BRITISH STANDARD

BS EN ISO 5801:2008
BS 848-1:2007

Incorporating corrigenda July 2008 & May 2009

Industrial fans — Performance testing using standardized airways

ICS 23.120

British Standards

National foreword

This British Standard is the UK implementation of EN ISO 5801:2008. It is identical to ISO 5801:2007, incorporating corrigendum 1:2008. It supersedes BS ISO 5801:2007 which is withdrawn.

The UK participation in its p eparation was entrusted to Technical Committee MCE/17, Fans for general purposes.

A list of organizations represented on this committee can be obtained on request to its secretary.

This publication does not purport to include all the necessary provisions of a contract. Users are responsible for its correct application.

Compliance with a British Standard cannot confer immunity from legal obligations.

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31 May 2009	This corrigendum renumbers BS ISO 5801:2007 as BS EN ISO 5801:2008
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EUROPEAN STANDARD

NORME EUROPÉENNE

EUROPÄISCHE NORM

EN ISO 5801

October 2008

ICS 23.120

English Version

Industrial fans - Performance testing using standardized airways (ISO 5801:2007 including Cor 1:2008)

Ventilateurs industriels - Essais aérauliques sur circuits normalisés (ISO 5801:2007, Cor 1:2008 inclus)

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Foreword

The text of ISO 5801:2007 including Cor 1:2008 has been prepared by Technical Committee ISO/TC 117 “Industrial fans” of the International Organization for Standardization (ISO) and has been taken over as EN ISO 5801:2008 by Technical Committee CEN/TC 156 “Ventilation for buildings” the secretariat of which is held by BSI.

This European Standard shall be given the status of a national standard, either by publication of an identical text or by endorsement, at the latest by April 2009, and conflicting national standards shall be withdrawn at the latest by April 2009.

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Endorsement notice

The text of ISO 5801:2007 including Cor 1:2008 has been approved by CEN as a EN ISO 5801:2008 without any modification.

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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.

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 2.

The main task of technical committees is to prepare International Standards. 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.

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights.

ISO 5801 was prepared by Technical Committee ISO/TC 117, *Industrial fans*.

This second edition cancels and replaces the first edition (ISO 5801:1997), which has been technically revised.

Introduction

This International Standard is the result of almost 30 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 Technical Committee 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 successive 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 this International Standard many setups from various national codes in order to authorize their future use. This explains the sheer volume of this document.

Essential features of this International Standard are as follows:

a) Categories 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 categories should be recognized (see 18.2).

A fan adaptable to more than one installation category will have more than one standardized performance characteristic. Users should select the installation category closest to their 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 fans where the outlet swirl is less than 15°, i.e. centrifugal, cross-flow or vane-axial fans, it is possible to use a simplified outlet duct without straightener when discharging to the atmosphere or to a measuring chamber. If there is any doubt about the degree of swirl, then a test should be performed to establish how much is present.

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 a duct of length $3D$ on the outlet side. Results obtained in this way may differ to some extent from those obtained using the normal category 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) Flow rate measurement

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

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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 referenced documents are indispensable for the application of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

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

ISO 5167-1, *Measurement of fluid flow by means of pressure differential devices inserted in circular cross-section conduits running full — Part 1: General principles and requirements*

ISO 5168, *Measurement of fluid flow — Procedures for the evaluation of uncertainties*

ISO 5221, *Air distribution and air diffusion — Rules to methods of measuring air flow rate in an air handling duct*

IEC 60034-2:1972, *Rotating electrical machines — Part 2: Methods for determining losses and efficiency of rotating electrical machinery from tests (excluding machines for traction vehicles)*

IEC 60051-2, *Direct acting indicating analogue electrical measuring instruments and their accessories — Part 2: Special requirements for ammeters and voltmeters*

IEC 60051-3, *Direct acting indicating analogue electrical measuring instruments and their accessories — Part 3: Special requirements for wattmeters and varmeters*

IEC 60051-4, *Direct acting indicating analogue electrical measuring instruments and their accessories — Part 4: Special requirements for frequency meters*

3 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 5168 and the following apply.

NOTE 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

NOTE Fan 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

NOTE 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

NOTE Temperature is expressed in degrees Celsius.

3.5
absolute temperature
Θ
thermodynamic temperature

$$\Theta = T + 273{,}15$$

NOTE In this document, Θ represents the absolute temperature in kelvin and T the temperature in degrees Celsius.

3.6
specific gas constant
R
for an ideal dry gas, the equation of state is written

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

NOTE For dry air, $R = 287\ \mathrm{J \cdot kg^{-1} \cdot K^{-1}}$.

3.7
isentropic exponent
κ
for an ideal gas and an isentropic process

$$\kappa = \gamma = \frac{c_p}{c_V}$$

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

NOTE For atmospheric air, $\kappa = 1{,}4$.

3.8
specific heat capacity at constant pressure
c_p
for an ideal gas

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

NOTE Specific heat capacity is normally expressed in joules per (kilogram kelvin).

3.9
specific heat capacity at constant volume
c_V
for an ideal gas

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

NOTE Specific heat capacity is normally expressed in joules per (kilogram kelvin).

3.10
compressibility factor
Z

NOTE 1 For an ideal gas, $Z = 1$.

NOTE 2 For a real gas,

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

where

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

NOTE 1 The stagnation temperature is constant along an airway and, for an inlet duct, is equal to the absolute ambient temperature in the test enclosure.

NOTE 2 Stagnation temperature is expressed in degrees Celsius.

NOTE 3 For Mach numbers less than 0,122 obtained for standard air with duct velocities less than 40 m/s, the stagnation temperature is virtually the same as the total temperature.

3.12
fluid temperature at a point
static temperature at a point
Θ
absolute temperature registered by a thermal sensor moving at the fluid velocity

NOTE 1 For real gas flow

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

where v is the fluid velocity, in metres per second, at a point.

NOTE 2 These temperatures are expressed in degrees Celsius.

NOTE 3 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

NOTE This temperature is expressed in degrees Celsius.

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

NOTE 1 When properly measured, it is a close approximation to the temperature of adiabatic saturation.

NOTE 2 This temperature is expressed in degrees Celsius.

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

NOTE This temperature is expressed in kelvin.

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

NOTE This temperature is expressed in kelvin.

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

NOTE This pressure is normally expressed in pascals.

3.18
atmospheric pressure
p_a
absolute pressure of the free atmosphere at the mean altitude of the fan

NOTE This pressure is normally expressed in pascals.

3.19
gauge pressure
p_e
value of the pressure when the datum pressure is the atmospheric pressure at the point of measurement

NOTE 1 Gauge pressure may be negative or positive

$$p_e = p - p_a$$

NOTE 2 This pressure is normally expressed in pascals.

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 given by the following equation:

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

NOTE 1 Ma is the Mach number at this point (see 3.23).

NOTE 2 This pressure is normally expressed in pascals.

NOTE 3 For Mach numbers less than 0,122 obtained for standard air with duct velocities less than 40 m/s, the stagnation pressure is virtually the same as the total pressure.

3.21
Mach factor
f_{Mx}
correction factor applied to the dynamic pressure at a point, given by the expression

$$f_{Mx} = \frac{p_{sg} - p}{p_d}$$

NOTE The Mach factor may be calculated by:

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

3.22
dynamic pressure at a point
p_d
pressure calculated from the velocity and the density ρ of the air at the point given by the following equation:

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

NOTE This pressure is normally expressed in pascals.

3.23
Mach number at a point
Ma
ratio of the gas velocity at a point to the velocity of sound given by the following equation:

$$Ma = \frac{v}{\sqrt{\kappa R_{\mathrm{w}} \Theta}} = \frac{v}{c}$$

where

c is the velocity of sound,

$$c = \sqrt{\kappa R_{\mathrm{w}} \Theta}$$

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}, given by the following equation:

$$p_{\mathrm{esg}} = p_{\mathrm{sg}} - p_{\mathrm{a}}$$

NOTE This pressure is normally expressed in pascals.

3.25
mass flow rate
q_m
mean value, over time, of the mass of air which passes through the specified airway cross-section per unit of time

NOTE 1 The mass flow will be the same at all cross-sections within the fan airway system excepting leakage.

NOTE 2 Mass flow rate is expressed in kilograms per second.

3.26
average gauge pressure at a section x
mean gauge pressure at a section x
$p_{\mathrm{e}x}$
mean value, over time, of the gauge pressure averaged over the area of the specified airway cross-section

NOTE This pressure is normally expressed in pascals.

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 given by the following equation:

$$p_x = p_{\mathrm{e}x} + p_{\mathrm{a}}$$

NOTE This pressure is normally expressed in pascals.

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

$$\rho_x = \frac{p_x}{R_\mathrm{w}\Theta_x}$$

where R_w is the gas constant of humid gas

NOTE Density is expressed in kilograms per cubic metre.

3.29
volume flow rate at a section x

q_{Vx}

mass flow rate at the specified airway cross-section divided by the corresponding mean value, over time, of the average density at that section given by the following equation:

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

NOTE Volume flow rate is expressed in cubic metres per second.

3.30
average velocity at a section x

$v_{\mathrm{m}x}$

volume flow rate at the specified airway cross-section divided by the cross-sectional area, A_x, given by the following equation:

$$v_{\mathrm{m}x} = \frac{q_{Vx}}{A_x}$$

NOTE 1 This is the mean value, over time, of the average component of the gas velocity normal to that section.

NOTE 2 Average velocity is expressed in metres per second.

3.31
conventional dynamic pressure at a section x

$p_{\mathrm{d}x}$

dynamic pressure calculated from the average velocity and the average density at the specified airway cross-section given by the following equation:

$$p_{\mathrm{d}x} = \rho_x \frac{v_{\mathrm{m}x}^2}{2} = \frac{1}{2\rho_x}\left(\frac{q_m}{A_x}\right)^2$$

NOTE 1 The conventional dynamic pressure will be less than the average of the dynamic pressures across the section.

NOTE 2 Dynamic pressure is expressed in pascals.

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 given by the following equation:

$$Ma_x = v_{\mathrm{m}x} \Big/ \sqrt{\kappa R_{\mathrm{w}} \Theta_x}$$

NOTE The Mach number is dimensionless.

3.33
average stagnation pressure at a section x
$p_{\mathrm{sg}x}$
sum of the conventional dynamic pressure $p_{\mathrm{d}x}$ corrected by the Mach factor coefficient $f_{\mathrm{M}x}$ at the section and the average absolute pressure p_x given by the following equation:

$$p_{\mathrm{sg}x} = p_x + p_{\mathrm{d}x} f_{\mathrm{M}x}$$

NOTE 1 The average stagnation pressure may be calculated by the equation:

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

NOTE 2 Average stagnation pressure is expressed in pascals.

3.34
gauge stagnation pressure at a section x
$p_{\mathrm{esg}x}$
difference between the average stagnation pressure, $p_{\mathrm{sg}x}$, at a section and the atmospheric pressure, p_{a}, given by the following equation:

$$p_{\mathrm{esg}x} = p_{\mathrm{sg}x} - p_{\mathrm{a}}$$

NOTE Gauge stagnation pressure is expressed in pascals.

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

NOTE 1 In this case, it is possible to consider the stagnation temperature as equal to the ambient temperature, Θ_{a}, given by the following equation:

$$\Theta_{\mathrm{sg1}} = \Theta_{\mathrm{a}} = T_{\mathrm{a}} + 273{,}15$$

NOTE 2 Inlet stagnation absolute temperature is expressed in kelvins.

3.36
stagnation density
ρ_{sg1}
density calculated from the inlet stagnation pressure, p_{sg1}, and the inlet stagnation temperature, Θ_{sg1}, given by the following equation:

$$\rho_{\mathrm{sg1}} = \frac{p_{\mathrm{sg1}}}{R_{\mathrm{w}} \Theta_{\mathrm{sg1}}}$$

NOTE Stagnation density is expressed in kilograms per cubic metre.

3.37
inlet stagnation volume flow rate

$q_{V\text{sg1}}$

mass flow rate divided by the inlet stagnation density given by the formula:

$$q_{V\text{sg1}} = \frac{q_m}{\rho_{\text{sg1}}}$$

NOTE Inlet stagnation volume flow rate is expressed in cubic metres per second.

3.38
fan pressure

p_f

difference between the stagnation pressure at the fan outlet and the stagnation pressure at the fan inlet given by the equation:

$$p_\text{f} = p_{\text{sg2}} - p_{\text{sg1}}$$

NOTE 1 When the Mach number is less than 0,15, it is possible to use the relationship:

$$p_\text{f} = p_{\text{tf}} = p_{\text{t2}} - p_{\text{t1}}$$

NOTE 2 It is possible to refer the fan pressure to the installation category A, B, C or D.

NOTE 3 Fan pressure is expressed in pascals.

3.39
dynamic pressure at the fan outlet

p_{d2}

conventional dynamic pressure at the fan outlet calculated from the mass flow rate, the average gas density at the outlet and the fan outlet area

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

NOTE Fan dynamic pressure is expressed in pascals.

3.40
fan static pressure

p_{sf}

conventional quantity defined as the fan pressure minus the fan dynamic pressure corrected by the Mach factor as given by the following equation:

$$p_{\text{sf}} = p_{\text{sg2}} - p_{\text{d2}} \cdot f_{\text{M2}} - p_{\text{sg1}} = p_2 - p_{\text{sg1}}$$

NOTE 1 It is possible to refer the fan static pressure to the installation category A, B, C or D.

NOTE 2 Fan static pressure is expressed in pascals.

3.41
mean density

ρ_m

arithmetic mean value of inlet and outlet densities

$$\rho_\text{m} = \frac{\rho_1 + \rho_2}{2}$$

NOTE Mean density is expressed in kilograms per cubic metre.

3.42
mean stagnation density
ρ_{msg}
arithmetic mean value of inlet and outlet stagnation densities given by the following equation:

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

NOTE Mean stagnation pressure is expressed in pascals.

3.43
fan work per unit mass
W_m
increase in mechanical energy per unit mass of fluid passing through the fan given by the following equation:

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

NOTE 1 It is possible to calculate W_m as in 3.47, as follows:

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

NOTE 2 The value obtained differs by only a few parts per thousand from the value given by the above expression.

NOTE 3 It is possible to refer the fan work per unit mass to the installation category A, B, C or D.

NOTE 4 Fan work is expressed in joules per kilogram.

3.44
fan static work per unit mass
W_{ms}
increase in mechanical energy per unit mass of fluid passing through the fan minus the kinetic energy per unit mass imparted to the fluid, given by the following equation:

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

NOTE 1 It is possible to refer the fan static work per unit mass to the installation category A, B, C or D.

NOTE 2 Fan static work is expressed in joules per kilogram.

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 as given by the following equation:

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

NOTE The fan pressure ratio is dimensionless.

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; k_p is given by the equation:

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

where

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

NOTE 1 The work done is derived from the impeller power on the assumption of polytropic compression with no heat transfer through the fan casing.

NOTE 2 k_p and ρ_{ms1}/ρ_{msg} differ by less than 2×10^{-3}.

NOTE 3 The compressibility coefficient is dimensionless.

NOTE 4 A second method of calculation is shown in 30.2.3.4.2, section b).

3.47
fan air power
P_u
conventional output power which is the product of the mass flow rate q_m and the fan work per unit mass W_m, or the product of the inlet volume flow rate q_{Vsg1}, the compressibility coefficient k_p and the fan pressure p_f given by the following equation:

$$P_u = q_m W_m = q_{Vsg1} \cdot p_f \cdot k_p$$

NOTE 1 It is possible to refer the fan air power to the installation category A, B, C or D.

NOTE 2 Fan air power is expressed in watts when q_m is in kilograms per second and W_m is in joules per kilogram.

NOTE 3 Fan air power is expressed in watts when q_{Vsg1} is in cubic metres per second and p_f is in pascals.

3.48
fan static air power
P_{us}
conventional output power which is the product of the mass flow rate q_m and the fan static work per unit mass W_{ms}, or the product of the inlet volume flow rate q_{Vsg1}, the compressibility coefficient k_{ps} and the fan static pressure p_{sf}; k_{ps} is calculated using $r = p_2/p_{sg1}$

$$P_{us} = q_m W_{ms} = q_{Vsg1} \cdot k_{ps} \cdot p_{sf}$$

NOTE 1 It is possible to refer the fan static air power to the installation category A, B, C or D.

NOTE 2 The fan static air power is expressed in watts when q_m is in kilograms per second and W_{ms} is in joules per kilogram.

3.49
impeller power
P_r
mechanical power supplied to the fan impeller

NOTE Impeller power is expressed in watts.

3.50
fan shaft power
P_a
mechanical power supplied to the fan shaft

NOTE Fan shaft power is expressed in watts.

3.51
motor output power
P_o
shaft power output of the motor or other prime mover

NOTE Motor output power is expressed in watts.

3.52
motor input power
P_e
electrical power supplied at the terminals of an electric motor drive

NOTE Motor input power is expressed in watts.

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 second

3.55
tip speed of the impeller
v_p
peripheral speed of the impeller blade tips

NOTE Tip speed is expressed in metres per second.

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 the fan inlet given by the following equation:

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

3.57
fan impeller efficiency
η_r
fan air power divided by the impeller power P_r as follows:

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

NOTE 1 It is possible to refer the fan impeller efficiency to the installation category A, B, C or D.

NOTE 2 Fan impeller efficiency may be expressed as a proportion of unity or as a percentage.

3.58
fan impeller static efficiency
η_{sr}
fan static power divided by the impeller power given by the equation:

$$\eta_{sr} = \frac{P_{us}}{P_r}$$

NOTE 1 It is possible to refer the fan impeller static efficiency to the installation category A, B, C or D.

NOTE 2 Fan impeller static efficiency may be expressed as a proportion of unity or as a percentage.

3.59
fan shaft efficiency
η_a
fan air power divided by the fan shaft power given by the equation:

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

NOTE 1 Fan shaft power includes bearing losses, while fan impeller power does not.

NOTE 2 It is possible to refer the fan shaft efficiency to the installation category A, B, C or D.

NOTE 3 Fan shaft efficiency may be expressed as a proportion of unity or as a percentage.

3.60
fan motor shaft efficiency
η_o
fan air power P_u divided by the motor output power P_o as given by the equation:

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

NOTE 1 It is possible to refer the fan motor shaft efficiency to the installation category A, B, C or D.

NOTE 2 Fan motor shaft efficiency may be expressed as a proportion of unity or as a percentage.

3.61
overall efficiency
η_e
fan air power divided by the motor input power for the fan and motor combination given by the equation:

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

NOTE 1 It is possible to refer the overall efficiency to the fan category A, B, C or D.

NOTE 2 Fan overall efficiency is expressed as a proportion of unity or as a percentage.

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 given by the following equation:

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

NOTE k_ρ is dimensionless.

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 and given by the following equation:

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

where

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

v_n is the local velocity, in metres per second, normal to the cross-section.

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 and given by the following equation:

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

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

NOTE It is the product of the local velocity, the local density and a relevant scale length (duct diameter, blade chord), divided by the dynamic viscosity as given by the following equation:

$$Re_{Dx} = \frac{v_{\mathrm{m}x} D_x}{\nu_x} = \frac{4 q_m}{\pi \mu D_x}$$

3.66
friction-loss coefficient
$(\xi_{\mathrm{x-y}})_y$
dimensionless coefficient for friction losses between planes x and y of a duct, calculated for the velocity and density at section y; for incompressible flow, the formula is given by:

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

3.67
hydraulic diameter
D_{h}
hydraulic diameter of a rectangular section of a duct given by the equation:

$$D_{\mathrm{h}} = \frac{4A}{2(b+h)}$$

where

A is the cross-sectional area;

b is the rectangular section width;

h is the rectangular section height.

3.68
flow coefficient
Φ
dimensionless number given by the equation:

$$\Phi = \frac{q_m}{\rho_{\mathrm{m}} D_r^2 u}$$

3.69
pressure coefficient
Ψ
dimensionless number given by the equation:

$$\Psi = \frac{p_{\mathrm{f}}}{\rho_{\mathrm{m}} u^2}$$

3.70
fan power coefficient
λ
dimensionless number given by:

$$\lambda = \frac{\phi \psi}{\eta}$$

4 Symbols and units

4.1 Symbols

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

Symbol	Represented quantity	Definition ref	SI Unit
A_x	Area of the conduit at section x	3.1	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	3.23	m/s
c_p	Specific heat capacity at constant pressure	3.8	J/kg/K
c_V	Specific heat capacity at constant volume	3.9	J/kg/K
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	3.67	m
D_x	Internal diameter of a circular conduit in section x	—	m
D_r	Outside diameter of the impeller	—	m
$f_{\mathrm{M}x}$	Mach factor for correction of dynamic pressure at section x	3.21	—
g	Acceleration due to gravity	—	m/s^2
h	Height of the rectangular section of a duct	—	m
h_u	Relative humidity p_V/p_sat	—	—
$i_{\mathrm{k}x}$	Kinetic index at section x	3.64	—
k_c	Resulting coefficient used in the conversion of test results	—	—
k_cs	Resulting coefficient used in the conversion of static pressure test results	—	—
k_ρ	Inlet to mean density ratio	3.62	—
k_p	Compressibility coefficient for the calculation of fan air power P_u	3.46	—
k_ps	Compressibility coefficient for the calculation of fan static air power	—	—
Ma	Mach number	3.23	—
Ma_x	Mach number at section x	3.32	—
$Ma_{x,\mathrm{ref}}$	Reference Mach number at section x at inlet stagnation conditions	—	—
Ma_u	Peripheral Mach number	3.56	—
m	Area ratio of an orifice plate $(d/D)^2$	—	—
n	Rotational frequency of impeller	—	r/s
N	Rotational speed of impeller	—	r/min
p	Absolute pressure of the fluid	3.17	Pa
p_a	Atmospheric pressure at the mean altitude of the fan	3.18	Pa

p_{e}	Gauge pressure	3.19	Pa
p_{sg}	Absolute stagnation pressure at a point	3.20	Pa
p_{esg}	Gauge stagnation pressure at a point	3.24	Pa
$p_{\mathrm{esg}x}$	Gauge stagnation pressure at section x	3.34	Pa
p_{d}	Dynamic pressure at a point	3.22	Pa
p_x	Mean absolute pressure in space and time of the fluid at section x	3.27	Pa
$p_{\mathrm{e}x}$	Mean gauge pressure in space and time at section x	3.26	Pa
$p_{\mathrm{sg}x}$	Mean stagnation pressure at section x	3.33	Pa
$p_{\mathrm{d}x}$	Conventional dynamic pressure at section x	3.31	Pa
p_{sat}	Saturation vapour pressure	12.2	Pa
p_{v}	Partial pressure of water vapour	12.2	Pa
p_{f}	Fan pressure	3.38	Pa
p_{sf}	Fan static pressure	3.40	Pa
p_{d2}	Fan outlet dynamic pressure	3.39	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	3.50	W
P_{e}	Motor input power	3.52	W
P_{o}	Motor output power	3.51	W
P_{r}	Mechanical power supplied to the impeller of the fan	3.49	W
P_{u}	Fan air power	3.47	W
P_{us}	Fan static air power	3.48	W
q_m	Mass flow rate	3.25	kg/s
q_V	Volume flow rate	—	$\mathrm{m^3/s}$
$q_{V\mathrm{sg1}}$	Inlet stagnation volume flow rate	3.37	$\mathrm{m^3/s}$
q_{Vx}	Volume flow rate at section x	3.29	$\mathrm{m^3/s}$
r	Fan pressure ratio	3.45	—
r_{d}	Pressure ratio for a flowmeter $r_{\mathrm{d}} = p_{\mathrm{do}}/p_{\mathrm{u}}$	—	—
$r_{\Delta p}$	$\Delta p/p_{\mathrm{do}}$ for a flowmeter	—	—
R	Gas constant of dry air or gas	3.6	J/kg/K
R_{w}	Gas constant of humid air or gas	—	J/kg/K
Re_{Dx}	Reynolds number at section x	3.65	—
T_{a}	Ambient temperature	—	°C
T_{b}	Barometer temperature	—	°C
T_{d}	Dry bulb temperature	3.13	°C
T_{w}	Wet bulb temperature	3.14	°C
T_x	Static temperature at section x	—	°C
$T_{\mathrm{sg}x}$	Stagnation temperature at section x	—	°C
u_x	Relative uncertainty of x	—	%

U_x	Absolute uncertainty of x	—	same as X
v	Velocity of gas at a point	—	m/s
$v_{\mathrm{m}x}$	Average velocity of the gas at section x	3.30	m/s
v_{p}	Peripheral velocity, or tip speed, of the impeller	3.55	m/s
W_m	Fan work per unit mass	3.43	J/kg
$W_{m\mathrm{s}}$	Fan static work per unit mass	3.44	J/kg
Z	Compressibility factor in equation of state	3.10	—
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)	—	—
z_x	Mean altitude of section x	—	m
α	Flow rate 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	3.63	—
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	—	—
$(\xi_{\mathrm{x-y}})_y$	Conventional friction loss coefficient between planes x and y calculated for section y	3.66	—
η	Efficiency	—	—
η_{s}	Static efficiency	—	—
η_{a}	Fan shaft efficiency	3.59	—
η_{e}	Overall efficiency	3.61	—
η_{o}	Fan motor shaft efficiency	3.60	—
η_{r}	Fan impeller efficiency	3.57	—
η_{sr}	Fan impeller static efficiency	3.58	—
$\Theta_{\mathrm{sg}x}$	Stagnation temperature at section x	3.15	K
Θ_x	Static or fluid temperature at section x	3.16	K
Θ_{a}	Ambient temperature	—	K
Θ_{u}	Temperature upstream of an in-line flowmeter	—	K
κ	Isentropic exponent for an ideal gas	3.7	—
Λ	Specific friction-loss coefficient for a length of one diameter of a straight duct	—	—
μ	Dynamic viscosity	—	Pa·s
ρ	Density of gas	—	kg/m^3
ρ_x	Mean density of gas at section x	3.28	kg/m^3

ρ_m	Mean density of gas in the fan	3.41	kg/m^3
Φ	Flow coefficient	3.68	—
Ψ	Pressure coefficient	3.69	—
λ	Fan power coefficient	3.70	—
ω	Angular velocity	—	rad/s
ν	Kinematic viscosity	12.3	m/s

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/kg corresponding to an increase in fan pressure approximately equal to 30 000 Pa for a mean density in the fan of 1,2 kg/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 categories of installation:

— category A: free inlet, free outlet;

— category B: free inlet, ducted outlet;

— category C: ducted inlet, free outlet;

— category 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 flow rate 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 flow rates, 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, a simplified method is 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 categories 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 category 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 category 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 setup described in 28.2.5 with a duct of length $3D$ 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 1.

