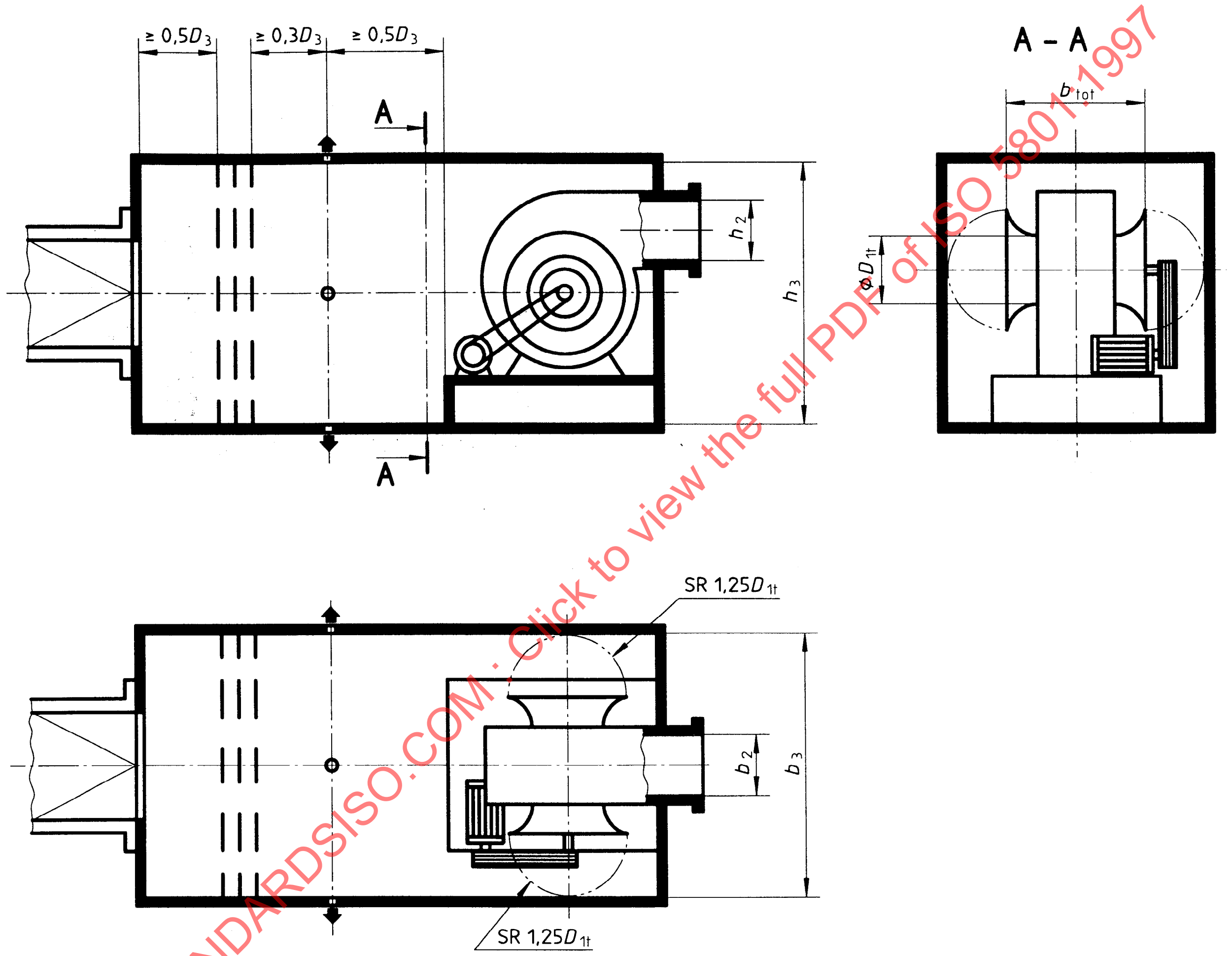
Chamber diameter $D_3 \geq 2{,}5 D_{1t}$
Equivalent chamber diameter $D_3 = \sqrt{h_3 b_3} \geq 2{,}5 D_{1t}$
h_3 min. $\geq 2 D_{1t}$
b_3 max. $\leq 1{,}5 h_3$

a) Dimensions of inlet test chamber



b) Minimum dimensions of extended test chamber with motor on the inlet side

Figure 67 — Inlet-side test chamber type 2



c) Minimum dimensions of extended test chamber for installation of two-flow fans

Figure 67 — Inlet-side test chamber type 2

31.1.4 Multiple nozzles

Multiple nozzles shall be located as symmetrically as possible. The centreline of each nozzle shall be at least 1,5 nozzle throat diameters from the chamber wall. The minimum distance between centres of any two nozzles in simultaneous use shall be three times the throat diameter of the largest nozzle.

The distance from the exit face of the largest nozzle to the downstream settling means shall be a minimum of 2,5 times the throat diameter of the largest nozzle.

The distance between the inlet plane of the nozzles and the upstream and downstream pressure taps is 38 mm ± 6 mm.

31.1.5 Orifice plate in chamber

The orifice shall be coaxial within the chamber within ± 1° and ± $0{,}005D_h$ (see 26.2). The distance between the upstream face of the orifice plate and the exit of the upstream settling means shall be a minimum of $0{,}4D_h$, where D_h is the hydraulic diameter of the chamber.

The distance between the exit face of the orifice plate and the downstream settling means shall be a minimum of $0{,}5D_h$.

The distance between the inlet plane of the orifice plate and the upstream and downstream pressure taps is $0{,}05D_h \pm 0{,}01D_h$.

The orifice plate shall be in accordance with the conditions described in 26.2.

31.2 Variable supply and exhaust systems

A means of varying the point of operation shall be provided in a laboratory set-up.

31.2.1 Throttling devices

Throttling devices may be used to control the point of operation of the fan. Such devices shall be located on the end of the duct or chamber and should be symmetrical about the duct or chamber axis.

31.2.2 Auxiliary fans

Auxiliary fans may be used to control the point of operation of the test fan. They shall be designed to produce sufficient pressure at the desired flowrate to overcome losses through the test set-up. Flow adjustment means, such as dampers, pitch control, or speed control may be required. Auxiliary fans shall not create surge or pulse flow during tests.

31.3 Standardized inlet test chambers

31.3.1 Test chamber

Three types of inlet test chamber are described in this International Standard (see figures 66, 67 and 68).

31.3.1.1 Inlet test chamber type 1

The test chamber cross-section may be circular with inside diameter D_3, square $D_3 \times D_3$ or rectangular with D_3 the shorter side.

The length should be sufficient to accommodate any fan to be tested without infringing on the minimum distance between chamber pressure taps and fan casing or motor, as shown in figure 66.

31.3.1.2 Inlet test chamber type 2

The test chamber cross-section may be circular with inside diameter D_3, square $D_3 \times D_3$ or rectangular $b_3 \times h_3$, with $b_3 \leqslant 1{,}5h_3$ and the equivalent chamber diameter:

$$D_3 = \sqrt{b_3 h_3}$$

For fans with an inlet-side drive or for two-flow fans or double-inlet fans, where a corresponding minimum distance is necessary in the chamber between the pressure tapping and the next segment of the fan depending on the installation conditions, it will be necessary to use a test chamber extended in length compared with the minimum dimensions indicated in figure 67.

Dimensions in millimetres

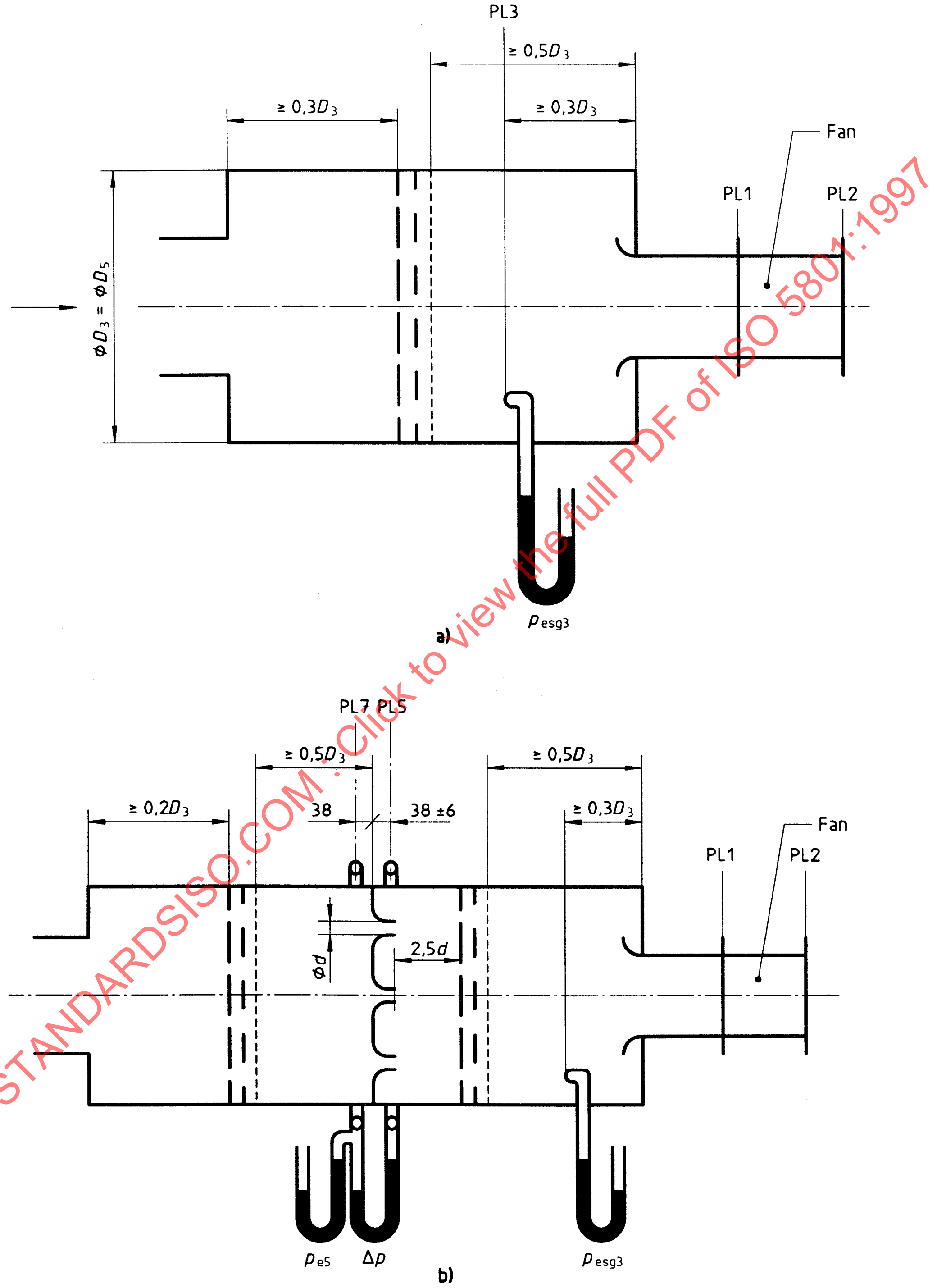


Figure 68 — Inlet-side test chamber type 3

31.3.1.3 Inlet test chamber type 3

The dimension D_3 of the inlet test chamber type 3 is the inside diameter of a circular chamber or the equivalent diameter of a rectangular chamber with inside transverse dimensions $h_3\, b_3$

where

$$D_3 = \sqrt{\frac{4h_3 b_3}{\pi}}$$

The pressure-measuring plane 3 is:

— at least $0{,}3D_3$ upstream of the downstream end of the chamber;

— at least $0{,}2D_3$ downstream of the flow-settling means.

Inlet chambers of type 3 may be fitted with nozzles for flowrate measurement (see figure 68).

31.3.2 Fan under test

31.3.2.1 Inlet chamber type 1

The fan under test may have any inlet throat area A_{1t} not exceeding

$$\frac{D_3^2}{8}$$

or

$$D_3^2 > 8A_{1t}$$

for a circular chamber, where A_{1t} is the inlet throat area provided the inlet is coaxial with the chamber. Where this is not practicable, the total throat area of the inlet or inlets shall not exceed

$$\frac{D_3^2}{16}$$

and the inlets should be so located that the flow remains as symmetrical about the chamber axis as possible.

Examples of fans with maximum inlet sizes are shown in figures 66 and 67.

31.3.2.2 Inlet chamber type 2

The fan under test may have any inlet throat diameter D_{1t} not exceeding $D_3/2{,}5$

or

$$A_{1t} \leqslant A_3/6{,}25$$

or

$$A_3 \geqslant 6{,}25\, A_{1t}$$

When testing a double-inlet fan, the minimum width of the chamber shall be capable of accommodating both inlets. It is expedient to choose a chamber with square or rectangular cross-section, of which the total width b_3 is the sum of the fan width b and an open space surrounding the two intake openings corresponding to a hemisphere of radius equal to $1{,}25D_{1t}$ as shown in figure 67.

31.3.2.3 Inlet chamber type 3

Inlet chambers shall have a cross-sectional area five times the fan inlet throat area

$$A_3 \geqslant 5A_1$$

They may be fitted with or without multinozzles for flowrate measurement (see figure 68).

31.4 Standardized outlet test chambers

31.4.1 Test chamber (see figure 69)

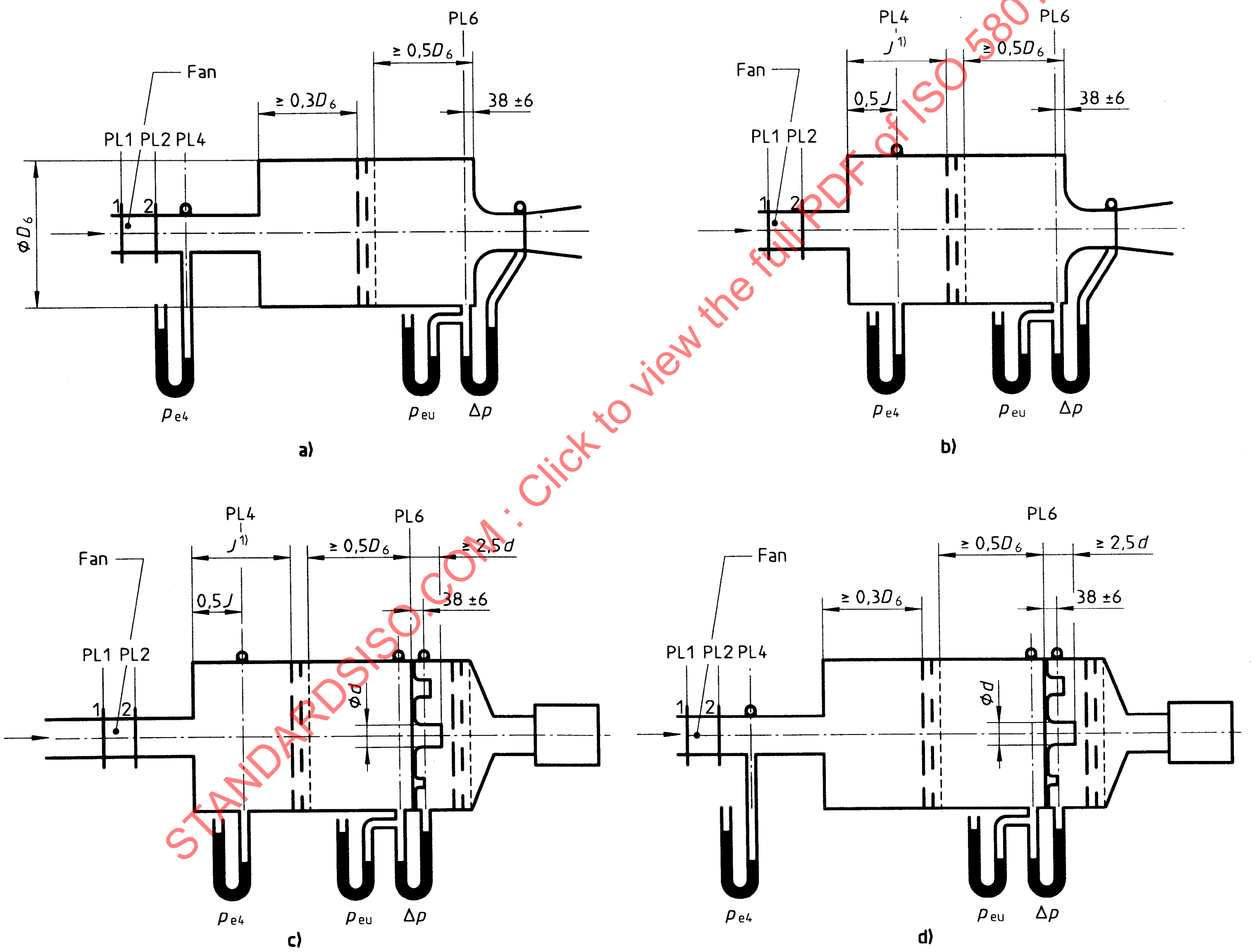
The test chamber cross-section may be circular with inside diameter D_6, square $D_6 \times D_6$ or rectangular $h_6 \times b_6$.

The dimension D_6 of the chamber is the inside diameter of a circular chamber or the equivalent diameter of a rectangular chamber with inside dimensions h_6 and b_6 where

$$D_6 = \sqrt{\frac{4h_6b_6}{\pi}}$$

The fan outlet pressure p_{e4} may be measured either in the fan outlet duct or in the chamber. Outlet chambers may be fitted with or without multinozzles for flowrate measurement (see figure 69).

Dimensions in millimetres



1) The distance J shall be equal to at least the diameter of the outlet duct for fans with axis of rotation perpendicular to the discharge flow, and to at least twice the diameter of the outlet duct for fans with axis of rotation parallel to the discharge flow.

Figure 69 — Outlet-side test chambers

31.4.2 Fan under test

An outlet test chamber (see figure 69) shall have a cross-sectional area at least nine times the area of the fan outlet or outlet duct for fans with axis of rotation at right angles to the discharge flow ($A_6 \geqslant 9A_2$) and a cross-sectional area at least 16 times the area of the fan outlet or outlet duct, for fans with axis of rotation parallel to the discharge flow: ($A_6 \geqslant 16A_2$).

32 Standard methods with test chambers — Type A installations

32.1 Types of fan setup

Two general setups of fan on chamber are shown:

a) inlet-side test chamber setup;

b) outlet-side test chamber setup.

Eleven methods of controlling and measuring the flowrate in the case of inlet chamber set-up and two methods in the case of outlet chamber setup are shown. The method of flowrate measurement is specified in each case, together with the clauses and figures detailing the flow measurement procedure.

A common procedure, comprising measurements to be taken and quantities to be calculated, allowing the determination of fan performance in type A installations, with 11 methods for determining flowrate in the case of inlet chamber setup and two methods in the case of outlet chamber setup are given in 32.2 and 32.3.

The chambers are assumed to be large enough that effects of Mach numbers are negligible.

The procedure is generally valid for all fans conforming with this International Standard.

However two simplified procedures may be followed when

— the reference Mach number Ma_{2ref} at the fan outlet is less than 0,15 and the pressure ratio is more than 1,02;

— the reference Mach number Ma_{2ref} is less than 0,15 and the pressure ratio is less than 1,02.

Procedures for theses cases are given in 32.2.4.1 and 32.2.4.2.

32.2 Inlet-side test chambers

32.2.1 Flowrate determination

The flowrate is determined by:

— Inlet ISO Venturi nozzle, see figure 70 a);

— quadrant inlet nozzle, see figure 70 a);

— conical inlet, see figure 70 a);

— inlet orifice with corner taps, see figure 70 b);

— inlet orifice with wall taps, see figure 70 b);

— in-duct orifice with D and $D/2$ taps, see figure 70 c);

— in-duct orifice with corner taps, see figure 70 c)

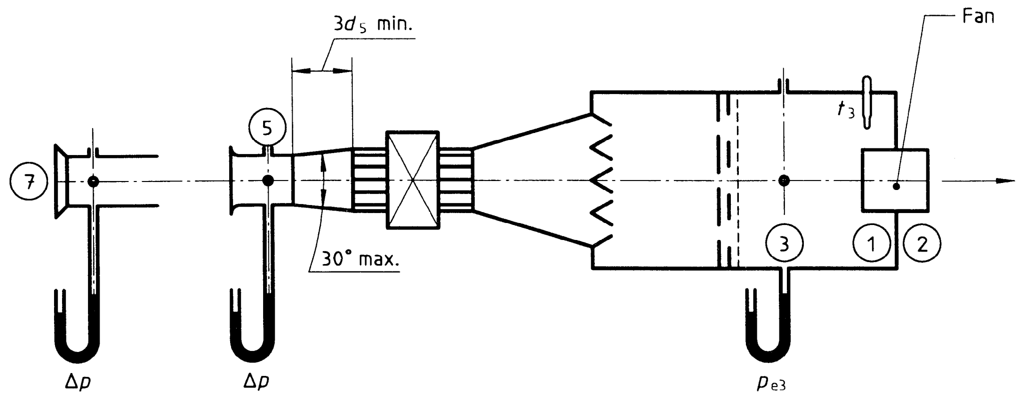
— in-duct ISO Venturi nozzle, see figure 70 d);

— Pitot-static tube traverse, see figure 70 e);

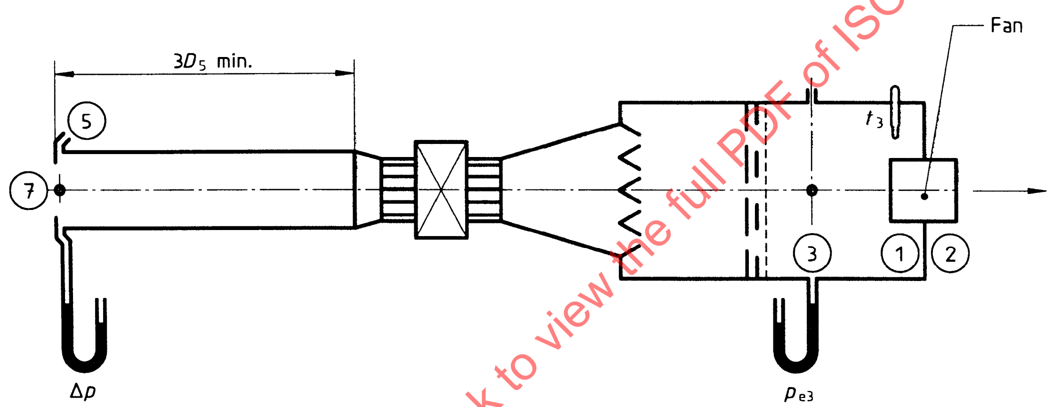
— in-duct Venturi nozzle, see figure 70 f);

— multiple nozzles in chamber, see figure 70 g);

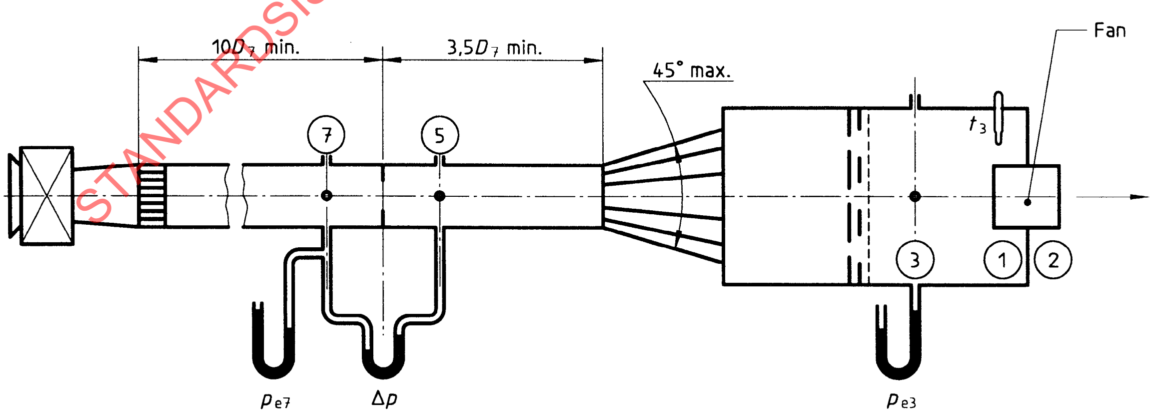
— orifice plate in chamber, see figures 70 g) and 23 h).



a) Flowrate determination using ISO Venturi nozzle or conical inlet or quadrant inlet nozzle