The test fans shown in the figures for each of the test installations are of one type (e.g. an axial fan). However, a test fan of another type could be used.

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 mbar) or to the nearest 1 mmHg. 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 pascals, 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 the 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).

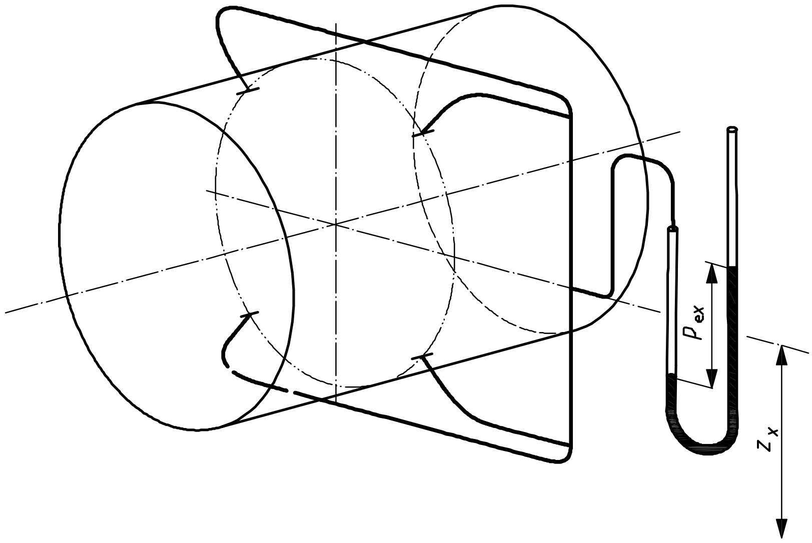


Figure 1 — Tapping connections to obtain average static pressure and altitude of manometer

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.

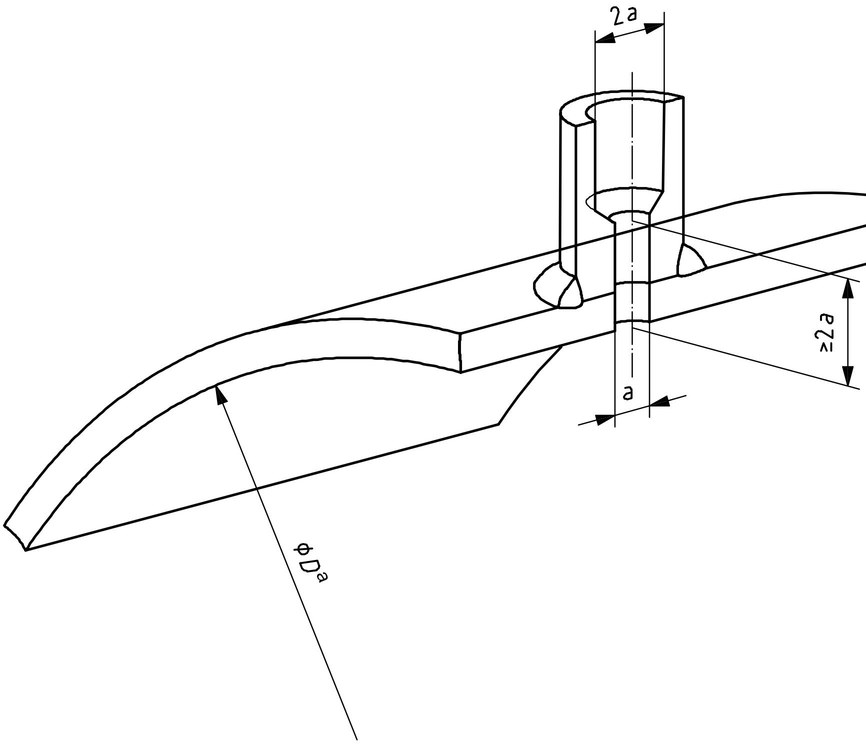
To determine the pressure difference between planes of pressure measurement on opposite sides of the fan, either or both sides of the manometer may be connected between sets of four tapping connections arranged as recommended in 7.4.

7.2 Use of wall tappings

At each of the sections for pressure measurement in the standardized airways specified in Clauses 21 to 25 and in Clauses 30 to 33, 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.

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.



a airway diameter (D)

Figure 2 — Construction of wall pressure tappings

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

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 D 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 flow rate approaching the maximum of the series. If any one of the four readings lies outside a range equal to 5 % for p_{ex} < 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 equally spaced pressure tappings should be used.

NOTE “Mean gauge pressure” here denotes 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.

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, usually slightly closer to the stagnation value.

If the air velocity is equal to 25 m/s, the difference between stagnation and static temperatures is 0,31 °C; at 35 m/s, 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/s, 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 0 m/s and 25 m/s 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 situated symmetrically 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/s. 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 Acceptable instruments

Instruments should have an uncertainty of not more than 0,5 % (i.e. accuracy class index of 0,5 in accordance with IEC 60051-4).

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 no greater 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 60034-2. For this purpose, measurements of voltage, current, speed and, in the case of alternating current (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 meters

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 direct current (d.c.) motors, by measurement of the input voltage and current.

The equipment used for standardized airway tests shall be of class index 0,5 in accordance with IEC 60051-2 and IEC 60051-3 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

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.

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 from the formula:

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

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 at section x

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

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

where

Θ_a is the absolute ambient temperature, in kelvins, given by

$$\Theta_a = T_a + 273{,}15$$

where $T_a = T_d$ (dry bulb temperature, in degrees Celsius);

p_a is the atmospheric pressure;

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

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 For standard air:

$\rho_a = 1{,}2$ kg/m^3

$\Theta_a = 293{,}15$ K

$p_a = 101\ 325$ Pa

$h_u = 0{,}40$

$R_w = 288$ J/(kg·K)

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

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

12.2 Determination of vapour pressure

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

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

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 when T_w is between 0 °C and 150 °C;

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

$(p_{sat})_{T_w}$ is the pressure of saturated vapour at the wet bulb temperature T_w.

Table 1 lists values of saturated vapour pressure (p_{sat}) over the temperature range –4 °C to 49,5 °C.

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
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

Table 1 (*continued*)

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**
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

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

$$p_{sat} = \exp\left(\frac{17{,}438\ T_w}{239{,}78 + T_w} + 6{,}414\ 7\right)$$

or, between 0 °C and 100 °C:

$$p_{sat} = 610{,}8 + 44{,}442\ T_w + 1{,}413\ 3\ T_w^2 + 0{,}027\ 68\ T_w^3 + 2{,}556\ 67 \times 10^{-4}\ T_w^4 + 2{,}891\ 66 \times 10^{-6}\ T_w^5$$

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

$$p_v = h_u (p_{sat})_{T_d}$$

where $(p_{sat})_{T_d}$ is the saturation vapour pressure at the dry bulb temperature T_d calculated using the above formula with T_d instead of T_w.

12.3 Determination of air viscosity

The following formula can be used in the range –20 °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 equation:

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

13 Determination of flow rate

13.1 General

The measurement of flow rate may be carried out in accordance with ISO 5167-1 and ISO 3966, and any flow rate measurement obtained in this way conforms 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 multi-Venturi nozzles, the orifice plate, and the conical or bellmouth inlet.

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 or bellmouth inlet may only be used at the inlet to an airway, drawing air from free space. Multi-nozzles are only used within a test chamber.

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

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

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

where

q_m is the mass flow rate, 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.

$$\rho_\mathrm{u} = \frac{p_\mathrm{u}}{R_\mathrm{w} \Theta_\mathrm{u}}$$

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

$$\Theta_{\mathrm{u}} = \Theta_{\mathrm{sgu}} - \frac{q_m^2}{2A_{\mathrm{u}}^2 \rho_{\mathrm{u}}^2 c_p} + \frac{P_{\mathrm{r}x} \text{ or } P_{\mathrm{e}x}}{q_m c_p}$$

where

$P_{\mathrm{r}x}$ or $P_{\mathrm{e}x}$ is the power provided by any auxiliary fan;

A_{u} is the area of the duct upstream of the flowmeter;

$A_{\mathrm{u}} = \infty$ for an inlet orifice.

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

$$\Theta_{\mathrm{u}} = \Theta_{\mathrm{sg1}} + \frac{P_{\mathrm{r}} \text{ or } P_{\mathrm{e}}}{q_m c_p} - \frac{q_m^2}{2A_{\mathrm{u}}^2 \rho_{\mathrm{u}}^2 c_p}$$

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 25, Tables 4 and 5, and Figures 19 and 21 to 23.

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.

The values for the uncertainties of the flow coefficient associated with each flow-metering element are given in Clauses 22 to 25. 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.

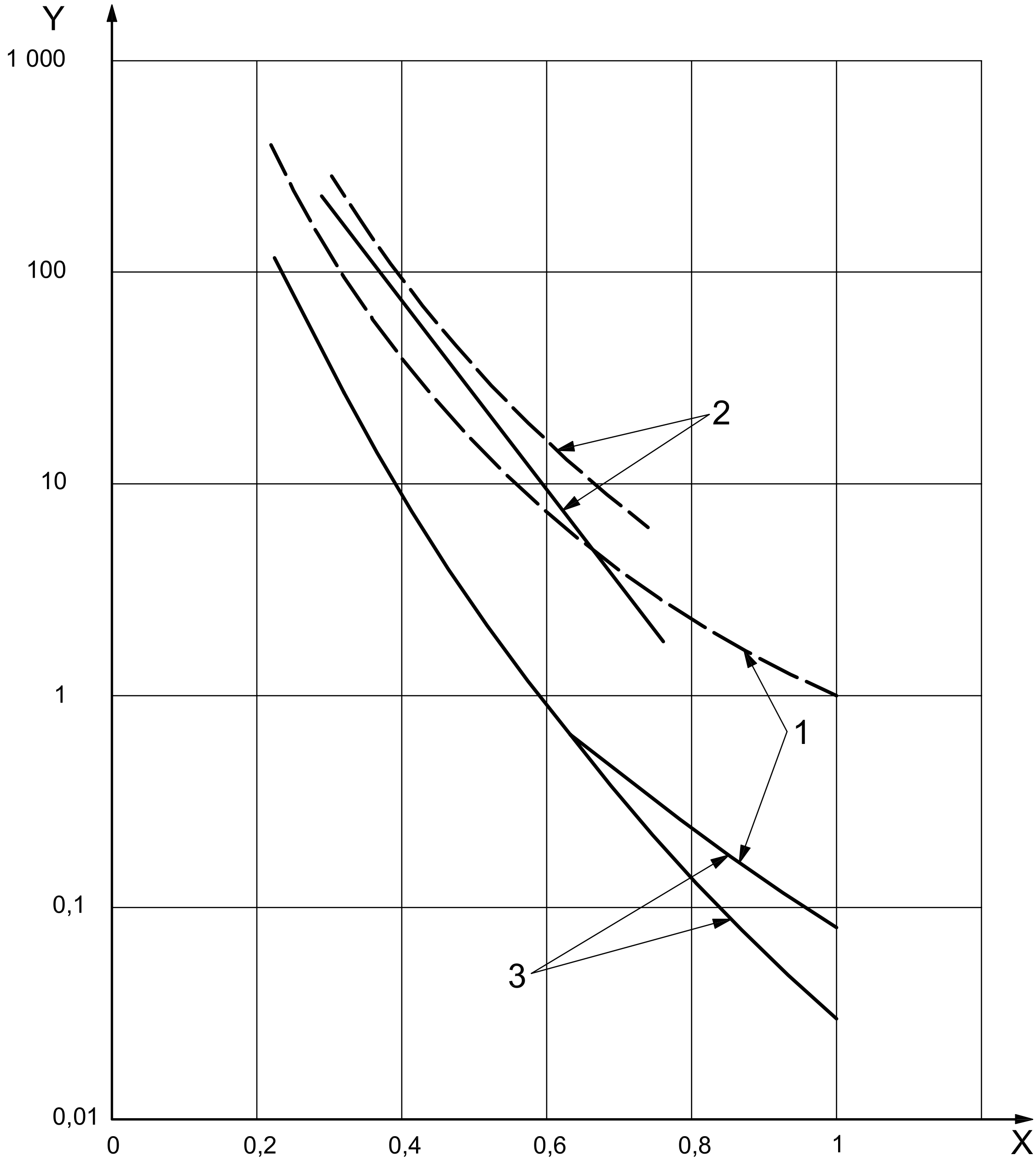
13.2.4 In order to facilitate the selection of type and size of flowmeter, the losses associated with each type are given in Figure 3. Approximate values for the pressure difference (expressed as a multiple of the dynamic pressure in the downstream airway) which will be registered across each device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

13.2.5 Multi-Venturi nozzles have 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.

13.3 Traverse methods

The local velocity shall 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 flow rate (see Clauses 11 and 25).

In standardized airways, a Pitot-static tube conforming to the requirements of ISO 3966 (see Figure 24) shall be used.



________ pressure loss

------------- pressure difference

Key

X ratio of throat diameter to downstream duct diameter($b' = d/D$)

Y stagnation pressure loss or pressure difference relative to downstream dynamic pressure

1 conical or bellmouth inlet

2 orifice plate

3 both units with 15° angle diffuser included

NOTE The curves are from ISO 5167-1.

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 33 depending on the test method used.

The method of calculation in the general case of compressible fluid flow is explained in this clause.

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: pressure in pascals; power in watts; and volume flow in cubic metres per second.

14.3 Temperature

14.3.1 In this International Standard, the mean temperature measured at section x is assumed to be the stagnation value, $\Theta_{\mathrm{sg}x}$, 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 14.4.3.1 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, $\Theta_{\mathrm{sg}x}$, in all sections upstream of the fan tested shall be considered constant and equal to the ambient temperature at the test site, Θ_{a}:

$$\Theta_{\mathrm{sg1}} = \Theta_{\mathrm{sg3}} = \Theta_{\mathrm{a}}$$

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 rate, q_m, and the heat capacity of air at constant pressure, c_p.

$$\Theta_{\mathrm{sg2}} = \Theta_{\mathrm{sg1}} + \frac{P_{\mathrm{r}} \text{ or } P_{\mathrm{e}}}{q_m c_p} = \Theta_{\mathrm{sg4}}$$

NOTE In the above equation, c_p can be taken as 1 008 J/(kg·K) as a first approximation for air.

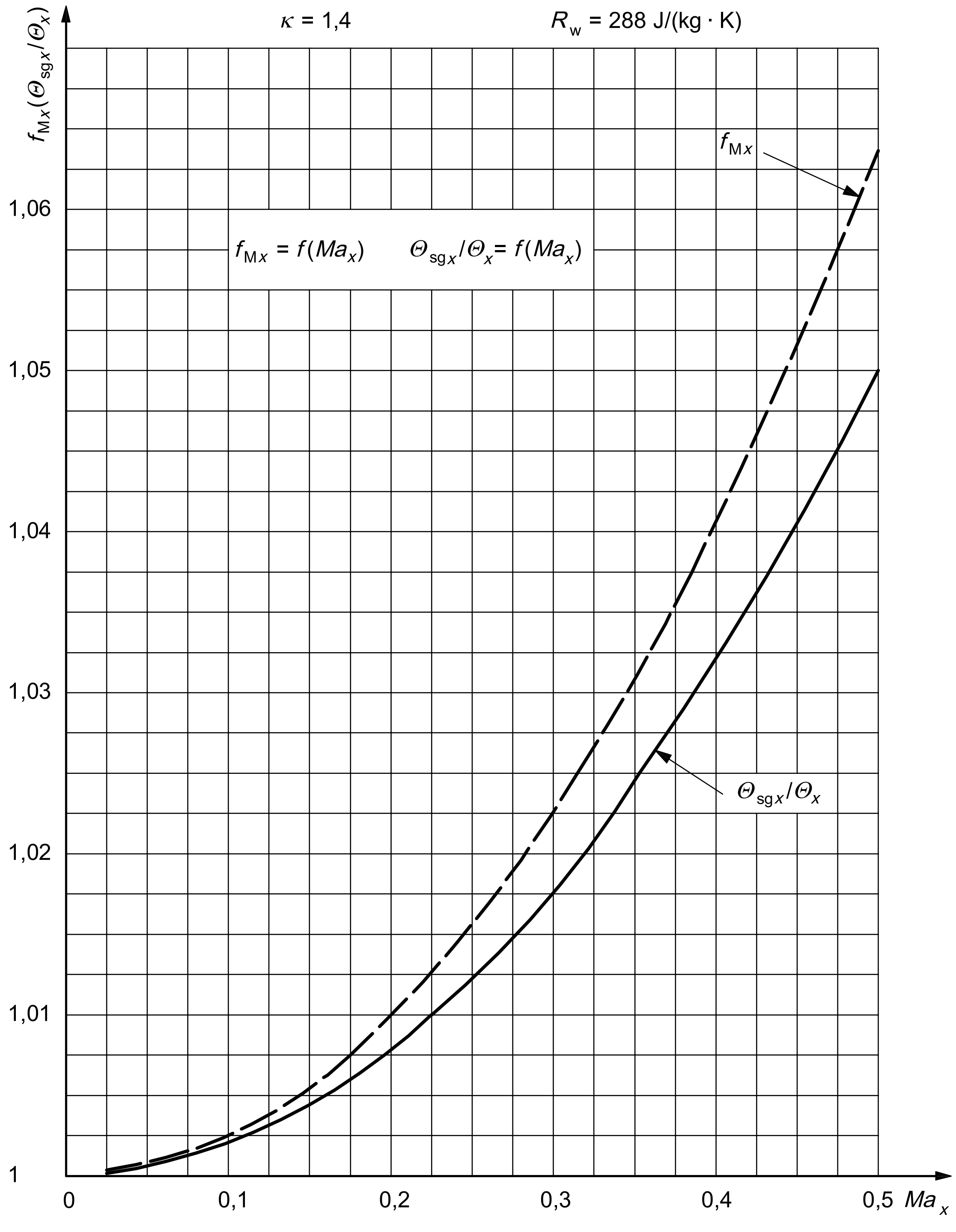
P_{r} should be replaced by the electric input power P_{e} when the motor is wholly immersed in the airstream.

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/s 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, $\Theta_{\mathrm{sg}x}$, as:

$$\frac{\Theta_{\mathrm{sg}x}}{\Theta_x} = 1 + \frac{\kappa - 1}{2} Ma_x^2$$

The ratio $\Theta_{\mathrm{sg}x}/\Theta_x$ is plotted in Figure 4 as a function of Ma_x.



Key

Ma_x Mach number at section x

f_{Mx} Mach factor

Θ_{sgx}/Θ_x temperature ratio

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

Because Ma_x is usually unknown, Θ_x shall be calculated from

— the mass flow rate, q_m,

— the stagnation temperature, $\Theta_{\mathrm{sg}x}$,

— the section area, A_x,

— the pressure, p_x, or the stagnation pressure at section x, $p_{\mathrm{sg}x}$, in accordance with 14.4.3.

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 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.

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, $Ma_{2\mathrm{ref}}$, is defined as:

$$Ma_{2\mathrm{ref}} = \frac{V_{\mathrm{m}2}}{c_{\mathrm{ref}}} = \frac{q_m}{A_2 \rho_{\mathrm{a}} \sqrt{\kappa R_{\mathrm{w}} \Theta_{\mathrm{sga}}}} = \frac{q_m}{A_2 \rho_{\mathrm{sg}1} \sqrt{\kappa R_{\mathrm{w}} \Theta_{\mathrm{sg}1}}}$$

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

This is defined as the mean velocity at section x, $v_{\mathrm{m}x}$, divided by the velocity of sound c_x at the same section, i.e.

$$Ma_x = \frac{v_{\mathrm{m}x}}{c_x} = \frac{q_m}{A_x \rho_x \sqrt{\kappa R_{\mathrm{w}} \Theta_x}}$$

where

$$\rho_x = \frac{p_x}{R_{\mathrm{w}} \Theta_x}$$

$$v_{\mathrm{m}x} = \frac{q_m}{A_x \rho_x}$$

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

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

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

The ratio Θ_{sgx}/Θ_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 $Ma_{sgx} < 0{,}45$

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

Figure 6 shows Ma_x/Ma_{sgx} plotted against Ma_{sgx}.

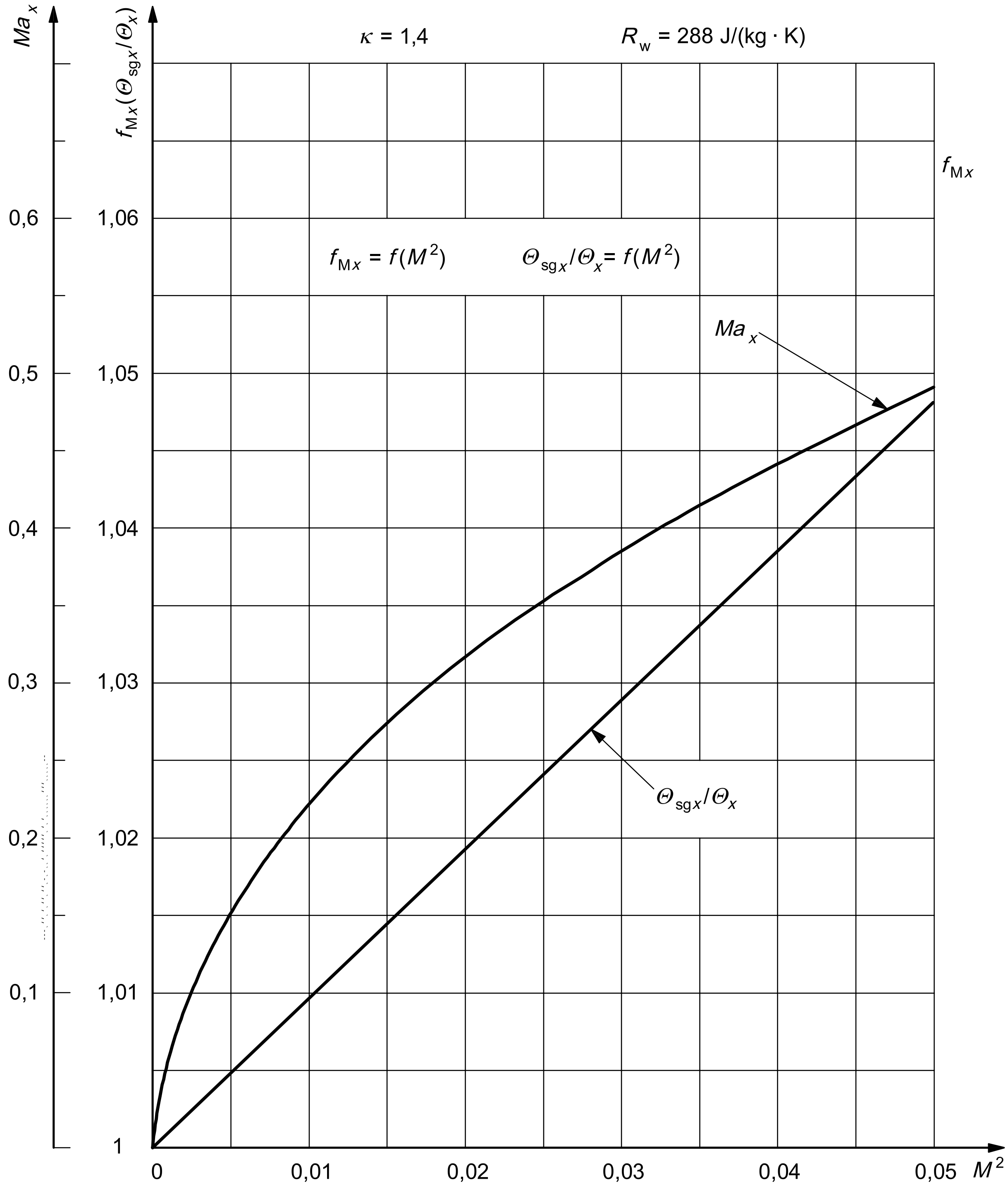


Figure 5 — Changes in Ma_x and the ratio Θ_{sgx}/Θ_x as functions of M^2

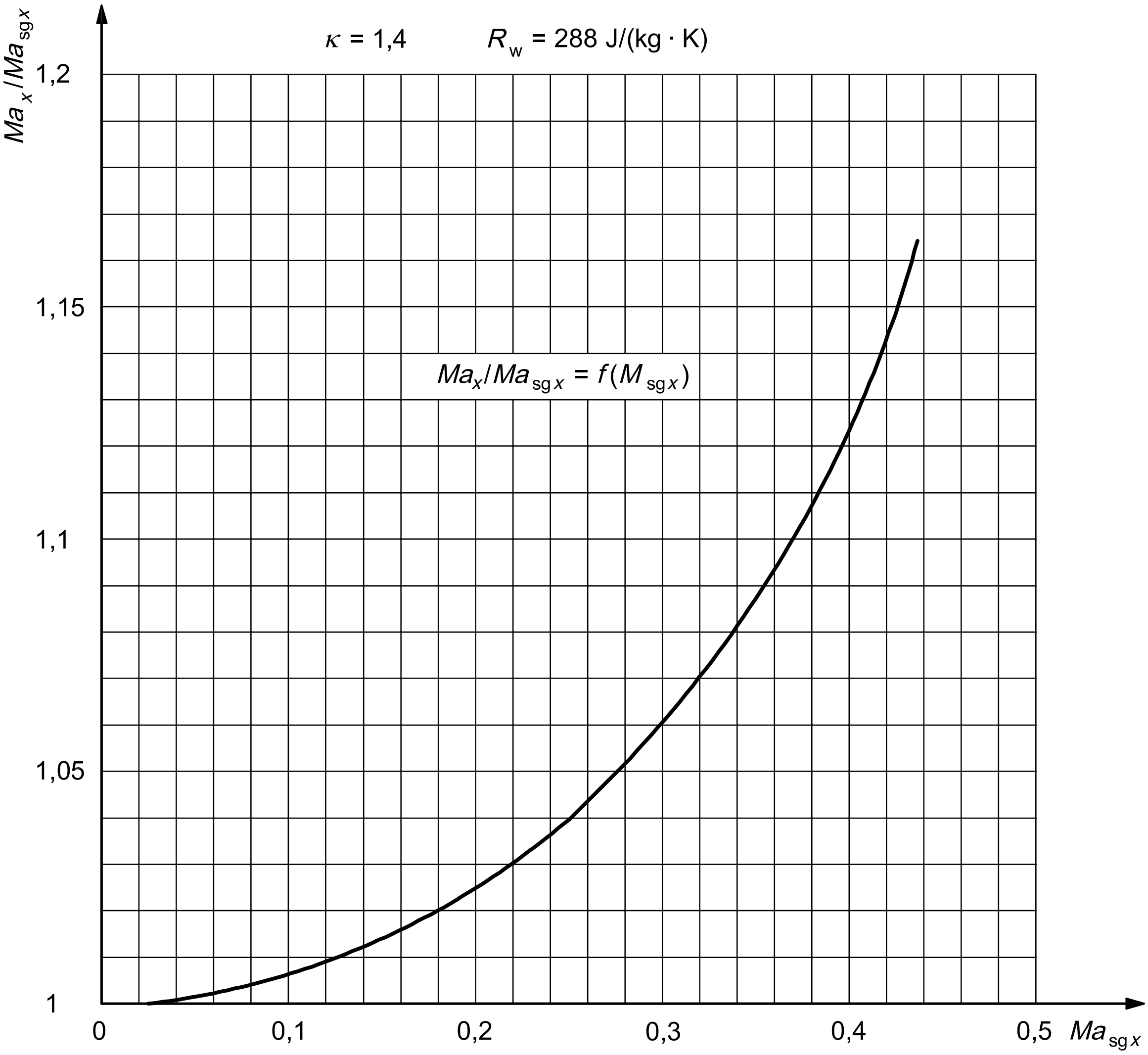


Figure 6 — Changes in the ratio Ma_x/Ma_{sgx} as a function of Ma_{sgx}

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 14.4.3.1 or 14.4.3.2, the ratio Θ_{sgx}/Θ_x is given by the following equation:

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

and

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

and

$$\frac{\rho_{\mathrm{sg}x}}{\rho_x} = \left(\frac{\Theta_{\mathrm{sg}x}}{\Theta_x}\right)^{\frac{1}{\kappa-1}}$$

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

$$v_{\mathrm{m}x} = \frac{q_m}{A_x \rho_x}$$

where

$$\rho_x = \frac{p_x}{R_\mathrm{w}\Theta_x} = \rho_{\mathrm{sg}x}\left(\frac{\Theta_{\mathrm{sg}x}}{\Theta_x}\right)^{\frac{-1}{k-1}} = \frac{p_{\mathrm{sg}x}}{R_\mathrm{w}\Theta_{\mathrm{sg}x}}\left(\frac{\Theta_{\mathrm{sg}x}}{\Theta_x}\right)^{\frac{-1}{\kappa-1}}$$

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.

$$p_\mathrm{f} = p_{\mathrm{sg}2} - p_{\mathrm{sg}1}$$

The stagnation pressure, $p_{\mathrm{sg}x}$, in any duct or chamber section x (with an area, A_x) is given by

$$p_{\mathrm{sg}x} = p_x + p_{\mathrm{d}x} f_{\mathrm{M}x}$$

where the conventional dynamic pressure, $p_{\mathrm{d}x}$, at section x is defined by

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

with

$$\rho_x = \frac{p_x}{R_\mathrm{w}\Theta_x}$$

The Mach factor, $f_{\mathrm{M}x}$, for pressure correction is given as a function of Ma_x by the equation

$$f_{\mathrm{M}x} = \frac{p_{\mathrm{sg}x} - p_x}{\frac{1}{2}\rho_x v_{\mathrm{m}x}^2} = 1 + \frac{Ma_x^2}{4} + \frac{Ma_x^4}{40} + \frac{Ma_x^6}{1\,600} + \ldots$$

for $\kappa = 1{,}4$ (see 3.4).