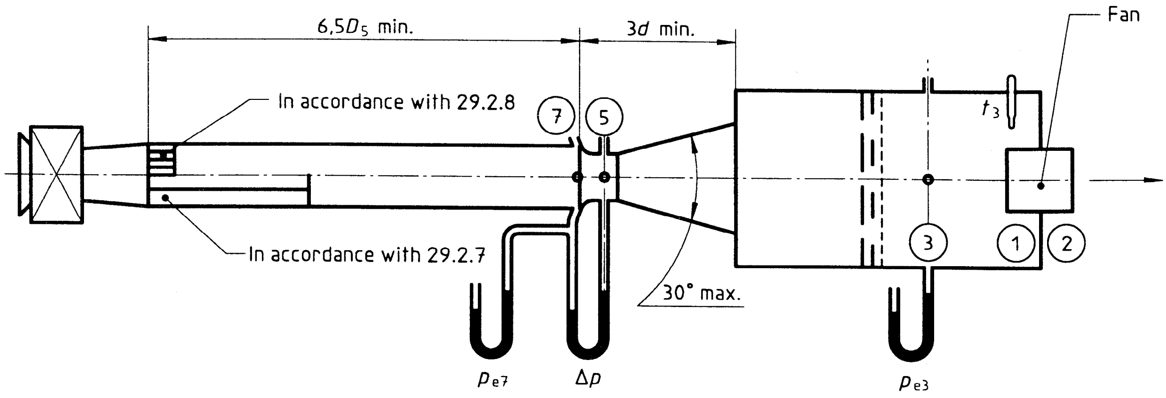


b) Flowrate determination using inlet orifice with corner taps or wall taps

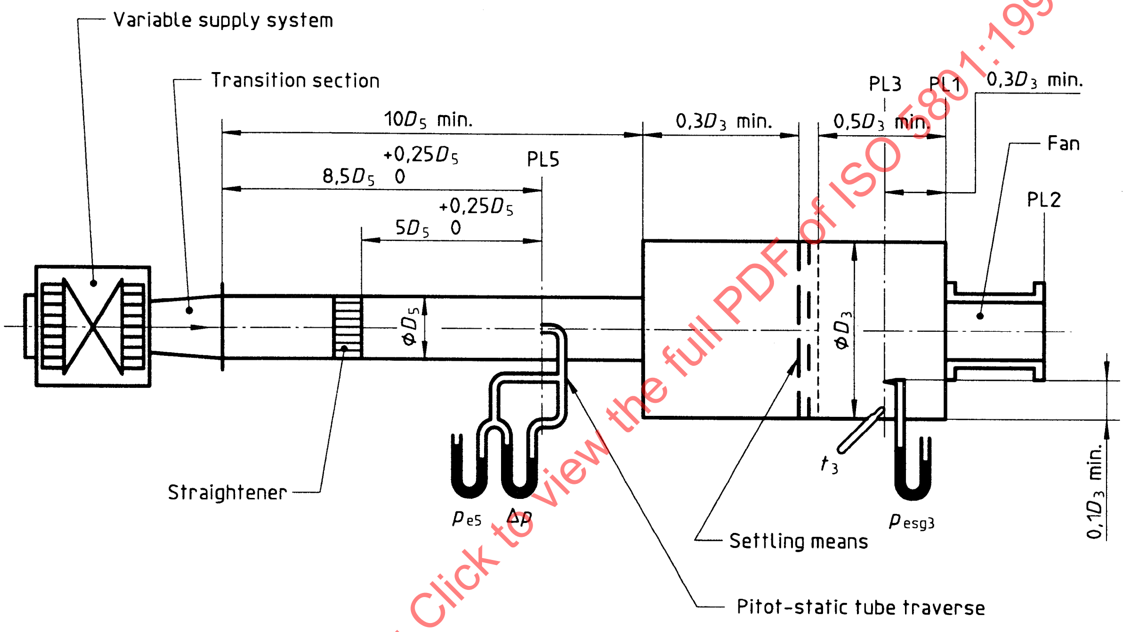


c) Flowrate determination using in-duct orifice with taps at D and 0,5D or in-duct orifice with corner taps

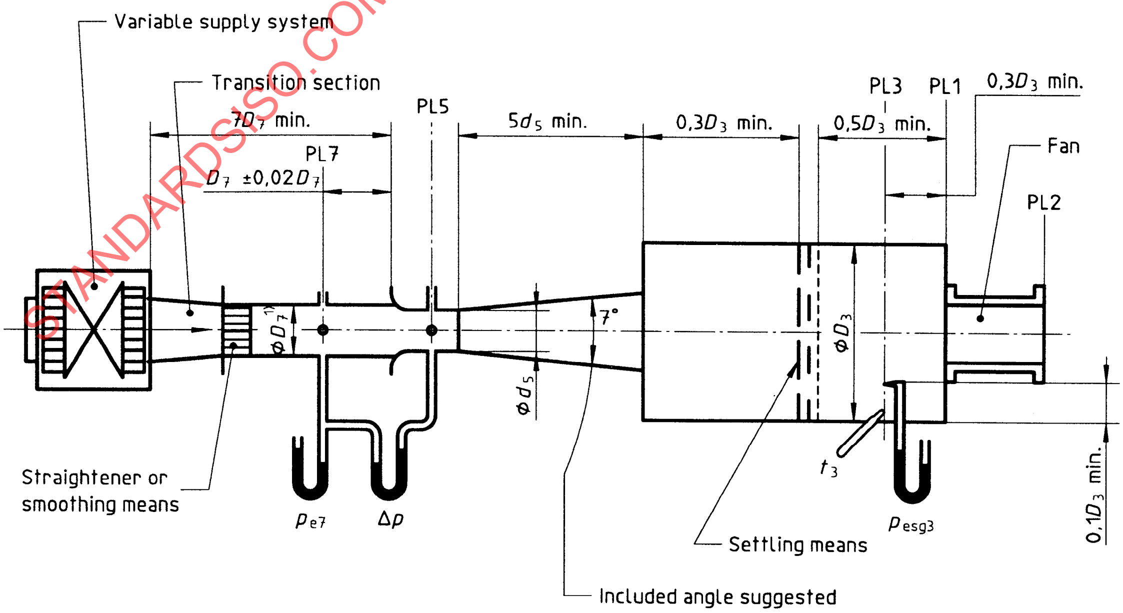
Figure 70 —Type A test installations (inlet-side test chamber)



d) Flowrate determination using in-duct Venturi nozzle



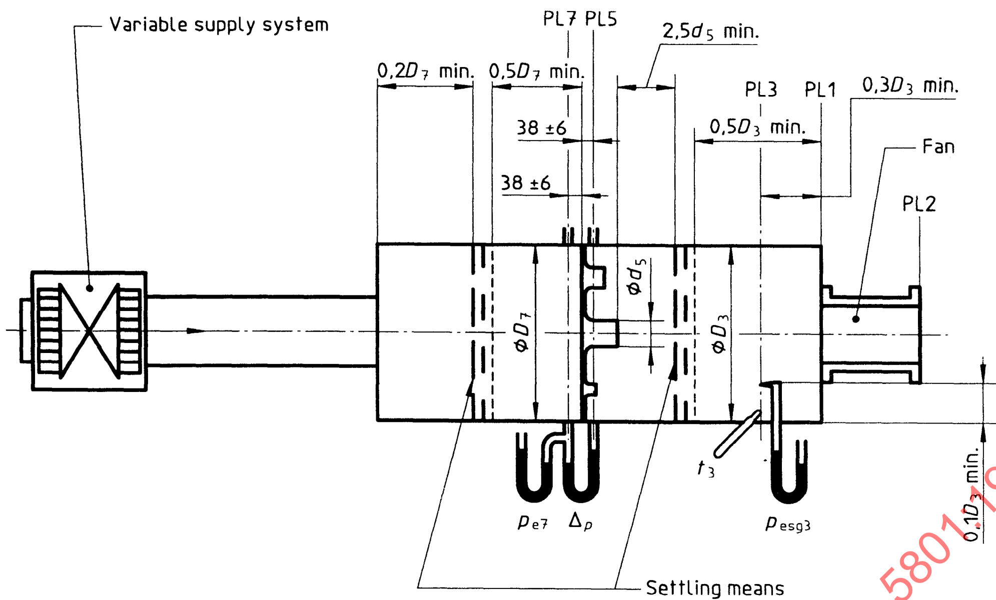
e) Flowrate determination using Pitot-static tube traverse



1) D_7 = 1,9d_5 min.

f) Flowrate determination using in-duct Venturi nozzle

Figure 70 — Type A test installations (inlet-side test chamber) *(continued)*



g) Flowrate determination using multiple nozzles in chamber

Figure 70 —Type A test installations (inlet-side test chamber) *(concluded)*

32.2.2 Measurements to be taken during tests (see clause 20)

Measure

— rotational speed, N, or rotational frequency, n;

— power, P_a, P_o or P_e and estimate impeller power (see 10.4) and power P_{ex} of an auxiliary fan;

— flowmeter differential pressure, Δp ;

— pressure, p_{e7} or p_{e5} upstream of the flowmeter;

— chamber pressure, p_{e3}, for figure 70 a) to d) and chamber stagnation pressure for figure 70 e, f) and g);

— chamber temperature, t_3.

In the test enclosure measure

— atmospheric pressure, p_a, at the mean altitude of the fan;

— ambient temperature near fan inlet, t_a;

— dry- and wet-bulb temperatures, t_d and t_w.

Determine the ambient air density, ρ_a, and the gas constant of humid air R_w (see clause 12).

32.2.3 General procedure for compressible fluid flow

This procedure should be applied when both the fan pressure ratio is more than 1,02 and the reference Mach number, Ma_{2ref}, is more than 0,15 (see 14.4.2).

32.2.3.1 Calculation of flowrate

32.2.3.1.1 The flowrate is determined by

— inlet ISO Venturi nozzle, see clause 22 and figure 70 a);

— quadrant inlet nozzle, see clause 24 and figure 70 a);

— conical inlet, see clause 25 and figure 70 a);

— inlet orifice with corner taps, see 26.10 and figure 70 b);

— inlet orifice with wall taps, see clause 26.11 and figure 70 b).

The in-line flowmeter is followed by a control device or an auxiliary fan with a control device.

Assuming that.

$$p_7 = p_a = p_u$$

$$\Theta_{sg7} = \Theta_7 = t_a + 273{,}15$$

$$\rho_7 = \frac{p_7}{R_w \Theta_7}$$

After calculation of the dynamic viscosity of air in accordance with 12.3 and a first approximation of the Reynolds number through the flowmeter, the flow coefficient α and the expansibility factor ε or the compound coefficient $\alpha\varepsilon$ may be determined in accordance with subclauses

22.3.2, 22.3.3 and 22.3.4 and figure 18 for an ISO Venturi nozzle;

24.4 for a quadrant inlet nozzle;

25.4 and figure 22 for a conical inlet;

26.10 and figures 26 and 28 for an inlet orifice with corner taps;

26.11 for an inlet orifice with wall taps.

The mass flowrate is given by the following expression:

$$q_m = \alpha \varepsilon \pi \frac{d_5^2}{4} \sqrt{2 \rho_7 \Delta p}$$

When α is a function of the Reynolds number Re_d or Re_D, the variation of α with Re_d or Re_D shall be taken into account in the expression above.

32.2.3.1.2 The flowrate is determined using

— in-duct orifice with taps at D and $D/2$, see 26.7 and figure 70 c);

— in-duct orifice corner taps, see 26.8 and figure 70 c);

— in-duct ISO Venturi nozzle, see clause 22 and figure 70 d).

A control device or an auxiliary fan with a control device is set upstream of the flowmeter.

Assuming that

$$p_7 = p_{e7} + p_a$$

$$\Theta_{sg7} = \Theta_{sg3} = t_3 + 273{,}15 = \Theta_a + \frac{P_{rx} \text{ or } P_{ex}}{q_m c_p}$$

$$\Theta_7 = \Theta_{sg7} - \frac{q_m^2}{2 A_7^2 \rho_7^2 c_p}$$

$$\rho_7 = \frac{p_7}{R_w \Theta_7}$$

the mass flowrate is determined by the following expression:

$$q_m = \alpha \varepsilon \frac{\pi d_5^2}{4} \sqrt{2 \rho_7 \Delta p}$$

The expansibility coefficient is determined in accordance with clause 22 and subclauses 26.7 and 26.8.

After an estimation of the flowmeter Reynolds number:

$$Re_{d5} = \frac{\alpha \varepsilon d_5 \sqrt{2\rho_7 \Delta p}}{(17{,}1 + 0{,}048 t_7)} \times 10^6$$

or

$$Re_{D7} = Re_{d5}\beta$$

the flow coefficient α or the compound coefficient $\alpha\varepsilon$ are determined in accordance with 26.7 and figure 24 for in-duct orifice with taps at D and $D/2$;

26.8 and figure 25 for in-duct orifice with corner taps;

clause 22 and figure 18 for in-duct ISO Venturi nozzles.

A first approximation of q_m may be obtained with $\Theta_7 = \Theta_{sg7}$, Θ_7 may be determined and a new value of α and q_m calculated.

Two iterations are sufficient for a calculation accuracy of 10^{-3}.

32.2.3.1.3 The flowrate is determined using a Pitot-static tube traverse, see clause 27 and figure 70 e).

A control device or an auxiliary fan with a control device is set upstream of the duct for flowrate measurement.

Assuming that

$$p_5 = p_{e5} + p_a$$

$$p_{e5} = \frac{1}{n} \sum_{j=1}^{n} p_{e5j}$$

$$\Theta_{sg5} = \Theta_{sg3} = t_3 + 273{,}15$$

The temperature t_5 in the test duct may be measured and considered as a stagnation temperature, but it is preferable to measure the temperature in the chamber t_3.

The mean differential pressure is given by:

$$\Delta p_m = \left[\frac{1}{n} \sum_{j=1}^{n} \Delta p_j^{0,5} \right]^2$$

see 27.5

and the mass flow is determined by the following expression:

$$q_m = \alpha \varepsilon A_5 \sqrt{2\rho_5 \Delta p_m}$$

where

$$\rho_5 = \frac{p_5}{R_w \Theta_5}$$

$$\Theta_5 = \Theta_{sg5} \left(\frac{p_5}{p_5 + \Delta p_m} \right)^{\frac{\kappa - 1}{\kappa}}$$

$$\varepsilon = \left[1 - \frac{1}{2\kappa} \frac{\Delta p_{\mathrm{m}}}{p_5} + \frac{\kappa + 1}{6\kappa^2} \left(\frac{\Delta p_{\mathrm{m}}}{p_5} \right)^2 \right]^{1/2}$$

α is a function of Reynolds number Re_{D5} very close to 0,99 (see 27.6).

A first approximation of q_m is calculated with $\alpha = 0{,}99$ and corrected for α variation.

32.2.3.1.4 The flowrate is determined using an in-duct Venturi nozzle, see clause 23 and figure 70 f).

A control device or an auxiliary fan with a control device is set upstream of the flowmeter.

Assuming that

$$\Theta_3 = \Theta_{\mathrm{sg3}} = \Theta_{\mathrm{sg7}} = t_3 + 273{,}15 = \Theta_{\mathrm{a}} + \frac{P_{\mathrm{rx}} \text{ or } P_{\mathrm{ex}}}{q_m c_p}$$

$$p_7 = p_{\mathrm{e7}} + p_{\mathrm{a}}$$

$$\Theta_7 = \Theta_{\mathrm{sg7}} - \frac{q_m^2}{2A_7^2 \rho_7^2 c_p}$$

The mass flowrate q_m is given by the following expression:

$$q_m = \alpha \varepsilon \pi \frac{d_5^2}{4} \sqrt{2\rho_7 \Delta p}$$

$$= C \varepsilon \pi \frac{d_5^2}{4} \frac{\sqrt{2\rho_7 \Delta p}}{\sqrt{1 - \alpha_{\mathrm{Au}} \beta^4}}$$

where

ε is the expansibility coefficient calculated in accordance with 23.4.3 and table 5.

$r_{\mathrm{d}} = 1 - \Delta p / p_7$

α is the flowrate coefficient equal to:

$$\frac{C}{\sqrt{1 - \alpha_{\mathrm{Au}} \beta^4}}$$

C is the nozzle discharge coefficient, a function of the Reynolds number Re_{d5} and of the shape of the nozzle (see 23.4.2 and table 5);

α_{Au} is a kinetic energy coefficient equal to 1 for a chamber approach and equal to 1,043 for a duct approach;

$\beta = d_5 / D_7$

$$Re_{d5} = \alpha \varepsilon d_5 \frac{\sqrt{2\rho_7 \Delta p}}{17{,}1 + 0{,}048\, t_7} \times 10^6$$

For a first approximation:

$$\alpha = \frac{0{,}95}{\sqrt{1 - \alpha_{\mathrm{Au}} \beta^4}}$$

and $\Theta_7 = \Theta_{\mathrm{sg7}}$

$$\rho_7 = \frac{p_7}{R_{\mathrm{w}} \Theta_7}$$

A first approximation of q_m may be calculated and with this value, Θ_7, ρ_7, Re_{d5} and α allow calculation of a new value of q_m.

Two or three calculation steps allow an accuracy of 10^{-3}.

32.2.3.1.5 The flowrate is determined using multiple nozzles in chamber, see clause 23 and figure 70 g).

A control device or an auxiliary fan with a control device is set upstream of the chamber.

Assuming that

$$\Theta_3 = \Theta_{sg3} = \Theta_{sg7} = \Theta_7 = t_3 + 273{,}15$$

$$p_7 = p_{e7} + p_a$$

$$\rho_7 = \frac{p_7}{R_w \Theta_7}$$

$$\beta = \frac{d_5}{D_7} \approx 0$$

The mass flowrate is given by the following expression in accordance with 23.4:

$$q_m = \varepsilon \pi \sum_{j=1}^{n} \left(C_j \frac{d_{5j}^2}{4} \right) \sqrt{2 \rho_7 \Delta p}$$

where

ε is the expansibility coefficient in accordance with 23.4.3 and table 6.

C_j is the discharge coefficient of the jth nozzle, which is a function of the nozzle throat Reynolds number Re_{d5j}.

$\beta = 0$ and $C_j = \alpha_j$.

$C_j = \alpha_j$ is calculated in accordance with 23.4.2 and table 5;

n is the number of nozzles.

For each nozzle, the throat Reynolds number Re_{d5} is estimated by the following expression:

$$Re_{d5j} = \frac{\varepsilon C_j d_{5j} \sqrt{2 \rho_7 \Delta p}}{17{,}1 + 0{,}048\, t_7} \times 10^6$$

with $C_j = 0{,}95$

After a first estimation of the mass flowrate, the discharge coefficients C_j are corrected.

32.2.3.1.6 The flowrate is determined using an orifice plate in the test chamber with wall tappings, see 26.9.1 and figures 70 g) and 23 h), i) and j).

An orifice plate is fitted instead of the multiple Venturi nozzles.

Assuming that

$$\Theta_3 = \Theta_{sg3} = \Theta_{sg7} = \Theta_7 = t_3 + 273{,}15$$

$$p_7 = p_{e7} + p_a$$

$$\rho_7 = \frac{p_7}{R_w \Theta_7}$$

$$\beta = \frac{d_5}{D_7} \leqslant 0{,}25$$

The mass flowrate is given by the following expression in accordance with 26.5:

$$q_m = \alpha \varepsilon \pi \frac{d_5^2}{4} \sqrt{2\rho_7 \Delta p}$$

where $\alpha\varepsilon$ is determined in accordance with 26.9 and 26.9.1.

32.2.3.2 Calculation of fan pressure

32.2.3.2.1 Fan inlet pressure

Assuming that

$$p_3 = p_{e3} + p_a$$

$$\Theta_3 = t_3 + 273,15 = \Theta_{sg3} = \Theta_{sg1}$$

$$\rho_3 = \frac{p_3}{R_w \Theta_3}$$

In accordance with 14.5 and 14.6:

$$p_{sg1} = p_3 + \rho_3 \frac{v_{m3}^2}{2} = p_3 + \frac{1}{2} \frac{q_m^2}{A_3^2 \rho_3} = p_{sg3}$$

or

$$p_{esg1} = p_{e3} + \rho_3 \frac{v_{m3}^2}{2} = p_{e3} + \frac{1}{2} \frac{q_m^2}{A_3^2 \rho_3} = p_{esg3}$$

This is valid for the setups in figure 70 a) to d). For the setups in figure 70 e), f) and g), the stagnation pressure p_{esg3} is measured by a Pitot-static tube and

$$p_{sg1} = p_{esg3} + p_a = p_{sg3}$$

$$p_{esg1} = p_{esg3}$$

$$p_{e3} \leqslant 0 \text{ and } p_{esg1} \leqslant 0$$

In accordance with 14.4.3.2 and 14.5.2 Ma_1, $\dfrac{\Theta_1}{\Theta_{sg1}}$ and p_1 may be determined.

The inlet static pressure p_1 is given by the following expression.

$$p_1 = p_{sg1} - p_{d1} F_{M1} = p_{sg1} - \frac{1}{2\rho_1} \left(\frac{q_m}{A_1} \right)^2 F_{M1}$$

or

$$p_{e1} = p_{esg1} - p_{d1} F_{M1} = p_{esg1} - \frac{1}{2\rho_1} \left(\frac{q_m}{A_1} \right)^2 F_{M1} = p_1 - p_a$$

the Mach factor F_{M1} being calculated in accordance with 14.5.1.

32.2.3.2.2 Fan outlet pressure

At the fan outlet p_2 is equal to the atmospheric pressure p_a and

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

Ma_2 and Θ_2 are calculated in accordance with 14.4.3.1:

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

$$p_{sg2} = p_2 + \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 F_{M2}$$

F_{M2} being calculated in accordance with 14.5.1.

It may also be written

$$p_{esg2} = \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 F_{M2}$$

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

$$k_\rho = \frac{\rho_1}{\rho_m}$$

32.2.3.2.3 Fan pressure

The fan static pressure p_{sFA} is given by the following expression:

$$p_{sFA} = p_2 - p_{sg1} = p_a - p_{sg1} = - p_{esg1}$$

and the fan pressure p_{FA} by

$$p_{FA} = p_{sg2} - p_{sg1} = p_a + \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 F_{M2} - p_{sg1}$$

$$p_{FA} = p_{esg2} - p_{esg1} = \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 F_{M2} - p_{esg1}$$

32.2.3.3 Calculation of volume flowrate

In the test conditions, the volume flowrate is calculated by the following expression:

$$q_{Vsg1} = \frac{q_m}{\rho_{sg1}} = \frac{q_m}{\left(\dfrac{p_{sg1}}{R_w \Theta_{sg1}}\right)}$$

32.2.3.4 Calculation of fan air power

32.2.3.4.1 Fan work per unit mass and fan air power

In accordance with 14.8.1, the fan static work per unit mass is given by the following expression:

$$y_{sA} = \frac{p_2 - p_1}{\rho_m} - \frac{v_{m1}^2}{2} = \frac{p_2 - p_1}{\rho_m} - \frac{1}{2}\left(\frac{q_m}{A_1\rho_1}\right)^2$$

and the fan work per unit mass by

$$y_A = \frac{p_2 - p_1}{\rho_m} + \frac{v_{m2}^2}{2} - \frac{v_{m1}^2}{2}$$

$$= \frac{p_2 - p_1}{\rho_m} + \frac{1}{2}\left[\left(\frac{q_m}{A_2\rho_2}\right)^2 - \left(\frac{q_m}{A_1\rho_1}\right)^2\right]$$

The fan static power and the fan air power P_{usA} and P_{uA} are given by the following expressions:

$$P_{usA} = q_m \cdot y_{sA}$$

$$P_{uA} = q_m \cdot y_A$$

32.2.3.4.2 Calculation of fan air power and compressibility coefficient

In accordance with 14.8.2:

$$P_{usA} = q_{Vsg1} \cdot p_{sFA} \cdot k_{ps}$$

$$P_{uA} = q_{Vsg1} \cdot p_{FA} \cdot k_p$$

The compressibility coefficients k_p and k_{ps} may be determined by two equivalent methods (see 14.8.2.1 and 14.8.2.2).

a) First method:

$$k_{ps} \text{ or } k_p = \frac{Z_k \log_{10} r}{\log_{10}\left[1 + Z_k(r-1)\right]}$$

where

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

for fan static air power, or

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

for fan air power, and

$$Z_k = \frac{\kappa - 1}{\kappa} \frac{P_r}{q_{Vsg1}\left(p_{sFA} \text{ or } p_{FA}\right)}$$

for fan static air power or fan air power.

b) Second method:

$$k_{ps} \text{ or } k_p = \frac{\ln(1+x)}{x} \frac{Z_p}{\ln\left(1 + Z_p\right)}$$

where

$$x = r - 1 = \frac{p_{sFA}}{p_{sg1}} \text{ or } \frac{p_{FA}}{p_{sg1}}$$

for fan static power or fan air power

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

32.2.3.5 Calculation of efficiencies

In accordance with 14.8.1, the efficiencies are given by the following expressions:

— Fan static efficiency:

$$\eta_{srA} = \frac{P_{usA}}{P_r}$$

— Fan efficiency:

$$\eta_{rA} = \frac{P_{uA}}{P_r}$$

— Fan static shaft efficiency:

$$\eta_{saA} = \frac{P_{usA}}{P_a}$$

— Fan shaft efficiency:

$$\eta_{aA} = \frac{P_{uA}}{P_a}$$

32.2.4 Simplified procedure

32.2.4.1 Reference Mach number Ma_{2ref} less than 0,15 and pressure ratio more than 1,02

The stagnation and static temperatures are considered as equal and the Mach factors F_M are equal to unity (see 14.9.1).

$$\Theta_x = \Theta_{sgx}$$

$$F_{M1} = F_{M2} = 1$$

32.2.4.1.1 Calculation of mass flowrate

The mass flowrate is determined in accordance with the methods described in 32.2.3.1.

However, the following simplifications can be applied to the calculations of setups in figure 70 c), d), e), f) and g).

The temperatures upstream of the flowmeter and in the chamber may be measured:

$$\Theta_u = t_u + 273{,}15 = \Theta_{sgu}$$

$$p_u = p_{eu} + p_a$$

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

There is no need for iterative procedures to take into account the difference between stagnation and static temperatures.

However, when α varies with the Reynolds number (see 32.2.3.1 page 118) the estimation of Reynolds number is necessary.

The flowrate is calculated with the estimated value of α and with $\Theta_u = \Theta_{sgu}$.

32.2.4.1.2 Calculation of fan pressure

32.2.4.1.2.1 Fan inlet pressure

In accordance with 14.9.1.2 and 14.9.1.3:

$$p_{sg1} = p_{sg3} = p_3 + \rho_3 \frac{v_{m3}^2}{2} = p_3 + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2$$

or

$$p_{esg1} = p_{esg3} = p_{e3} + \frac{v_{m3}^2}{2} = p_{e3} + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2$$

where

$$\rho_3 = \frac{p_3}{R_w \Theta_3} = \frac{p_{e3} + p_a}{R_w \Theta_3}$$

except in the cases of figure 70 e), f) and g) for which the stagnation pressure is measured:

$$p_{sg1} = p_{sg3}$$

or

$$p_{esg1} = p_{esg3}$$

$$p_1 = p_{sg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2$$

or

$$p_{e1} = p_{esg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2$$

where

$$\rho_1 = \frac{p_1}{R_w \Theta_1} = \frac{p_1}{R_w \Theta_{sg1}} = \frac{p_1}{R_w \Theta_{sg3}}$$

but p_1 is unknown, and an iterative procedure shall be used to determine p_1 and ρ_1. Two or three calculation steps are sufficient (see 14.9.1.4).

The pressure p_1 may be determined by the following expression (see 14.9.1.4):

$$p_1 = \frac{p_{sg1} + \sqrt{p_{sg1}^2 - 2\left(\frac{q_m}{A_1}\right)^2 R_w \Theta_{sg1}}}{2}$$

and

$$p_{e1} = p_1 - p_a$$

32.2.4.1.2.2 Fan outlet pressure

At the fan outlet, $p_2 = p_a$ or $p_{e2} = 0$.

$$\rho_2 = \frac{p_2}{R_w \Theta_{sg2}}$$

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

$$p_{sg2} = p_2 + \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 = p_a + \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2$$

$$p_{esg2} = \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2$$

32.2.4.1.2.3 Fan pressure

The fan static pressure p_{sFA} and the fan pressure p_{FA} are given by the following expressions:

$$p_{sFA} = p_2 - p_{sg1} = p_a - p_{sg1} = -\,p_{esg1}$$

$$p_{FA} = p_{sg2} - p_{sg1} = p_a + \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 - p_{sg1}$$

$$= \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 - p_{esg1}$$

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

$$k_\rho = \frac{\rho_1}{\rho_m}$$

NOTE 44 $p_{esg1} \leqslant 0$, $p_{e3} \leqslant 0$.

32.2.4.1.3 Calculation of volume flowrate

The volume flowrate is given by the following expression as in 32.2.3.3:

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

where

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

32.2.4.1.4 Calculation of fan air power

Fan air powers are determined in accordance with 14.8.1, 14.8.2, 14.8.3 and 32.2.3.4.

32.2.4.1.5 Calculation of fan efficiencies

Fan efficiencies are calculated in accordance with 14.8.1, 14.8.2, 14.8.3 and 32.2.3.5.

32.2.4.2 Reference Mach number Ma_{2ref} less than 0,15 and pressure ratio less than 1,02 (see 14.9.2)

The air flow through the fan may be considered as incompressible.

$$\Theta_1 = \Theta_{sg1} = \Theta_3 = \Theta_{sg3} = \Theta_2 = \Theta_{sg2}$$

$$\rho_1 = \rho_2$$

$$F_{M1} = F_{M2} = 1$$

$$k_p = 1$$

32.2.4.2.1 Calculation of mass flowrate

The mass flowrate is determined in accordance with 32.2.4.1.1.

32.2.4.2.2 Calculation of fan pressure

32.2.4.2.2.1 Fan inlet pressure

$$\rho_1 = \rho_{sg1} = \rho_{sg3} = \frac{p_3}{R_w \Theta_3}$$

$$p_{sg1} = p_3 + \frac{1}{2\rho_{sg1}}\left(\frac{q_m}{A_3}\right)^2$$

$$p_{esg1} = p_{e3} + \frac{1}{2\rho_1}\left(\frac{q_m}{A_3}\right)^2$$

$$p_1 = p_{sg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2$$

or

$$p_{e1} = p_{esg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2$$

except in the case of figure 70 e), f) and g) for which the stagnation pressure p_{esg3} is measured and $p_{esg1} = p_{esg3}$ or $p_{sg1} = p_{sg3}$.

32.2.4.2.2.2 Fan outlet pressure

At the fan outlet:

$$p_2 = p_a$$

$$p_{e2} = 0$$

$$p_{sg2} = p_2 + \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2$$

or

$$p_{esg2} = \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2$$

32.2.4.2.2.3 Fan pressure

The fan pressures are given by the following expressions:

$$p_{sFA} = p_2 - p_{sg1} = p_a - p_{sg1} = -p_{esg1}$$

$$p_{FA} = p_{sg2} - p_{sg1} = p_2 + \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2 - \left[p_3 + \frac{1}{2\rho_1}\left(\frac{q_m}{A_3}\right)^2\right]$$

$$= \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2 - \left[p_{e3} + \frac{1}{2\rho_1}\left(\frac{q_m}{A_3}\right)^2\right]$$

32.2.4.2.3 Calculation of volume flowrate

The volume flowrate at inlet conditions is determined by the following expression:

$$q_{Vsg1} = \frac{q_m}{\rho_{sg1}} = \frac{q_m}{\left(\dfrac{p_{sg1}}{R_w \Theta_{sg1}}\right)}$$

32.2.4.2.4 Calculation of fan air power

The fan air powers are determined by the following expressions:

$$P_{usA} = q_{Vsg1} \cdot p_{sFA}$$

$$P_{uA} = q_{Vsg1} \cdot p_{FA}$$

32.2.4.2.5 Calculation of fan efficiencies

Fan efficiencies are calculated in accordance with 14.8.1 and 32.2.4.1.5.

32.2.5 Fan performances under test conditions

Under test conditions, the fan performances are:

— inlet volume flow, q_{Vsg1};

— fan static pressure, p_{sFA};

— fan pressure, p_{FA};

— fan efficiency, η_{srA} or η_{rA}.

32.3 Outlet-side test chambers

32.3.1 Flowrate determination

The flowrate is determined using:

— Venturi nozzle on end of chamber, see clause 23 and figure 71 a);

— multiple nozzles in chamber, see clause 23, figure 71 b);

— orifice plate in chamber, see 26.9.1, figure 71 b).

32.3.2 Measurements to be taken during tests (see clause 20)

Measure:

— rotational speed, N, or rotational frequency, n;

— power input, P_a, P_o or P_e, and estimate impeller power (see 10.4);

— flowmeter differential pressure, Δp;

— upstream pressure, p_{e6};

— chamber pressure, p_{e4};

— chamber temperature, t_4.

In the test enclosure, measure:

— atmospheric pressure at the mean altitude of the fan, p_a;

— ambient temperature near the inlet, t_a;

— dry- and wet-bulb temperatures, t_d and t_w.

Determine the ambient air density ρ_a and gas constant of humid air R_w in accordance with clause 12.

32.3.3 General procedure for compressible fluid flow

This procedure should be applied when the reference Mach number Ma_{2ref} is more than 0,15 and the pressure ratio greater than 1,02.

32.3.3.1 Calculation of mass flowrate

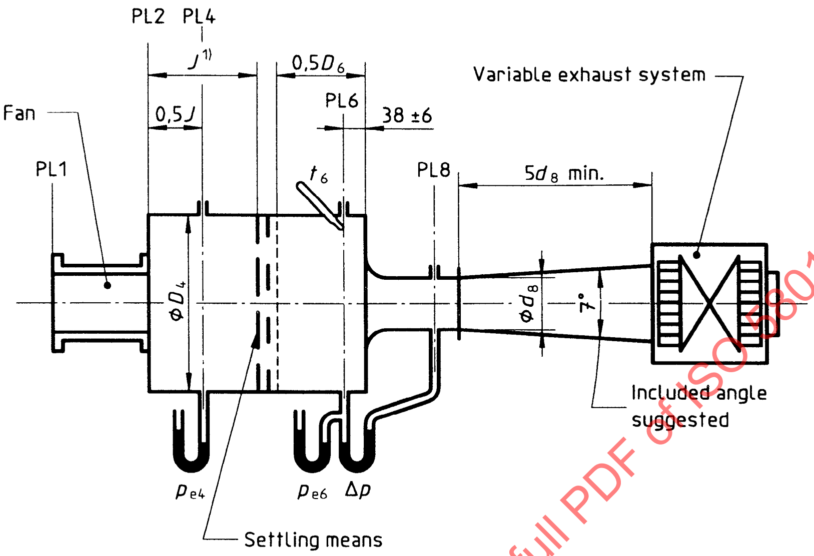
32.3.3.1.1 The mass flowrate is determined using

— Venturi nozzle, see clause 23 and figure 71 a);

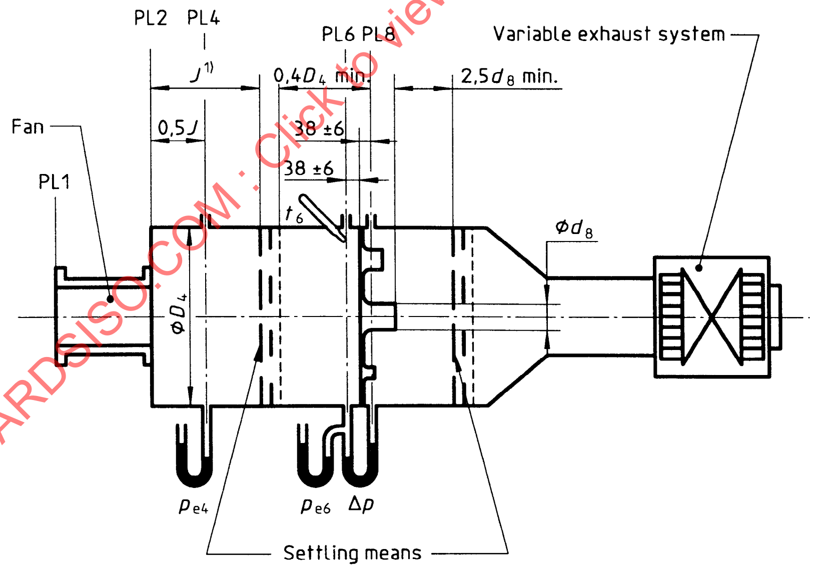
— multiple nozzles in chamber, see clause 23 and figure 71 b).

The chamber is followed by a control device or an auxiliary fan with a control device.

Dimensions in millimetres



a) Flowrate determination using Ventutri nozzle on end of chamber



b) Flowrate determination using multiple nozzles in chamber

1) The distance J shall be equal to at least the diameter of the outlet duct for fans with axis of rotation perpendicular to the discharge flow, and to at least twice the diameter of the outlet duct for fans with axis of rotation parallel to the discharge flow.

Figure 71 —Type A test installations (outlet-side test chamber)

Assuming that

$$p_6 = p_a + p_{e6}$$

$$\Theta_6 = t_6 + 273{,}15 = \Theta_{sg6}$$

$$\frac{d_8}{D_6} = \beta \approx 0$$

$$\rho_6 = \frac{p_6}{R_w \Theta_6}$$

The mass flowrate is given by the following expression in accordance with 23.4:

$$q_m = \varepsilon \pi \sum_{j=1}^{n} \left(C_j \frac{d_{8j}^2}{4} \right) \sqrt{2 \rho_6 \Delta p}$$

where

ε is the expansibility coefficient in accordance with 23.4.3 and table 6;

C_j is the discharge coefficient of the jth nozzle, as a function of the nozzle throat Reynolds number Re_{d8j}, see 23.4;

β = 0 and $C_j = \alpha$

$C_j = \alpha_j$ is calculated in accordance with 23.4 and table 5;

n is the number of nozzles, equal to 1 for a nozzle on end of chamber.

For each nozzle, the throat Reynolds number Re_{d8} is estimated using the following expression:

$$Re_{d8j} = \frac{\varepsilon C_j d_{8j} \sqrt{2 \rho_6 \Delta p}}{17{,}1 + 0{,}048\, t_6} \times 10^6$$

with $C_j = 0{,}95$

After a first estimation of the mass flowrate, the discharge coefficients C_j are corrected for the Reynolds number variations.

32.3.3.1.2 The mass flowrate is determined using an orifice plate in the test chamber with wall tappings, see 26.9.1, figures 71 b) and 23 h), i) and j).

Assuming that

$$p_6 = p_a + p_{e6}$$

$$\Theta_6 = t_6 + 273{,}15 = \Theta_{sg6}$$

$$\frac{d_8}{D_6} = \beta \leqslant 0{,}25$$

$$\rho_6 = \frac{p_6}{R_w \Theta_6}$$

The mass flowrate is given by the following expression in accordance with 26.5.

$$q_m = \alpha \varepsilon \pi \frac{d_8^2}{4} \sqrt{2 \rho_6 \Delta p}$$

where $\alpha\varepsilon$ is determined in accordance with 26.5 and 26.9.1.

32.3.3.2 Calculation of fan pressure

32.3.3.2.1 Fan outlet pressure

$$p_2 = p_4 = p_{e4} + p_a$$

$$\Theta_{sg2} = \Theta_{sg4} = t_4 + 273{,}15 = \Theta_{sg6}$$

The Mach number Ma_2 and the temperature Θ_2 are determined in accordance with 14.4.3.1 and figure 5.

$$\Theta_2 = \Theta_{sg2} \frac{\Theta_2}{\Theta_{sg2}}$$

$$\rho_2 = \frac{p_2}{R_w \Theta_2} = \frac{p_4}{R_w \Theta_2}$$

$$p_{sg2} = p_2 + \frac{1}{2\rho_2} \left(\frac{q_m}{A_2} \right)^2 F_{M2}$$

or

$$p_{esg2} = p_{e2} + \frac{1}{2} \rho_2 \left(\frac{q_m}{A_2} \right)^2 F_{M2}$$

F_{M2} being determined in accordance with 14.5.1.

32.3.3.2.2 Fan inlet pressure

$$p_{sg1} = p_a ; \quad p_{esg1} = 0$$

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

The Mach number Ma_1, the ratio

$$\frac{\rho_{sg1}}{\rho_1}$$

and the Mach factor F_{M1} are calculated in accordance with 14.4.3.2, 14.4.4 and 14.5.1.

$$p_1 = p_{sg1} - \frac{1}{2\rho_1} \left(\frac{q_m}{A_1} \right)^2 F_{M1}$$

or

$$p_{e1} = -\frac{1}{2\rho_1} \left(\frac{q_m}{A_1} \right)^2 F_{M1}$$

32.3.3.2.3 Fan pressure

The fan static pressure p_{sFA} and the fan pressure p_{FA} are given by the following expressions:

$$p_{sFA} = p_2 - p_{sg1} = p_2 - p_a = p_{e2}$$

$$p_{FA} = p_{sg2} - p_{sg1} = p_{esg2} - p_{esg1} = p_{esg2}$$

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

$$k_\rho = \frac{\rho_1}{\rho_m}$$

32.3.3.3 Calculation of the volume flowrate

Under the test conditions the volume flowrate is determined by the following expression:

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

where

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

32.3.3.4 Calculation of fan air power

32.3.3.4.1 Fan work per unit mass and fan air power

In accordance with 14.8.1, the fan static work per unit mass and the fan work per unit mass are given by the following expressions:

$$y_{sA} = \frac{p_2 - p_1}{\rho_m} - \frac{v_{m1}^2}{2} = \frac{p_2 - p_1}{\rho_m} - \frac{1}{2}\left(\frac{q_m}{A_1 \rho_1}\right)^2$$

$$y_A = \frac{p_2 - p_1}{\rho_m} + \frac{v_{m2}^2}{2} - \frac{v_{m1}^2}{2}$$

$$= \frac{p_2 - p_1}{\rho_m} + \frac{1}{2}\left(\frac{q_m}{A_2 \rho_2}\right)^2 - \frac{1}{2}\left(\frac{q_m}{A_1 \rho_1}\right)^2$$

The fan static air power and the fan air power are given by

$$P_{usA} = q_m y_{sA}$$

$$P_{uA} = q_m y_A$$

32.3.3.4.2 Calculation of fan air power and compressibility coefficients

In accordance with 14.8.2:

$$P_{usA} = q_{Vsg1}\, p_{sFA}\, k_{ps}$$

$$P_{uA} = q_{Vsg1}\, p_{FA}\, k_p$$

The compressibility coefficients k_p and k_{ps} may be determined by two equivalent methods (see 14.8.2.1 and 14.8.2.2).

a) First method:

$$k_{ps} \text{ or } k_p = \frac{Z_k \log_{10} r}{\log_{10}\left[1 + Z_k (r - 1)\right]}$$

where

$$r = 1 + \frac{p_{sFA}}{p_{sg1}} \text{ and } Z_k = \frac{\kappa - 1}{\kappa} \frac{P_r}{q_{Vsg1}\, p_{sFA}} \text{ for } k_{ps}$$

$$r = 1 + \frac{p_{FA}}{p_{sg1}} \text{ and } Z_k = \frac{\kappa - 1}{\kappa} \frac{P_r}{q_{Vsg1}\, p_{FA}} \text{ for } k_p$$

b) Second method:

$$k_{ps} \text{ or } k_p = \frac{\ln(1 + x)}{x} \frac{Z_p}{\ln(1 + Z_p)}$$

where

$$x = r - 1 = \frac{p_{sFA}}{p_{sg1}} \text{ for } k_{ps}$$

$$x = r - 1 = \frac{p_{FA}}{p_{sg1}} \text{ for } k_p$$

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

32.3.3.5 Calculation of efficiencies

In accordance with 14.8.1, the efficiencies are given by the following expressions:

— Fan static efficiency:

$$\eta_{srA} = \frac{P_{usA}}{P_r}$$

— Fan efficiency:

$$\eta_{rA} = \frac{P_{uA}}{P_r}$$

32.3.4 Simplified procedures

32.3.4.1 Reference Mach number Ma_{2ref} less than 0,15 but pressure ratio more than 1,02

The stagnation and static temperatures may be considered as equal and the Mach factors F_M are equal to 1 (see 14.9.1).

$$\Theta_x = \Theta_{sgx}$$

$$F_{M1} = F_{M2} = 1$$

32.3.4.1.1 Calculation of mass flowrate

The mass flowrate is calculated by the procedure described in 32.3.1.

32.3.4.1.2 Calculation of fan pressure

32.3.4.1.2.1 Fan outlet pressure

Assuming that

$$\Theta_2 = \Theta_{sg2} = \Theta_{sg4} = \Theta_4 = t_4 + 273{,}15$$

$$F_{M1} = F_{M2} = 1$$

$$\rho_2 = \frac{p_2}{R_w \Theta_2} = \frac{p_4}{R_w \Theta_4}$$

$$p_{sg2} = p_4 + \rho_2 \frac{v_{m2}^2}{2} = p_4 + \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2$$

or

$$p_{esg2} = p_{e4} + \rho_2 \frac{v_{m2}^2}{2} = p_{e4} + \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2$$

32.3.4.1.2.2 Fan inlet pressure

$$p_a = p_{sg1}$$

$$\rho_{sg1} = \rho_a = \frac{p_a}{R_w \Theta_{sg1}}$$

$$p_1 = p_{sg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2$$

or

$$p_{e1} = -\frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2$$

where

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

But p_1 is unknown and an interative procedure shall be used to determine p_1 and ρ_1 (see 14.9.1.4).

Two or three calculation steps are sufficient.

The pressure p_1 may be determined by the following expression:

$$p_1 = \frac{p_{sg1} + \sqrt{p_{sg1}^2 - 2\left(\frac{q_m}{A_1}\right)^2 R_w \Theta_{sg1}}}{2}$$

and

$$p_{e1} = p_1 - p_a$$

32.3.4.1.2.3 Fan pressure

The fan static pressure and the fan pressure are given by the following expressions:

$$p_{sFA} = p_2 - p_{sg1} = p_4 - p_a = p_{e4}$$

$$p_{FA} = p_{sg2} - p_{sg1} = p_4 + \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 - p_a$$

$$= p_{e4} + \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2$$

32.3.4.1.3 Calculation of volume flowrate

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

32.3.4.1.4 Calculation of fan air power

Fan work per unit mass and fan air power are calculated in accordance with 14.8.1, 14.8.2 and 32.3.3.4.

32.3.4.1.5 Calculation of efficiencies

Efficiencies are calculated in accordance with 32.3.3.5.

32.3.4.2 Reference Mach number less than 0,15 and pressure ratio less than 1,02 (see 14.9.2)

The flow through the fan and the test airway may be considered as incompressible.

$$\Theta_1 = \Theta_{sg1} = \Theta_2 = \Theta_{sg2} = \Theta_4 = \Theta_{sg4} = \Theta_u = \Theta_a = t_a + 273,15$$

$$p_1 = p_2$$

$$F_{M1} = F_{M2} = 1$$

$$k_p = 1$$

32.3.4.2.1 Calculation of mass flowrate

The mass flowrate is determined in accordance with 32.3.3.1.

32.3.4.2.2 Calculation of fan pressure

32.3.4.2.2.1 Fan outlet pressure

$$\rho_1 = \rho_{sg1} = \rho_2 = \rho_{sg2} = \rho_4 = \rho_u = \rho_a = \frac{p_a}{R_w \Theta_a}$$

$$p_{sg2} = p_4 + \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2$$

or

$$p_{esg2} = p_{e4} + \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2$$

32.3.4.2.2.2 Fan inlet pressure

$$p_1 = p_{sg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2$$

$$p_{e1} = -\frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2$$

32.3.4.2.2.3 Fan pressure

The fan pressures are given by the following expressions:

$$p_{sFA} = p_2 - p_{sg1} = p_4 - p_a = p_{e4}$$

$$p_{FA} = p_{sg2} - p_{sg1} = p_4 + \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2 - p_a$$

$$= p_{e4} + \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2 = p_{esg4}$$

32.3.4.2.3 Calculation of volume flowrate

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

32.3.4.2.4 Calculation of fan air power

The fan air powers are determined by the following expressions:

$$P_{usA} = q_{Vsg1} \cdot p_{sFA}$$

$$P_{uA} = q_{Vsg1} \cdot p_{FA}$$

32.3.4.2.5 Calculation of fan efficiencies

Fan efficiencies are calculated in accordance with 14.8.1 and 32.3.3.5.

32.3.5 Fan performance under test conditions

Under the test conditions the fan performances are as follows:

— inlet volume flow, q_{Vsg1};

— fan static pressure, p_{sFA};

— fan pressure, p_{FA};

— fan efficiency, η_{srA} or η_{rA}.