$f_{\mathrm{M}x}$ is plotted in Figure 4 as a function of Ma_x.

NOTE 1 The difference between the gauge stagnation pressure, $p_{\mathrm{esg}x}$, at section x of the test airway and the total pressure, $p_{\mathrm{t}x}$, used in earlier standards is very small at low velocities when the Mach number, Ma_x, is less than 0,15 (= 0,006 $p_{\mathrm{d}x}$).

NOTE 2 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.

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

where $p_\mathrm{e1} < 0$.

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.

$$p_\mathrm{sf} = p_2 - p_\mathrm{sg1}$$

When $p_{\mathrm{sg}x}$, $\Theta_{\mathrm{sg}x}$, q_m and A_x are known for a section x, p_x is calculated by the following method.

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

$$p_x = \frac{p_{\mathrm{sg}x}}{\left(1 + \dfrac{\kappa - 1}{2} Ma_x^2\right)^{\frac{1}{\kappa - 1}}} = p_{\mathrm{sg}x} \frac{p_x}{p_{\mathrm{sg}x}}$$

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

and is shown in Figure 7 as a function of Ma_x

and

$$p_x = p_{\mathrm{sg}x} - p_{\mathrm{d}x} f_{\mathrm{M}x} = p_{\mathrm{sg}x} - \frac{1}{2\rho_x}\left(\frac{q_m}{A_x}\right)^2 f_{\mathrm{M}x}$$

with $f_{\mathrm{M}x}$ being determined in accordance with 14.5.1 and Figure 4.

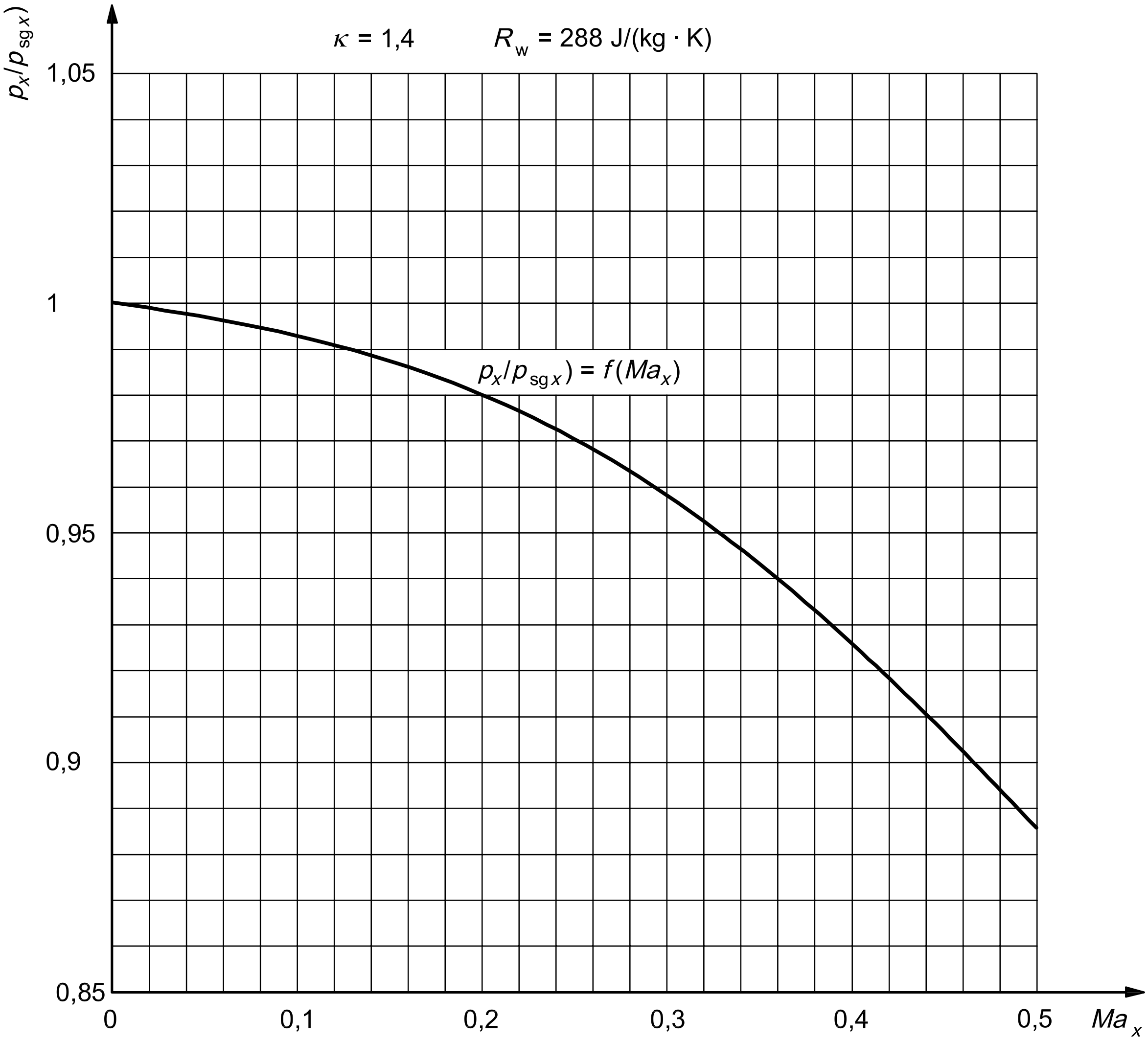


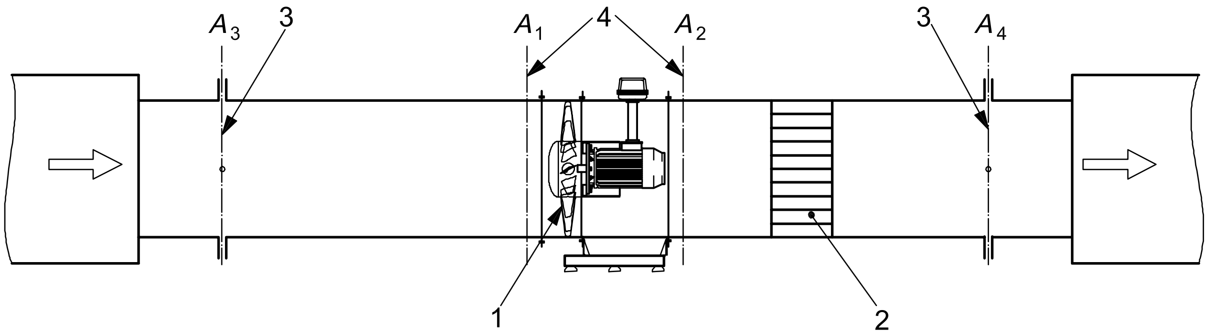
Figure 7 — Changes in the ratio p_x/p_{sgx} as a function of Ma_x

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

Assume that

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

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



Key

1 fan
2 flow straightener
3 static pressure measuring planes
4 reference planes

Figure 8 — Measurement and reference planes

The absolute pressure at section x is given by:

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

and, in accordance with 14.4.3.2,

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

Ma_x and Θ_x are calculated in accordance with 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

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

where

$(\xi_{n-x})_x$ is the energy loss coefficient between section n and section x calculated for section x in accordance with 28.6;

$(\xi_{n-x})_x > 0$ for an outlet test duct;

$(\xi_{n-x})_x < 0$ for an inlet test duct.

NOTE 1 p_{ex} is negative for an inlet test duct or an inlet chamber.

NOTE 2 It is possible to write:

$$p_{\mathrm{esg}n} = p_{\mathrm{e}x} + \frac{1}{2}\rho_x v_{\mathrm{m}x}^2 f_{\mathrm{M}x}\left[1 + \left(\xi_{n-x}\right)_x\right]$$

The fluid or static pressure in a reference section of the fan, p_{sf}, is calculated in accordance with 14.5.2 from $p_{\mathrm{sg}n}$, $\Theta_{\mathrm{sg}n}$ and A_n.

14.7 Inlet volume flow rate

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

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

$$q_{v\mathrm{sg1}} = \frac{q_m}{\rho_{\mathrm{sg1}}}$$

where

$$\rho_{\mathrm{sg1}} = \frac{p_{\mathrm{sg1}}}{R_{\mathrm{w}}\Theta_{\mathrm{sg1}}}$$

14.8 Fan air power and efficiency

Three methods are proposed:

— the first derived from the concept of work per unit mass;

— the two others based on the concepts of volume flow rate 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

We can say:

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

where

$$\rho_{\mathrm{m}} = \frac{\rho_1 + \rho_2}{2}$$

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 $q_m W_m$.

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 flow rate and fan pressure

We can say:

$$P_u = q_{V\mathrm{sg1}}\, p_f\, k_p$$

where

$q_{V\mathrm{sg1}}$ is the volume flow rate at inlet stagnation conditions;

p_f is the fan pressure, $p_{\mathrm{sg2}} - p_{\mathrm{sg1}}$;

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 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

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

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 function of the pressure ratio r and of the coefficient Z_k.

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

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

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

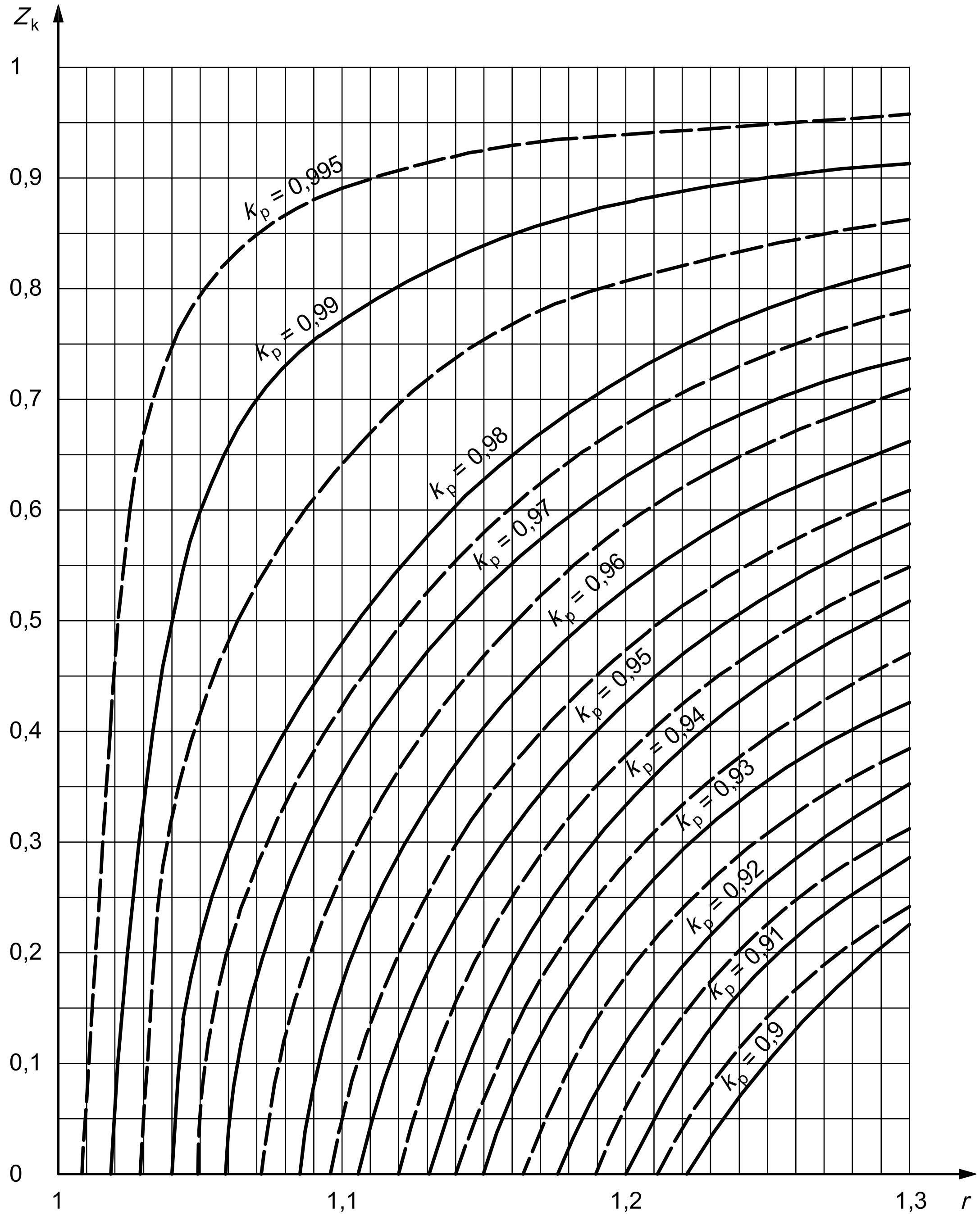
where

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

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.

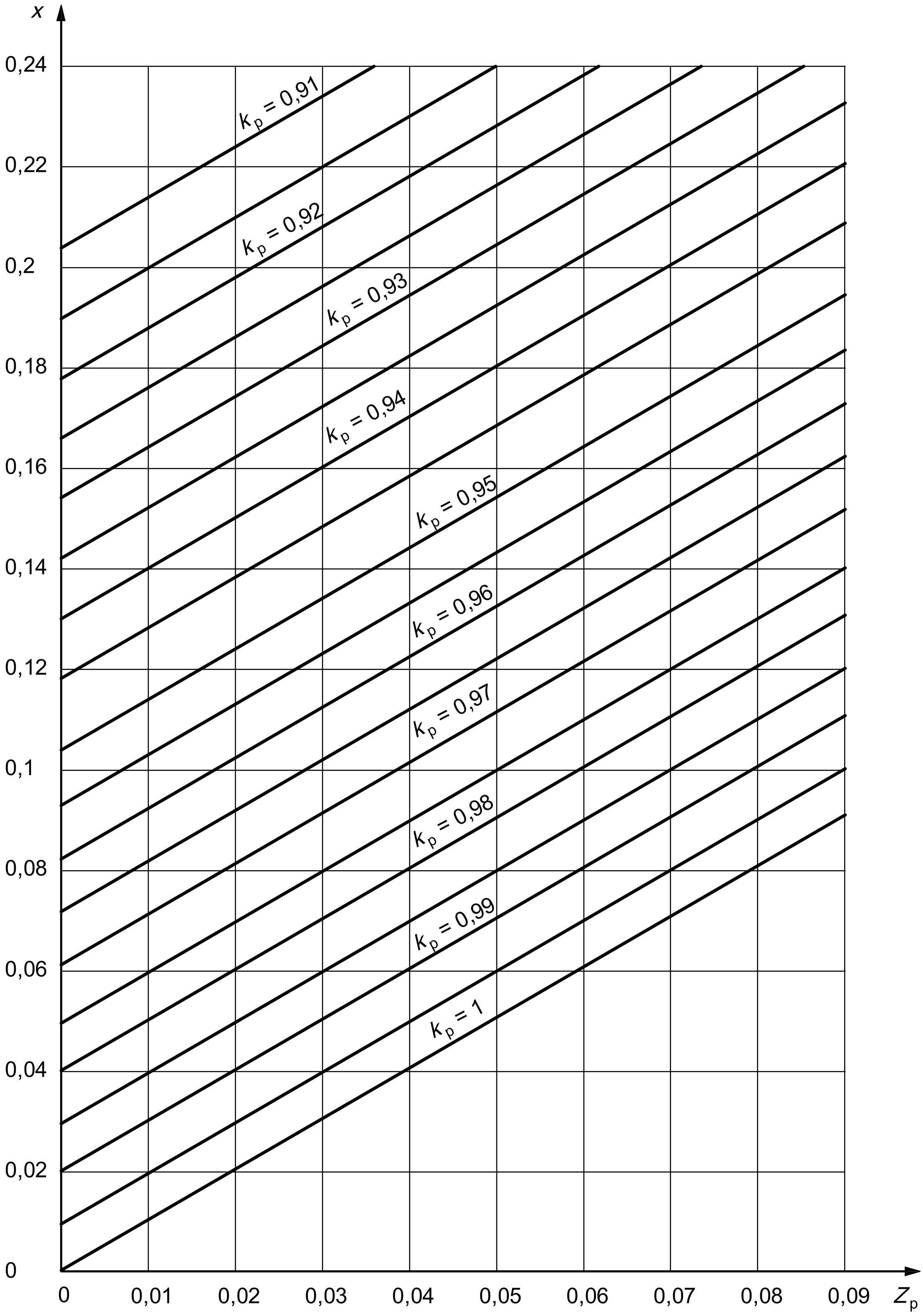


Key

r pressure ratio

Z_k compressibility factor, k_p, calculation coefficient (first method)

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



Key

Z_p compressibility factor, k_p, calculation coefficient (second method)

Figure 10 — Graph for the determination of the 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 W_m may be determined using the following equation:

$$W_m = \frac{P_{\mathrm{u}}}{q_m} = \frac{P_{\mathrm{u}}}{q_{V\mathrm{sg1}}\,\rho_{\mathrm{sg1}}}$$

where

$$P_{\mathrm{u}} = q_{V\mathrm{sg1}}\,p_{\mathrm{f}}\,k_{\mathrm{p}}$$

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

We can say:

$$W_{m\mathrm{s}} = \frac{p_2 - p_1}{\rho_{\mathrm{m}}} - \frac{v_{\mathrm{m1}}^2}{2}$$

where

$$\rho_{\mathrm{m}} = \frac{\rho_1 + \rho_2}{2}$$

The fan static air power is equal to the product $q_m W_{m\mathrm{s}}$ therefore

$$P_{\mathrm{us}} = q_m W_{m\mathrm{s}}$$

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 flow rate and fan static pressure

The fan static power is given by the following equation:

$$P_{\mathrm{us}} = q_{V\mathrm{sg1}}\,p_{\mathrm{sf}}\,k_{\mathrm{ps}}$$

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

and

$$r = 1 + \frac{p_{\mathrm{sf}}}{p_{\mathrm{sg1}}}$$

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

and

$$Z_{\mathrm{k}} = \frac{\kappa - 1}{\kappa}\,\frac{\rho_{\mathrm{sg1}} P_{\mathrm{r}}}{q_m\,p_{\mathrm{sf}}}$$

and

$$Z_{\mathrm{p}} = \frac{\kappa - 1}{\kappa} \frac{P_{\mathrm{r}}}{q_{V\mathrm{sg1}}\, p_{\mathrm{sg1}}}$$

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

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

14.8.3.3 Determination of the fan static work per unit mass, $W_{m\mathrm{s}}$, from the fan static air power, P_{us}

The fan static work per unit mass, $W_{m\mathrm{s}}$, is determined using the following equation:

$$W_{m\mathrm{s}} = \frac{P_{\mathrm{us}}}{q_m} = \frac{P_{\mathrm{us}}}{q_{V\mathrm{sg1}}\, \rho_{\mathrm{sg1}}}$$

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

The kinetic index, $i_{\mathrm{k}x}$, is given by the following equations:

— at the fan inlet:

$$i_{\mathrm{k1}} = \frac{v_{\mathrm{m1}}^2}{2W_{m\mathrm{s}}}$$

— at the fan outlet:

$$i_{\mathrm{k2}} = \frac{v_{\mathrm{m2}}^2}{2W_{m\mathrm{s}}}$$

14.8.5 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 $f_{\mathrm{M}x}$ 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:

$$\Theta_1 = \Theta_{\mathrm{sg1}} = \Theta_2 = \Theta_{\mathrm{sg2}} = \Theta_3 = \Theta_{\mathrm{sg3}} = \Theta_{\mathrm{u}} = \Theta_{\mathrm{a}} = T_{\mathrm{a}} + 273{,}15$$

— 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 the planes labelled 3 and 4 in Figure 8.

14.8.5.1 Determination of mass flow rate

$$p_{\mathrm{u}} = p_{\mathrm{eu}} + p_{\mathrm{a}}$$

$$\rho_{\mathrm{u}} = \frac{p_{\mathrm{u}}}{R_{\mathrm{w}} \Theta_{\mathrm{u}}}$$

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

14.8.5.2 Determination of the stagnation pressure at section x, $p_{\mathrm{sg}x}$

$$p_x = p_{\mathrm{e}x} + p_{\mathrm{a}}$$

$$p_{\mathrm{sg}x} = p_x + \frac{1}{2}\rho_1 v_{\mathrm{m}x}^2 = p_x + \frac{1}{2\rho_1}\left(\frac{q_m}{A_x}\right)^2$$

or

$$p_{\mathrm{esg}x} = p_{\mathrm{e}x} + \frac{1}{2\rho_1}\left(\frac{q_m}{A_x}\right)^2$$

where

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

$$\rho_1 = \frac{p_{\mathrm{a}}}{R_{\mathrm{w}} \Theta_{\mathrm{sg1}}} = \frac{p_{\mathrm{a}}}{R_{\mathrm{w}} \Theta_{\mathrm{a}}} = \rho_{\mathrm{a}}$$

with an auxiliary fan upstream of the test fan,

$$\rho_1 = \rho_2 = \rho_3 = \rho_4 = \frac{p_3}{R_{\mathrm{w}}\, \Theta_3}$$

14.8.5.3 Determination of the stagnation pressure at a reference section of the fan from the gauge pressure measured at section x, $p_{\mathrm{e}x}$

In accordance with 14.8.5

$$p_{\mathrm{sg}n} = p_x + \frac{1}{2}\rho_1 v_{\mathrm{m}x}^2 \left[1 + \left(\xi_{n-x}\right)_x\right]$$

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

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

$$p_{\mathrm{esg}n} = p_{\mathrm{e}x} + \frac{1}{2\rho_1}\left(\frac{q_m}{A_x}\right)^2 \left[1 + \left(\xi_{n-x}\right)_x\right]$$

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

In accordance with 14.8.5 and 14.8.5.2

$$p_n = p_{\mathrm{sg}n} \frac{1}{2} \rho_1 v^2_{\mathrm{m}n}$$

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

which may also be written

$$p_{\mathrm{e}n} = p_{\mathrm{esg}n} - \frac{1}{2\rho_1} \left(\frac{q_m}{A_x} \right)^2 \left(\frac{A_x}{A_n} \right)^2 = p_{\mathrm{esg}n} - \frac{1}{2\rho_1} \left(\frac{q_m}{A_n} \right)^2$$

14.8.5.5 Calculation of fan pressure

The fan pressure, p_{f}, and the fan static pressure, p_{sf}, are given by the following equations:

$$p_{\mathrm{f}} = p_{\mathrm{sg2}} - p_{\mathrm{sg1}} = p_{\mathrm{esg2}} - p_{\mathrm{esg1}}$$

$$p_{\mathrm{sf}} = p_2 - p_{\mathrm{sg1}} = p_{\mathrm{e2}} - p_{\mathrm{esg1}}$$

14.8.5.6 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 equations:

$$P_{\mathrm{u}} = q_{V\mathrm{sg1}}\, p_{\mathrm{f}}$$

$$P_{\mathrm{us}} = q_{V\mathrm{sg1}}\, p_{\mathrm{sf}}$$

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

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 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.

15.1.2 Reynolds number similarity

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

$$Re_{\mathrm{u}} = \frac{u D_{\mathrm{r}}\, \rho_{\mathrm{sg1}}}{\mu} = \frac{u D_{\mathrm{r}}\, p_{\mathrm{sg1}}}{\mu R_{\mathrm{w}}\, \Theta_{\mathrm{sg1}}}$$

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.