33 Standard test methods with outlet-side test ducts — Type B installations

33.1 Types of fan setup

Two general setups of fan are shown:

a) outlet test duct with antiswirl device, the pressure being measured downstream of the antiswirl device;

b) outlet duct of the short type; 2 or 3 equivalent diameters long without antiswirl device, in which no measurements are taken, followed by an outlet chamber and a flowmeter. The results obtained in this way may differ to some extent from those obtained using common airways on the outlet side.

Eight methods of controlling and measuring the flowrate in the test duct are shown in the first case, and two methods in the second case. The method of flowrate measurement is specified in each case, together with the clauses and figures detailing the flow measurement procedure.

A common procedure, comprising measurements to be taken and quantities to be calculated allowing the determination of fan performance in type B installations, is given in 33.2.3 to 33.2.3.5. It is generally valid for all fans in accordance with this International Standard.

However, two simplified procedures may be followed when:

— the reference Mach number Ma_{2ref} is less than 0,15 but the pressure ratio more than 1,02;

— the reference Mach number Ma_{2ref} is less than 0,15 and the pressure ratio less than 1,02.

In these circumstances, the procedures which are given in 14.9.1, 14.9.2, 33.2.4 and 33.3.4 may be followed.

33.2 Outlet-side test ducts with antiswirl device

33.2.1 Mass flowrate determination

The mass flowrate is determined using:

— in-duct ISO Venturi nozzle, see clause 22 and figure 72 a);

— outlet orifice with wall taps, see 26.9 and figure 72 b);

— in-duct orifice with D and $D/2$ taps, see 26.7 and figure 72 c);

— in-duct orifice with corner taps, see 26.8 and figure 72 c);

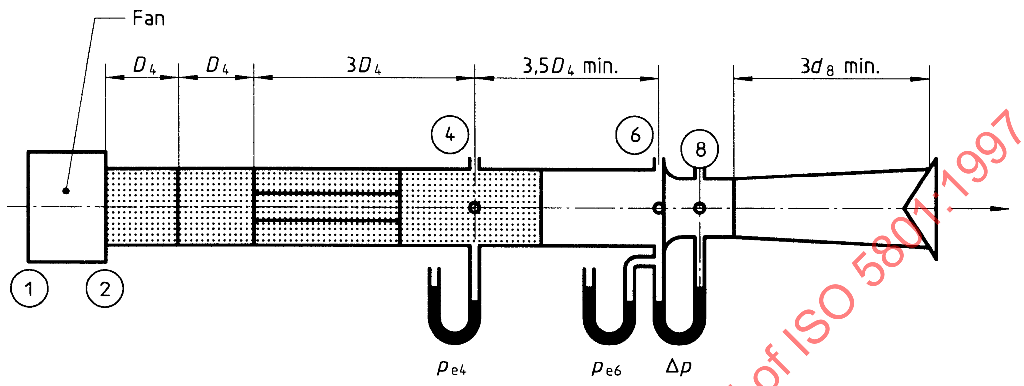
— Pitot-static tube traverse, see clause 27 and figure 72 d) and e);

— in-duct Venturi nozzle, see clause 23 and figure 72 f);

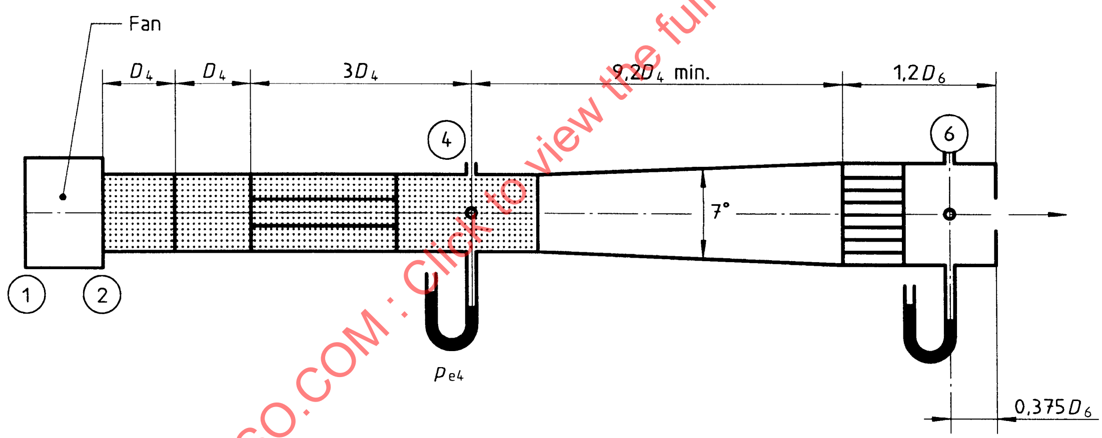
— outlet Venturi nozzle on chamber, see clause 23 and figure 72 g);

— multiple nozzles in chamber, see clause 23 and figure 72 h).

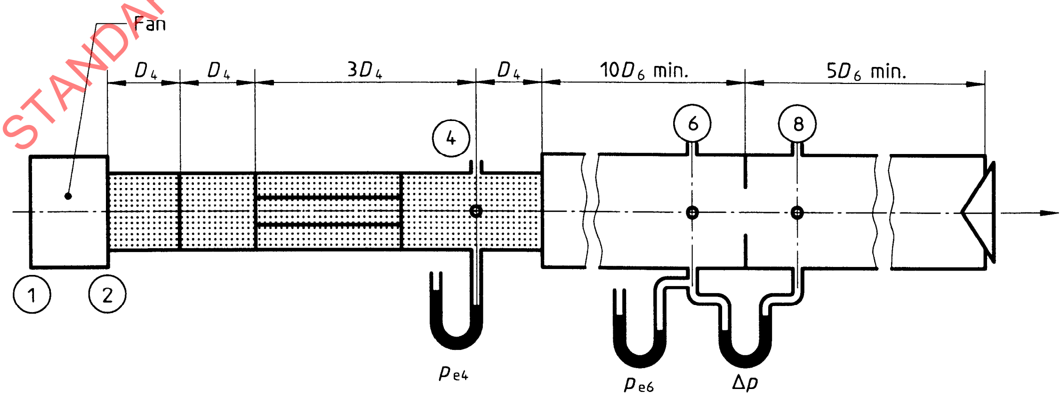
Dimensions in millimetres



a) Flowrate determination using ISO Venturi nozzle



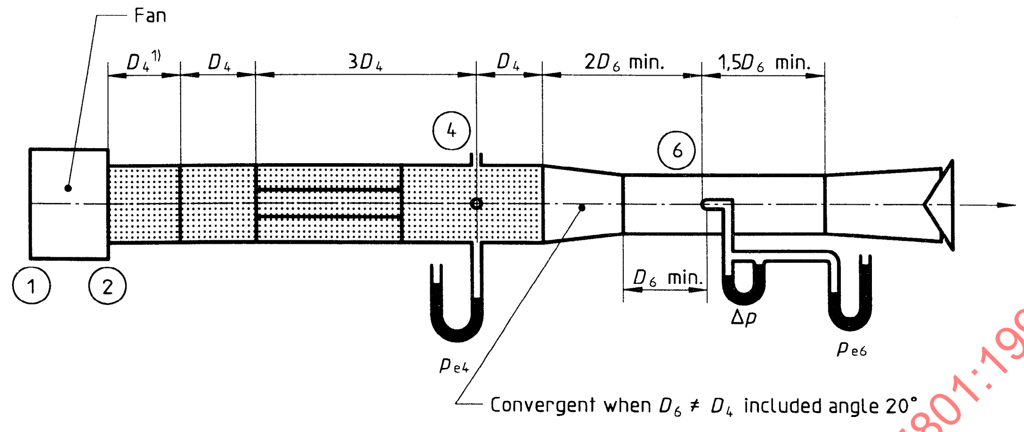
b) Flowrate determination using outlet orifice



c) Flowrate determination using in-duct orifice with taps at D and $0,5D$ or in-duct orifice with corner taps

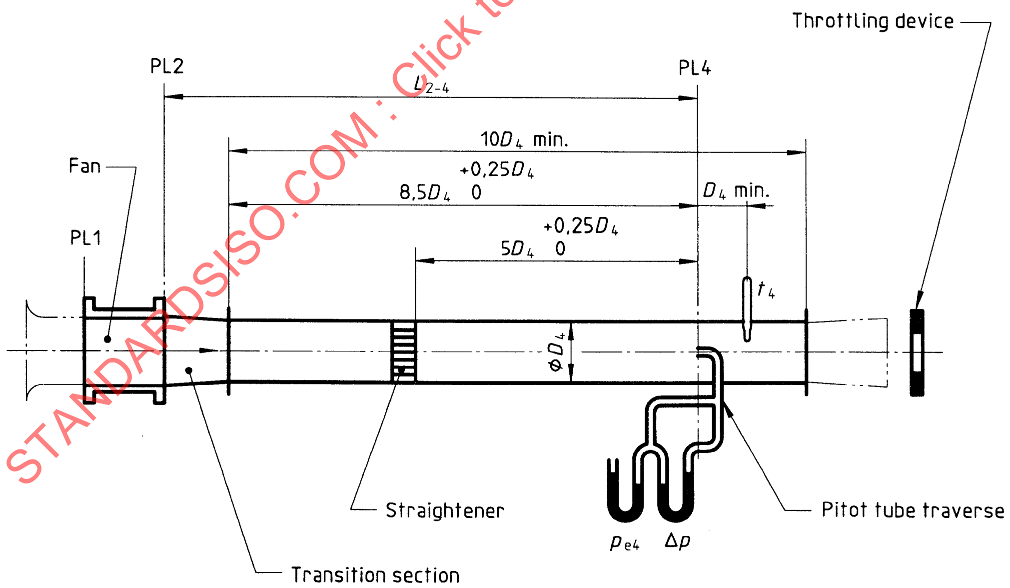
Figure 72 — Type B test installations (with antiswirl device)

Dimensions in millimetres



1) This cylindrical airway section of length D_4 may be replaced by a transition section in accordance with clause 30 when required to accommodate a change in area and/or shape.

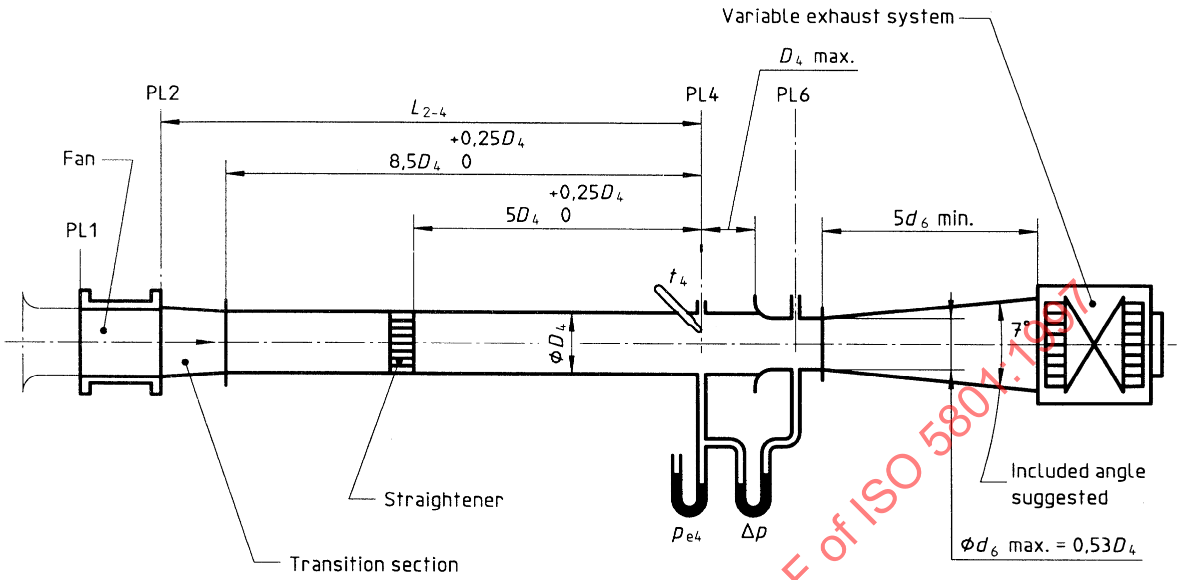
d) Flowrate determination using Pitot-static tube traverse



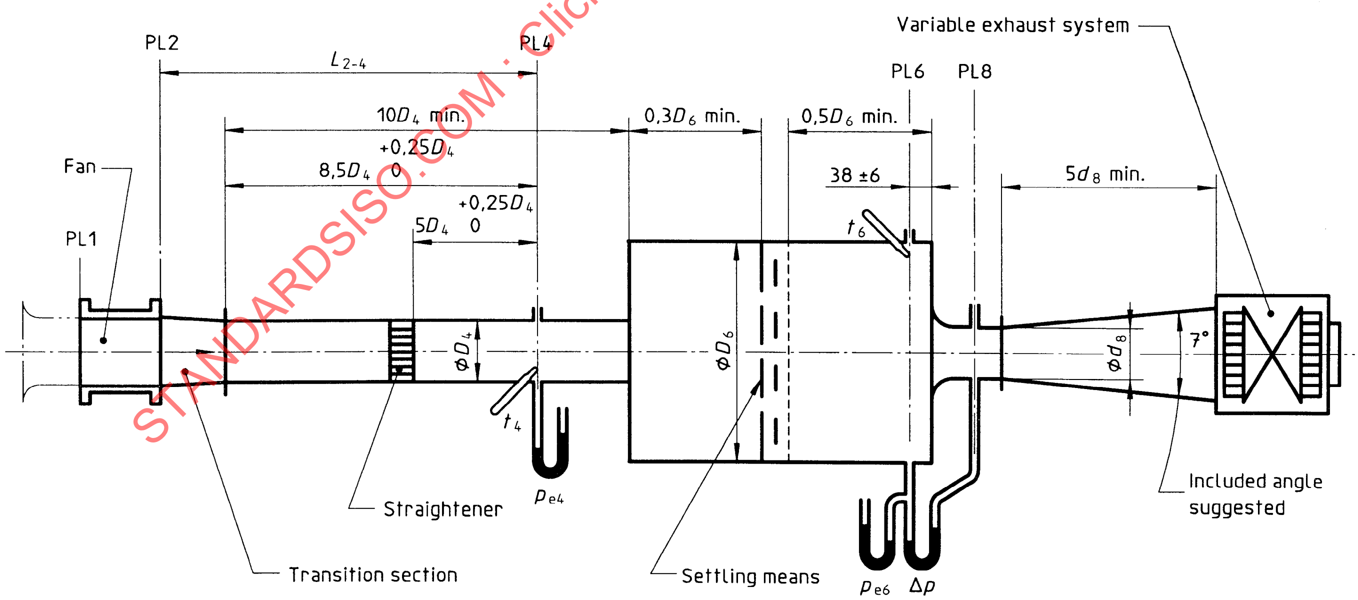
e) Flowrate determination using Pitot-static tube traverse

Figure 72 — Type B test installations (with antiswirl device) *(continued)*

Dimensions in millimetres



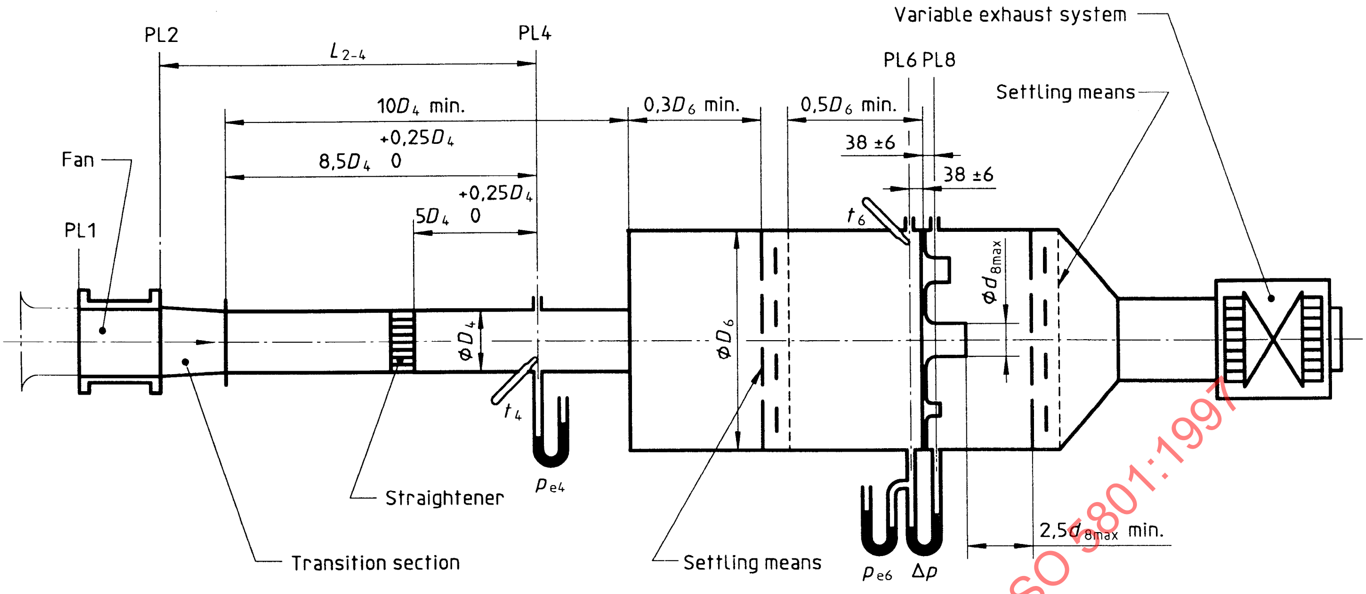
f) Flowrate determination using in-duct Venturi nozzle



g) Flowrate determination using nozzle on end of chamber

Figure 72 — Type B test installations (with antiswirl device) *(continued)*

Dimensions in millimetres



h) Flowrate determination using multiple nozzles in chamber

Figure 72 — Type B test installations (with antiswirl device) (*concluded*)

33.2.2 Measurements to be taken during tests (see clause 20)

Measure:

— rotational speed, N, or rotational frequency, n;

— power input, P_a, P_o or P_e, and estimate impeller power (see 10.4);

— outlet pressure, p_{e4};

— pressure, p_{e6}, upstream of flowmeter;

— differential pressure, Δp;

— chamber temperature, t_6.

In the test enclosure, measure

— atmospheric pressure, p_a, at the mean fan altitude;

— ambient temperature near fan inlet, t_a;

— dry- and wet-bulb temperatures, t_d and t_w.

Determine the ambient air density, ρ_a, and the gas constant of humid air, R_w (see clause 12).

33.2.3 General procedure for compressible fluid flow

This procedure should be applied when both the reference Mach number Ma_{2ref} is more than 0,15 and the pressure ratio is more than 1,02.

33.2.3.1 Calculation of mass flowrate

33.2.3.1.1 The mass flowrate is determined using:

— in-duct ISO Venturi nozzle, see clause 22 and figure 72 a);

— outlet orifice with wall taps, see 26.9 and figure 72 b);

— in-duct orifice with taps at D and $D/2$, see 26.7 and figure 72 c);

— in-duct orifice with corner taps, see 26.8 and figure 72 c).

The outlet test ducts for pressure and flowrate measurements are followed by a control device or an auxiliary fan with a control device.

a) The temperature in the test duct t_4 or t_6 is not measured.

This is the normal procedure.

Assuming that

$$p_6 = p_{e6} + p_a$$

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

$$\Theta_{sg6} = \Theta_{sg1} + \frac{P_r \text{ or } P_e}{q_m c_p}$$

$$\Theta_6 = \Theta_{sg6} - \frac{q_m^2}{2A_6^2 \rho_6^2 c_p} = \Theta_{sg1} + \frac{P_r \text{ or } P_e}{q_m c_p} - \frac{q_m^2}{2A_6^2 \rho_6^2 c_p}$$

$$\rho_6 = \frac{p_6}{R_w \Theta_6}$$

but Θ_6, Θ_{sg6}, q_m are unknown.

The mass flowrate is determined by the following expression:

$$q_m = \alpha \varepsilon \pi \frac{d_8^2}{4} \sqrt{2\rho_6 \Delta p}$$

where

ε is the expansibility coefficient determined in accordance with 22.3.3, 26.7, 26.8, 26.9;

α is the flow coefficient function of Reynolds number Re_{d8} or Re_{D6} estimated by the following expressions:

$$Re_{d8} = \frac{\alpha \varepsilon d_8 \sqrt{2\rho_6 \Delta p}}{17,1 + 0,048 t_6} \times 10^6$$

or

$$Re_{D6} = \frac{\alpha \varepsilon \beta d_6 \sqrt{2\rho_6 \Delta p}}{17,1 + 0,048 t_6} \times 10^6$$

α or the compound coefficient $\alpha\varepsilon$ is determined in accordance with 22.3, 26.7, 26.8, 26.9 and figures 18, 24, 25, 26 and 27.

An iterative procedure should be applied to calculate Θ_6, ρ_6, Re_{d8} or Re_{D6}, α and q_m from a first value of Θ_6: $\Theta_6 = \Theta_{sg6} = \Theta_{sg1}$

Three or four iterations are sufficient to obtain q_m with a calculation accuracy of 10^{-3}.

b) The temperature t_6 is measured. It is considered as a stagnation temperature Θ_{sg6}:

$$\Theta_6 = t_6 + 273,15 - \frac{q_m^2}{2A_2^2 \rho_6^2 c_p}$$

and the above procedure is applied.

33.2.3.1.2 Flowrate is determined using a Pitot-static tube traverse, see clause 27, figure 72 d) and e).

NOTE 45 For the installation in figure 72 e), plane 4 and plane 6 are identical.

The outlet ducts for pressure and flowrate measurements are followed by a control device or an auxiliary fan with a control device.

Assuming that

$$p_{e6} = \frac{1}{n}\sum_{j=n}^{n} p_{e6j}$$

$$p_6 = p_{e6} + p_a$$

$$\Theta_{sg6} = \Theta_{sg1} + \frac{P_r \text{ or } P_e}{q_m c_p}$$

$$\Theta_6 = \Theta_{sg6}\left(\frac{p_6}{p_6 + \Delta p_m}\right)^{\frac{\kappa - 1}{\kappa}}$$

where

$$\Delta p_m = \left(\frac{1}{n}\sum_{j=1}^{n}\Delta p_j^{0,5}\right)^2$$

$$= \left[\frac{1}{n}\left(\sqrt{\Delta p_1} + \sqrt{\Delta p_2} + \ldots\sqrt{\Delta p_n}\right)\right]^2$$

$$\rho_6 = \frac{p_6}{R_w \Theta_6}$$

The mass flowrate q_m is determined by the following expression:

$$q_m = \alpha\varepsilon A_6\sqrt{2\rho_6\Delta p_m}$$

where

α is a flowrate coefficient function of the Reynolds number Re_{D6}, very close to 0,99 (see 27.6).

$$Re_{D6} = \alpha\varepsilon D_6 \frac{\sqrt{2\rho_6\Delta p_m}}{17{,}1 + 0{,}048\, t_6} \times 10^6$$

ε is the expansibility coefficient (see 27.5):

$$\varepsilon = \left[1 - \frac{1}{2\kappa}\frac{\Delta p_m}{p_6} + \frac{\kappa + 1}{6\kappa^2}\left(\frac{\Delta p_m}{p_6}\right)^2\right]^{1/2}$$

A first approximation of q_m is calculated with $\alpha = 0{,}99$ and ε is calculated by the expression above.

This value of q_m allows calculation of Re_{D6}, α and a second value of q_m.

Two or three iterations are sufficient to determine the mass flowrate with a calculation accuracy of 10^{-3}.

33.2.3.1.3 The mass flowrate is determined using an in-duct Venturi nozzle, see clause 23 and figure 72 f).

The outlet ducts for pressure and flowrate measurements are followed by a control device or an auxiliary fan with a control device.

Assuming that

$$p_4 = p_{e4} + p_a$$

$$\rho_4 = \frac{p_4}{R_w \Theta_4}$$

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

$$\Theta_4 = \Theta_{sg1} + \frac{P_r \text{ or } P_e}{q_m c_p} - \frac{q_m{}^2}{2A_4^2 \rho_4^2 c_p}$$

Θ_{sg4}, Θ_4, q_m are unknown.

$$\beta = \frac{d_6}{D_4}$$

The mass flowrate q_m is given by the following expression (see clause 23):

$$q_m = \alpha \varepsilon \pi \frac{d_6^2}{4} \sqrt{2\rho_4 \Delta p}$$

$$q_m = \varepsilon C \pi \frac{d_6^2}{4} \frac{\sqrt{2\rho_4 \Delta p}}{\sqrt{1 - \alpha_{Au} \beta^4}}$$

where

α is the flow coefficient of nozzle equal to:

$$\frac{C}{\sqrt{1 - \alpha_{Au} \beta^4}}$$

C is the discharge coefficient of the nozzle, a function of the throat Reynolds number Re_{d6} (see 23.4.2 and table 5):

$$Re_{d6} = \frac{\alpha \varepsilon d_6 \sqrt{2\rho_4 \Delta p}}{17{,}1 + 0{,}048 t_4} \times 10^6$$

α_{Au} is a kinetic energy coefficient equal to 1,043 for a duct approach and 1 for a chamber approach;

ε is the expansibility coefficient calculated in accordance with 23.4.3 and table 6.

For a first approximation,

$$\alpha = \frac{0{,}95}{\sqrt{1 - \alpha_{Au} \beta^4}}$$

$$\Theta_4 = \Theta_{sg1}$$

The value of q_m calculated allows calculation of new values of Θ_4, ρ_4, Re_{d6}, α and q_m.

Two or three iterations are sufficient for a calculation accuracy 10^{-3} for q_m.

33.2.3.1.4 The mass flowrate is determined using

— outlet Venturi nozzle on chamber, see clause 23 and figure 72 g) and h),

— multiple nozzles in chamber, see clause 23 and figure 72 g) and h).

The outlet ducts for pressure and flowrate measurements are followed by a control device or an auxiliary fan with a control device.

The temperature t_6 in the chamber may be measured:

$$p_6 = p_{e6} + p_a$$

$$\Theta_6 = \Theta_{sg6} = t_6 + 273{,}15$$

$$\beta = \frac{d_8}{D_6} = 0$$

$$\rho_6 = \frac{p_6}{R_w \Theta_6}$$

The mass flowrate is given by the following expression:

$$q_m = \varepsilon \pi \sum_{j=1}^{n} \left(C_j \frac{d_{8j}^2}{4} \right) \sqrt{2 \rho_6 \Delta p}$$

where

ε is the expansibility coefficient in accordance with 23.4.3 and table 6;

C_j is the discharge coefficient of the jth nozzle, as a function of the nozzle throat Reynolds number

Re_{d8j}, see 23.4;

$\beta = 0$ and $C_j = \alpha_j$

$C_j = \alpha_j$ is calculated in accordance with 23.4 and table 5;

n is the number of nozzles, equal to 1 for a nozzle on end of chamber.

For each nozzle, the throat Reynolds number Re_{d8} is estimated using the following expression:

$$Re_{d8j} = \frac{\varepsilon C_j d_{8j} \sqrt{2 \rho_6 \Delta p}}{17{,}1 + 0{,}048\, t_6} \times 10^6$$

with $C_j = 0{,}95$

After a first estimation of the mass flowrate, the discharge coefficients C_j are corrected for the Reynolds number variations.

33.2.3.2 Calculation of fan pressure

33.2.3.2.1 Fan outlet pressure

Assuming that

$$p_4 = p_{e4} + p_a$$

$$\Theta_{sg4} = \Theta_{sg2} = \Theta_{sg1} + \frac{P_r \text{ or } P_e}{q_m c_p} = \Theta_{sg6} = t_6 + 273{,}15$$

The Mach number in section 4 and the ratio Θ_{sg4}/Θ_4 are determined in accordance with 14.4.3.1 and figure 5.

$$\Theta_4 = \Theta_{sg4} \frac{\Theta_4}{\Theta_{sg4}}$$

$$\rho_4 = \frac{p_4}{R_w \Theta_4}$$

$$F_{M4} = 1 + \frac{Ma_4^2}{4} + \frac{Ma_4^4}{40} + \frac{Ma_4^6}{1600} \text{ (see 14.5.1)}$$

The friction loss coefficient between sections 2 and 4 $(\zeta_{2-4})_4$ is calculated in accordance with 30.6 and figure 65.

The stagnation pressure at fan outlet p_{sg2} is given by the following expression:

$$p_{sg2} = p_4 + \frac{\rho_4 v_{m4}^2}{2} F_{M4} \left[1 + \left(\zeta_{2-4} \right)_4 \right]$$

or

$$p_{esg2} = p_{e4} + \frac{\rho_4 v_{m4}^2}{2} F_{M4} \left[1 + \left(\zeta_{2-4} \right)_4 \right]$$

The static density ρ_2 and the pressure p_2 are calculated in accordance with 14.5.2, Ma_2 being determined in accordance with 14.4.3.2 and figure 6.

$$p_2 = p_{sg2} - \rho_2 \frac{v_{m2}^2}{2} F_{M2}$$

$$= p_{sg2} - \frac{1}{2\rho_2} \left(\frac{q_m}{A_2} \right)^2 F_{M2}$$

or

$$p_{e2} = p_{esg2} - \frac{1}{2\rho_2} \left(\frac{q_m}{A_2} \right)^2 F_{M2}$$

33.2.3.2.2 Fan inlet pressure

At the fan inlet $p_{sg1} = p_a$, $\Theta_{sg1} = \Theta_a$, and p_1 may be determined in accordance with 14.5.2 and 14.4.3.2.

$$p_1 = p_{sg1} - \rho_1 \frac{v_{m1}^2}{2} F_{M1} = p_{sg1} - \frac{1}{2\rho_1} \left(\frac{q_m}{A_1} \right)^2 F_{M1}$$

Ma_1 and ρ_1 being calculated in accordance with 14.4.3.2 and 14.5.2 and figures 4, 5 and 6.

We have also:

$$p_{esg1} = 0$$

$$p_{e1} = -\frac{1}{2\rho_1} \left(\frac{q_m}{A_1} \right)^2 F_{M1}$$

33.2.3.2.3 Fan pressure

The fan pressure p_{FB} and the fan static pressure p_{sFB} may be calculated using the following expression:

$$p_{FB} = p_{sg2} - p_{sg1} = p_{esg2}$$

$$p_{sFB} = p_2 - p_{sg1} = p_{e2}$$

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

and

$$k_\rho = \rho_1 / \rho_m$$

33.2.3.3 Calculation of volume flowrate

The volume flowrate is calculated by the following expression:

$$q_{vsg1} = \frac{q_m}{\rho_{sg1}} = \frac{q_m}{\left(\dfrac{p_{sg1}}{R_w \Theta_{sg1}} \right)}$$

33.2.3.4 Calculation of fan air power

33.2.3.4.1 Fan work per unit mass and fan air power

In accordance with 14.8.1, the fan work per unit mass y_B and the fan static work per unit mass y_{sB} are given by the following expressions:

$$y_B = \frac{p_2 - p_1}{\rho_m} + \frac{v_{m2}^2}{2} - \frac{v_{m1}^2}{2}$$

$$= \frac{p_2 - p_1}{\rho_m} + \frac{1}{2}\left(\frac{q_m}{A_2\rho_2}\right)^2 - \frac{1}{2}\left(\frac{q_m}{A_1\rho_1}\right)^2$$

$$= \frac{p_{e2} - p_{e1}}{\rho_m} + \frac{v_{m2}^2}{2} - \frac{v_{m1}^2}{2}$$

$$y_{sB} = \frac{p_2 - p_1}{\rho_m} - \frac{v_{m1}^2}{2} = \frac{p_2 - p_1}{\rho_m} - \frac{1}{2}\left(\frac{q_m}{A_1\rho_1}\right)^2$$

$$= \frac{p_{e2} - p_{e1}}{\rho_m} - \frac{v_{m1}^2}{2}$$

The fan air power P_{uB} and the fan static power P_{usB} are given by the following expressions:

$$P_{uB} = q_m y_B$$

$$P_{usB} = q_m y_{sB}$$

33.2.3.4.2 Calculation of fan air power and compressibility coefficients

In accordance with 14.8.2:

$$P_{uB} = q_{Vsg1}\, p_{FB}\, k_p$$

$$P_{usB} = q_{Vsg1}\, p_{sFB}\, k_{ps}$$

The compressibility coefficients k_p and k_{ps} may be determined by two equivalent methods (see 14.8.2.1 and 14.8.2.2).

a) First method:

$$k_{ps} \text{ or } k_p = \frac{Z_k \log_{10} r}{\log_{10}\left[1 + Z_k(r-1)\right]}$$

where

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

for k_p or

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

for k_{ps} and

$$Z_k = \frac{\kappa - 1}{\kappa}\,\frac{P_r}{q_{Vsg1}\, p_{FB}}$$

for k_p or

$$Z_k = \frac{\kappa - 1}{\kappa} \frac{P_r}{q_{Vsg1} \, p_{sFB}}$$

for k_{ps}.

b) Second method:

$$k_{ps} \text{ or } k_p = \frac{\ln(1+x)}{x} \frac{Z_p}{\ln(1+Z_p)}$$

where

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

or

$$x = \frac{p_{sFB}}{p_{sg1}}$$

and

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

33.2.3.5 Calculation of efficiencies

In accordance with 14.8.1, the efficiencies are calculated by the following expressions:

— Fan efficiency:

$$\eta_{rB} = \frac{P_{uB}}{P_r}$$

— Fan static efficiency:

$$\eta_{srB} = \frac{P_{usB}}{P_r}$$

— Fan shaft efficiency:

$$\eta_{sB} = \frac{P_{uB}}{P_a}$$

— Fan static shaft efficiency:

$$\eta_{saB} = \frac{P_{usB}}{P_a}$$

33.2.4 Simplified procedures

33.2.4.1 Reference Mach number Ma_{2ref} less than 0,15 and pressure ratio more than 1,02

At a section of the test duct the stagnation and static temperature are considered as equal:

$$\Theta_x = \Theta_{sgx} = t_x + 273,15$$

The Mach factors F_{M1} and F_{M2} are equal to 1.

33.2.4.1.1 Calculation of mass flowrate

The mass flowrate is determined in accordance with the methods described in 33.2.3.1.

However, the following simplification applies:

The temperature upstream of the flowmeter may be measured:

$$\Theta_u = \Theta_{sgu} = t_u + 273{,}15 = \Theta_{sg4} = \Theta_4$$

$$p_u = p_{eu} + p_a$$

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

There is no need for an iterative procedure to take into account the difference between stagnation and static temperatures, but the Reynolds number effect on α or C should be applied.

33.2.4.1.2 Calculation of fan pressure

33.2.4.1.2.1 Fan outlet pressure

In accordance with 14.9.1.2, 14.9.1.3 and 30.6

$$p_{sg2} = p_4 + \rho_4 \frac{v_{m4}^2}{2}\left[1 + \left(\zeta_{2-4}\right)_4\right]$$

or

$$p_{esg2} = p_{e4} + \rho_4 \frac{v_{m4}^2}{2}\left[1 + \left(\zeta_{2-4}\right)_4\right]$$

where

$$\rho_4 = \frac{p_4}{R_w \Theta_4} = \frac{p_4}{R_w \Theta_{sg4}}$$

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

p_2 may be determined by the following method (see 14.9.1.4):

$$(p_2)_1 = p_{sg2} - \rho_4 \frac{v_{m4}^2}{2}\left(\frac{A_4}{A_2}\right)^2$$

$$(\rho_2)_1 = \frac{(p_2)_1}{R_w \Theta_{sg2}}$$

$$p_2 = p_{sg2} - \frac{1}{2}\frac{q_m^2}{A_2^2 (\rho_2)_1}$$

$$p_{e2} = p_{esg2} - \frac{1}{2}\frac{q_m^2}{A_2^2 (\rho_2)_1}$$

Two or three iterations are sufficient; p_2 may be also determined by the following expression:

$$p_2 = \frac{1}{2}\left[p_{sg2} + \sqrt{p_{sg2}^2 - 2\left(\frac{q_m}{A_2}\right)^2 R_w \Theta_{sg2}}\right]$$

$$\rho_2 = \frac{p_2}{R_w \Theta_{sg2}}$$

33.2.4.1.2.2 Fan inlet pressure

At the fan inlet

$$p_{sg1} = p_a$$

$$p_{esg1} = 0$$

The static pressure p_1 may be determined by one of the two methods applied in 33.2.4.1.2.1.

33.2.4.1.2.3 Fan pressure

The fan pressure p_{FB} and the fan static pressure p_{sFB} are given by the following expressions:

$$p_{FB} = p_{sg2} - p_{sg1} = p_{sg2} - p_a = p_{esg2}$$

$$p_{sFB} = p_2 - p_{sg1} = p_2 - p_a = p_{e2}$$

33.2.4.1.3 Calculation of volume flowrate

The volume flowrate is given by the following expression:

$$q_{Vsg1} = \frac{q_m}{\rho_{sg1}} = \frac{q_m}{\left(\dfrac{p_a}{R_w \Theta_{sg1}}\right)}$$

33.2.4.1.4 Calculation of fan air power

Fan air powers are determined in accordance with 14.8.1, 14.8.2, 14.8.3 and 33.2.3.4.

33.2.4.1.5 Calculation of fan efficiencies

Fan efficiencies are calculated in accordance with 14.8.1, 14.8.2, 14.8.3 and 33.2.3.5.

33.2.4.2 Reference Mach number Ma_{2ref} less than 0,15 and pressure ratio less than 1,02 (see 14.9.2)

The air flow through the fan and the test airway may be considered as incompressible.

$$\Theta_1 = \Theta_{sg1} = \Theta_2 = \Theta_{sg2} = \Theta_a = t_a + 273,15$$

$$\rho_1 = \rho_2 = \rho_4 = \rho_6 = \rho_a = \frac{p_a}{R_w \Theta_a}$$

$$F_{M1} = F_{M2} = 1$$

$$k_p = 1$$

33.2.4.2.1 Calculation of mass flowrate

The mass flowrate is determined in accordance with 33.2.4.1.1, with $\rho_u = \rho_a = \dfrac{p_a}{R_w \Theta_a}$

33.2.4.2.2 Calculation of fan pressure

33.2.4.2.2.1 Fan outlet pressure

According to the assumption above,

$$\rho_1 = \rho_{sg1} = \rho_2 = \rho_{sg2} = \rho_4 = \rho_6 = \rho_a = \frac{p_a}{R_w \Theta_a}$$

$$p_{sg2} = p_4 + \frac{1}{2\rho_1}\left(\frac{q_m}{A_4}\right)^2 \left[1 + \left(\zeta_{2-4}\right)_4\right]$$

$$p_{esg2} = p_{e4} + \frac{1}{2\rho_1}\left(\frac{q_m}{A_4}\right)^2 \left[1 + \left(\zeta_{2-4}\right)_4\right]$$

$$p_2 = p_{sg2} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2$$

$$p_{e2} = p_{esg2} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2$$

33.2.4.2.2.2 Fan inlet pressure

$$p_{sg1} = p_a$$

$$p_{esg1} = 0$$

$$p_1 = p_{sg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2$$

$$p_{e1} = -\frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2$$

33.2.4.2.2.3 Fan pressure

The fan pressure p_{FB} and the fan static pressure p_{sFB} may be determined by the following expressions:

$$p_{FB} = p_{sg2} - p_{sg1} = p_{sg2} - p_a = p_{esg2}$$

$$p_{sFB} = p_2 - p_{sg1} = p_2 - p_a = p_{e2}$$

33.2.4.2.3 Calculation of volume flowrate

The volume flowrate is given by the following expression, as in 33.2.4.1.3:

$$q_{Vsg1} = \frac{q_m}{\rho_{sg1}} = \frac{q_m}{\left(\dfrac{p_a}{R_w \Theta_{sg1}}\right)}$$

33.2.4.2.4 Calculation of fan air power

In accordance with 14.9.2.6

$$P_{uB} = q_{Vsg1}\, p_{FB}$$

$$P_{usB} = q_{Vsg1}\, p_{sFB}$$

33.2.4.2.5 Calculation of fan efficiencies

Fan efficiencies are determined from P_{uB} or P_{usB} as in 33.2.3.5.

33.2.5 Fan performance under test conditions

Under tests conditions, the fan performances are the following:

- inlet volume flowrate, q_{Vsg1};
- fan pressure, p_{FB};
- fan static pressure, p_{sFB};
- fan efficiency, η_{rB} or η_{srB}.

33.3 Outlet chamber test ducts without antiswirl device

It is recommended to use the setups below only for fans without outlet swirling flow.

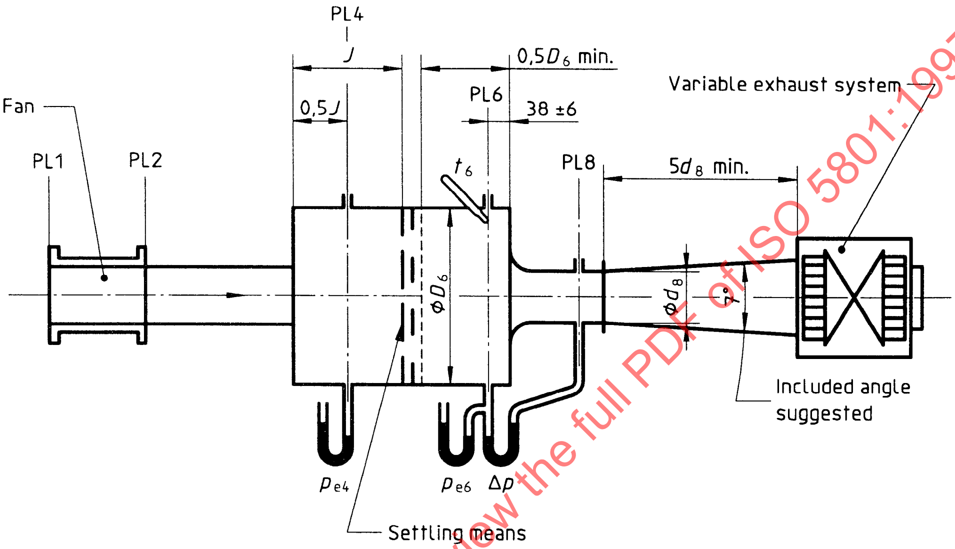
33.3.1 Mass flowrate determination

The mass flowrate is determined using

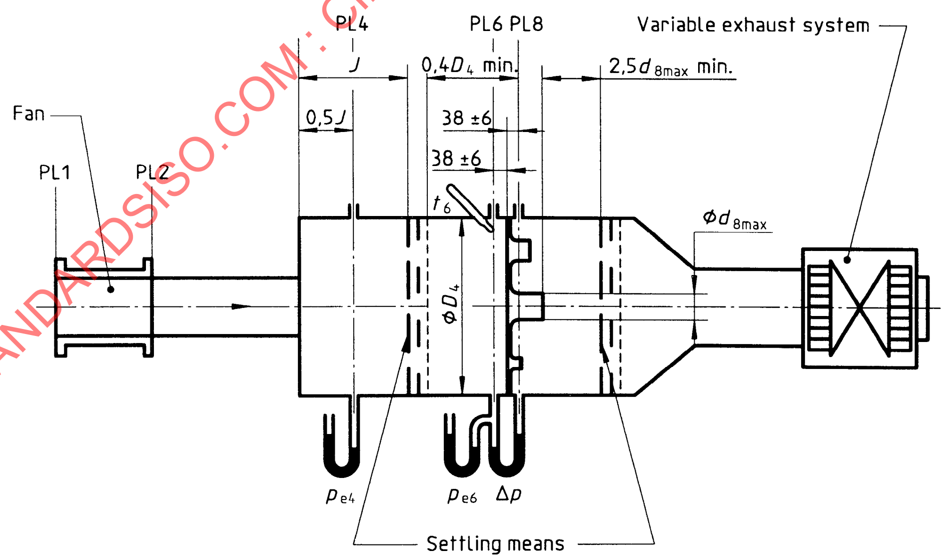
— Venturi nozzle on end of chamber, see clause 23 and figure 73 a);

— multiple nozzles in chamber, see clause 23 and figure 73 b).

Dimensions in millimetres



a) Flowrate determination using Venturi nozzle on end of chamber



b) Flowrate determination using multiple nozzles in chamber

NOTE — Dimension J shall be equal to at least the fan equivalent discharge diameter for fans with axis of rotation perpendicular to the discharge flow, and to at least twice the fan equivalent discharge diameter for fans with axis of rotation parallel to the discharge flow.

Figure 73 — Type B test installations (without antiswirl device)

33.3.2 Measurements to be taken during test (see clause 20)

Measure

- rotational speed, N, or rotational frequency, n;
- power input, P_a, P_o or P_e, and estimate impeller power (see 10.4);
- outlet pressure, p_{e4};
- pressure, p_{e6}, upstream of the flowmeter;
- differential pressure, Δp;
- outlet temperature, t_6.

In the test enclosure, measure

- atmospheric pressure, p_a, at the mean altitude of the fan;
- ambient temperature, t_a, near fan inlet;
- dry- and wet-bulb temperatures, t_d and t_w.

Determine the ambient air density, ρ_a, and the gas constant of humid air, R_w (see clause 12).

33.3.3 General procedure for compressible fluid flow

This procedure should be applied when both the reference Mach number Ma_{2ref} is more than 0,15 and the pressure ratio is more than 1,02.

33.3.3.1 Calculation of mass flowrate

33.3.3.1.1 The mass flowrate is determined using

- Venturi nozzle on end of chamber, see clause 23 and figure 73 a);
- multiple nozzles in chamber, see clause 23 and figure 73 b).

The outlet ducts for pressure and flowrate measurements are followed by a flowrate control device or an auxiliary fan with a flowrate control device.

The temperature t_6 in the chamber may be measured:

$$p_6 = p_{e6} + p_a$$

$$\Theta_6 = \Theta_{sg6} = t_6 + 273{,}15$$

$$\beta = \frac{d_8}{D_6} \approx 0$$

$$\rho_6 = \frac{p_6}{R_w \Theta_6}$$

The mass flowrate is given by the following expression:

$$q_m = \varepsilon \pi \sum_{j=1}^{n} \left(C_j \frac{d_{8j}}{4} \right) \sqrt{2 \rho_6 \Delta p}$$

where

ε is the expansibility coefficient in accordance with 23.4.3 and table 6;

C_j is the discharge coefficient of the jth nozzle, and is dependent upon the nozzle throat Reynolds number Re_{d8j}.

$\beta = 0$ and $C_j = \alpha j$

$C_j = \alpha_j$ is calculated in accordance with 23.4 and table 5;

n is the number of nozzles and is equal to 1 for a nozzle on end of chamber.

For each nozzle, the throat Reynolds number, Re_{d8}, is estimated with the following expression:

$$Re_{d8j} = \frac{\varepsilon C_j d_{8j} \sqrt{2\rho_6 \Delta p}}{17{,}1 + 0{,}048 t_6} 10^6$$

with $C_j = 0{,}95$.

It is recommended to use these setups only for fans without outlet swirling flow.

After a first estimation of the mass flowrate, the discharge coefficients C_j are determined and corrected.

33.3.3.2 Calculation of fan pressure

33.3.3.2.1 Fan outlet pressure

Assuming that

$$p_4 = p_{e4} + p_a$$

$$\Theta_{sg4} = \Theta_{sg2} = \Theta_{sg6} = t_6 + 273{,}15 = \Theta_{sg1} + \frac{P_r \text{ or } P_e}{q_m c_p}$$

the section 2.4 being the section of the outlet duct at the entrance to the chamber ($A_{2.4} = A_2$ if there is no outlet duct simulation), the Mach number at section 2.4, $Ma_{2.4}$, and the ratio $\Theta_{sg4}/\Theta_{2.4}$ are determined in accordance with 14.4.3.1 and figure 5.

$$\Theta_{2.4} = \Theta_{sg4} \frac{\Theta_{2.4}}{\Theta_{sg4}}$$

$$\rho_{2.4} = \frac{p_4}{R_w \Theta_{2.4}}$$

$$F_{M2.4} = 1 + \frac{Ma_{2.4}^2}{4} + \frac{Ma_{2.4}^4}{40} + \frac{Ma_{2.4}^6}{1\,600}$$

(see 14.5.1 and figure 4).

There is no loss allowance for this test duct, and the stagnation pressure at section 2 is given by the following expression:

$$p_{sg2} = p_4 + \frac{1}{2\rho_{2.4}} \left(\frac{q_m}{A_{2.4}} \right)^2 F_{M2.4}$$

or

$$p_{esg2} = p_{e4} + \frac{1}{2\rho_{2.4}} \left(\frac{q_m}{A_{2.4}} \right)^2 F_{M2.4}$$

The pressure p_2 and the static temperature Θ_2 in section 2 are determined in accordance with 14.5.2, Ma_2 being calculated in accordance with 14.4.3.2.