15.1.3 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

$$Ma_{\mathrm{u}} = \frac{v_{\mathrm{p}}}{\sqrt{\kappa R_{\mathrm{w}}\, \Theta_{\mathrm{sg1}}}}$$

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 the 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 There are never shock waves in fans: Ma, 0,7.

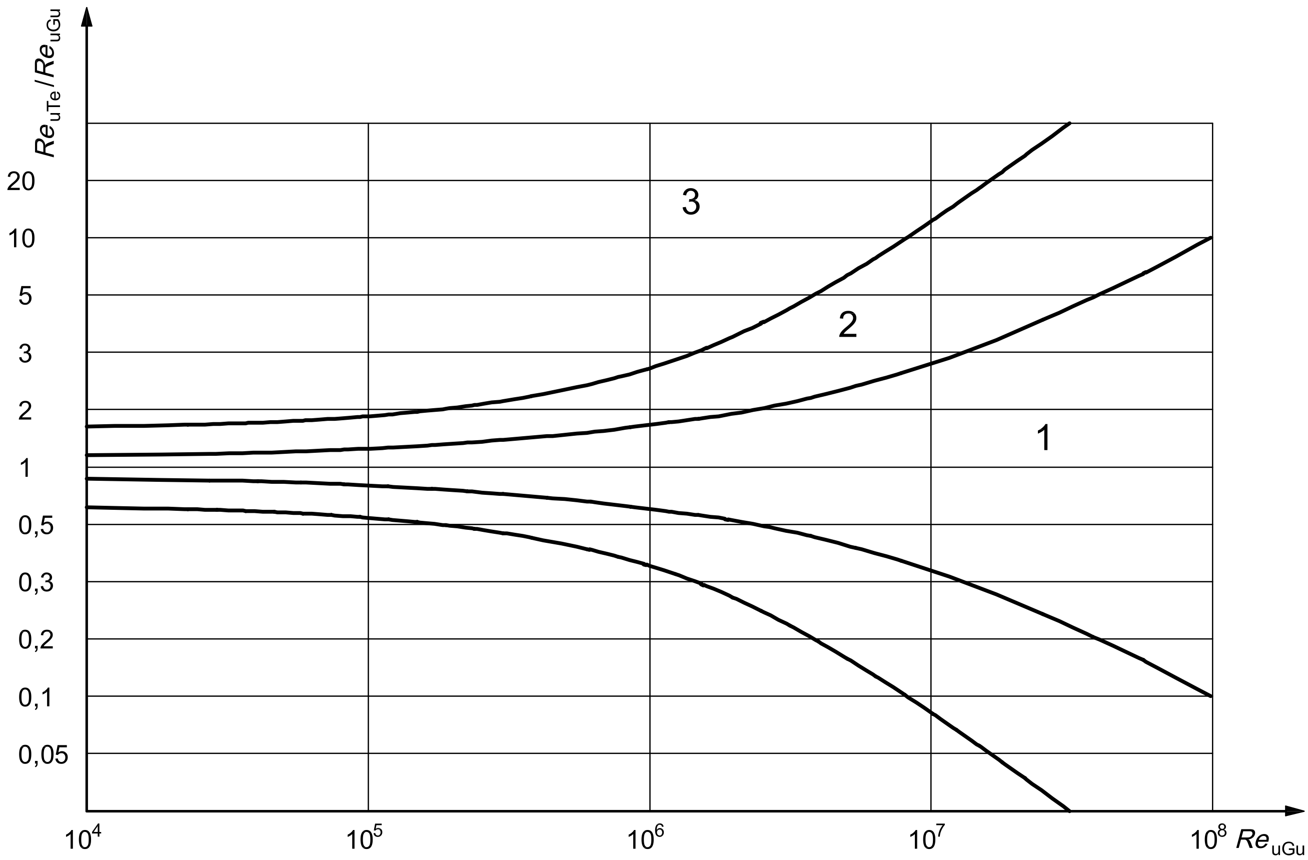
15.2 Conversion rules

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, Re_{uTe}/Re_{uGu}, as a function of Re_{uGu}, and Figure 12 gives an indication of the variations of the ratio, n_{Gu}/n_{Te}, as a function of k_{pGu} and Δk_p,

where

$$\Delta k_p = k_{pGu} - k_{pTe}$$



Key

1 permissible zone

2 limit zone

3 unacceptable zone

Figure 11 — Permissible variations of Re_{uTe}/Re_{uGu} as function of Re_{uGu}

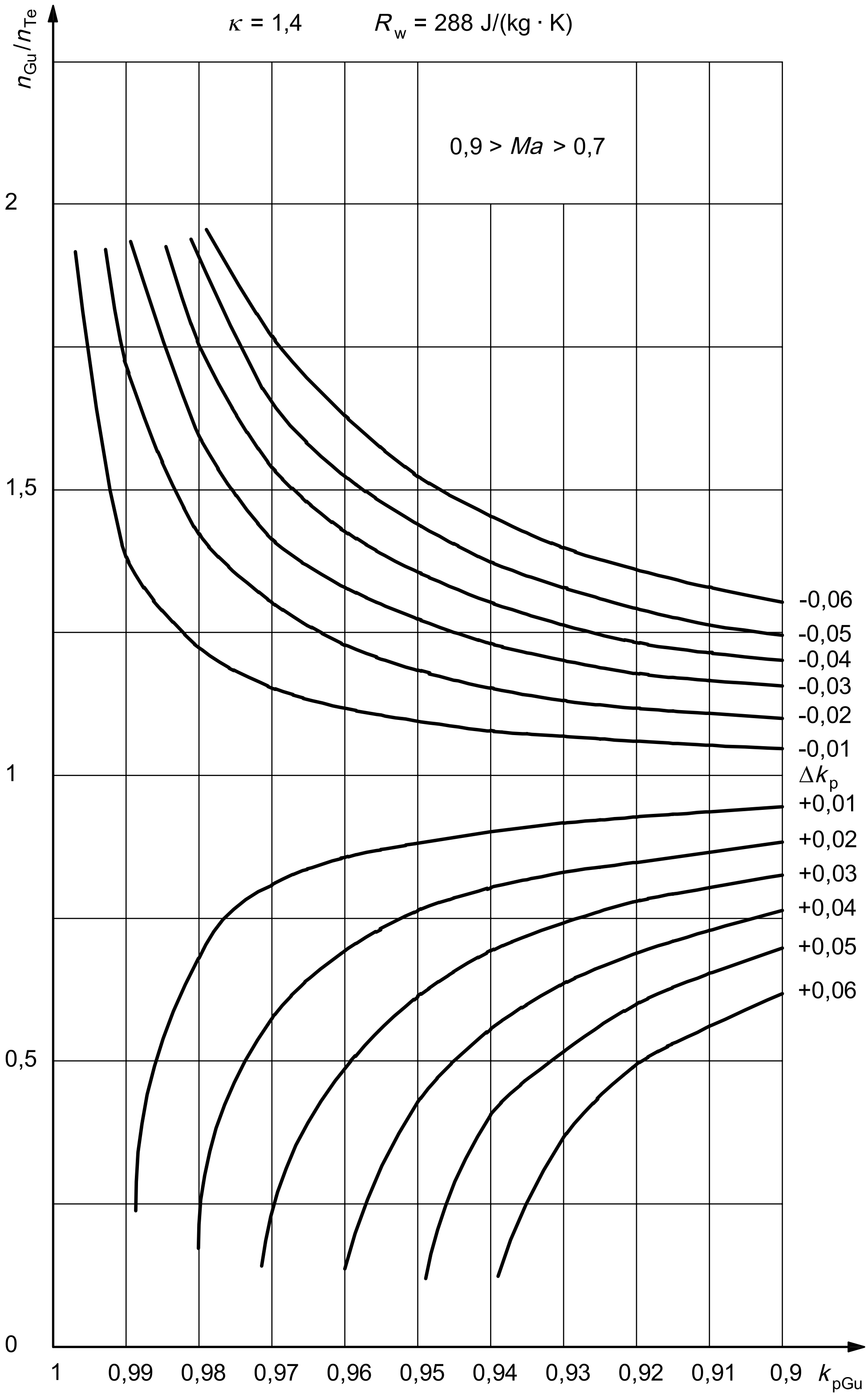


Figure 12 — Variation of n_{Gu}/n_{Te} as a function of k_{pTe} and Δk_p

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 $\pm$ 0,01 and which may be as great as 0,06.

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

These conversion rules can be represented by the following equations, in which q is an exponent which may vary from one design to another. Values from 0 to –0,5 have been demonstrated, but it is the responsibility of the manufacturer to test sufficient sizes and types to determine the exact value.

A type-test is recommended (which may be at model scale) to determine the range of pressure ratio, r, and the range of fan characteristics 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 coefficients k_{pGu} and k_{psGu} after conversion may be found from the following approximate equations, which are correct to within a few parts per 1 000:

$$\frac{1 - k_{pGu}}{1 - k_{pTe}} = \left(\frac{n_{Gu}\, D_{rGu}}{n_{Te}\, D_{rTe}}\right)^2 \left(\frac{R_{wTe}\, \Theta_{sg1\,Te}}{R_{wGu}\, \Theta_{sg1\,Gu}}\right) \frac{\kappa_{Te}}{\kappa_{Gu}} \left[\frac{1-\kappa_{Gu}\left(1-\eta_r\right)}{1-\kappa_{Te}\left(1-\eta_r\right)}\right] = k_c{}^2$$

$$\frac{1 - k_{psGu}}{1 - k_{psTe}} = \left(\frac{n_{Gu}\, D_{rGu}}{n_{Te}\, D_{rTe}}\right)^2 \left(\frac{R_{wTe}\, \Theta_{sg1\,Te}}{R_{wGu}\, \Theta_{sg1\,Gu}}\right) \frac{\kappa_{Te}}{\kappa_{Gu}} \left[\frac{1-\kappa_{Gu}\left(1-\eta_{sr}\right)}{1-\kappa_{Te}\left(1-\eta_{sr}\right)}\right] = k_{cs}{}^2$$

where η is η_r or η_{sr}.

The fan performance after conversion may then be found by the following equations:

$$\frac{q_{Vsg1\,Gu}}{q_{Vsg1\,Te}} = \frac{n_{Gu}}{n_{Te}} \left(\frac{D_{rGu}}{D_{rTe}}\right)^3 \left(\frac{k_{pGu}}{k_{pTe}}\right)^q$$

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

$$\frac{p_{sfGu}}{p_{sfTe}} = \left(\frac{n_{Gu}}{n_{Te}}\right)^2 \left(\frac{D_{rGu}}{D_{rTe}}\right)^2 \left(\frac{\rho_{sg1Gu}}{\rho_{sg1Te}}\right) \left(\frac{k_{psGu}}{k_{psTe}}\right)^{-1}$$

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

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 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, and for incompressible flow, the simplified conversion rules may be used as detailed in 15.2.2.

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.

$$\frac{q_{V\mathrm{sg1\,Gu}}}{q_{V\mathrm{sg1\,Te}}} = \left(\frac{n_{\mathrm{Gu}}}{n_{\mathrm{Te}}}\right)\left(\frac{D_{\mathrm{rGu}}}{D_{\mathrm{rTe}}}\right)^3$$

$$\frac{p_{\mathrm{fGu}}}{p_{\mathrm{fTe}}} = \left(\frac{n_{\mathrm{Gu}}}{n_{\mathrm{Te}}}\right)^2\left(\frac{D_{\mathrm{rGu}}}{D_{\mathrm{rTe}}}\right)^2\left(\frac{\rho_{\mathrm{sg1Gu}}}{\rho_{\mathrm{sg1Te}}}\right) = \frac{p_{\mathrm{sfGu}}}{p_{\mathrm{sfTe}}}$$

$$\frac{P_{\mathrm{rGu}}}{P_{\mathrm{rTe}}} = \left(\frac{n_{\mathrm{Gu}}}{n_{\mathrm{Te}}}\right)^3\left(\frac{D_{\mathrm{rGu}}}{D_{\mathrm{rTe}}}\right)^5\left(\frac{\rho_{\mathrm{sg1\,Gu}}}{\rho_{\mathrm{sg1\,Te}}}\right)$$

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

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

and

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

in order to carry out the conversion specified in 15.2.

However, the error incurred by assuming

$$\frac{P_{\mathrm{rGu}}}{P_{\mathrm{rTe}}} = \frac{P_{\mathrm{aGu}}}{P_{\mathrm{aTe}}}$$

will not exceed the following, as a percentage,

$$\frac{200\left(n_{\mathrm{Gu}} - n_{\mathrm{Te}}\right) P_{\mathrm{b}}}{n_{\mathrm{Te}}\, P_{\mathrm{a}}}$$

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.

Graphs 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 $\rho_2/\rho_{\mathrm{sg1}}$ or $k_\rho = \rho_1/\rho_m$.

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}.

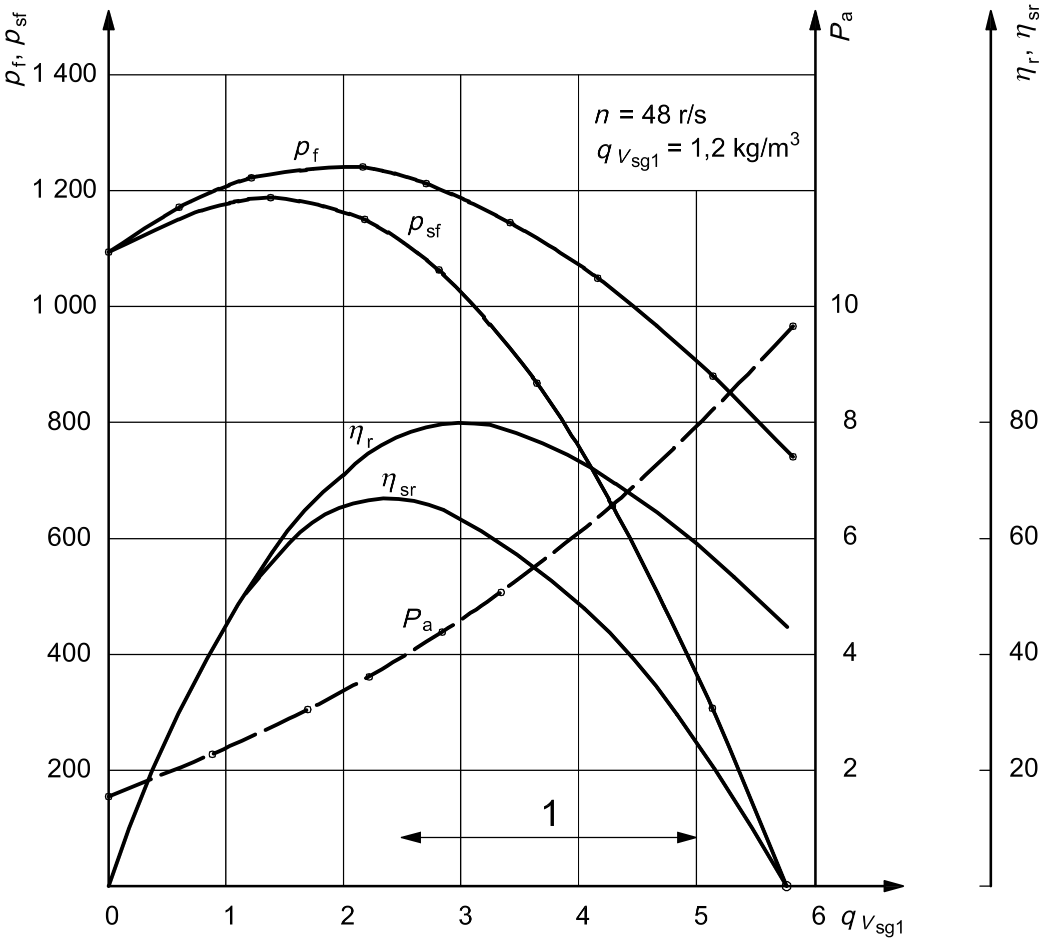
The fan pressure, p_{f}, and the fan static pressure, p_{sf}, or either one of them together with the fan dynamic pressure corrected for the Mach number effect, $p_{\mathrm{d2}}f_{\mathrm{M2}}$, shall be plotted against the inlet volume flow rate, $q_{V\mathrm{sg1}}$. The fan efficiency, η_{r}, and/or 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 flow rate. 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.



Key

1 working range

P_a fan shaft power, in kilowatts

p_f fan pressure, in pascals

p_{sf} fan static pressure, in pascals

$q_{V\text{sg1}}$ fan inlet volume flow rate, in cubic metres per second

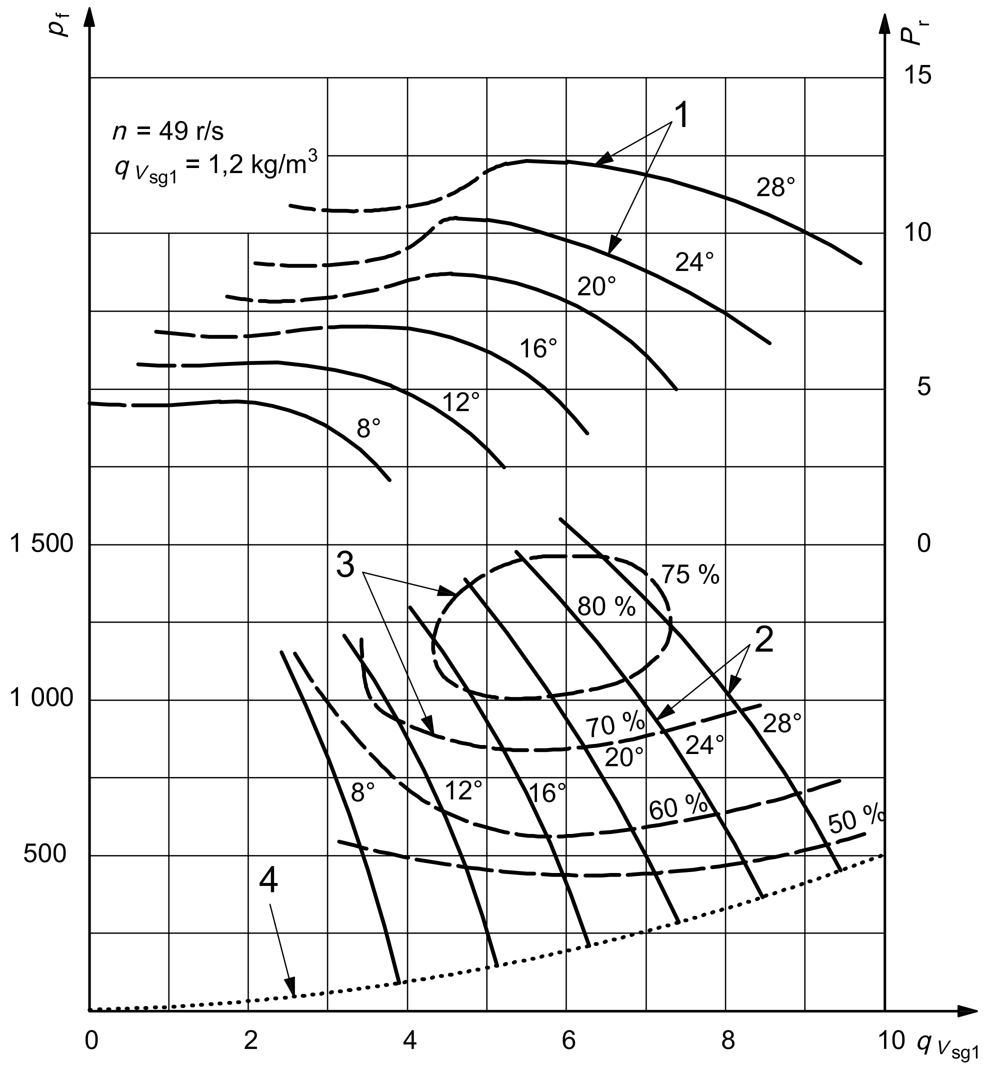
η_r fan efficiency, as a percentage

η_{sr} fan static efficiency, as a percentage

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

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 flow rates. 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.



Key

1 impeller power curves at different blade pitch angles

2 fan pressure volume curves at different blade pitch angles

3 fan total efficiency curves, η_r

4 fan dynamic pressure at outlet, p_{d2}

p_f fan pressure, in pascals

P_r impeller power, in kilowatts

q_{Vsg1} fan inlet volume flow rate, in cubic metres per second

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

16.6 Complete fan characteristic curve

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

Only part of this curve is normally used however, and it is recommended that the supplier should state the range of inlet volume flow rates 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 flow rates, the uncertainty of measurement is liable to increase and unsatisfactory flow patterns may develop at inlet or outlet.

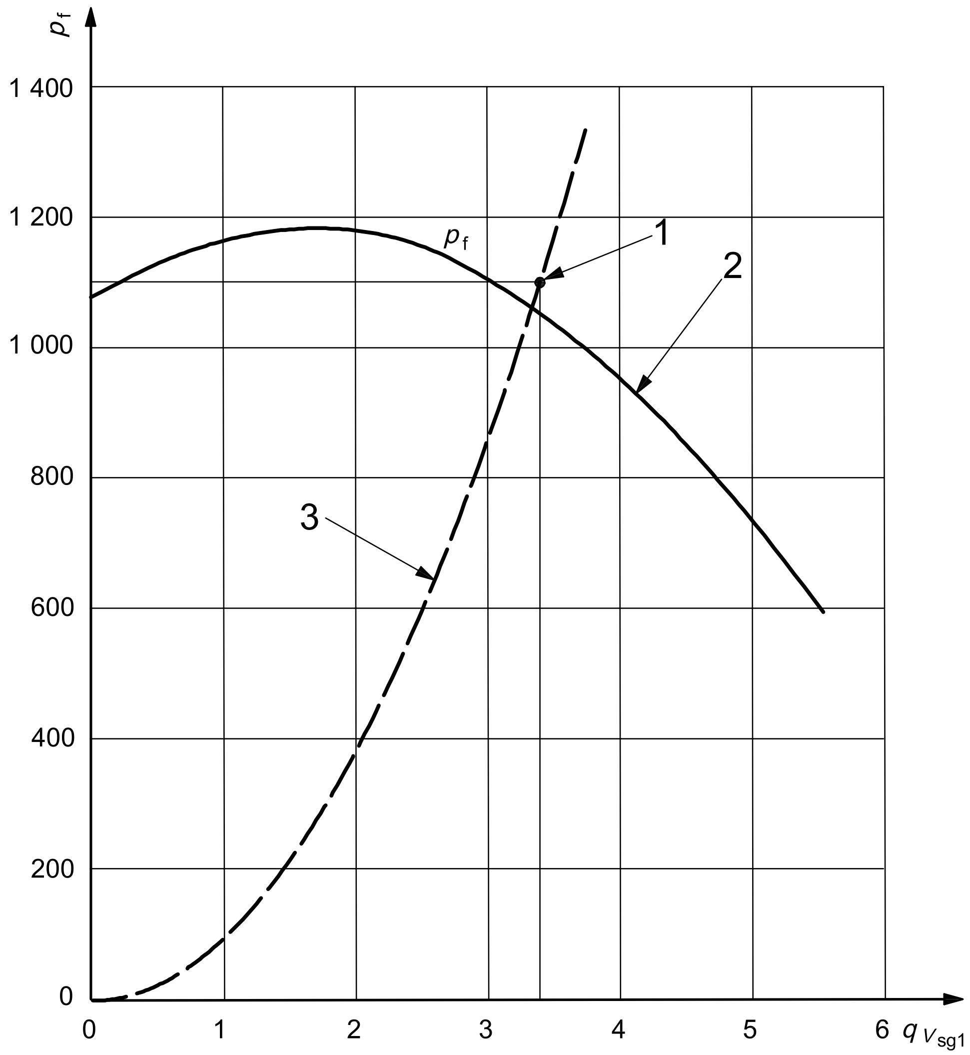
16.7 Test for a specified duty

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 flow rate 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 flow rate (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.

Deviations or tolerances should be determined in accordance with the planned International Standard concerning fan tolerances.



Key

1 specified duty: 3,4 m^3/s at 1 100 Pa

2 fan pressure volume characteristic curve

3 system resistance curve, $p_f \propto q^2_{Vsg1}$

p_f fan pressure, in pascals

q_{Vsg1} fan inlet volume flow rate, in cubic metres per second

Figure 15 — Example of test for a specified duty

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 flow rate 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.

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 (RSS) 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 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 of the parameter;

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

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) $R = q_V = 5\ \text{m}^3/\text{s}$

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

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

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

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

then

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

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 the 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.

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 (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 flow rate	q_m	u_{qm}		22 to 25

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 requirements of 17.6.

The uncertainties in Table 3 are based on the 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.

Table 3 — Maximum allowable uncertainty for the results

Parameter	Symbol	Relative uncertainty	Remarks
Ambient density	ρ_a	$u_{\rho a} = \pm 0{,}4$ %	$\sqrt{u_{\Theta a} + u_{hu} + u_{pa}^2}$
Fan temperature rise	$\Delta\Theta$	$u_{\Delta\Theta} = \pm 2{,}8$ %	$\sqrt{u_{pr} + u_{qm}^2}$
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}	$u_{p\,d2} = \pm 4$ %	$\sqrt{4u_{qm} + 4u_A + u_{\rho 2}^2}$
Fan pressure	p_f	$u_{pf} = \pm 1{,}4$ %	$= u_{pe}$
Fan air power	P_u	$u_{Pu} = \pm 2{,}5$ %	$\sqrt{u_{qm} + u_{p\,f}^2}$
Fan efficiency	η_r	$u_{\eta r} = \pm 3{,}2$ %	$\sqrt{u_{Pu} + u_{Pr}^2}$
Fan flow rate	q_m or q_V	u_{qm} or $u_{qV} = \pm 2$ %	See individual clauses for various flow-measurement methods

18 Selection of test method

18.1 Classification

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

18.2 Installation categories

The four categories of installation are as follows:

— category A: free inlet, free outlet;

— category B: free inlet, ducted outlet;

— category C: ducted inlet, free outlet;

— category 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 into 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.

18.3 Test report

All references to fan performance stated to be in accordance with this International Standard shall also state the installation category to which they refer. This is because a fan adaptable for use in all four installation categories 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 30 to 33 shall also be stated, but this is not necessary for catalogue data or contracts of sale since the alternative methods permissible within each installation category may be expected to give results falling within the uncertainty of measurement.