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

$$p_2 = p_{sg2} - \rho_2 \frac{v_{m2}^2}{2} F_{M2} = p_{sg2} - \frac{1}{2\rho_2} \left(\frac{q_m}{A_2} \right)^2 F_{M2}$$

or

$$p_{e2} = p_{esg2} - \frac{1}{2\rho_2} \left(\frac{q_m}{A_2} \right)^2 F_{M2}$$

33.3.3.2.2 Fan inlet pressure

At the fan inlet

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

$$p_{sg1} = p_a$$

and p_1 may be determined in accordance with 14.5.2, Ma_1 and Θ_1 being calculated in accordance with 14.4.3.2.

$$p_1 = p_{sg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2 F_{M1}$$

$$p_{e1} = -\frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2 F_{M1}$$

33.3.3.2.3 Fan pressure

The fan pressure, p_{FB}, and the fan static pressure p_{sFB} may be calculated by the following expressions:

$$p_{FB} = p_{sg2} - p_{sg1} = p_{sg2} - p_a = p_{sg2}$$

$$p_{sFB} = p_2 - p_{sg1} = p_2 - p_a = p_{e2}$$

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

and

$$k_\rho = \frac{\rho_1}{\rho_m}$$

33.3.3.3 Calculation of volume flowrate

The volume flowrate is calculated using the following expression:

$$q_{Vsg1} = \frac{q_m}{\rho_{sg1}} = \frac{q_m}{\left(\dfrac{p_{sg1}}{R_w \Theta_{sg1}}\right)}$$

33.3.3.4 Calculation of fan air power

33.3.3.4.1 Fan work per unit mass and fan air power

In accordance with 14.8.1, the fan work per unit mass y_B and the fan static work per unit mass y_{sB} are given by the following expressions:

$$y_B = \frac{p_2 - p_1}{\rho_m} + \frac{v_{m2}^2}{2} - \frac{v_{m1}^2}{2}$$

$$= \frac{p_2 - p_1}{\rho_m} + \frac{1}{2}\left(\frac{q_m}{A_2\rho_2}\right)^2 - \frac{1}{2}\left(\frac{q_m}{A_1\rho_1}\right)^2$$

$$= \frac{p_{e2} - p_{e1}}{\rho_m} + \frac{v_{m2}^2}{2} - \frac{v_{m1}^2}{2}$$

$$y_{sB} = \frac{p_2 - p_1}{\rho_m} - \frac{v_{m1}^2}{2} = \frac{p_2 - p_1}{\rho_m} - \frac{1}{2}\left(\frac{q_m}{A_1\rho_1}\right)^2$$

$$= \frac{p_{e2} - p_{e1}}{\rho_m} - \frac{v_{m1}^2}{2}$$

The fan air power P_{uB} and the fan static power P_{usB} are given by the following expressions:

$$P_{uB} = q_m y_B$$

$$P_{usB} = q_m y_{sB}$$

33.3.3.4.2 Calculation of fan air power and compressibility coefficient

In accordance with 14.8.2:

$$P_{uB} = q_{Vsg1}\, p_{FB}\, k_p$$

$$P_{usB} = q_{Vsg1}\, p_{sFB}\, k_{ps}$$

The compressibility coefficients k_p and k_{ps} may be determined by two equivalent methods (see 14.8.2.1 and 14.8.2.2).

a) First method:

$$k_{ps} \text{ or } k_p = \frac{Z_k \log_{10} r}{\log_{10}\left[1 + Z_k(r-1)\right]}$$

where

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

for k_p or

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

for fan static air power and

$$Z_k = \frac{\kappa - 1}{\kappa} \frac{P_r}{q_{Vsg1}\, p_{FB}}$$

for fan air power or

$$Z_k = \frac{\kappa - 1}{\kappa} \frac{P_r}{q_{Vsg1}\, p_{sFB}}$$

for fan static air power.

b) Second method:

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

where

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

or

$$x = \frac{p_{sFB}}{p_{sg1}}$$

and

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

33.3.3.5 Calculation of efficiencies

In accordance with 14.8.1, the efficiencies are given by the following expressions:

— Fan efficiency:

$$\eta_{rB} = \frac{P_{uB}}{P_r}$$

— Fan static efficiency:

$$\eta_{srB} = \frac{P_{usB}}{P_r}$$

— Fan shaft efficiency:

$$\eta_{aB} = \frac{P_{uB}}{P_a}$$

— Fan static shaft efficiency:

$$\eta_{saB} = \frac{P_{usB}}{P_a}$$

33.3.4 Simplified procedures

33.3.4.1 Reference Mach number Ma_{2ref} less than 0,15 and pressure ratio more than 1,02

At a section of the test duct the stagnation and static temperatures are considered as equal:

$$F_{M1} = F_{M2} = 1$$

$$\Theta_x = \Theta_{sgx}$$

33.3.4.1.1 Calculation of mass flowrate

The mass flowrate is determined in accordance with the methods described in 33.3.3.1.

$$\Theta_u = \Theta_{sgu} = t_6 + 273{,}15 = \Theta_6 = \Theta_{sg6}$$

$$p_u = p_{e6} + p_a$$

$$\rho_6 = \frac{p_6}{R_w \Theta_6} = \frac{p_6}{R_w \Theta_{sg6}}$$

C and α shall be corrected for the Reynolds number effect.

33.3.4.1.2 Calculation of fan pressure

33.3.4.1.2.1 Fan outlet pressure

In accordance with 14.9.1, 14.9.1.3 and 33.3.3.2

$$p_4 = p_{e4} + p_a$$

$$p_{sg2} = p_4 + \rho_4 \frac{v_{m2.4}^2}{2} = p_4 + \frac{1}{2\rho_4}\left(\frac{q_m}{A_{2.4}}\right)^2$$

or

$$p_{esg2} = p_{e4} + \rho_4 \frac{v_{m2.4}^2}{2}$$

where

$$\rho_4 = \frac{p_4}{R_w \Theta_4} = \frac{p_4}{R_w \Theta_{sg4}}$$

p_2 may be determined by the following method (see 14.9.1.4):

$$(p_2)_1 = p_{sg2} - \rho_4 \frac{v_{m2.4}^2}{2} \left(\frac{A_{2.4}}{A_2} \right)^2$$

$$(\rho_2)_1 = \frac{(p_2)_1}{R_w \Theta_{sg2}}$$

$$p_2 = p_{sg2} - \frac{1}{2} \frac{q_m^2}{A_2^2 \rho_2}$$

$$p_{e2} = p_{esg2} - \frac{1}{2} \frac{q_m^2}{A_2^2 \rho_2}$$

Two or three iterations are sufficient; p_2 may be also determined by the following expression:

$$p_2 = \frac{1}{2} \left[p_{sg2} + \sqrt{p_{sg2}^2 - 2 \left(\frac{q_m}{A_2} \right)^2 R_w \Theta_{sg2}} \right]$$

$$\rho_2 = \frac{p_2}{R_w \Theta_{sg2}}$$

33.3.4.1.2.2 Fan inlet pressure

At the fan inlet

$$p_{sg1} = p_a$$

$$p_{esg1} = 0$$

The pressure p_1 and the density ρ_1 may be determined by one of the two methods developed for the calculation of p_2.

33.3.4.1.2.3 Fan pressure

The fan pressure p_{FB} and the fan static pressure p_{sFB} are given by the following expressions:

$$p_{FB} = p_{sg2} - p_{sg1} = p_{sg2} - p_a = p_{esg2}$$

$$p_{sFB} = p_2 - p_{sg1} = p_2 - p_a = p_{e2}$$

33.3.4.1.3 Calculation of volume flowrate

The volume flowrate is given by the following expression:

$$q_{vsg1} = \frac{q_m}{\rho_{sg1}} = \frac{q_m}{\left(\dfrac{p_a}{R_w \Theta_{sg1}} \right)}$$

33.3.4.1.4 Calculation of fan air power

Fan air powers are determined in accordance with 14.8.1, 14.8.2, 14.8.3 or 14.9.1.6 and 33.3.3.4.

33.3.4.1.5 Calculation of fan efficiencies

Fan efficiencies are calculated in accordance with 14.8.1, 14.8.2, 14.8.3 and 33.3.3.5.

33.3.4.2 Reference Mach number Ma_{2ref} less than 0,15 and pressure ratio less than 1,02 (see 14.9.2)

The air flow through the fan and the test airway may be considered incompressible.

$$\Theta_1 = \Theta_{sg1} = \Theta_2 = \Theta_{sg2} = \Theta_4 = \Theta_{sg4} = \Theta_6 = \Theta_{sg6} = \Theta_a = t_a + 273{,}15$$

$$\rho_1 = \rho_2 = \rho_4 = \rho_6 = \rho_a = \frac{p_a}{R_w \Theta_a}$$

$$F_{M1} = F_{M2} = 1$$

$$k_p = 1$$

33.3.4.2.1 Calculation of mass flowrate

The mass flowrate is calculated in accordance with 33.3.4.1.1.

33.3.4.2.2 Calculation of fan pressure

33.3.4.2.2.1 Fan outlet pressure

In accordance with the assumptions above,

$$\rho_1 = \rho_{sg1} = \rho_2 = \rho_{sg2} = \rho_a$$

$$p_{sg2} = p_4 + \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2$$

$$p_{esg2} = p_{e4} + \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2$$

$$p_2 = p_{sg2} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2 = p_4$$

$$p_{e2} = p_{esg2} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2 = p_{e4}$$

33.3.4.2.2.2 Fan inlet pressure

$$p_{sg1} = p_a$$

$$p_{esg1} = 0$$

$$p_1 = p_{sg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2$$

$$p_{e1} = -\frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2$$

33.3.4.2.2.3 Fan pressure

The fan pressure p_{FB} and the fan static pressure p_{sFB} may be determined by the following expressions:

$$p_{FB} = p_{sg2} - p_{sg1} = p_{sg2} - p_a = p_{esg2}$$

$$p_{sFB} = p_2 - p_{sg1} = p_2 - p_a = p_{e2}$$

33.3.4.2.3 Calculation of volume flowrate

The volume flowrate is given by the following expression as in 33.2.4.1.3:

$$q_{Vsg1} = \frac{q_m}{\rho_{sg1}} = \frac{q_m}{\left(\dfrac{p_a}{R_w \Theta_{sg1}}\right)}$$

33.3.4.2.4 Calculation of fan air power

In accordance with 14.9.2.6:

$$P_{uB} = q_{Vsg1}\, p_{FB}$$

$$P_{usB} = q_{Vsg1}\, p_{sFB}$$

33.3.4.2.5 Calculation of fan efficiencies

Fan efficiencies are determined from P_{uB} or P_{usB} as in 33.2.4.1.5.

33.3.5 Fan performance under test conditions

Under test conditions, the fan performances are the following:

— inlet volume flow, q_{Vsg1};

— fan pressure, p_{FB};

— fan static pressure, p_{sFB};

— fan efficiency, η_{rB} or η_{srB}.

34 Standard test methods with inlet-side test ducts or chambers — Type C installations

34.1 Types of fan setup

Two general types of setup of fan are shown:

a) inlet duct, where the inlet pressure is measured in the test duct;

b) inlet chamber with, at the end of the chamber, inlet duct simulation. The inlet pressure is measured in the chamber.

Nine methods of controlling and measuring the flowrate in the test duct are shown. The method of flowrate measurement is specified in each case, together with the clauses and figures detailing the flowrate measurement procedure.

A common procedure, comprising measurement to be taken and quantities to be calculated, allowing the determination of fan performance in type C installations with nine methods for determining flowrate, is given in 34.2.3.1.1 to 34.2.3.1.3 and 34.3.3.1.1 to 34.3.3.1.4. This procedure is generally valid for all fans conforming to this International Standard.

However, two simplified procedures may be followed when

— the reference Mach number is less than 0,15 and the pressure ratio is more than 1,02;

— the reference Mach number is less than 0,15 and the pressure ratio is less than 1,02.

In these circumstances, the procedures given in 34.2.4 and 34.3.4 may be followed.

34.2 Inlet-side test ducts

34.2.1 Mass flowrate determination

The mass flowrate is determined by

— inlet ISO Venturi nozzle, see figure 74 a);

— quadrant inlet nozzle, see figure 74 a);

— conical inlet, see figure 74 a);

— inlet orifice with corner taps, see figure 74 b);

— inlet orifice with corner taps, see figure 74 c);

— inlet orifice with wall taps, see figure 74 d);

— in-duct orifice with D and $D/2$, see figure 74 e);

— in-duct orifice with corner taps, see figure 74 e);

— Pitot-static tube traverse, see figure 74 f);

— Pitot-static tube traverse, see figure 74 g).

34.2.2 Measurements to be taken during tests (see clause 20)

Measure

— rotational speed, N, or rotational frequency, n;

— power input, P_a, P_o or P_e, and estimate impeller power (see 10.3) and power input P_{ex} of an auxiliary fan;

— flowmeter differential pressure, Δp;

— pressure, p_{e7} or p_{e3}, upstream of the flowmeter;

— inlet static pressure, p_{e3}.

In the test enclosure, measure

— atmospheric pressure, p_a, at the mean fan altitude;

— ambient temperature, t_a, near fan duct inlet;

— dry- and wet-bulb temperatures, t_d and t_w.

Determine the ambient air density, ρ_a, and the gas constant of humid air, R_w (see clause 12).

34.2.3 General procedure for compressible fluid flow

This procedure should be applied when both the reference Mach number Ma_{2ref} (see 14.4.2) is more than 0,15 and the pressure ratio is more than 1,02.

34.2.3.1 Calculation of mass flowrate

34.2.3.1.1 The mass flowrate is determined using

— inlet ISO Venturi nozzle, see clause 22 and figure 74 a);

— quadrant inlet nozzle, see clause 24 and figure 74 a);

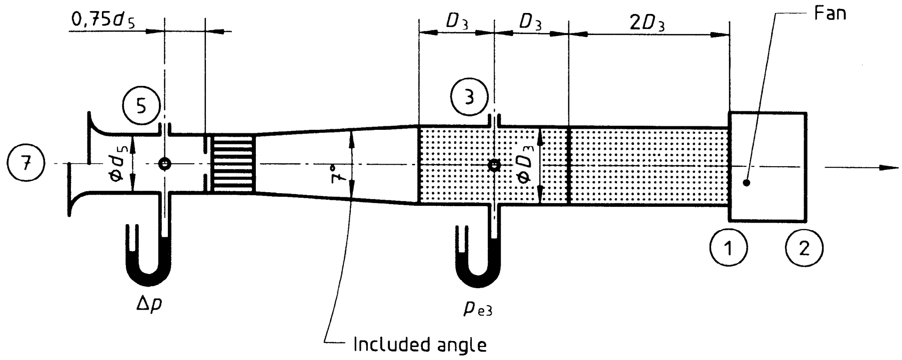
— conical inlet, see clause 25 and figure 74 a);

— inlet orifice with corner taps, see 26.10 and figure 74 b) and c);

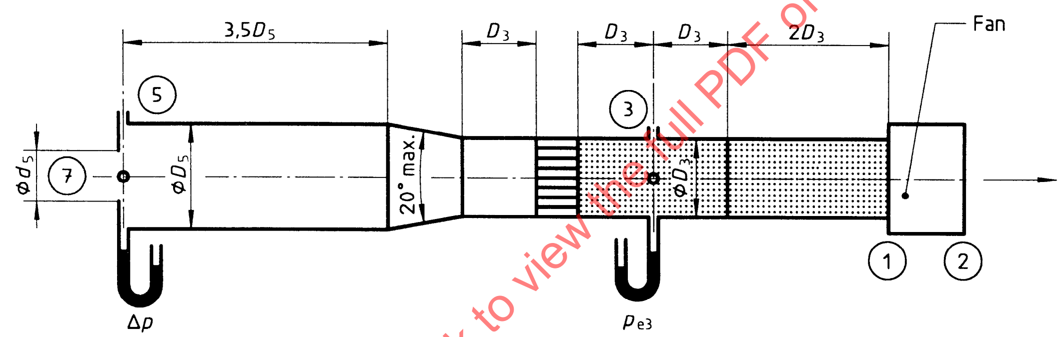
— inlet orifice with wall tappings, see 26.11 and figure 74 c) and d);

The flowrate is controlled by an adjustable screen loading [see figure 74 a) and 25.2], by the orifice plate [see figure 74 b) and d)], or by an auxiliary fan with a control device [figure 74 c)].

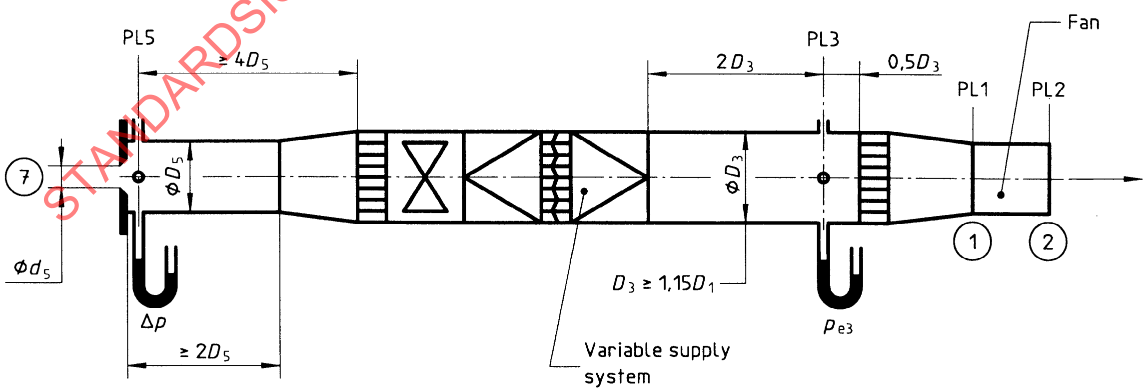
Dimensions in millimetres



a) Flowrate determination using Venturi nozzle, quadrant inlet nozzle, conical inlet



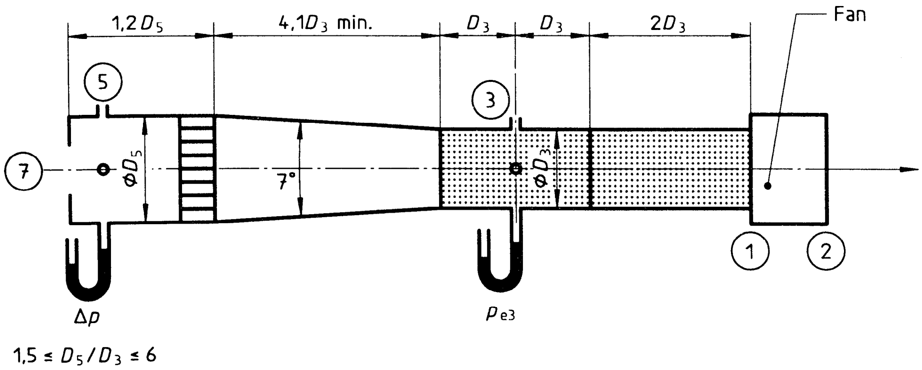
b) Flowrate determination using inlet orifice with corner taps



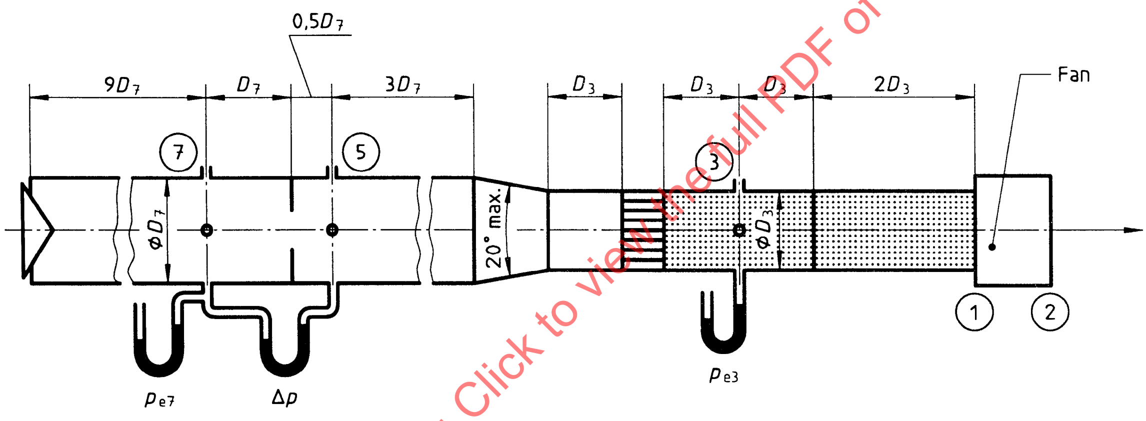
c) Flowrate determination using inlet orifice with wall tappings

Figure 74 — Type C test installations (inlet-side test duct)

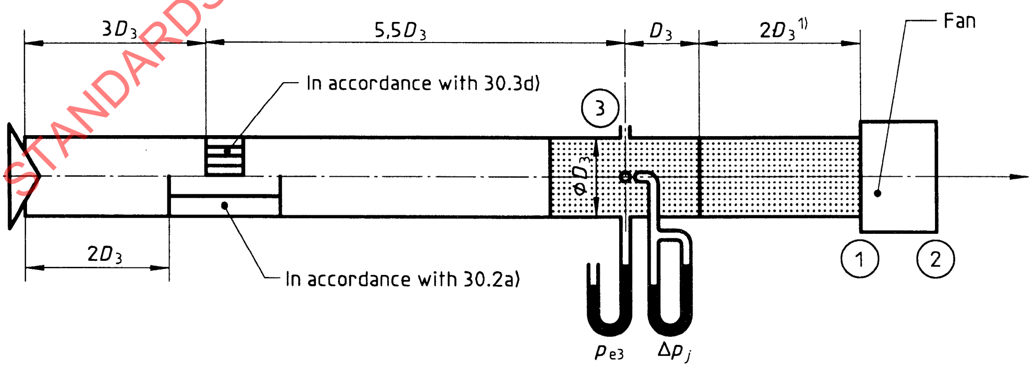
Dimensions in millimetres



d) Flowrate determination using inlet orifice with wall tappings



e) Flowrate determination using in-duct orifice with taps at *D* and 0,5*D* or in-duct orifice with corner taps

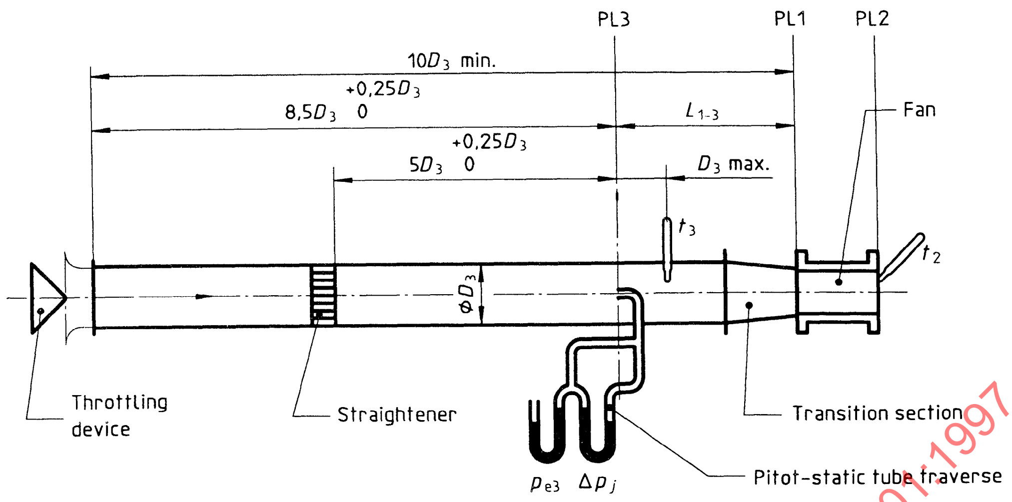


1) This cylindrical airway section of length 2D_3 may be replaced by a transition section in accordance with clause 30 when required to accommodate a change in area and/or shape.

f) Flowrate determination using Pitot-static tube traverse

Figure 74 — Type C test installations (inlet-side test duct) *(continued)*

Dimensions in millimetres



g) Flowrate determination using Pitot-static tube traverse

Figure 74 — Type C test installations (inlet-side test duct) *(concluded)*

Assuming that

$$p_{e7} = 0$$

$$p_7 = p_a$$

$$\Theta_7 = \Theta_{sg7} = t_a + 273,15$$

$$\rho_7 = \frac{p_7}{R_w \Theta_7}$$

The mass flowrate is given by the following expression:

$$q_m = \alpha \varepsilon \pi \frac{d_5^2}{4} \sqrt{2 \rho_7 \Delta p}$$

where

α is the flowrate coefficient function of the Reynolds number Re_{d5} estimated by the following expression, in which the value of α is a mean value:

$$Re_{d5} = \frac{\alpha \varepsilon d_5 \sqrt{2 \rho_7 \Delta p}}{17,1 + 0,048 t_a} \times 10^6$$

ε is the expansibility coefficient.