18.4 User installations

In selecting a category 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 28.3) 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 $2D$ or $3D$ 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 category, the alternative methods available differ only in the method of flow rate measurement and control. The relative merits of nozzle, orifice and traverse methods of flow rate 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 30, 31, 32 and 33 and Figures 40 to 46.

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 28.

Standardized airways designed for category A installation tests may be adapted to provide tests for category B, C or D installations. It follows that the inlet-side or outlet-side test chambers described in Clause 29, 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 category B or C installation tests may be adapted to provide tests for category 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/s. 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 $2D$ from the inlet and $5D$ on the 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 substantially 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 flow rate 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 flow rate.

20.7 Tests for a fan characteristic curve

Tests 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 flow rate

Four methods of flow rate measurement are listed in 21.1 to 21.4 and described in Clauses 22 to 25.

21.1 Multiple nozzle

— Multiple nozzle in test chamber

21.2 Conical or bellmouth inlet

21.3 Orifice plate

— Inlet orifice plate

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

— Outlet orifice plate

— Orifice plate in chamber

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

22 Determination of flow rate using multiple nozzles

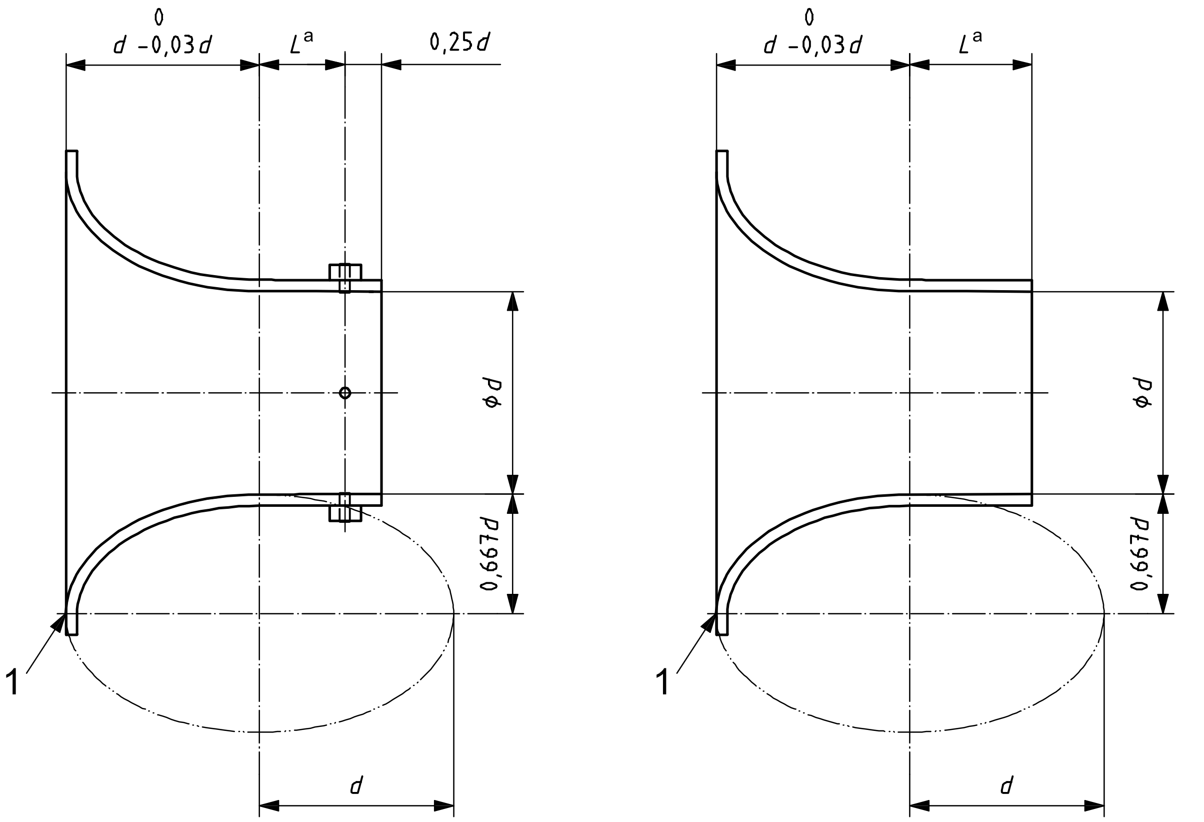
22.1 Installation

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.

22.2 Geometric form

22.2.1 Multiple-nozzle dimensions and tolerances are shown in Figure 16.

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 $0{,}5d \pm 0{,}005d$.



a) Nozzle with throat-pressure tappings **b) Nozzle without throat-pressure tappings**

Key

1 fairing radius approximately $0{,}05d$, if necessary

[a] Length L should be $0{,}5d$ or $0{,}6d$; $0{,}6d$ is recommended for new constructions.

Figure 16 — Nozzle geometry

22.2.2 Nozzles shall have an elliptical form as shown in Figure 16, 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.

22.2.3 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.

22.2.4 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.

22.2.5 Where nozzles are used in a chamber, the type shown in Figure 16 shall be used.

22.3 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.

22.4 Multiple-nozzle characteristics

22.4.1 A multiple-nozzle installation manufactured in accordance with the requirements in 22.3 may be used uncalibrated for pressure ratios r_{d}·0,9 (i.e. Δp, 10 kPa).

22.4.2 The nozzle flow rate coefficient, α, is obtained from Table 4 or may be calculated by the following equations:

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

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

or

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

for $L/d = 0{,}5$

where

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

$$Re_d = 0{,}95\varepsilon d \frac{\sqrt{2\rho_{\mathrm{u}}\,\Delta p}}{\left(17{,}1 + 0{,}048T_{\mathrm{u}}\right)} \times 10^6$$

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) ($\beta <$ 0,525 for an in-duct nozzle);

C is the nozzle discharge coefficient.

Table 4 — Flow rate coefficients for nozzles used in a chamber

Nozzle flow rate coefficient	Reynolds number, Re_d		Nozzle flow rate coefficient	Reynolds number, Re_d	
α	$L/d = 0{,}5$	$L/d = 0{,}6$	α	$L/d = 0{,}5$	$L/d = 0{,}6$
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			

22.4.3 The expansibility factor is obtained from Table 5 or may be calculated from:

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

where

$$r_\mathrm{d} = \frac{p_\mathrm{u} - \Delta p}{p_\mathrm{u}} = 1 - \frac{\Delta p}{p_\mathrm{u}}$$

This expression may be replaced by:

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

Table 5 — 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

22.4.4 The mass flow rate for a multiple nozzle is given by:

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

and, for a Venturi nozzle, by:

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

where

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

ρ_{u} is the upstream density.

22.5 Uncertainty

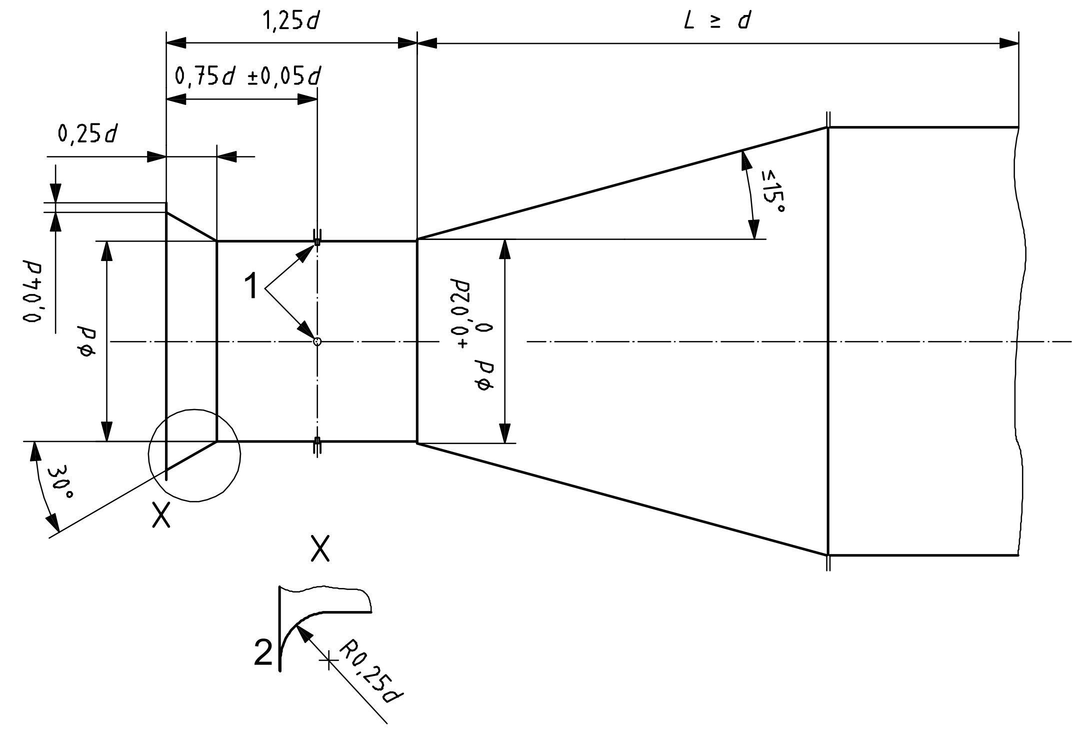
The uncertainty in the discharge coefficient C is ±1,2 % for $Re_d > 1{,}2 \times 10^4$.

23 Determination of flow rate using a conical or bellmouth inlet

The conical or bellmouth inlet shall only be used when drawing air from an open (free) space.

23.1 Geometric form

23.1.1 The conical or bellmouth inlet dimensions and tolerances are given in Figure 17. 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.



Key

1 four wall pressure tappings

2 alternative bellmouth inlet

NOTE The four wall pressure tappings are as specified in Clause 7.

Figure 17 — Geometry of conical or bellmouth inlet

23.1.2 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.

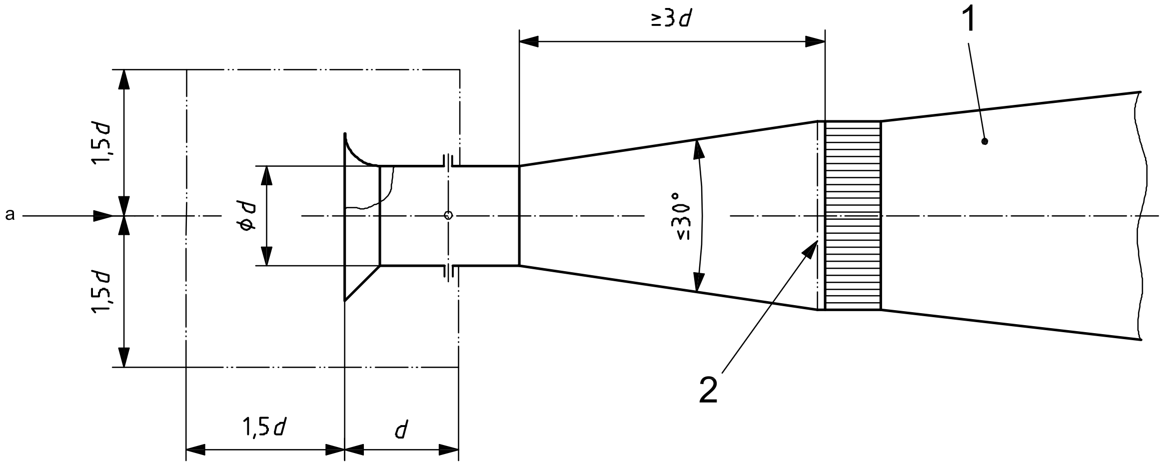
23.1.3 The pressure tappings shall conform to the requirements of Clause 7.

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

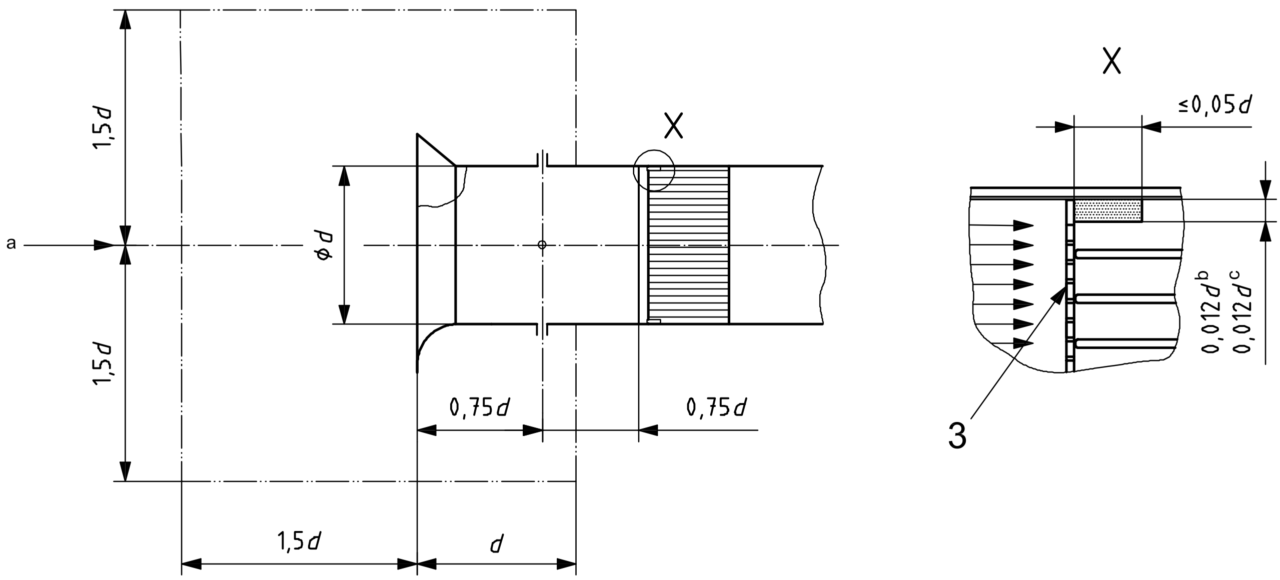
23.1.5 Except where otherwise specified, the included angle of the divergent section may lie anywhere in the range $\theta < 30°$. The divergent or cylindrical connection piece shall be not less than 3d long.

23.2 Screen loading

23.2.1 Adjustable screen loading in accordance with Figure 18 is permissible with the conical or bellmouth inlet, but the uncertainty of the flow rate coefficient α is increased (see 23.6.3).



a) Conical or bellmouth inlet



b) Conical or bellmouth inlet with adjustable screen loading

Key

1 duct expander, shape transition, sudden expansion

2 resistance screen, if required

3 screen loading and support-ring in accordance with Clause 22

[a] The inlet zone shall be clear from obstruction.

[b] = 6 mm.

[c] = 3 mm.

Figure 18 — Conical or bellmouth inlet flow-metering installations

23.2.2 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.

23.2.3 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 than 2 %. The supports shall ensure that the screens are not allowed to bow in the middle.

NOTE An antiswirl device makes an excellent screen support, see Figure 18 b).

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 to eliminate leakage at the wall.

23.3 Inlet zone

23.3.1 Within the inlet zone defined in Figure 18, 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.

23.3.2 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.

23.4 Conical inlet performance

23.4.1 A conical or bellmouth inlet manufactured in accordance with the above requirements may be used uncalibrated for pressure ratios $r_{\mathrm{d}} > 0{,}96$, i.e. $\Delta p < 4\,000$ Pa.

23.4.2 The compound coefficient $\alpha\varepsilon$ is dependent on the Reynolds number Re_d and is plotted in Figure 19.

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

23.4.3 The mass flow rate is given by the following equation:

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

where ρ_{u} is the upstream density.

23.5 Bellmouth inlet performance

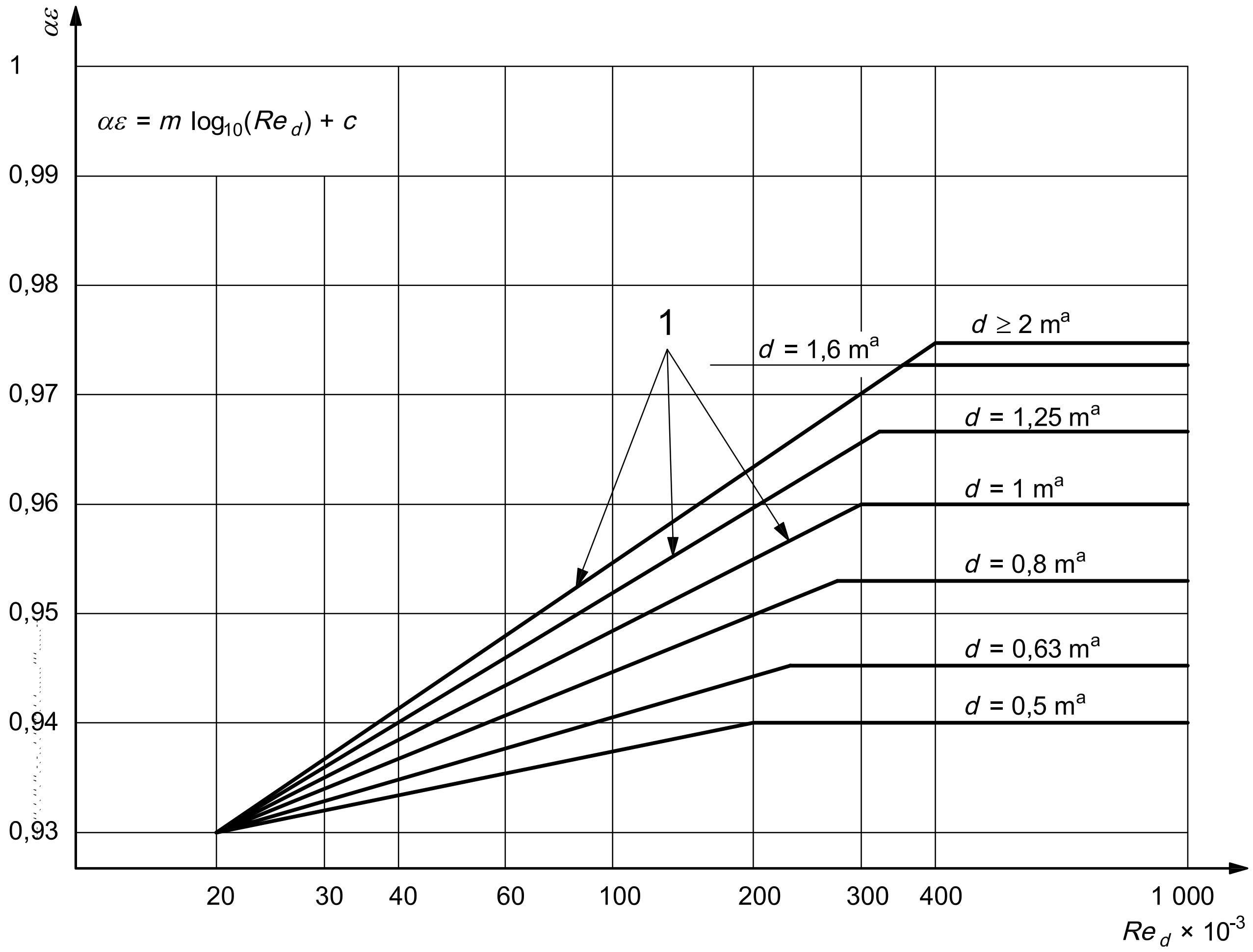
23.5.1 A bellmouth inlet manufactured in accordance with the requirements of 23.4 may be used uncalibrated for pressure ratios $r_{\mathrm{d}} > 0{,}96$, i.e. $\Delta p < 4\,000$ Pa.

23.5.2 For a bellmouth inlet the compound coefficient $\alpha\varepsilon$ is equal to 1.0.

23.5.3 The mass flow rate is given by the following equation:

$$q_m = \pi \frac{d^2}{4} \sqrt{2 \rho_{\mathrm{u}} \Delta p}$$

where ρ_{u} is the upstream density.



Key

1 curves for different duct diameters

$\alpha\varepsilon$ compound coefficient

$Re_d \times 10^{-3}$ Reynolds number

NOTE 1 For $d \leqslant 0{,}5$ m: $m = 0{,}010\ 00$; $c = 0{,}887\ 0$; $\alpha\varepsilon$ max. $= 0{,}94$.

NOTE 2 For $0{,}5$ m $< d \leqslant 2$ m: $m = -0{,}009\ 63 + 0{,}047\ 83d - 0{,}012\ 86d^2$; $c = 0{,}971\ 5 - 0{,}205\ 8d + 0{,}055\ 33d^2$; $\alpha\varepsilon$ max. $= 0{,}913\ 1 + 0{,}062\ 3d - 0{,}015\ 67d^2$.

NOTE 3 For $d \geqslant 2$ m: $m = 0{,}034\ 59$; $c = 0{,}781\ 2$; $\alpha \cdot \varepsilon$ max. $= 0{,}975$.

[a] Duct diameter.

Figure 19 — Compound coefficient $\alpha\varepsilon$ for conical or bellmouth inlets

23.6 Uncertainties

23.6.1 The uncertainty in the compound coefficient $\alpha\varepsilon$ and that in the flow coefficient α are the same. The basic uncertainty, applicable when $Re_d > 3 \times 10^5$, 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.

23.6.2 The additional uncertainty, as a percentage, due to low Re_d (i.e. $2 \times 10^4 < Re_d < 3 \times 10^5$) is as follows:

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

23.6.3 The additional uncertainty due to the presence of a uniform screen complying with 23.2 is 0,5 % and shall be added arithmetically.

23.6.4 These uncertainties may be reduced if a calibrated value of $\alpha\varepsilon$ is used in place of the value given in Figure 19. 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 flow rate coefficient not exceeding 1,0 %. The overall uncertainty of mass or volume flow rate measurement with screen loading in accordance with Figure 18 b) may then be taken as ± 2 %.

24 Determination of flow rate using an orifice plate

24.1 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.

24.2 Orifice plate

24.2.1 The orifice plate and the associated pressure tappings shall conform to the dimensions shown in Figure 20, to the additional requirements of this clause and to ISO 5167-1.

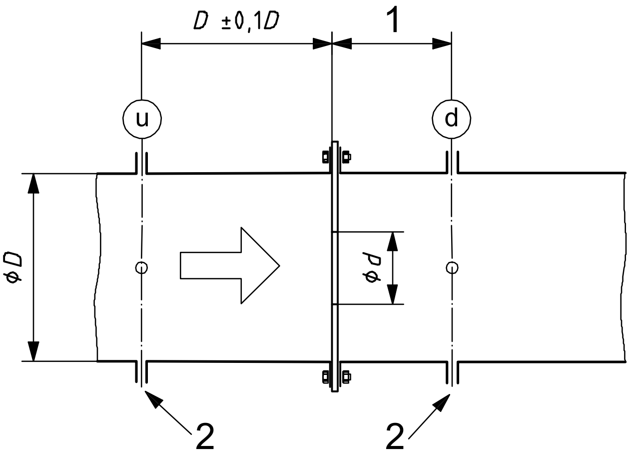
Two alternative types of tapping are available, the piezometer ring being generally the more convenient for small ducts and the wall tapping for larger sizes, although neither usage is exclusive.

24.2.2 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 0,000 $4d$. 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.

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

24.2.4 The upstream face of the orifice plate shall be flat to within 1 mm per 100 mm and its roughness, R_a, should not exceed 0,000 $1d$. Any gasket sealing of the plate and the duct flange shall not project internally.

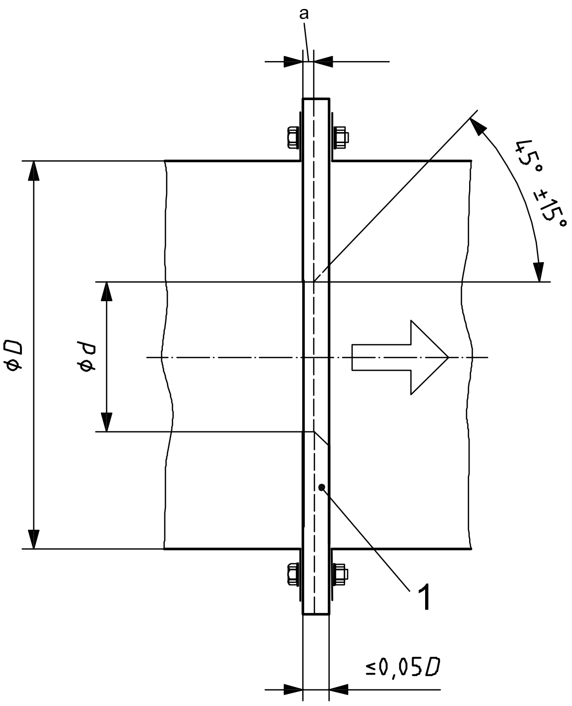


Key

1 $L_d = 0{,}5D \pm 0{,}02D$ for $\beta \leqslant 0{,}6$
$L_d = 0{,}5D \pm 0{,}01D$ for $\beta \geqslant 0{,}6$

2 wall tappings shall comply with Clause 7

a) In-duct orifice with taps at D and 0,5D

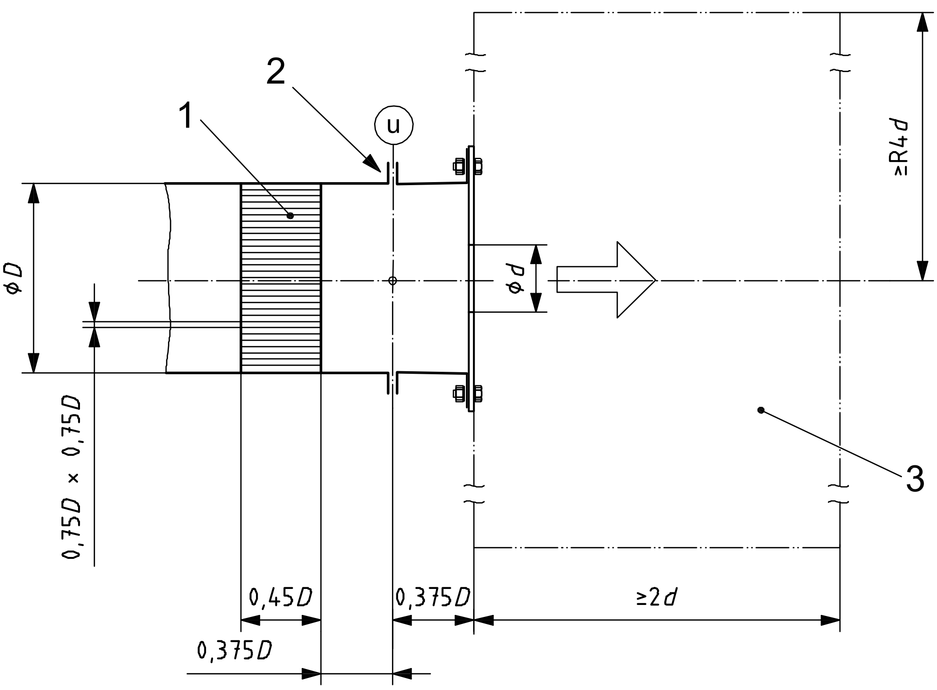


Key

1 additional thickness, if required, to stiffen the orifice plate

[a] Dimension e given by: $0{,}005D \leqslant e \leqslant 0{,}02D$.

b) Details of orifice plate



Key

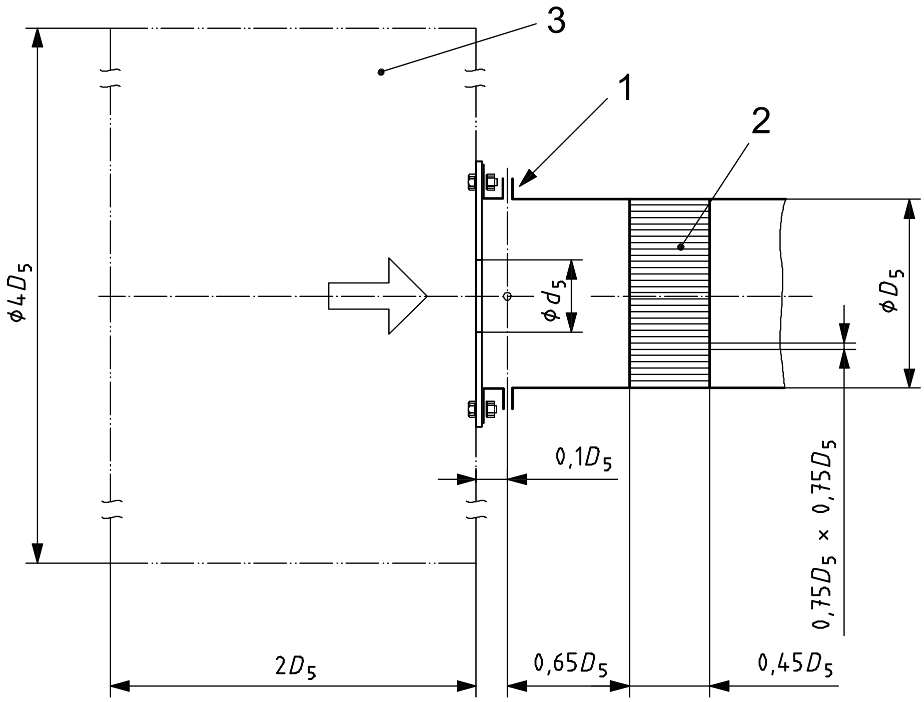
1 flow straightener (cell-type shown), see Clause 27

2 wall tappings complying with Clause 7

3 no obstacles within this space

c) Outlet orifice with wall tappings

Figure 20 — Orifice plates and assemblies



Key

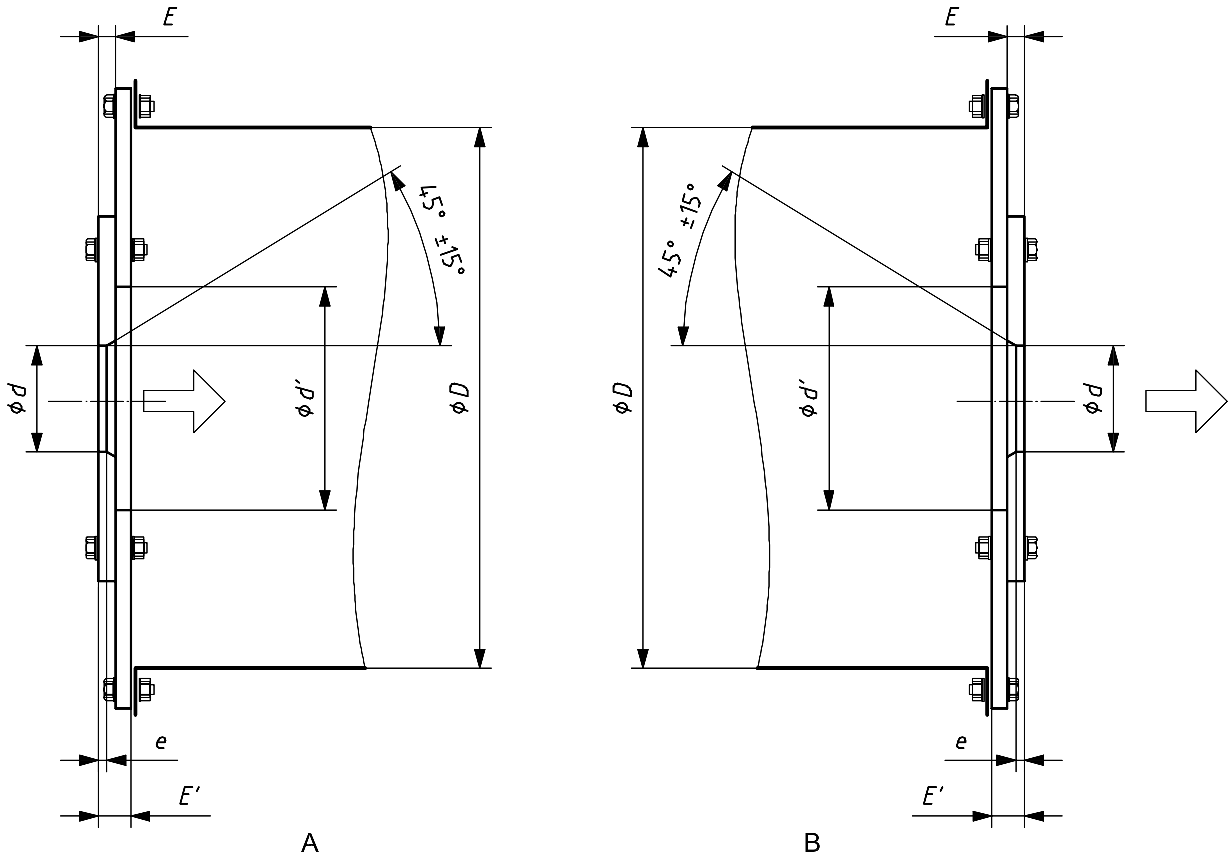
1 wall tappings complying with Clause 7
2 flow straightener (cell-type shown), see Clause 27
3 no obstacles within this space

NOTE 1 If the orifice plate is held in place by a clip then the internal diameter is $\geqslant D_5$ and the thickness $\leqslant 0{,}01D_5$.

NOTE 2 If the orifice plate is held in place by a collar then the internal diameter is $\leqslant D_5$ and the radial obstruction $\leqslant 0{,}01D_5$.

d) Inlet orifice with wall tappings

Figure 20 (*continued*)



Key

A inlet orifice

B outlet orifice

E $E \geqslant 0{,}003d'$

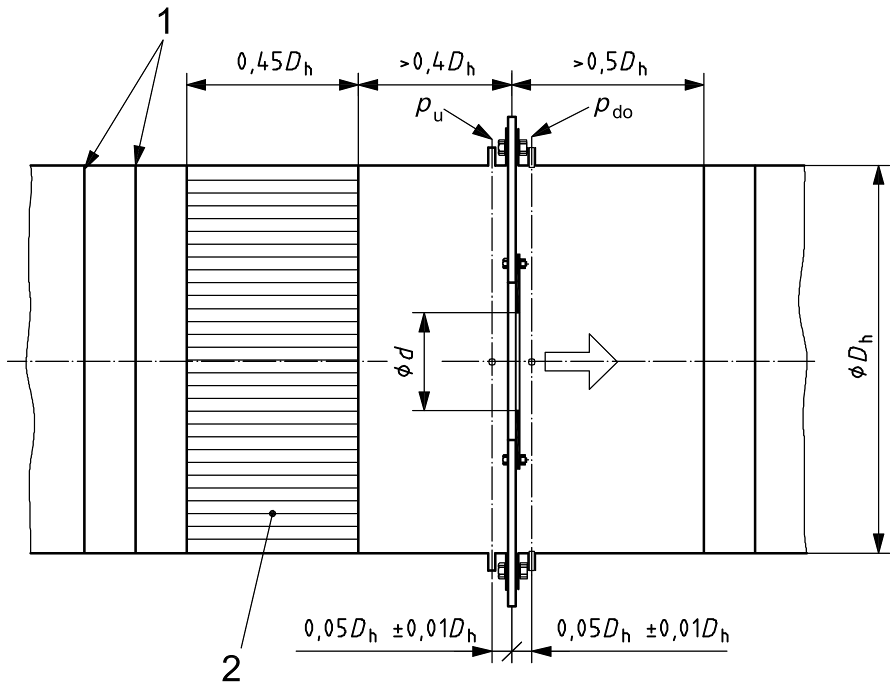
e $e \leqslant 0{,}01d$

NOTE 1 Chamfer for $E \geqslant 0{,}01d$.

NOTE 2 Where the orifice plate is bolted to a supplementary plate: $d' \geqslant 1{,}25d + 4E'$.

e) Detail of inlet or outlet orifice plates for wall tappings, see 24.8, 24.8.1, and 24.8.2

Figure 20 (*continued*)



Key

1 flow settling screens

2 flow straightener (cell-type shown), see Clause 27

f) Orifice plate in test chamber (inlet side or outlet side), see 24.8.1, 29.4 and 29.3

Figure 20 (*continued*)

24.3 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 ISO 5167-1.

A flow straightener in accordance with Clause 27 shall be fitted in the upstream duct. The length of the upstream and downstream ducts and the installation conditions are given in ISO 5167-1.

24.4 Pressure tappings

Wall tappings shall be four in number, in accordance with Clause 7, and in the locations shown in Figure 20. The axis of each tap should intersect the duct axis at right angles.

The dimensions of the wall tapping holes shall conform to the dimensions shown in Figure 2. Any gasket shall be included in these dimensions.

24.5 Calculation of mass flow rate

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

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.

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

— The flow rate coefficient, α, depends on the orifice diameter ratio $\beta = d/D$ and on the duct Reynolds number Re_D (see 24.6). 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 24.7, 24.8, and Figure 22.

24.6 Reynolds number

The Reynolds numbers required for calculating orifice flow rate 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 equation:

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

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 an inlet orifice, it may be sufficient to take, for the dynamic viscosity, the 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.

24.7 In-duct orifice with D and $D/2$ taps [see Figure 20 a) and ISO 5167-1]

The following conditions shall apply:

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

$p_{do}/p_u > 0,75$

ρ_u is the air density at the upstream tapping;

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

The flow rate coefficient, α, is given by the Stolz formula:

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

and is shown in Figure 21.

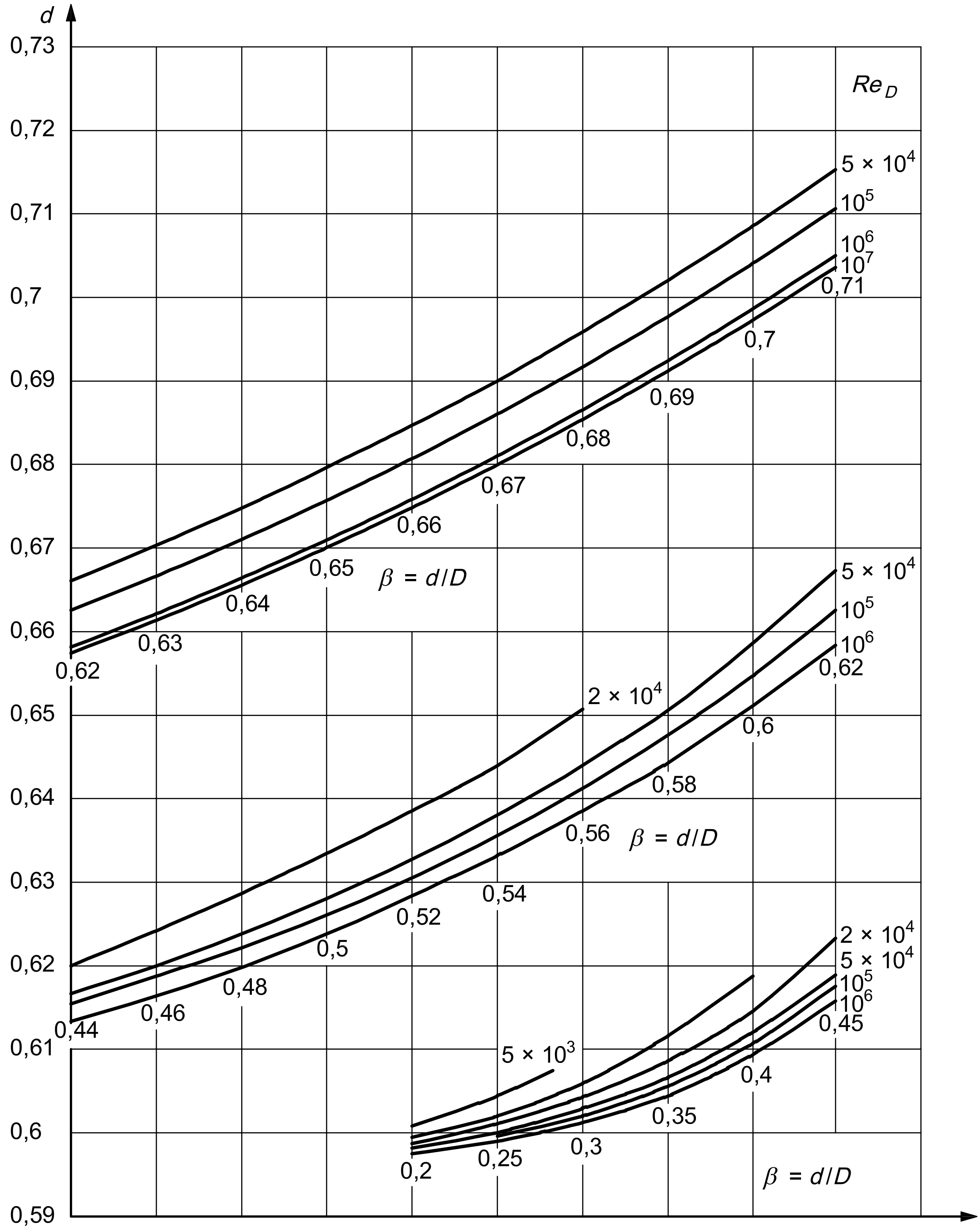
The expansibility factor, ε, is given by

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

and is shown in Figure 22.

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

The uncertainty in ε, as a percentage, is $4(\Delta p/p_u)$.

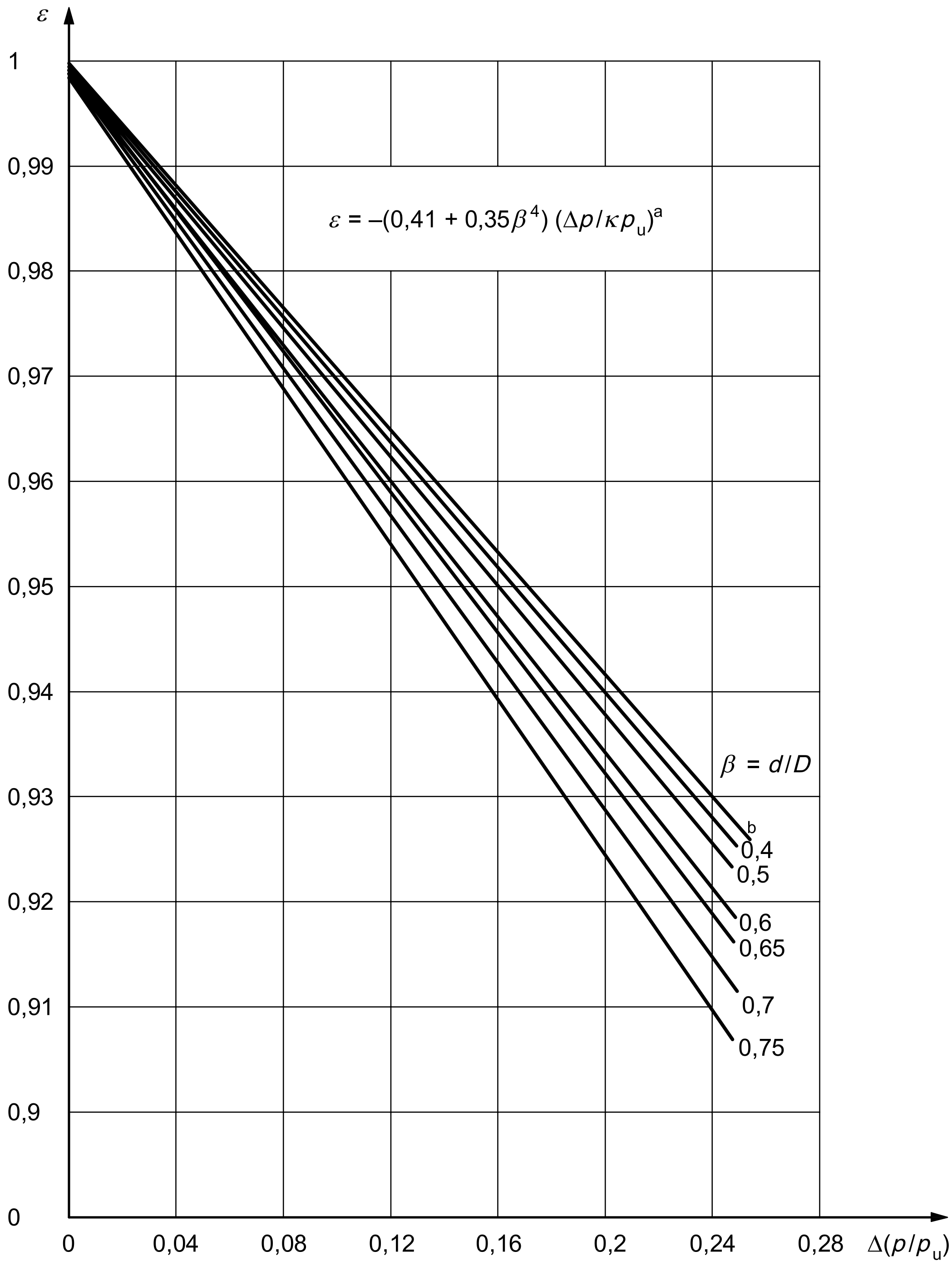


Key

α flow rate coefficient

Re_D Reynolds number

Figure 21 — Flow rate coefficient, α, of in-duct orifice with taps at D and D/2 (see 24.7)



Key

ε expansibility factor

$\Delta p/p_u$ differential pressure ratio

[a] See 24.7.

[b] 0,1 to 0,2.

Figure 22— Expansibility factor, ε, for orifice plates in atmospheric air (see 24.7 and 24.8)

24.8 Outlet orifice with wall tappings [see Figure 20 c) and e)]

The following conditions shall apply:

$$\Delta p = \rho_{\text{u}} - p_{\text{a}} = p_{\text{eu}} = p_{\text{e6}}$$

where

p_{a} is the ambient atmospheric pressure;

ρ_{u} is the air density at the upstream tapping;

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

$\alpha\varepsilon$ is given by the following equation and plotted in Figure 23 as function of

$$\frac{p_{\text{e6}}}{p_{\text{a}}} = \frac{p_{\text{eu}}}{p_{\text{a}}} = r_{\Delta p} = \frac{\Delta p}{p_{\text{u}} - \Delta p}$$

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

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 $\beta < 0{,}5$

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

24.8.1 Orifice plate with wall tappings in the test chamber [see Figure 20 e) and f)]

The following conditions shall apply:

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

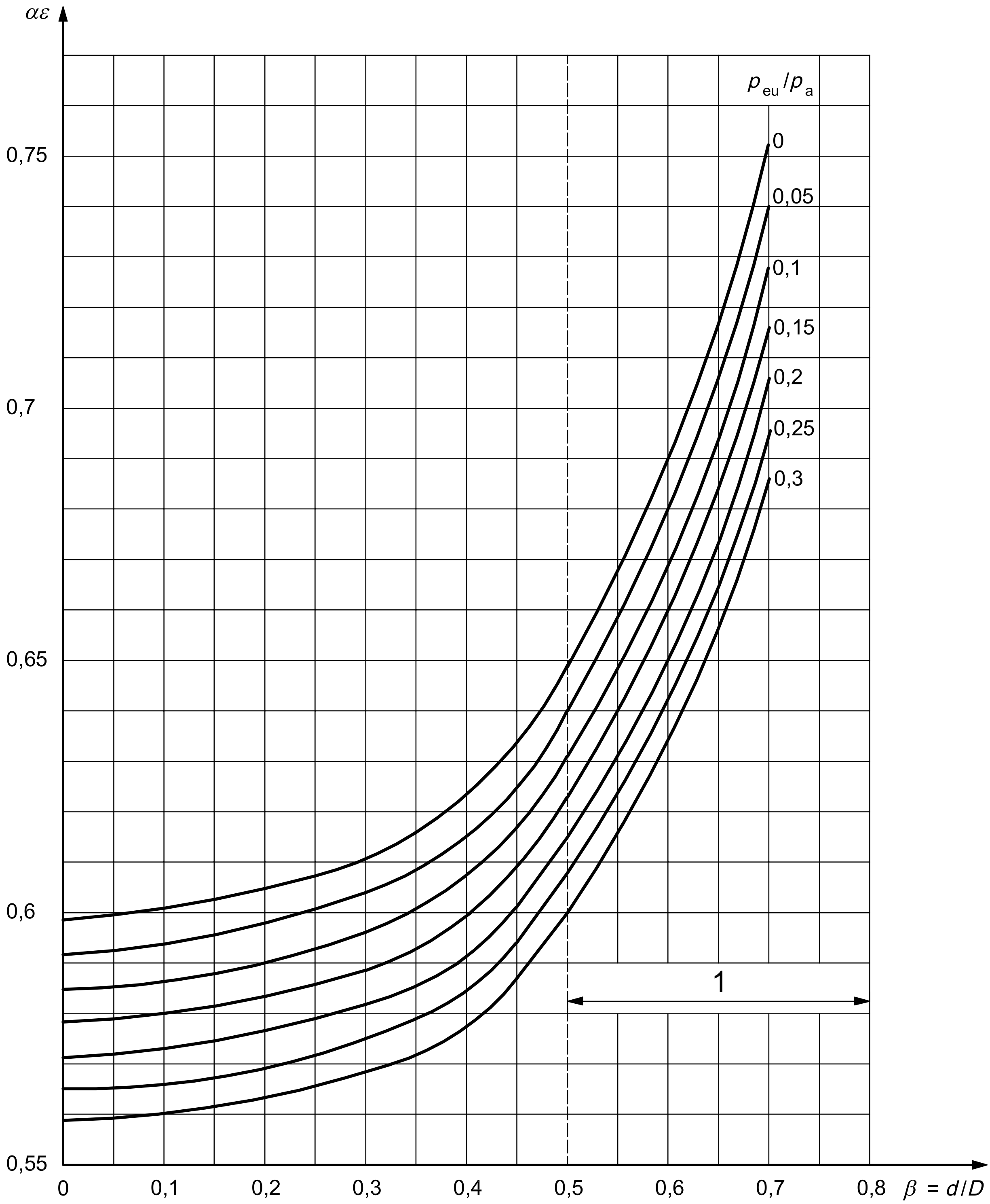
The temperature, T_{u}, is measured in the test chamber.

$$\Theta_{\text{u}} = \Theta_{\text{sgu}} = T_{\text{u}} + 273{,}15$$

$\beta = d/D_{\text{h}}$ shall not exceed 0,25;

$\alpha\varepsilon$ is determined in accordance with 24.8.

The other remarks of 24.8 shall apply.



Key

1	zone of reduced accuracy
$\alpha\varepsilon$	compound flow rate coefficient
$\beta = d/D$	diameter ratio

Figure 23 — Compound flow rate coefficient, $\alpha\varepsilon$, of outlet orifices with wall taps (see 24.8)

24.8.2 Inlet orifice with wall tappings [see Figure 20 d) and e)]

The following conditions shall apply:

$$\Delta p = p_{\mathrm{a}} - p_{\mathrm{do}} = p_{\mathrm{e5}}$$

where p_{a} is the ambient atmospheric pressure.

$$p_{\mathrm{u}} = \rho_{\mathrm{a}}$$

where ρ_{a} is 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 24.5.

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

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

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

The uncertainty with which α is known may be taken as $\pm$ 1,0 % provided that $Re_D \geqslant 5 \times 10^4$ and $r_{\Delta p} = \Delta p/(p_{\mathrm{u}}\ 2\ \Delta p) \leqslant 0{,}3$.

25 Determination of flow rate using a Pitot-static tube traverse

25.1 General

For standardized airway tests, only traverses using Pitot-static tube in cylindrical ducts are recognized. The locations of the traverse planes shall be those shown in: Figures 42 c) and d); Figure 44 e) and f); Figure 45 a); and Figure 46 g). The working fluid is normally 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 25.6 as a function of Reynolds number to determine the average velocity at the section with an uncertainty of ±2 %.

25.2 Pitot-static tube

The instrument shall conform to the requirements of ISO 3966. The external diameter of the tube, d, shall not exceed D/48, where D is the diameter of the airway. The diameter of the stagnation pressure hole shall not be less than 1 mm.