α, ε or the compound coefficient, $\alpha\varepsilon$ are determined in accordance with 22.3.2, 22.3.3, 24.4, 25.4, 26.10 a) and b), 26.11 and figures 18, 22, 27, 28 and 26 after estimation of Re_{d5}.

34.2.3.1.2 The mass flowrate is determined using an in-duct orifice with taps at D and $D/2$ or corner taps [see 26.7, 26.8 and figure 74 e)]

Assuming that

$$p_7 = p_{e7} + p_a$$

$$\Theta_{sg7} = \Theta_a = t_a + 273,15$$

$$\Theta_7 = \Theta_{sg7} - \frac{v_{m7}^2}{2c_p} = \Theta_{sg7} - \frac{1}{2c_p} \left(\frac{q_m}{A_7 \rho_7} \right)^2$$

$$\rho_7 = \frac{p_7}{R_w \Theta_7}$$

The mass flowrate is given by the following expression:

$$q_m = \alpha \varepsilon \pi \frac{d_5^2}{4} \sqrt{2 \rho_7 \Delta p}$$

where

ε is the expansibility coefficient, a function of the ratio $\Delta p / p_7$, and the Reynolds number Re_{D7};

$$Re_{D7} = \frac{\alpha \varepsilon \beta d_5 \sqrt{2 \rho_7 \Delta p}}{17{,}1 + 0{,}048 t_7} \times 10^6$$

α is the flowrate coefficient varying with:

$$\beta = \frac{d_5}{D_7}$$

Θ_7, ρ_7 and q_m being unknown, q_m is determined by an iterative procedure taking $\Theta_7 = \Theta_{sg7}$ for the first approximation.

Two or three iterations are sufficient for a calculation accuracy of 10^{-3}.

34.2.3.1.3 The mass flowrate is determined using a Pitot-static tube traverse [see clause 27 and figure 74 f) and g)]

Assuming that

$$p_3 = p_{e3} + p_a$$

when the pressure p_{e3} is measured by the Pitot-static tube:

$$p_{e3} = \frac{1}{n} \sum_{j=1}^{n} p_{e3j}$$

$$\Theta_{sg3} = t_a + 273{,}15$$

$$\Delta p_m = \left(\frac{1}{n} \sum_{j=1}^{n} \Delta p_j^{0,5} \right)^2$$

$$\Theta_3 = \Theta_{sg3} \left(\frac{p_3}{p_3 + \Delta p_m} \right)^{\frac{\kappa - 1}{\kappa}}$$

$$\rho_3 = \frac{p_3}{R_w \Theta_3}$$

The location of measuring points j is given in 27.4 and figure 30.

The mass flowrate q_m is given by the following expression (see 27.5):

$$q_m = \alpha \varepsilon \pi \frac{D_3^2}{4} \sqrt{2 \rho_3 \Delta p_m}$$

where

ε is the expansibility factor (see 27.5);

α is the correction factor or flow coefficient (see 27.6) depending upon the Reynolds number Re_{D3}:

$$Re_{D3} = \frac{4\, q_m}{\pi D_3 (17{,}1 + 0{,}048\, t_3)} \times 10^6$$

α varies between

$0{,}990 + 0{,}002$ for $Re_{D3} = 3 \times 10^6$

and

$0{,}990 - 0{,}004$ for $Re_{D3} = 3 \times 10^4$

A first approximation of q_m is obtained with $\alpha = 0{,}990$ and corrected for the value of Re_{D3} (see 27.6).

34.2.3.2 Determination of fan pressure

34.2.3.2.1 Fan inlet pressure

The two following cases should be considered:

— there is no auxiliary fan between planes 5 and 3;

— there is an auxiliary fan between planes 5 and 3.

a) There is no auxiliary fan between planes 5 and 3.

$$p_3 = p_{e3} + p_a$$

$$\Theta_{sg3} = \Theta_{sg5} = \Theta_{sg7} = \Theta_a = \Theta_{sg1} = t_a + 273{,}15$$

The Mach number Ma_3 and the ratio

$$\frac{\Theta_3}{\Theta_{sg3}}$$

are calculated in accordance with 14.4.3.1.

$$\Theta_3 = \Theta_{sg3} \frac{\Theta_3}{\Theta_{sg3}}$$

$$\rho_3 = \frac{p_3}{R_w \Theta_3}$$

The inlet stagnation pressure p_{sg1} is given by the following expression (see 14.6.1):

$$p_{sg1} = p_3 + \frac{1}{2} \rho_3\, v_{m3}^2 F_{M3} \left[1 + (\zeta_{3-1})_3\right]$$

$$= p_3 + \frac{1}{2\rho_3} \left(\frac{q_m}{A_3}\right)^2 F_{M3} \left[1 + (\zeta_{3-1})_3\right]$$

where

$(\zeta_{3-1})_3 \leqslant 0$ is the conventional coefficient calculated in accordance with 30.6.4, 30.6.5 and 30.6.6;

F_{M3} is the Mach factor determined in accordance with 14.5.1.

p_{e3} is always negative.

$$p_{esg1} = p_{e3} + \frac{1}{2\rho_3} \left(\frac{q_m}{A_3}\right)^2 F_{M3} \left[1 + (\zeta_{3-1})_3\right]$$

b) There is an auxiliary fan between planes 5 and 3 [see figure 74 c)].

In this case, $(\zeta_{3-1})_3 \leqslant 0$ is determined by test and is not defined by this International Standard.

If the impeller power of the auxiliary fan P_{rx} or the motor input power of the auxiliary fan P_{ex} (in the case of an immersed motor) may be determined:

$$\Theta_{sg3} = \Theta_{sg7} + \frac{P_{rx} \text{ or } P_{ex}}{q_m c_p} = \Theta_a + \frac{P_{rx} \text{ or } P_{ex}}{q_m c_p} = \Theta_{sg1}$$

In other cases, the temperature t_3 should be measured and the quantity $t_3 + 273{,}15$ assumed to be a stagnation temperature.

The static temperature Θ_3 is determined in accordance with 14.4.3.1 and the stagnation pressure p_{sg1} calculated in the same way as in the first case.

The pressure p_1 is determined after the calculation of the Mach number Ma_1 and of the ratio

$$\frac{\Theta_1}{\Theta_{sg1}}$$

in accordance with 14.4.3.2.

The density ρ_1 is calculated in accordance with 14.4.4 and the static pressure p_1 is given by the following expression (see 14.5.2):

$$p_1 = p_{sg1} - \frac{1}{2}\rho_1 v_{m1}^2 F_{M1} = p_{sg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2 F_{M1}$$

or

$$p_{e1} = p_{esg1} - \frac{1}{2}\rho_1 v_{m1}^2 F_{M1} = p_{esg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2 F_{M1}$$

34.2.3.2.2 Fan outlet pressure

The static pressure at fan outlet p_2 is equal to the atmospheric pressure p_a.

$$p_2 = p_a$$

The stagnation temperature at fan outlet Θ_{sg2} is given by the following relation:

$$\Theta_{sg2} = \Theta_{sg3} + \frac{P_r \text{ or } P_e}{q_m c_p}$$

The Mach number Ma_2 and the ratio

$$\frac{\Theta_2}{\Theta_{sg2}}$$

are determined in accordance with 14.4.3.1.

$$\Theta_2 = \Theta_{sg2}\frac{\Theta_2}{\Theta_{sg2}}$$

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

and p_{sg2} is given by the following expression (see 14.5.1):

$$p_{sg2} = p_2 + \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 F_{M2} = p_a + \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 F_{M2}$$

$$p_{esg2} = \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 F_{M2}$$

34.2.3.2.3 Fan pressure

The fan pressure p_{FC} is given by the following expression:

$$p_{FC} = p_{sg2} - p_{sg1} = p_a + \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 F_{M2} - \left\{p_3 + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 F_{M3}\left[1 + \left(\zeta_{3-1}\right)_3\right]\right\}$$

$$= \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 F_{M2} - \left\{p_{e3} + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 F_{M3}\left[1 + \left(\zeta_{3-1}\right)_3\right]\right\}$$

The fan static pressure p_{sFC} is given by the following expression:

$$p_{sFC} = p_2 - p_{sg1} = p_a - \left\{p_3 + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 F_{M3}\left[1 + \left(\zeta_{3-1}\right)_3\right]\right\}$$

$$= -\left\{p_{e3} + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 F_{M3}\left[1 + \left(\zeta_{3-1}\right)_3\right]\right\}$$

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

$$k_\rho = \frac{\rho_1}{\rho_m}$$

34.2.3.3 Determination of volume flowrate

The volume flowrate at stagnation inlet conditions is given by the following expression:

$$q_{Vsg1} = \frac{q_m}{\rho_{sg1}} = \frac{q_m}{\left(\dfrac{p_{sg1}}{R_w \Theta_{sg1}}\right)}$$

34.2.3.4 Determination of fan air power

34.2.3.4.1 Fan work per unit mass and fan air power

According to 14.8.1, the fan work per unit mass y_C and the fan static work per unit mass y_{sC} are given by the following expressions:

$$y_C = \frac{p_2 - p_1}{\rho_m} + \frac{v_{m2}^2}{2} - \frac{v_{m1}^2}{2}$$

$$= \frac{p_2 - p_1}{\rho_m} + \frac{1}{2}\left(\frac{q_m}{A_2\rho_2}\right)^2 - \frac{1}{2}\left(\frac{q_m}{A_1\rho_1}\right)^2$$

$$= \frac{p_{e2} - p_{e1}}{\rho_m} + \frac{1}{2}\left(\frac{q_m}{A_2\rho_2}\right)^2 - \frac{1}{2}\left(\frac{q_m}{A_1\rho_1}\right)^2$$

$$y_{sC} = \frac{p_2 - p_1}{\rho_m} - \frac{v_{m1}^2}{2} = \frac{p_2 - p_1}{\rho_m} - \frac{1}{2}\left(\frac{q_m}{A_1\rho_1}\right)^2$$

$$= \frac{p_{e2} - p_{e1}}{\rho_m} - \frac{1}{2}\left(\frac{q_m}{A_1\rho_1}\right)^2$$

The fan power P_{uC} and the fan static power P_{usC} are given by the following expressions:

$$P_{uC} = y_C \, q_m$$

$$P_{usC} = y_{sC} \, q_m$$

34.2.3.4.2 Calculation of fan air power and compressibility coefficients

In accordance with 14.8.2

$$P_{uC} = q_{Vsg1}\, p_{FC}\, k_p$$

$$P_{usC} = q_{Vsg1}\, p_{sFC}\, k_{ps}$$

The compressibility coefficients k_p and k_{ps} may be determined by two equivalent methods (see 14.8.2.1 and 14.8.2.2).

a) First method:

$$k_{ps} \text{ or } k_p = \frac{Z_k \log_{10} r}{\log_{10}\left[1 + Z_k (r - 1)\right]}$$

where

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

for k_p or

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

for k_{ps} and

$$Z_k = \frac{\kappa - 1}{\kappa} \frac{P_r}{q_{Vsg1}\, p_{FC}}$$

for k_p or

$$Z_k = \frac{\kappa - 1}{\kappa} \frac{P_r}{q_{Vsg1}\, p_{sFC}}$$

for k_{ps}.

b) Second method:

$$k_{ps} \text{ or } k_p = \frac{\ln (1 + x)}{x} \frac{Z_p}{\ln (1 + Z_p)}$$

where

$$x = r - 1 = \frac{p_{FC}}{p_{sg1}} \text{ for } k_p$$

or

$$x = \frac{p_{sFC}}{p_{sg1}} \text{ for } k_{ps}$$

and

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

34.2.3.5 Calculation of efficiencies

In accordance with 14.8.1 and 14.8.2, the efficiencies are calculated using the following expressions:

— Fan efficiency:

$$\eta_{rC} = \frac{P_{uC}}{P_r}$$

— Fan static efficiency:

$$\eta_{srC} = \frac{P_{usC}}{P_r}$$

— Fan shaft efficiency:

$$\eta_{aC} = \frac{P_{uC}}{P_a}$$

— Fan static shaft efficiency:

$$\eta_{saC} = \frac{P_{usC}}{P_a}$$

34.2.4 Simplified procedures

34.2.4.1 Reference Mach number Ma_{2ref} less than 0,15 and pressure ratio more than 1,02

At a section of the inlet test duct, the stagnation and static temperatures may be considered as equal:

$$\Theta_x = \Theta_{sgx} = t_x + 273,15$$

The Mach factors F_{M1} and F_{M2} are equal to 1.

34.2.4.1.1 Calculation of mass flowrate

For corresponding setups, the procedures described in 34.2.3.1 are followed.

For the procedure described in 34.2.3.1.2, there is no need for an iterative procedure to determine Θ_7 because:

$$\Theta_7 = \Theta_{sg7} = t_7 + 273,15 = t_a + 273,15$$

t_7 being measured in the test duct:

$$\rho_7 = \frac{p_7}{R_w \Theta_7} = \frac{p_7}{R_w \Theta_{sg7}}$$

In the same way, for 34.2.3.1.3:

$$\Theta_3 = \Theta_{sg3} = t_3 + 273,15\ t_a + 273,15$$

However, in any case, the correction of α as a function of the Reynolds number should be applied.

34.2.4.1.2 Calculation of fan pressure

34.2.4.1.2.1 Fan inlet pressure

Assuming that

without auxiliary fan:

$$\Theta_7 = \Theta_3 = \Theta_1 = \Theta_{sg1} = \Theta_a = t_a + 273,15$$

with auxiliary fan:

$$\Theta_3 = \Theta_{sg3} = \Theta_1 = \Theta_{sg1} = t_3 + 273,15$$

and $(\zeta_{3-1})_3$ is determined by test.

$$\rho_3 = \frac{p_3}{R_w \Theta_3}$$

In accordance with 14.9.1

$$p_{sg1} = p_3 + \frac{1}{2}\rho_3 v_{m3}^2 \left[1 + \left(\zeta_{3-1}\right)_3\right]$$

$$= p_3 + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 \left[1 + \left(\zeta_{3-1}\right)_3\right]$$

or

$$p_{esg1} = p_{e3} + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 \left[1 + \left(\zeta_{3-1}\right)_3\right]$$

where p_{e3} and $(\zeta_{3-1})_3$ are negative [see 34.2.3.2.1 b)].

The pressure p_1 may be determined by the following procedure:

$$p_1 = p_{sg1} - \rho_1 \frac{v_{m1}^2}{2} = p_{sg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2$$

where

$$\rho_1 = \frac{p_1}{R_w \Theta_1} = \frac{p_1}{R_w \Theta_{sg1}}$$

A first value $(\rho_1)_1$ is obtained with $(\rho_1)_1 = \rho_{sg1}$

$$(\rho_1)_1 = \frac{p_{sg1}}{R_w \Theta_1}$$

$$(p_1)_1 = p_{sg1} - \frac{1}{2(\rho_1)_1}\left(\frac{q_m}{A_1}\right)^2$$

$$p_{e1} = p_1 - p_a$$

Two or three calculation steps are sufficient for a calculation accuracy of 10^{-3} on p_{e1}.

The pressure p_1 may also be determined by the following expression:

$$p_1 = \frac{1}{2}\left[p_{sg1} + \sqrt{p_{sg1}^2 - 2\left(\frac{q_m}{A_1}\right)^2 R_w \Theta_{sg1}}\right]$$

$$p_{e1} = p_1 - p_a$$

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

34.2.4.1.2.2 Fan outlet pressure

At the fan outlet, $p_2 = p_a$ and the stagnation pressure p_{sg2} may be determined by the following expression:

$$p_{sg2} = p_2 + \frac{1}{2}\rho_2 v_{m2}^2 = p_2 + \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2$$

$$= p_a + \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2$$

$$p_{esg2} = \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2$$

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

$$\Theta_2 = \Theta_{sg2} = \Theta_{sg1} + \frac{P_r \text{ or } P_e}{q_m c_p}$$

34.2.4.1.2.3 Fan pressure

The fan pressure p_{FC} and the fan static pressure p_{sFC} are given by:

$$p_{FC} = p_{sg2} - p_{sg1} = p_a + \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 - \left\{p_3 + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 \left[1 + (\zeta_{3-1})_3\right]\right\}$$

$$= \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 - \left\{p_{e3} + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 \left[1 + (\zeta_{3-1})_3\right]\right\}$$

$$p_{sFC} = p_2 - p_{sg1} = p_a - p_{sg1} = - p_{esg1}$$

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

$$k_\rho = \frac{\rho_1}{\rho_m}$$

34.2.4.1.3 Determination of volume flowrate

The volume flowrate at inlet stagnation conditions is given by the following expression:

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

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

34.2.4.1.4 Calculation of fan air power

The fan work per unit mass and the fan air power are determined in accordance with 14.8.1, 14.8.2 and also 34.2.3.4.

34.2.4.1.5 Calculation of efficiencies

Fan efficiencies are determined in accordance with 34.2.3.5.

34.2.4.2 Reference Mach number Ma_{2ref} less than 0,15 and pressure ratio less than 1,02

$$\Theta_1 = \Theta_{sg1} = \Theta_2 = \Theta_{sg2} = \Theta_3 = \Theta_{sg3}$$

The temperature in the test duct may be measured and:

$$F_{M1} = F_{M2} = F_{M3} = 1$$

$$k_p = 1$$

The air flow through the fan and the test airway may be considered as incompressible, except with an auxiliary fan.

34.2.4.2.1 Calculation of mass flowrate

The mass flowrate is determined in accordance with 34.2.4.1.1 with $\rho_u = \rho_a$.

34.2.4.2.2 Determination of fan pressure

34.2.4.2.2.1 Fan inlet pressure

Assuming that

without an auxiliary fan $\Theta_1 = \Theta_{sg1} = \Theta_2 = \Theta_{sg2} = \Theta_3 = \Theta_{sg3} = t_a + 273{,}15$

$\Theta_3 = t_a + 273{,}15$

When there is an auxiliary fan between planes 7 and 3, the temperature t_3 in the test duct may be measured:

$\Theta_1 = \Theta_{sg1} = \Theta_2 = \Theta_{sg2} = \Theta_3 = \Theta_{sg3} = t_3 + 273{,}15$

$p_3 = p_{e3} + p_a$

In accordance with 14.9

$$p_{sg1} = p_3 + \frac{1}{2}\rho_3 v_{m3}^2 \left[1 + (\zeta_{3-1})_3\right]$$

$$= p_3 + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 \left[1 + (\zeta_{3-1})_3\right]$$

$$p_{esg1} = p_{e3} + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 \left[1 + (\zeta_{3-1})_3\right]$$

where

p_{e3} and $(\zeta_{3-1})_3 \leqslant 0$ [see 34.2.3.2.1 b)]

$$\rho_3 = \frac{p_3}{R_w \Theta_3} = \frac{p_3}{R_w \Theta_{sg3}}$$

The pressure p_1 is given by the following expression:

$$p_1 = p_{sg1} - \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 \left(\frac{A_3}{A_1}\right)^2 = p_{sg1} - \frac{1}{2\rho_3}\left(\frac{q_m}{A_1}\right)^2$$

$$p_{e1} = p_{esg1} - \frac{1}{2\rho_3}\left(\frac{q_m}{A_1}\right)^2$$

34.2.4.2.2.2 Fan outlet pressure

At the fan outlet

$p_2 = p_a$

$p_{e2} = 0$

and the stagnation pressure p_{sg2} is given by

$$p_{sg2} = p_a + \frac{1}{2}\rho_3 v_{m2}^2 = p_a + \frac{1}{2\rho_3}\left(\frac{q_m}{A_2}\right)^2$$

$$p_{esg2} = \frac{1}{2\rho_3}\left(\frac{q_m}{A_2}\right)^2$$

34.2.4.2.2.3 Fan pressure

The fan pressure p_{FC} and the fan static pressure p_{sFC} are given by the following expressions:

$$p_{FC} = p_{sg2} - p_{sg1} = p_a + \frac{1}{2\rho_3}\left(\frac{q_m}{A_2}\right)^2 - \left\{p_3 + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2\left[1+\left(\zeta_{3-1}\right)_3\right]\right\}$$

$$= \frac{1}{2\rho_3}\left(\frac{q_m}{A_2}\right)^2 - \left\{p_{e3} + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2\left[1+\left(\zeta_{3-1}\right)_3\right]\right\}$$

$$p_{sFC} = p_2 - p_{sg1} = p_a - p_{sg1} = -\,p_{esg1}$$

$$= -\left\{p_{e3} + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2\left[1+\left(\zeta_{3-1}\right)_3\right]\right\}$$

34.2.4.2.3 Determination of volume flowrate

The volume flowrate in the inlet stagnation conditions is given by

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

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

34.2.4.2.4 Calculation of fan air power

In accordance with 14.9.2.5

$$P_{uC} = q_{Vsg1}\, p_{FC}$$

$$P_{usC} = q_{Vsg1}\, p_{sFC}$$

34.2.4.2.5 Calculation of fan efficiencies

Efficiencies are determined in accordance with 14.8.1 and 34.2.3.5.

34.2.5 Fan performances under test conditions

Under test conditions, the fan performances are the following:

— fan pressure, p_{FC}

— fan static pressure, p_{sFC}

— inlet volume flowrate, q_{Vsg1}

— fan efficiency, η_{rC}

— fan static efficiency, η_{srC}

34.3 Inlet-side test chambers

34.3.1 Mass flowrate determination

The mass flowrate is determined by

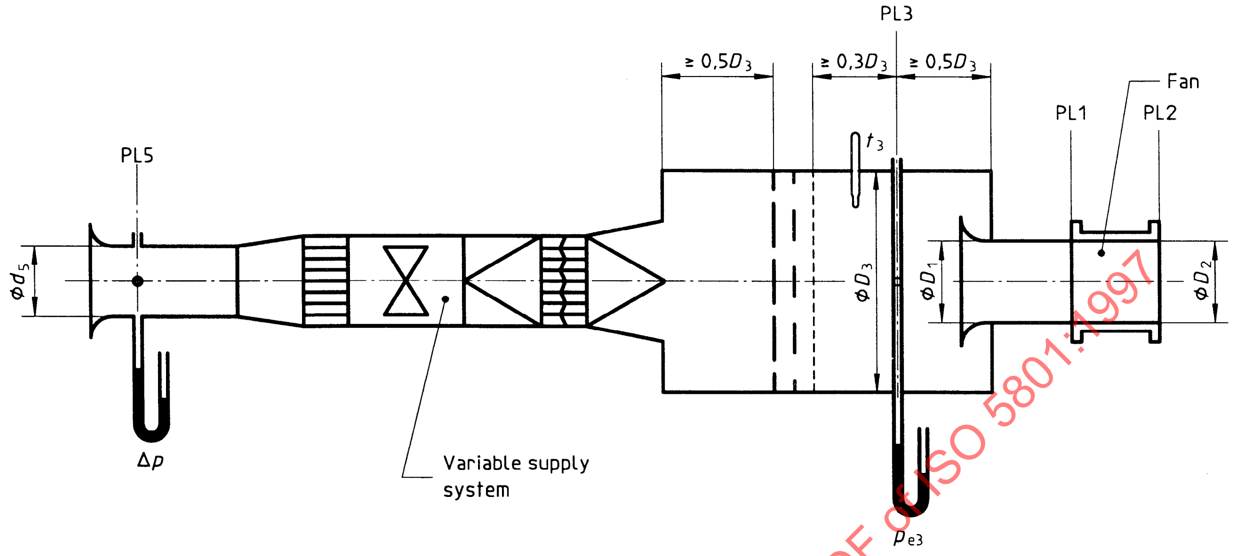
— quadrant inlet nozzle, see clause 24 and figure 75 a);

— Pitot-static tube traverse, see clause 27 and figure 75 b);

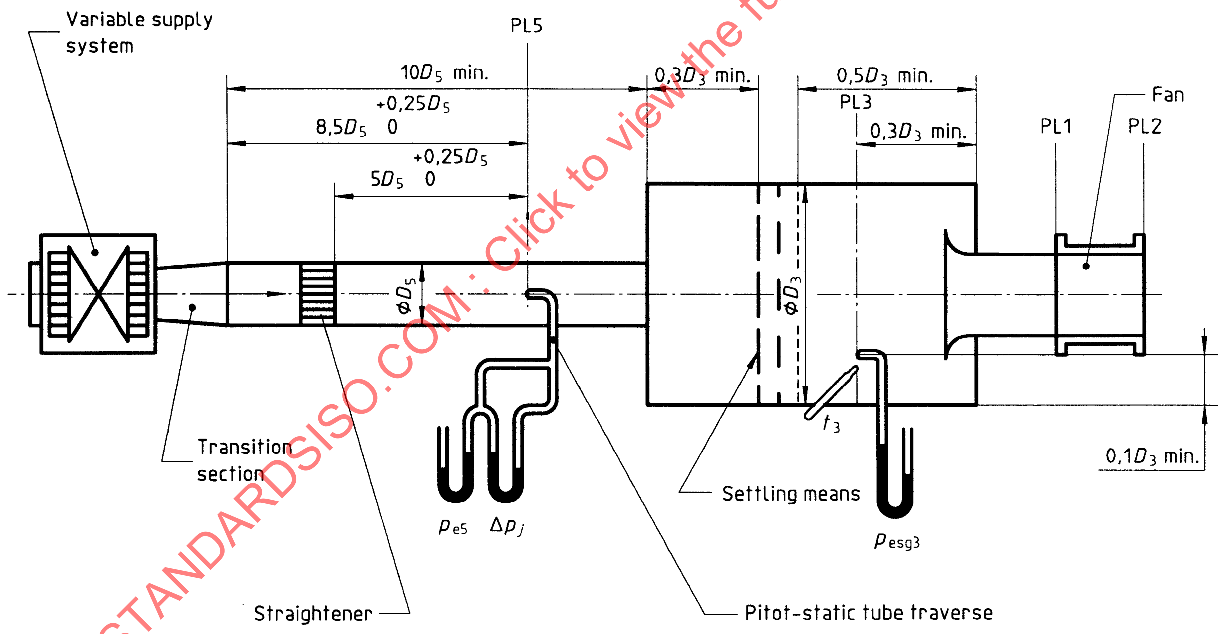
— in-duct Venturi nozzle upstream of the chamber, see clause 23 and figure 75 c);

— multiple nozzles in chamber, see clause 23 and figure 75 d).