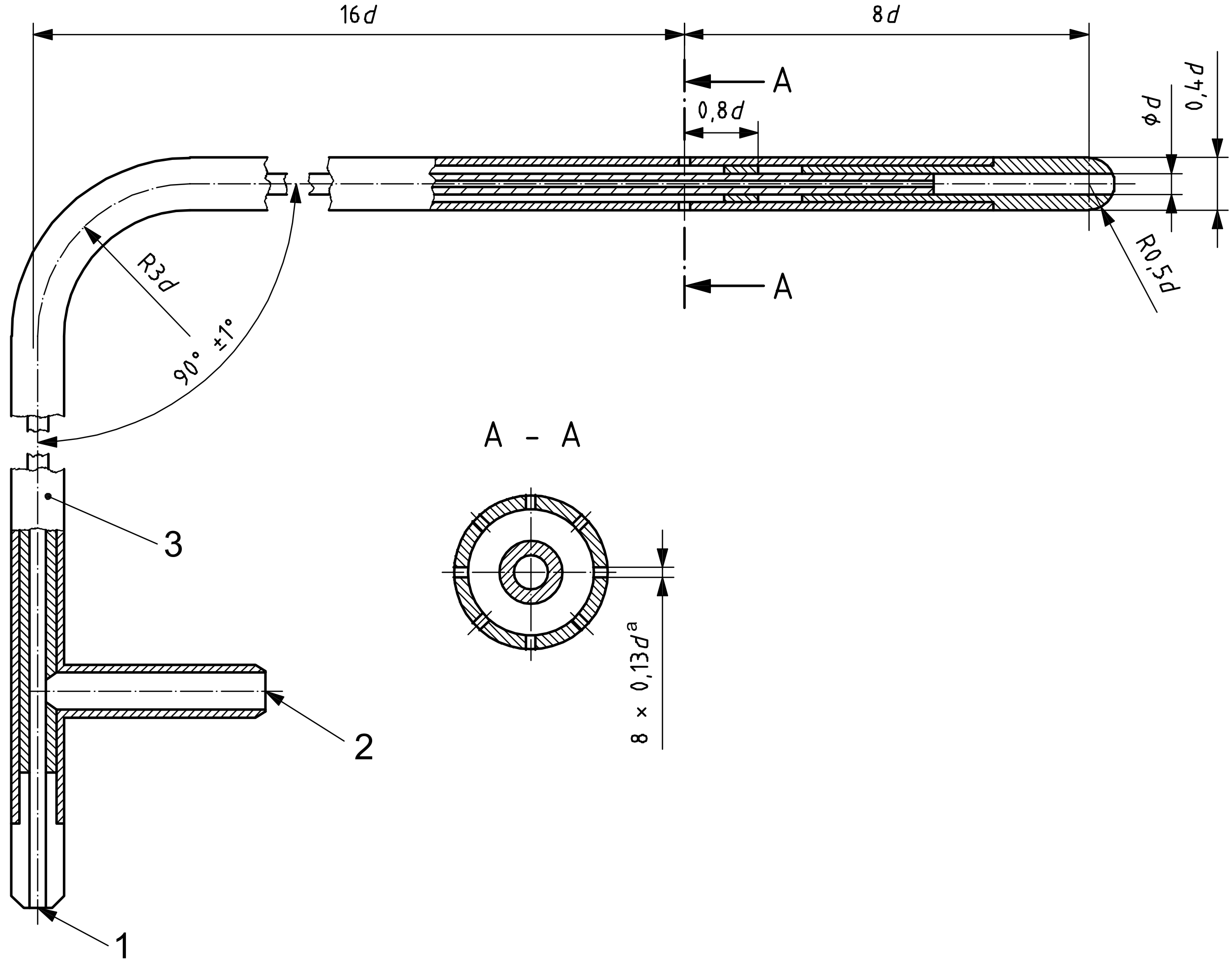
Four types of Pitot-static tube may be used:

— Air Movement and Control Association (AMCA) type, see Figure 24 a);

— modified National Physical Laboratory (NPL) ellipsoidal nose type, see Figure 24 b);

— Centre Technique des Industries Aérauliques et Thermiques (CETIAT) type, see Figure 24 c);

— Deutsches Zentrum für Luft- und Raumfahrt (DLR) type, see Figure 24 d).



Key

1 stagnation pressure connection

2 static pressure connection

3 main stem

[a] Drilled holes shall not exceed 1 mm diameter; they shall be equally spaced and free from burrs. The hole depth shall not be less than the hole diameter.

NOTE 1 The Pitot tube head shall be free from nicks and burrs.

NOTE 2 All dimensions shall be within ± 2 %.

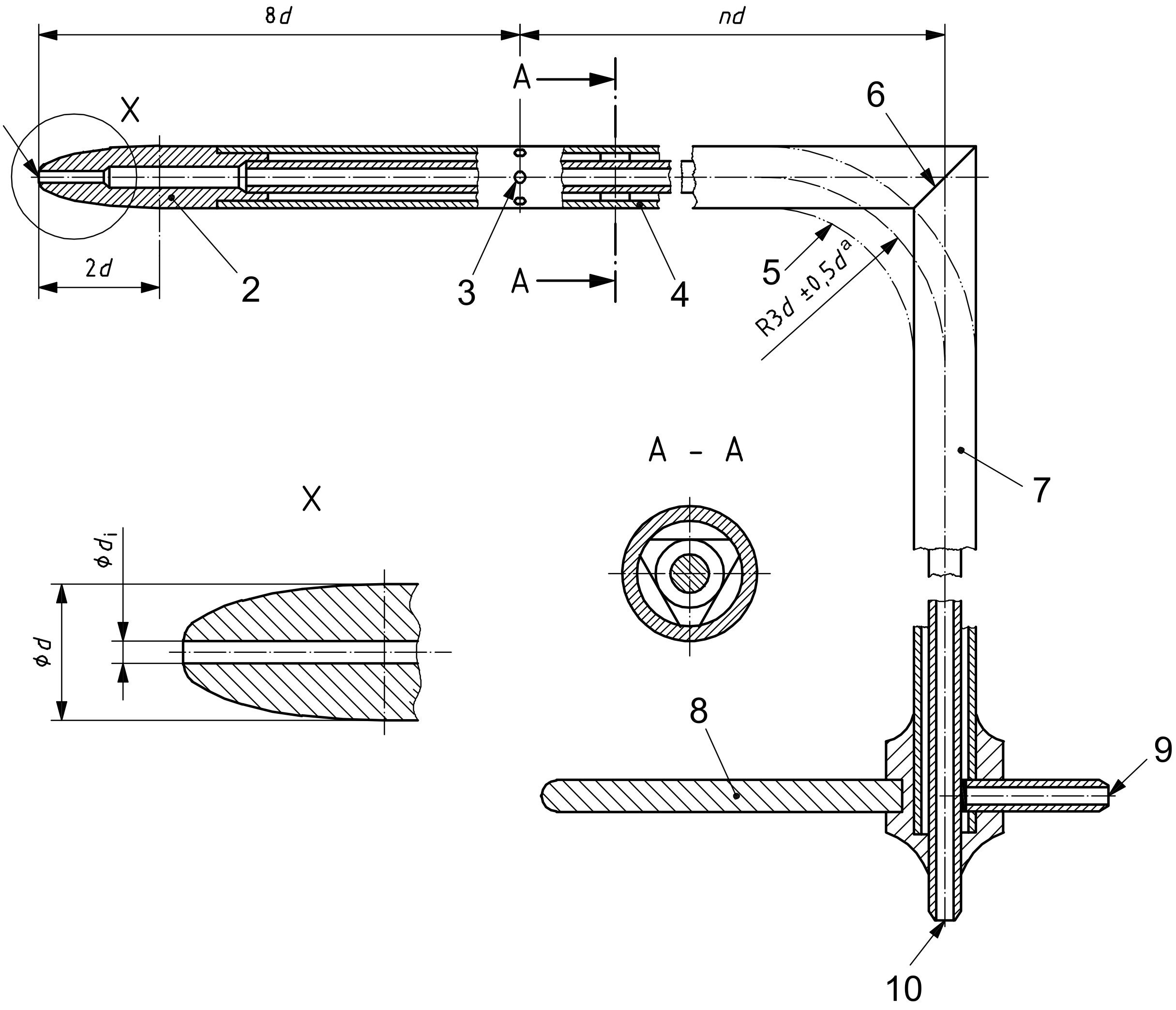
NOTE 3 Surface roughness shall be 0,8 µm or better.

NOTE 4 The static orifices shall not exceed 1 mm in diameter.

NOTE 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 24 — Types of Pitot-static tubes



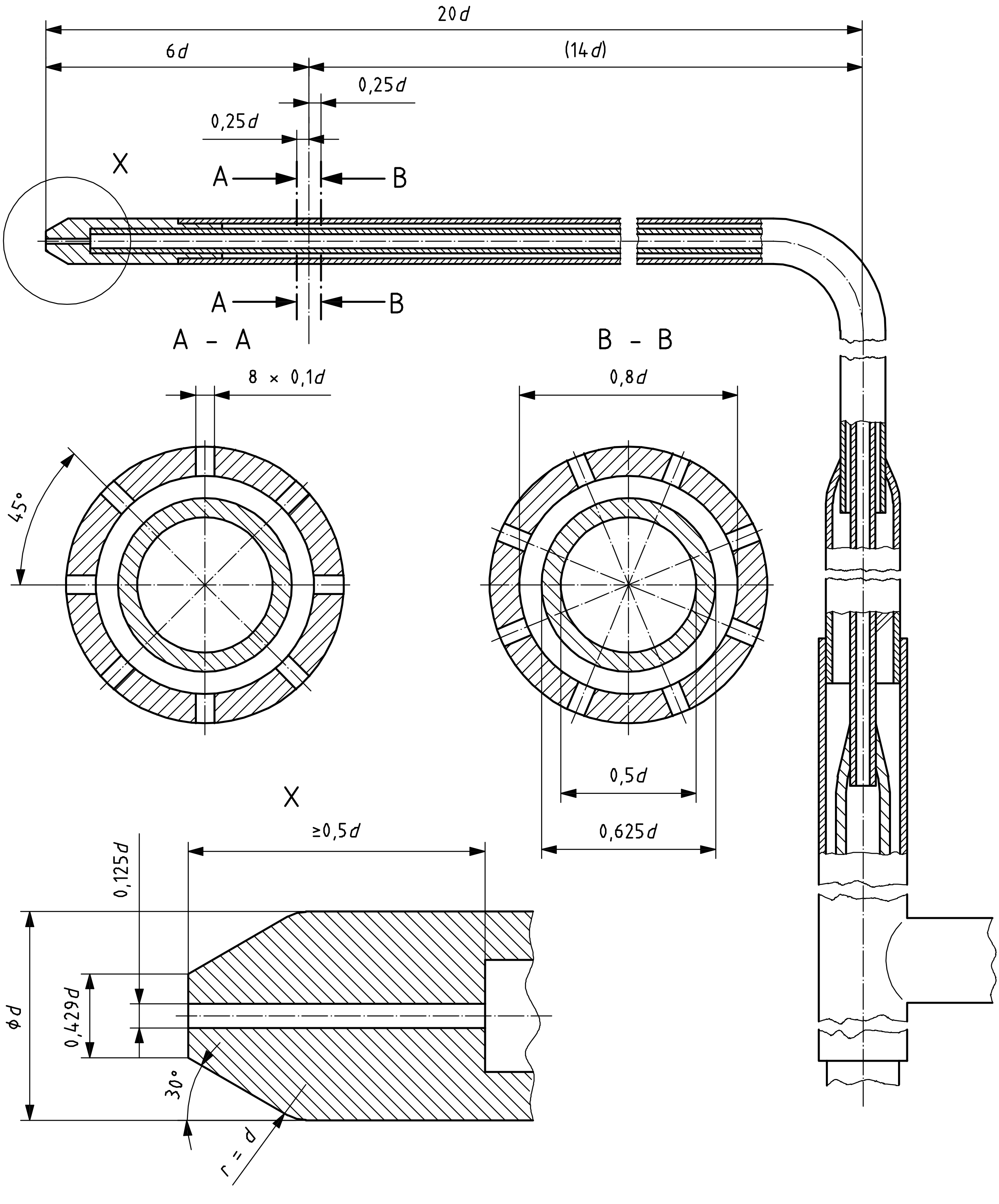
Key

1	stagnation pressure hole	6	mitred junction
2	modified ellipsoidal nose	7	main stem
3	static pressure holes	8	alignment arm
4	internal spacer	9	static pressure connection
5	alternative curved junction	10	stagnation pressure connection

[a] Mean radius of curved option.

b) NPL type with modified ellipsoidal nose

Figure 24 (*continued*)

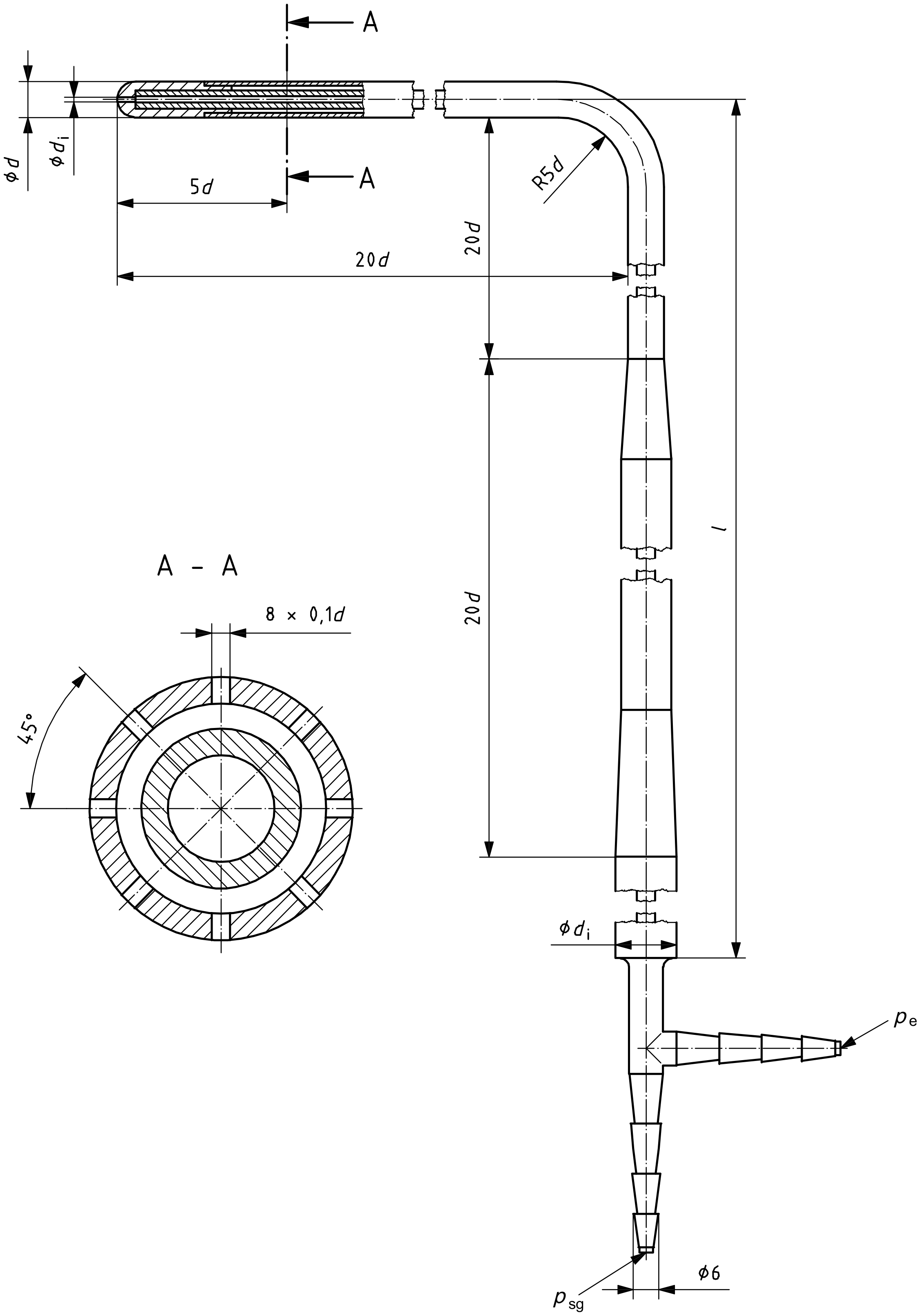


NOTE Static pressure taps may be limited to those indicated on section A-A, in which case section A-A shall be placed at 6d from the tube tip.

c) CETIAT type

Figure 24 (*continued*)

Dimensions in millimetres



d) DLR type

Figure 24 (*continued*)

25.3 Limits of air velocity

The Mach number of the flow past the tube should not exceed 0,25 (85 m/s in 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 $v = 3/d_i$.

25.4 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 25.

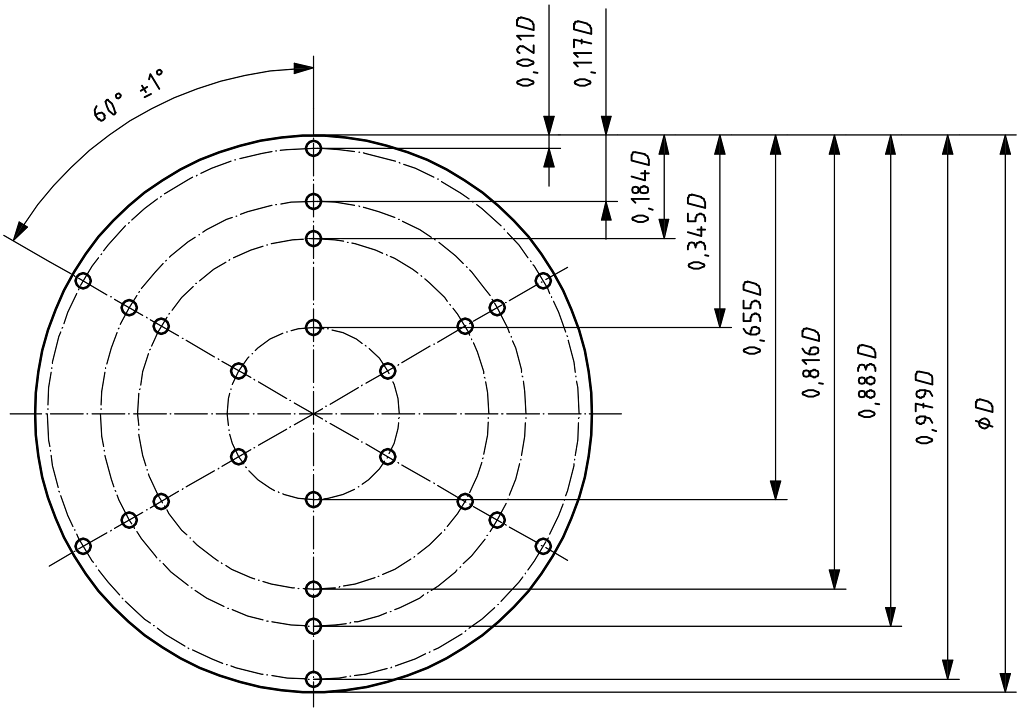


Figure 25 — Positions for traverse measurements in standardized airways

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

The distance of the measurement points (when there are eight 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.

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

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

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

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

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

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

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

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

25.5 Determination of flow rate

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 equation:

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

$$\Delta p_{\mathrm{m}} = \left[\frac{1}{n} \left(\sqrt{\Delta p_1} + \sqrt{\Delta p_2} + \dots + \sqrt{\Delta p_{\mathrm{n}}} \right) \right]^2$$

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

$$p_{\mathrm{e}x} = \frac{1}{n} \left(p_{\mathrm{e}x1} + p_{\mathrm{e}x2} + \dots + p_{\mathrm{e}xn} \right)$$

$$p_x = p_{\mathrm{e}x} + p_{\mathrm{a}}$$

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

$$\Theta_x = \Theta_{\mathrm{sg}x} \left[\frac{p_x}{p_x + \Delta p_{\mathrm{m}}} \right]^{\frac{\kappa - 1}{\kappa}}$$

$$\rho_x = \frac{p_x}{R_{\mathrm{w}} \Theta_x}$$

The mass flow rate, q_m, is given by:

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

where

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

is the expansibility factor et α is the correction factor or flow rate coefficient given in 25.6.

25.6 Flow rate coefficient

The flow rate 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 International Standard. The coefficient α is dependent on the Reynolds number which is derived from the diameter D_x and average velocity $v_{\mathrm{m}x}$ at the section x as shown below.

$$Re_{Dx} = \frac{\rho_x V_{\mathrm{m}x} D_x}{\mu} = \frac{4 q_m}{\pi D_x \mu} \approx 71 \times 10^3 \frac{q_m}{D_x}$$

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

25.7 Uncertainty of measurement

The use of an average value for α involves disregarding systematic errors which may reach ± 0,8 % of volume flow rate or mass flow rate. Random uncertainties of measurement total ± 1,1 %. Therefore the uncertainty of flow rate 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. The manometer calibration required for air with a density of 1,2 kg/m^3 is shown below for different flow velocities:

± 1,5 Pa	± 1 Pa	± 0,5 Pa	± 0,25 Pa
16 m/s	13 m/s	9 m/s	6 m/s

26 Installation and setup categories

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

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

The test installation shall reproduce these working conditions as closely as possible, therefore four categories of test setup have been defined.

26.1 Category A: free inlet and free outlet

In order to qualify for installation category 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 29.3 and 29.4.

26.2 Category B: free inlet and ducted outlet

In order to qualify for installation category B, an outlet duct with straightener shall be used, which shall 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. The duct and antiswirl device form a common segment at the fan outlet (see 28.2).

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 28.2.5) may be used between fan and chamber.

26.3 Category C: ducted inlet and free outlet

In order to qualify for installation category 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 28.3).

An inlet test chamber may be used (see 29.3). 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 category C fan tests.

26.4 Category D: ducted inlet and ducted outlet

In order to qualify for installation category 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 28.2 and 28.3 respectively.

When an inlet or outlet chamber is used, the outlet duct may be of the short variety described in 28.2.5 when there is no swirl at the fan outlet.

For large fans ($\geqslant$ 800 mm diameter), 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 28.2.5 and 28.4 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.

26.5 Test installation type

To identify the performance, the symbols of the characteristics influenced by the installation category, 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}.

27 Flow straighteners

The swirl energy at the fan outlet can only be partially recovered in a straight uniform duct, and only over very long distances ($> 100D$). In the presence of swirl, simple measurements of effective pressure or volume flow are impossible, and it must therefore be removed when tests are to be taken in a duct on the outlet side of the fan. An effective flow straightener will achieve this.

27.1 Types of straightener

There are two designs of flow straightener which can be used. Details are given below.

27.1.1 AMCA cell straightener

The AMCA cell straightener is used only to prevent the growth of swirl in a normally axial flow. It does not improve asymmetric velocity distributions. The device, shown in Figure 26 consists of a nest of equal cells of square cross-section. It has a very low pressure loss and is typically used either side of an auxiliary booster fan where this is necessary to overcome the resistance of the airway when a complete characteristic is required.

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 $0{,}005D_4$.

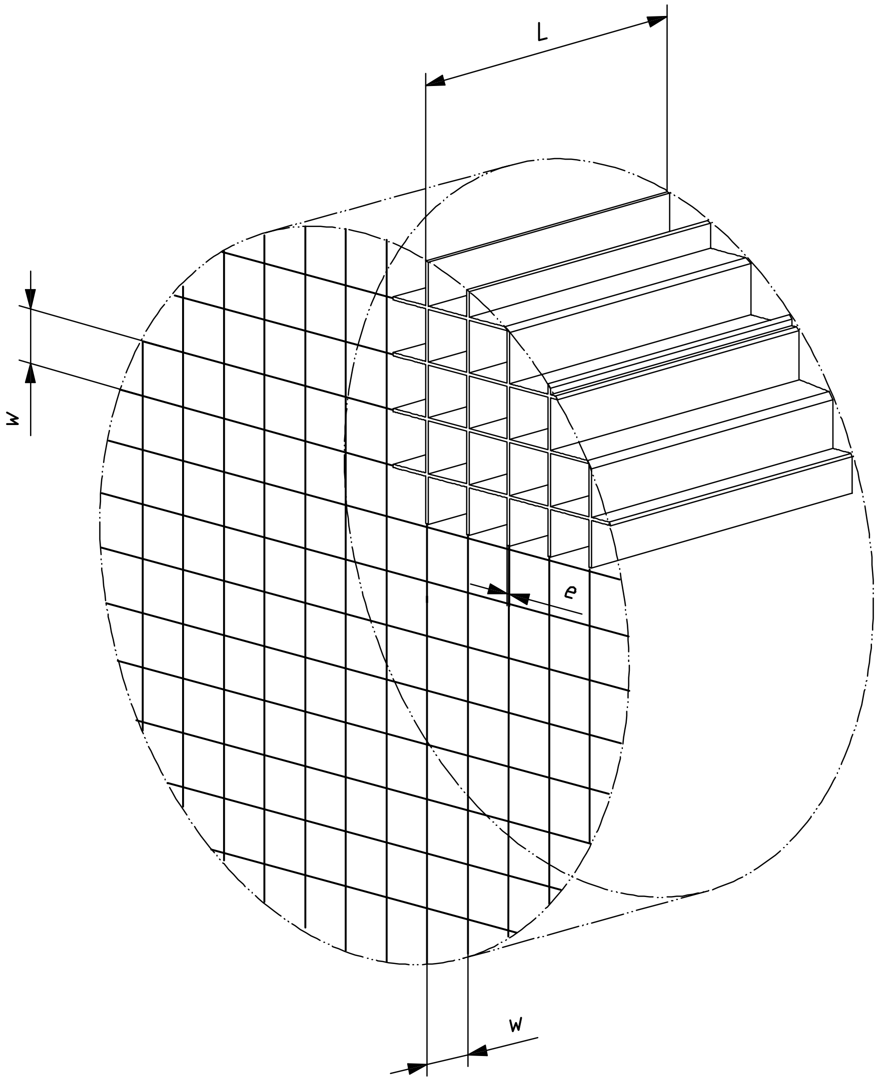


Figure 26 — Cell type flow straightener

For a cell duct straightener (see Figure 26):

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

$L = 0{,}45D_4$

$e \leqslant 0{,}005D_4$

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

27.1.2 Star straightener

The star straightener is also designed to eliminate swirl but is of little use in equalization of asymmetric velocity distributions. The eight radial plates should be of adequate thickness to provide sufficient strength but should not exceed 0,007D_4 for pressure loss considerations. This straightener has a similar pressure drop to the cell type straightener, but is easier to manufacture. More importantly, and unlike the cell type, it allows the static pressure to equalize radially as the air flows through it, making it the preferred type of flow straightener.

The star straightener, as shown in Figure 27, is constructed of eight radial blades of length 2D_4 (with a $\pm$ 1 % tolerance) and of thickness not greater than 0,007D_4. The blades will be arranged to be equidistant on the circumference with the angular deviation being no greater than 5° between adjacent plates.