Dimensions in millimetres



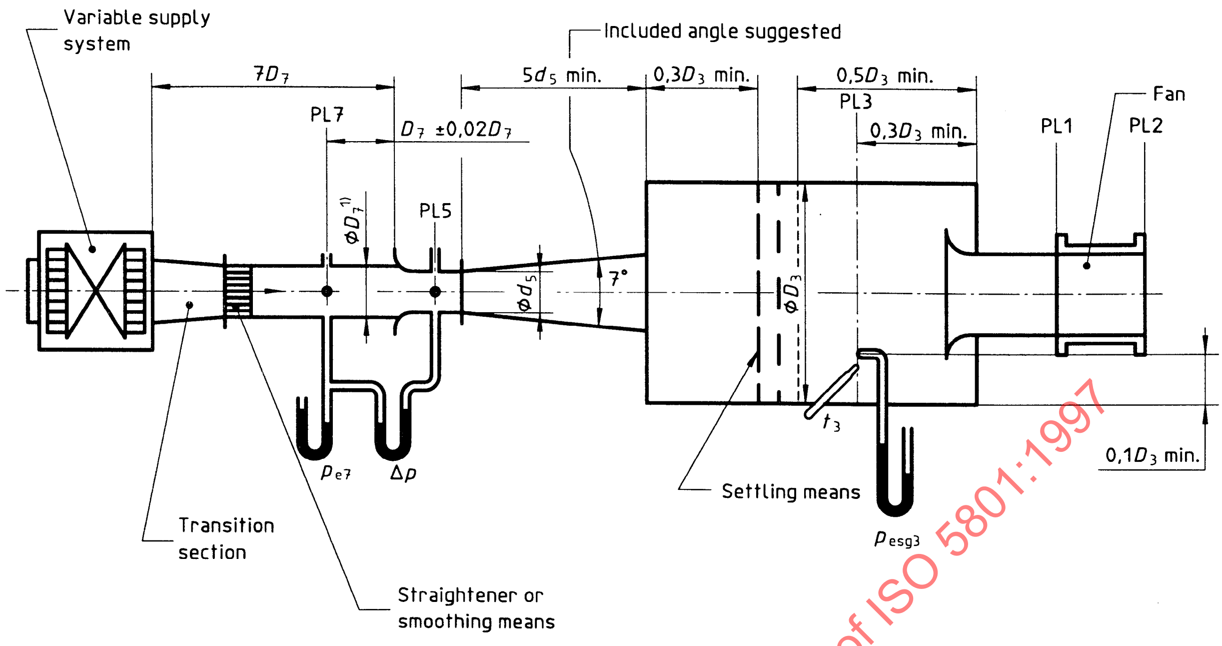
a) Flowrate determination using quadrant inlet nozzle



b) Flowrate determination using Pitot-static tube traverse

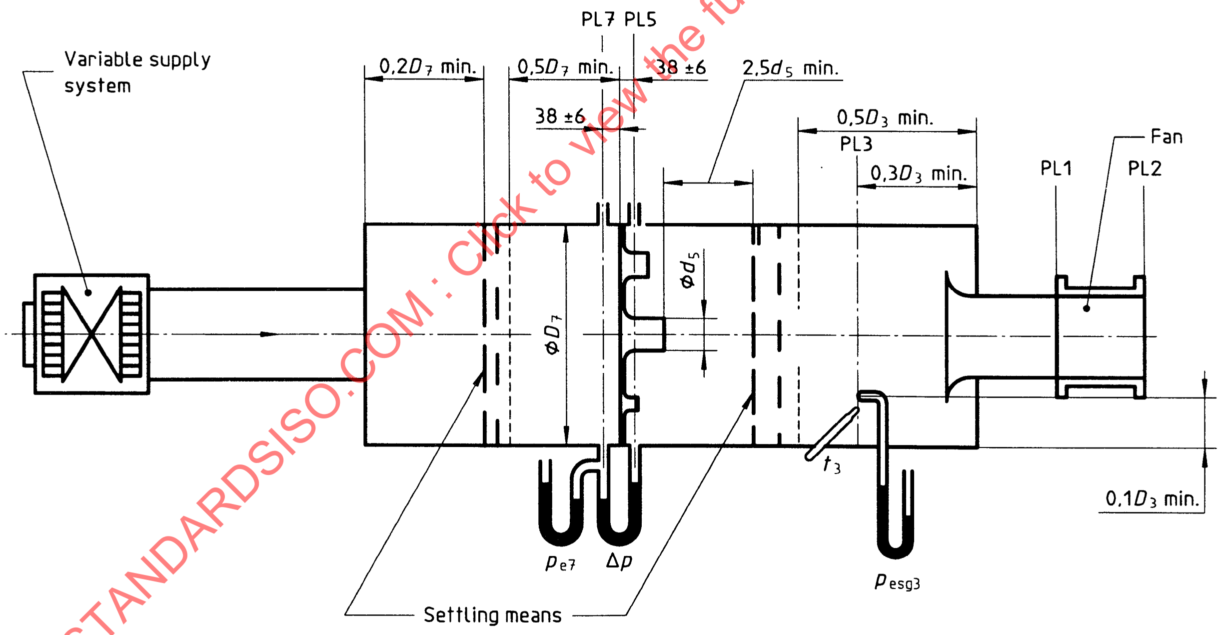
Figure 75 — Type C test installations (inlet-side test chamber)

Dimensions in millimetres



1) $D_7 = 1,9d_5$ min.

c) Flowrate determination using in-duct Venturi nozzle



d) Flowrate determination using multiple nozzles in chamber

Figure 75 — Type C test installations (inlet-side test chamber) *(concluded)*

34.3.2 Measurements to be taken during tests (see clause 20)

Measure

- rotational speed, N, or rotational frequency, n;
- power input, P_a, P_o or P_e, and estimate impeller power (see 10.4) and the power input P_{ex} of an auxiliary fan;
- flowmeter differential pressure, Δp;
- pressure, p_{e7} or p_{e5}, upstream of the flowmeter;

— chamber stagnation or static pressure p_{e3} or p_{esg3};

— Chamber temperature, t_3.

In the test enclosure, measure

— atmospheric pressure, p_a, at the mean fan altitude;

— ambient temperature, t_a, near the fan inlet;

— dry- and wet-bulb temperatures, t_d and t_w.

Determine the ambient air density, ρ_a, and the gas constant of humid air, R_w (see clause 12).

34.3.3 General procedure for compressible fluid flow

This procedure should be applied when both the reference Mach number Ma_{2ref} is more than 0,15 and the fan pressure ratio is more than 1,02.

34.3.3.1 Determination of mass flowrate

34.3.3.1.1 The mass flowrate is determined using a quadrant inlet nozzle, [see clause 24 and figure 75 a)]. It is controlled by an adjustable screen loading with an auxiliary fan.

Assuming that

$$p_{e7} = 0$$

$$p_7 = p_a$$

$$\Theta_7 = \Theta_{sg7} = t_a + 273,15$$

$$\rho_7 = \frac{p_7}{R_w \Theta_7}$$

the mass flowrate is given by the following expression:

$$q_m = \alpha \varepsilon \pi \frac{d_5^2}{4} \sqrt{2 \rho_7 \Delta p}$$

where

α is the flow coefficient function of the Reynolds number Re_{d5} estimated by the following expression, in which the value of α is a mean value equal to 1:

$$Re_{d5} = \frac{\alpha \varepsilon d_5 \sqrt{2 \rho_7 \Delta p}}{17,1 + 0,048\, t_7} \times 10^6$$

ε is the expansibility coefficient;

α and ε are determined in accordance with 24.4 from the value of Re_{d5} and of Δp.

34.3.3.1.2 The mass flowrate is determined using a Pitot-static tube traverse [see clause 27 and figure 75 b)].

Assuming that

$$p_{e5} = \frac{1}{n} \sum_{j=1}^{n} p_{e5j}$$

$$p_5 = p_{e5} + p_a$$

$$\Theta_{sg5} = t_3 + 273,15 = \Theta_a + \frac{P_{rx} \text{ or } P_{ex}}{q_m c_p}$$

$$\Delta p_{\mathrm{m}} = \left(\frac{1}{n} \sum_{j=1}^{n} \Delta p_j^{0,5} \right)^2$$

$$\Theta_5 = \Theta_{\mathrm{sg5}} \left(\frac{p_5}{p_5 + \Delta p_{\mathrm{m}}} \right)^{\frac{\kappa - 1}{\kappa}}$$

$$\rho_5 = \frac{p_5}{R_{\mathrm{w}} \Theta_5}$$

The location of the measuring points j is given in 27.4 and figure 30.

The mass flowrate is given by the following expression (see 27.5):

$$q_m = \alpha \varepsilon \pi \frac{D_5^2}{4} \sqrt{2 \rho_5 \Delta p_{\mathrm{m}}}$$

where

ε is the expansibility coefficient in accordance with 27.5;

α is the correction factor or flow coefficient (see 27.6), depending upon the Reynolds number Re_{D5}:

$$Re_{D5} = \frac{4 q_m}{\pi D_5 (17{,}1 + 0{,}048\, t_5)} \times 10^6$$

α varies between

$0{,}990 + 0{,}002$ for $Re_{D5} = 3 \times 10^6$

and

$0{,}990 - 0{,}004$ for $Re_{D5} = 3 \times 10^4$

A first approximation of q_m is obtained with $\alpha = 0{,}990$ and corrected for the value of Re_{D5} (see 27.6).

34.3.3.1.3 The mass flowrate is determined using an in-duct Venturi nozzle, see clause 23 and figure 75 c).

Assuming that

$$p_7 = p_{\mathrm{e7}} + p_{\mathrm{a}}$$

$$\Theta_{\mathrm{sg7}} = t_3 + 273{,}15 = \Theta_{\mathrm{a}} + \frac{P_{\mathrm{rx}} \text{ or } P_{\mathrm{ex}}}{q_m c_p}$$

$$\Theta_7 = \Theta_{\mathrm{sg7}} - \frac{1}{2 c_p} \left[\frac{q_m}{A_7 \rho_7} \right]^2$$

The mass flowrate q_m is given by the following expression:

$$q_m = \alpha \varepsilon \pi \frac{d_5^2}{4} \sqrt{2 \rho_7 \Delta p}$$

$$= C \varepsilon \pi \frac{d_5^2}{4} \frac{\sqrt{2 \rho_7 \Delta p}}{\sqrt{1 - \alpha_{\mathrm{Au}} \beta^4}}$$

where

ε is the expansibility coefficient calculated in accordance with 22.4.3 and table 6.

α is the flowrate coefficient of the nozzle equal to

$$\frac{C}{\sqrt{1 - \alpha_{\mathrm{Au}} \beta^4}}$$

C is the discharge coefficient of the nozzle, a function of the throat Reynolds number Re_{d5} (see 23.4.2 and table 5);

α_{Au} is a kinetic energy coefficient equal to 1,043 for a duct approach and to 1 for a chamber approach;

$\beta = d_5/D_7$

$$Re_{d5} = \alpha \varepsilon d_5 \frac{\sqrt{2\rho_7 \Delta p}}{17{,}1 + 0{,}048 t_3} 10^6$$

For a first approximation:

$$\alpha = \frac{0{,}95}{\sqrt{1 - \alpha_{Au}\beta^4}}$$

$$\Theta_7 = \Theta_{sg7}$$

The first approximation of q_m obtained in these conditions allows calculation of new values of Θ_7, ρ_7, Re_{d5}, α and thus q_m.

Two or three calculation steps are sufficient for a calculation accuracy of 10^{-3}.

34.3.3.1.4 The mass flowrate is determined using multiple nozzles in chamber, see clause 23 and figure 75 d).

Assuming that

$$p_7 = p_{e7} + p_a$$

$$\Theta_{sg7} = \Theta_7 = \Theta_3 = \Theta_{sg3} = t_3 + 273{,}15$$

$$\rho_7 = \frac{p_7}{R_w \Theta_7}$$

$$\beta = \frac{d_{5j}}{D_7} \approx 0$$

The mass flowrate is given by the following expression:

$$q_m = \varepsilon \pi \sum_{j=1}^{n} \left(C_j \frac{d_{5j}^2}{4} \right) \sqrt{2\rho_7 \Delta p}$$

where

ε is the expansibility coefficient in accordance with 23.4.3 and table 6.

C_j is the discharge coefficient of the jth nozzle, which is a function of the nozzle throat Reynolds number Re_{d5j}.

$\beta = 0$ and $C_j = \alpha_j$

$C_j = \alpha_j$ is calculated in accordance with 23.4 and table 5;

n is the number of nozzles.

For each nozzle, the throat Reynolds number Re_{d5} is estimated by the following expression:

$$Re_{d5j} = \frac{\varepsilon C_j d_{5j} \sqrt{2\rho_7 \Delta p}}{17{,}1 + 0{,}048\, t_7} 10^6$$

with $C_j = 0{,}95$

After a first estimation of the mass flowrate, the discharge coefficients C_j are corrected.

34.3.3.2 Determination of fan pressure

34.3.3.2.1 Fan inlet pressure

Figure 75 a) to c) shows two types of chamber pressure measurements, where

— the chamber pressure p_{e3} is a gauge pressure;

— the chamber pressure p_{esg3} is a gauge stagnation pressure.

a) The chamber pressure is a gauge pressure p_{e3}.

Assuming that

$$F_{M3} = 1$$

$$p_3 = p_{e3} + p_a$$

$$\Theta_{sg1} = \Theta_3 = \Theta_{sg3} = t_3 + 273{,}15$$

$$\rho_3 = \frac{p_3}{R_w \Theta_3}$$

The inlet stagnation pressure p_{sg3} is given by the following expression:

$$p_{sg3} = p_3 + \frac{1}{2}\rho_3 v_{m3}^2 = p_3 + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2$$

$$p_{esg3} = p_{e3} + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2$$

b) The chamber pressure is an absolute stagnation pressure p_{esg3}.

In these conditions

$$p_{sg3} = p_{esg3} + p_a$$

$$\Theta_3 = \Theta_{sg3} = \Theta_{sg1} = t_3 + 273{,}15$$

there is no friction loss allowance for the inlet simulation duct of length $1D_1$, or $2D_1$ and

$$p_{sg1} = p_{sg3}$$

$$p_{esg1} = p_{esg3}$$

When an inlet simulation duct longer than $1D_1$ or $2D_1$ is required, friction loss allowances may be taken into account.

At the inlet of the duct, downstream of the inlet bell mouth, index 3.1.

$$p_{sg3.1} = p_{sg3}$$

The stagnation pressure at fan inlet section 1 is given by the following expressions:

$$\rho_{3.1} = \rho_{sg3}\frac{\rho_{3.1}}{\rho_{sg3}}$$

$$p_{sg1} = p_{sg3} + \frac{1}{2\rho_{3.1}}\left(\frac{q_m}{A_1}\right)^2 F_{M3.1}\left(\zeta_{3-1}\right)_1$$

$$p_{esg1} = p_{esg3} + \frac{1}{2\rho_{3.1}}\left(\frac{q_m}{A_1}\right)^2 F_{M3.1}\left(\zeta_{3-1}\right)_1$$

where

$Ma_{3.1}$, $\rho_{3.1}$, $F_{M3.1}$ are determined in accordance with 14.4.3.2, 14.4.4 and 14.5.2.

$(\zeta_{3-1})_1 \leqslant 0$ is the conventional friction loss coefficient for the inlet simulation duct of diameter D_1 and length L in accordance with 30.6.

$$\left(\zeta_{3-1}\right)_1 = -\Lambda \frac{L}{D_1}$$

The static pressure p_1 is determined by the following expression:

$$p_1 = p_{sg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2 F_{M1}$$

or

$$p_{e1} = p_{esg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2 F_{M1}$$

where Ma_1, ρ_1, F_{M1} are determined in accordance with 14.4.3.2, 14.4.4 and 14.5.2.

34.3.3.2.2 Fan outlet pressure

At the fan outlet

$$p_2 = p_a$$

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

The Mach number Ma_2 and the density ρ_2 are determined in accordance with 14.4.3.1 and figure 5.

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

$$p_{sg2} = p_2 + \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 F_{M2} = p_a + \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 F_{M2}$$

or

$$p_{esg2} = \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 F_{M2}$$

34.3.3.2.3 Fan pressure

The fan pressure p_{FC} and the fan static pressure p_{sFC} are given by the following expressions:

$$p_{FC} = p_{sg2} - p_{sg1} = p_a + \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 F_{M2} - p_{sg1}$$

$$= \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 F_{M2} - p_{esg1}$$

$$p_{sFC} = p_2 - p_{sg1} = p_a - p_{sg1} = p_{esg1}$$

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

$$k_\rho = \frac{\rho_1}{\rho_m}$$

34.3.3.3 Determination of volume flowrate

The volume flowrate under inlet stagnation conditions is given by the following expression:

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

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

34.3.3.4 Determination of fan air power

34.3.3.4.1 Fan work per unit mass and fan air power

The fan work per unit mass y_C and the fan static work per unit mass y_{sC} are given by the following expressions:

$$y_C = \frac{p_2 - p_1}{\rho_m} + \frac{v_{m2}^2}{2} - \frac{v_{m1}^2}{2}$$

$$= \frac{p_2 - p_1}{\rho_m} + \frac{1}{2}\left(\frac{q_m}{A_2 \rho_2}\right)^2 - \frac{1}{2}\left(\frac{q_m}{A_1 \rho_1}\right)^2$$

$$y_{sC} = \frac{p_2 - p_1}{\rho_m} - \frac{v_{m1}^2}{2} = \frac{p_2 - p_1}{\rho_m} - \frac{1}{2}\left(\frac{q_m}{A_1 \rho_1}\right)^2$$

$$= \frac{p_{e2} - p_{e1}}{\rho_m} - \frac{1}{2}\left(\frac{q_m}{A_1 \rho_1}\right)^2$$

The corresponding fan air power P_{uC} and fan static air power P_{usC} are given by the following expressions:

$$P_{uC} = y_C \, q_m$$

$$P_{usC} = y_{sC} \, q_m$$

34.3.3.4.2 Calculation of fan air power and compressibility coefficient

In accordance with 14.8.2

$$P_{uC} = q_{Vsg1} \, p_{FC} \, k_p$$

$$P_{usC} = q_{Vsg1} \, p_{sFC} \, k_{ps}$$

The compressibility coefficients k_p and k_{ps} may be determined by two equivalent methods (see 14.8.2.1 and 14.8.2.2).

a) First method:

$$k_{ps} \text{ or } k_p = \frac{Z_k \log_{10} r}{\log_{10}\left[1 + Z_k (r - 1)\right]}$$

where

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

for k_p or

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

for k_{ps} and

$$Z_k = \frac{\kappa - 1}{\kappa} \frac{P_r}{q_{Vsg1}\, p_{FC}}$$

for k_p or

$$Z_k = \frac{\kappa - 1}{\kappa} \frac{P_r}{q_{Vsg1}\, p_{sFC}}$$

for k_{ps}.

b) Second method

$$k_{ps} \text{ or } k_p = \frac{\ln(1+x)}{x} \frac{Z_p}{\ln(1+Z_p)}$$

where

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

or

$$x = \frac{p_{sFC}}{p_{sg1}}$$

and

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

34.3.3.5 Calculation of efficiencies

In accordance with 14.8.1, the efficiencies are calculated using the following expressions:

— Fan efficiency:

$$\eta_{rC} = \frac{P_{uC}}{P_r}$$

— Fan static efficiency:

$$\eta_{srC} = \frac{P_{usC}}{P_r}$$

— Fan shaft efficiency:

$$\eta_{sC} = \frac{P_{uC}}{P_a}$$

— Fan shaft static efficiency:

$$\eta_{saC} = \frac{P_{usC}}{P_a}$$

34.3.4 Simplified procedures

34.3.4.1 Reference Mach number Ma_{2ref} less than 0,15 and pressure ratio more than 1,02

34.3.4.1.1 Determination of flowrate

$$\Theta_x = \Theta_{sgx}$$

$$F_{M1} = F_{M2} = F_{M3} = 1$$

The procedures described in 34.3.3.1.1 to 34.3.3.1.4 apply.

The temperature Θ_7 in the test chamber may be measured and there is no need for an iterative procedure to calculate Θ_7.

In any case, the correction of α as a function of the Reynolds number should be applied.

34.3.4.1.2 Determination of fan pressure

34.3.4.1.2.1 Fan inlet pressure

The absolute stagnation pressure in the chamber p_{esg3} is measured

$$\Theta_{sg3} = \Theta_3 = \Theta_1 = \Theta_{sg1} = t_3 + 273{,}15$$

$$p_{sg3} = p_{esg3} + p_a$$

or given by the following procedure:

$$p_3 = p_{e3} + p_a$$

$$\rho_3 = \frac{p_3}{R_w \Theta_3}$$

$$p_{sg3} = p_3 + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2$$

$$p_{esg3} = p_{e3} + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2$$

Normally

$$p_{sg1} = p_{sg3}$$

$$p_{esg1} = p_{esg3}$$

When friction loss allowances may be made, the pressure $p_{3.1}$ at the inlet of the simulation duct is determined by the following method:

Assuming that

$$\Theta_3 = \Theta_{sg3} = \Theta_{sg1}$$

$$p_{3.1} = p_{sg3} - \frac{1}{2\rho_{3.1}}\left(\frac{q_m}{A_1}\right)^2$$

For a first approximation $(p_{3.1})_1 = p_{sg3}$

$$(p_{3.1})_1 = p_{sg3} - \frac{1}{2(\rho_{3.1})_1}\left(\frac{q_m}{A_1}\right)^2$$

$$\rho_{3.1} = \frac{(p_{3.1})_1}{R_w \Theta_3}$$

$$p_{3.1} = p_{sg3} - \frac{1}{2\rho_{3.1}}\left(\frac{q_m}{A_1}\right)^2$$

Two or three calculation steps are sufficient; $p_{3.1}$ may also be calculated by the following expression:

$$p_{3.1} = \frac{1}{2}\left[p_{sg3} + \sqrt{p_{sg3}^2 - 2\left(\frac{q_m}{A_1}\right)^2 R_w \Theta_{sg3}}\right]$$