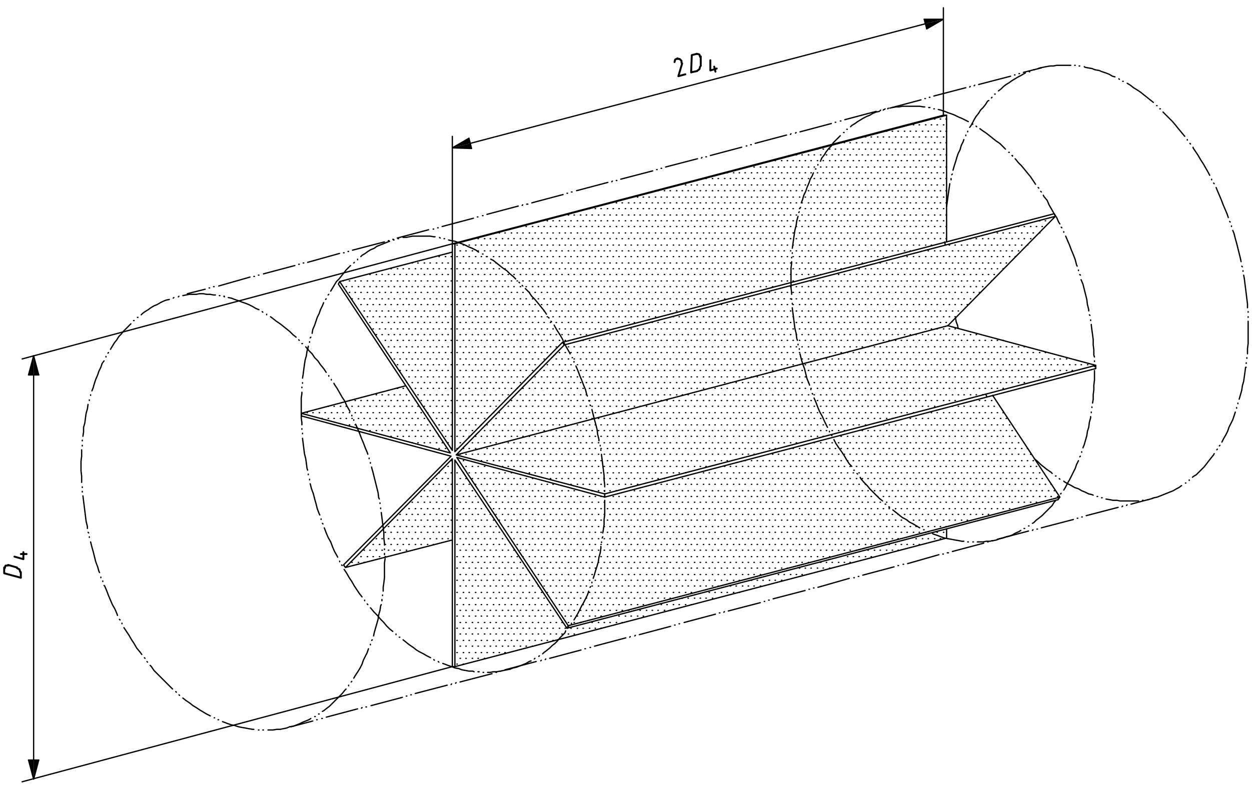


Figure 27 — Star type flow straightener

27.2 Rules for use of a straightener

For fans where the outlet swirl is greater than 15°, a flow straightener must always be used on the discharge side of the test fan. If there is any doubt about the degree of swirl, then a test should be performed to establish how much is present. For fans where the outlet swirl is less than 15°, that is: centrifugal, cross-flow or vane-axial fans, it is possible to use a simplified outlet duct without straightener when discharging to the atmosphere or to a measuring chamber. If there is any doubt about the degree of swirl, then a test should be performed to establish how much is present.

IMPORTANT — Even in cases where the swirl is less than 15°, a flow straightener must always be used upstream of a static or dynamic pressure measuring plane located within a test duct.

28 Common-segment airways for ducted fan installations

28.1 Common segments

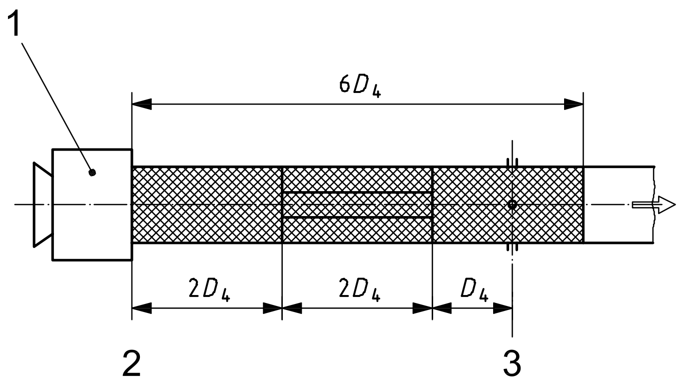
Standardized airways for category B, C or D ducted fan installations incorporate common segments adjacent to the fan inlet and/or outlet which are described in this clause.

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 category to another.

28.2 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 27.1.2 and Figure 28 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 28.2.2 and 28.2.3.

Figures 28, 29 and 30 show the recommended devices.



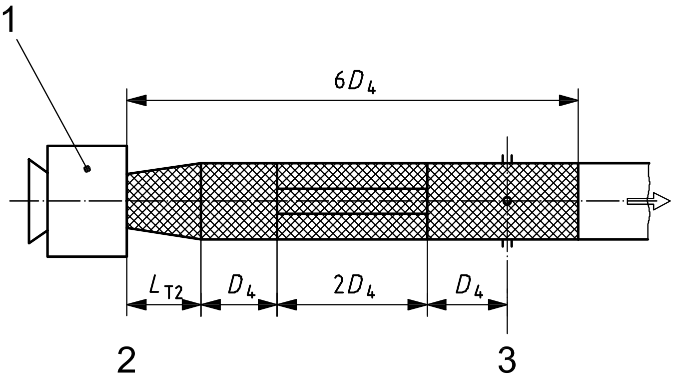
Key

1 fan on test

2 test fan outlet

3 pressure-measurement section

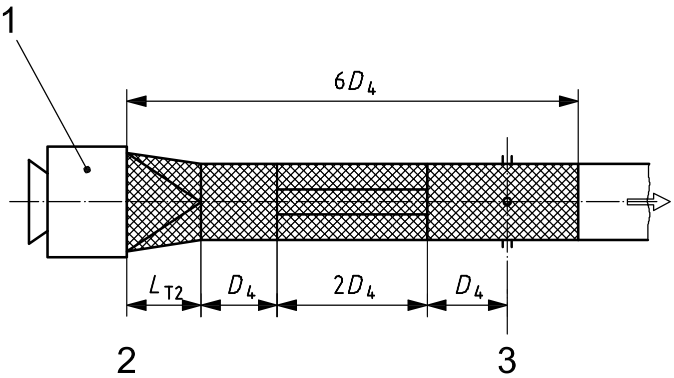
Figure 28 — Circular fan outlet for $D_2 = D_4$



Key

1 fan on test
2 test fan outlet
3 pressure-measurement section

Figure 29 — Circular fan outlet for $D_2 \neq D_4$



Key

1 fan on test
2 test fan outlet
3 pressure-measurement section

Figure 30 — Rectangular fan outlet where $b < h$

28.2.1 Circular fan outlet when $D_4 = D_2$ (see Figure 28).

28.2.2 Circular fan outlet when $D_4 \neq D_2$ (see Figure 29):

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

$L_{T2} = D_4$

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

28.2.3 Rectangular fan outlet, bh, where $b < h$ (see Figure 30)

$0{,}95 < \pi D_4^2 / 4b < 1{,}07$

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

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

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

28.2.4 Transition (see Figure 31)

The transition section is preferably formed from a single sheet of material as illustrated in Figure 31 in accordance with 28.2.

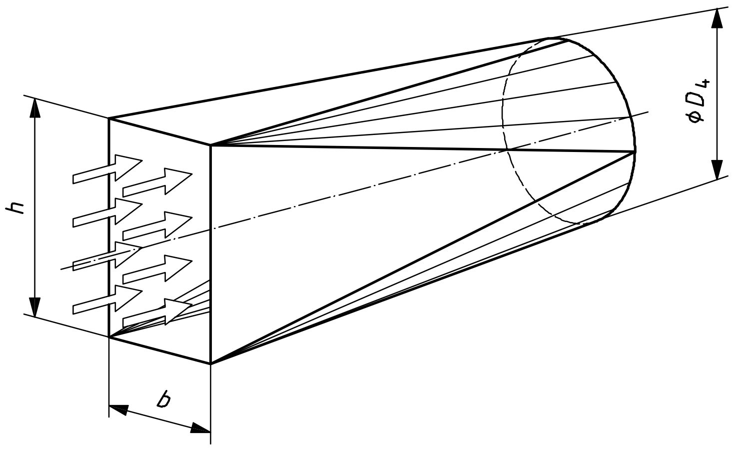


Figure 31 — Transition

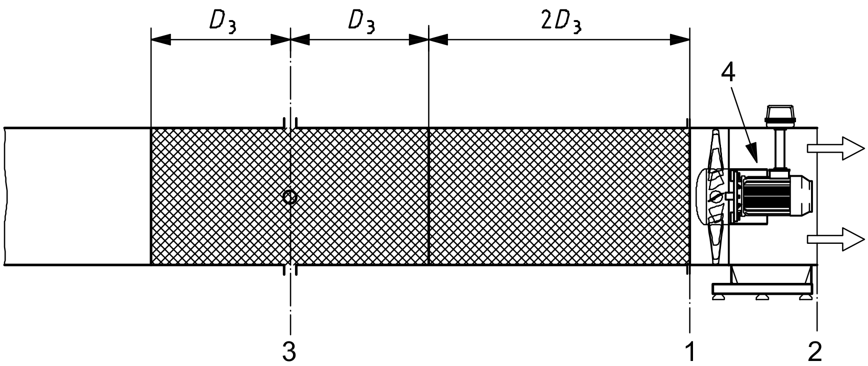
28.2.5 Short duct

In the particular case of tests on fans of category B or D without significant outlet swirl, such as a centrifugal, cross-flow or vane-axial 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 determined by the condition:

$$L \geqslant 3\sqrt{\frac{4b_2h_2}{\pi}}$$

28.3 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 32.



Key

1 test fan inlet

2 test fan outlet

3 pressure-measurement section

4 fan on test

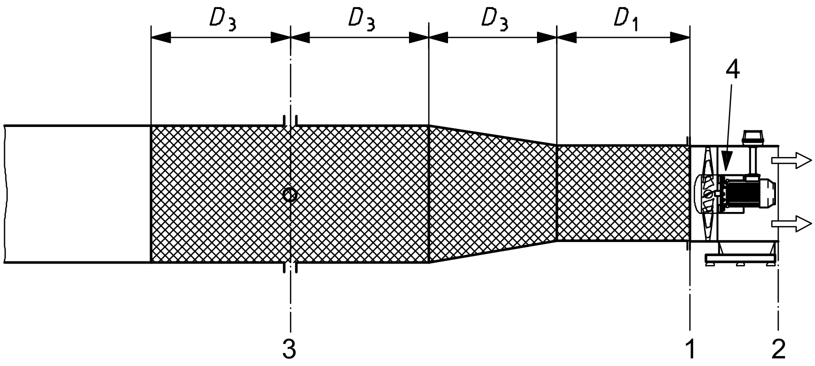
Figure 32 — Circular fan inlet for $D_3 = D_1$

A transition section may be used to accommodate a difference in area and/or shape within the limits specified in 28.3.1 and 28.3.2.

28.3.1 A circular fan inlet when $D_3 = D_1$ (see Figure 32)

28.3.2 Circular fan inlet where 0,975 $D_1 < D_3 <$ 1,5 D_1 (see Figure 33)

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



Key

1 test fan inlet

2 test fan outlet

3 pressure-measurement section

4 fan on test (tube-axial type shown)

Figure 33 — Circular fan inlet for $0{,}975D_1 < D_3 < 1{,}5D_1$

28.3.3 Rectangular fan inlet, bh (see Figure 34)

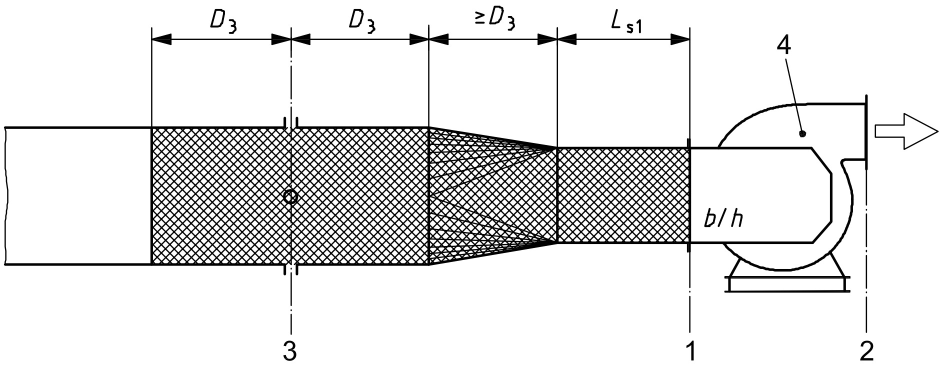
The section adjacent to the fan inlet has the same rectangular cross-section, bh, as the fan inlet to which it is attached and its length, L_{s1}, is given below:

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

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

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

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 28.2.5.



Key

1 test fan inlet
2 test fan outlet
3 pressure-measurement section
4 fan on test (centrifugal shown with an integral inlet box)

Figure 34 — Rectangular fan inlet

28.4 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 28.2, according to the particular case. 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.

In certain cases, 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 $3D$. This static pressure is taken to be equal to the atmospheric pressure.

28.5 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.

28.5.1 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 bellmouth 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 28.3.1 with a bellmouth entry at its inlet end.

28.5.2 Rectangular fan inlet

The simulation section should have the same rectangular cross-section, bh, as the fan inlet to which it is attached, and its length, L_{s1}, given by the following equation:

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

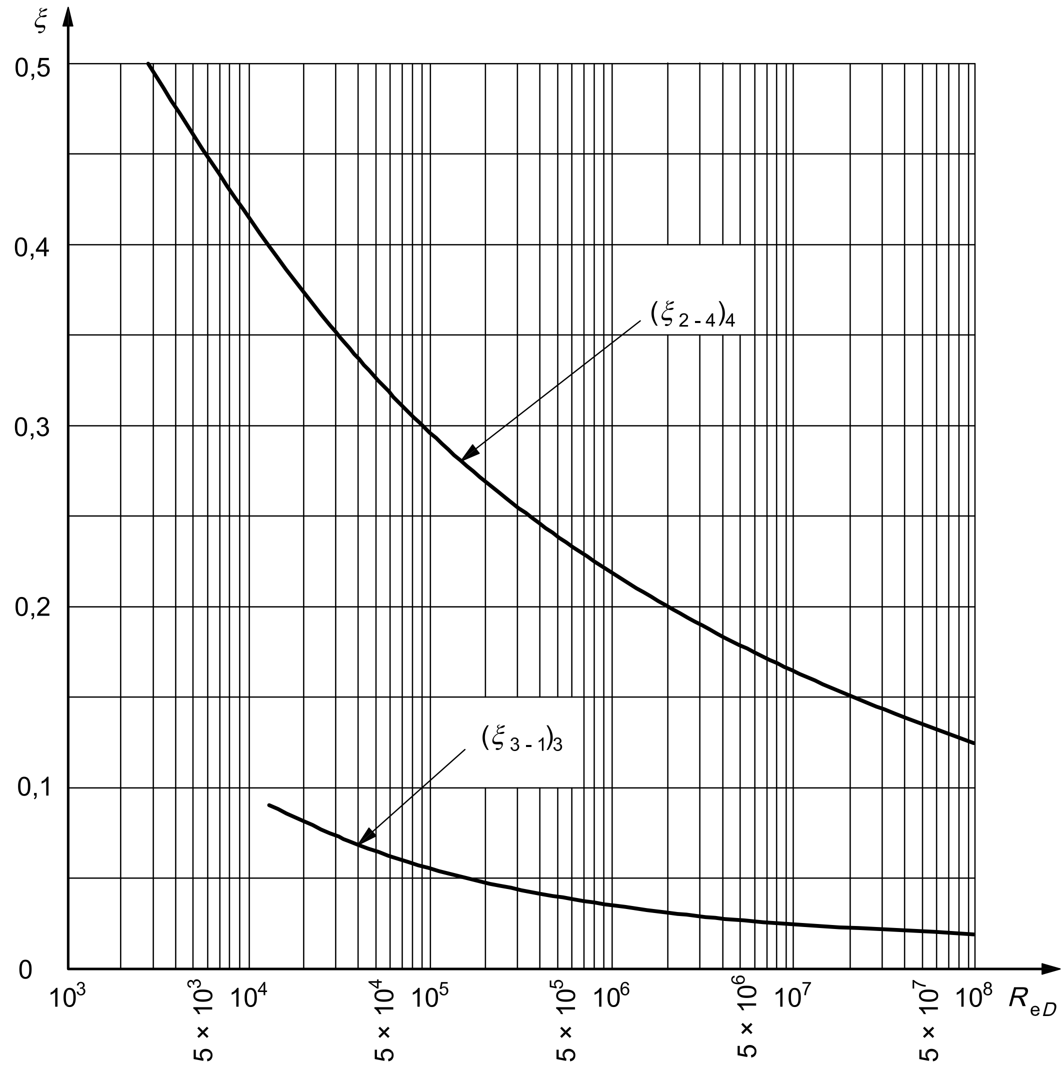
A bellmouth entry should be fitted.

28.6 Loss allowances for standardized airways

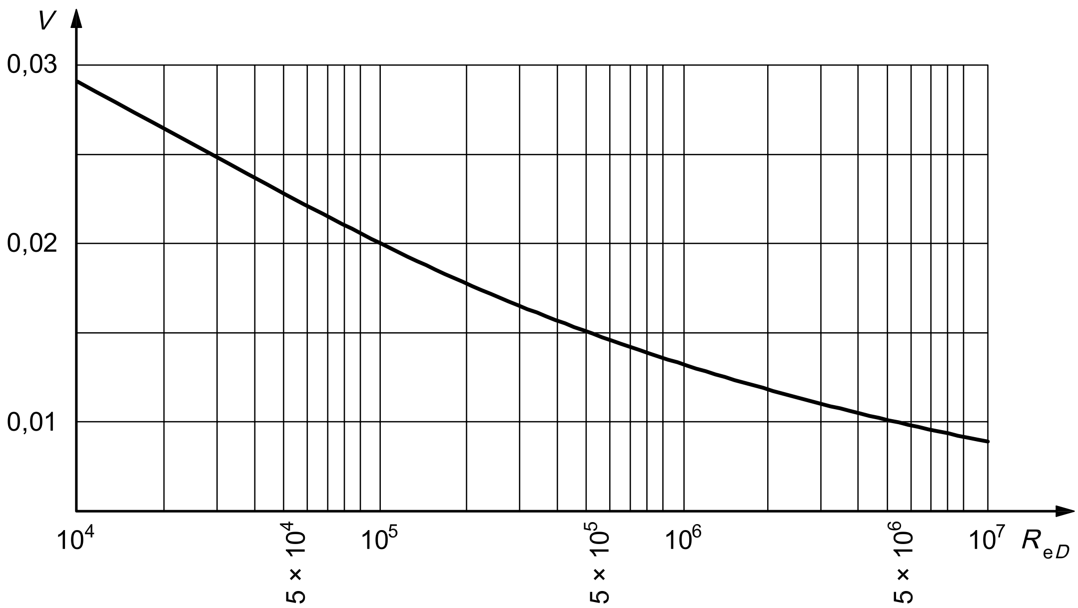
Conventional allowances given in this subclause shall be made for airway friction in tests with standardized airways. The friction allowance factors are shown in Figure 35.

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 28.2 and 28.3 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 28.5 is used (in which case they cover the bellmouth entry loss).



a) Conventional loss coefficient for standardized airways (28.6.1)



b) Friction-loss coefficients for ducts (28.6.2)

Figure 35 — Loss coefficients

28.6.1 Loss allowances for straight common outlet segments described in 28.2.1, 28.2.2 and 28.2.3

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

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

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

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

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

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

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

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 $(\xi_{2-4})_4$ is plotted against the Reynolds number, see Figure 35 a).

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

$$\Delta p_{2-4} = \left(\xi_{2-4}\right)_4 \frac{\rho_4 v_{\mathrm{m}4}^2}{2} f_{\mathrm{M}4}$$

28.6.2 Loss allowances for common outlet segments described in 28.2

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

$$\Lambda = 0{,}14\left(Re_{D\mathrm{h}4}\right)^{-0{,}17}$$

and is shown plotted in Figure 35 b).

The ratio of equivalent length of a cell-straightener to hydraulic diameter D_{h} ($D_{\mathrm{h}} = D_4$ for a circular duct) is given by the following equation:

$$\frac{L_{\mathrm{e}}}{D_{\mathrm{h}}} = \frac{15{,}04}{\left[1 - 26{,}65 \dfrac{e}{D_{\mathrm{h}}} + 184{,}6 \left(\dfrac{e}{D_{\mathrm{h}}}\right)^2\right]^{1{,}83}}$$

The conventional loss coefficient of the common outlet segment described in 28.2.1, 28.2.2 or 28.2.3 is given by the following equation [see Figure 35 a)]:

$$\left(\xi_{2-4}\right)_4 = \Lambda\left[\frac{L_{2-4}}{D_{\mathrm{h}4}} + \frac{L_{\mathrm{e}}}{D_{\mathrm{h}4}}\right]$$

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

28.6.3 Energy loss allowances for a short outlet duct described in 28.2.5

The duct friction shall not be considered.

28.6.4 Energy loss allowances for a common inlet segment described in 28.3

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

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

and

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

where

$$Re_{D3} = \frac{v_{\mathrm{m}3}\, D_3\, \rho_3}{\mu_3} \approx \frac{v_{\mathrm{m}3}\, D_3}{15} \times 10^6$$

in standard air.

The conventional loss coefficient

$$\left(\xi_{3-1}\right)_3 = -\left(\xi_{3-1}\right)_3$$

is always negative and is shown in Figure 35 a).

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

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

28.6.5 Energy loss allowances for inlet duct simulation described in 28.5

There are no losses allowed for this inlet duct, unless an inlet duct corresponding to the common segments described in 28.3 or other is required.

29 Standardized test chambers

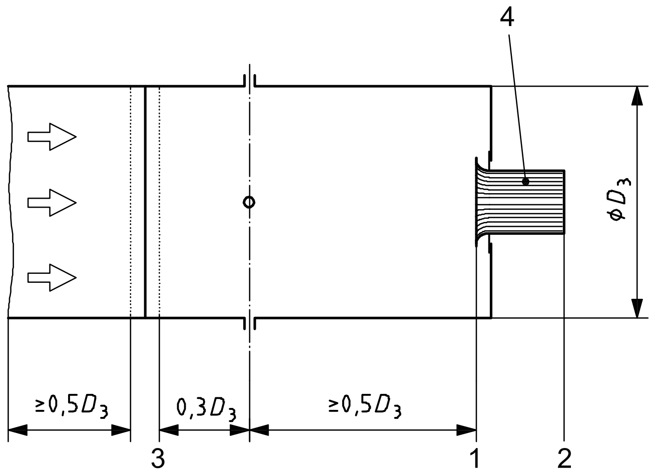
29.1 Test chamber

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

29.1.1 Dimensions

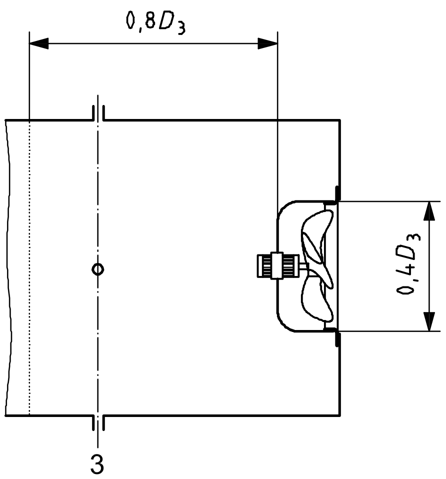
The test chamber cross-section may be circular, square or rectangular.

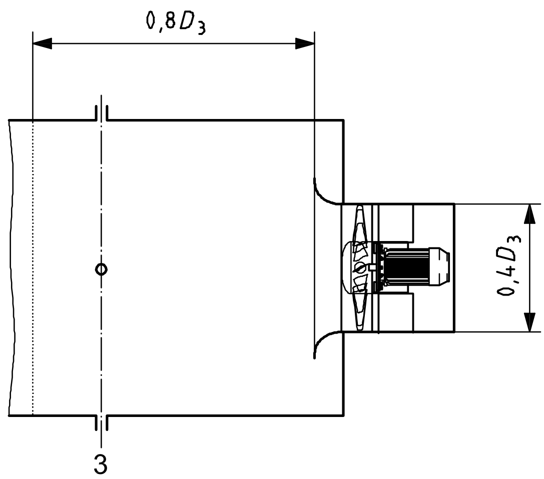
The length should be sufficient to accommodate any fan to be tested without infringing on the minimum distance shown in Figures 36 and 37.



NOTE For test airways for flow rate control and measurement, see Figure 40 and Clause 31.

a) Inlet chamber dimensions

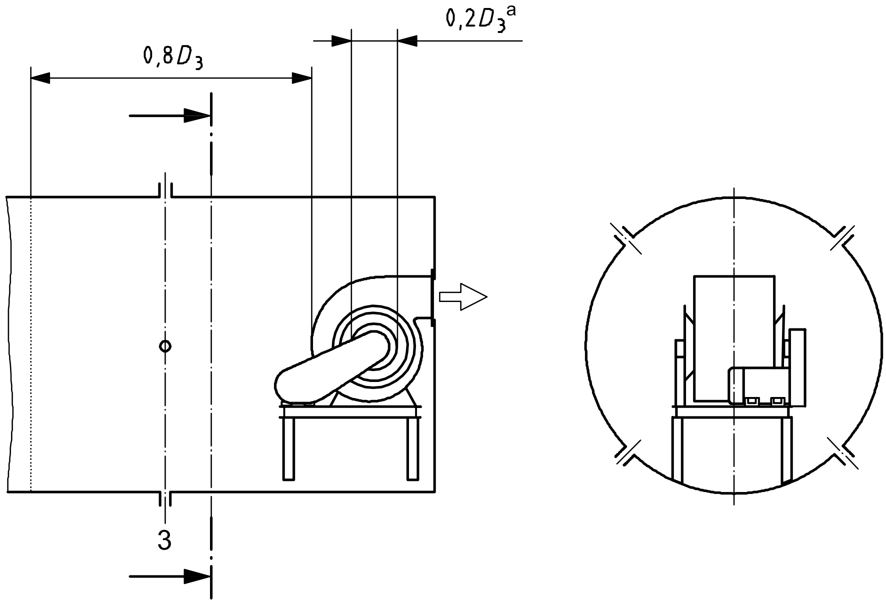




NOTE The fans illustrated have the maximum permissible dimensions.

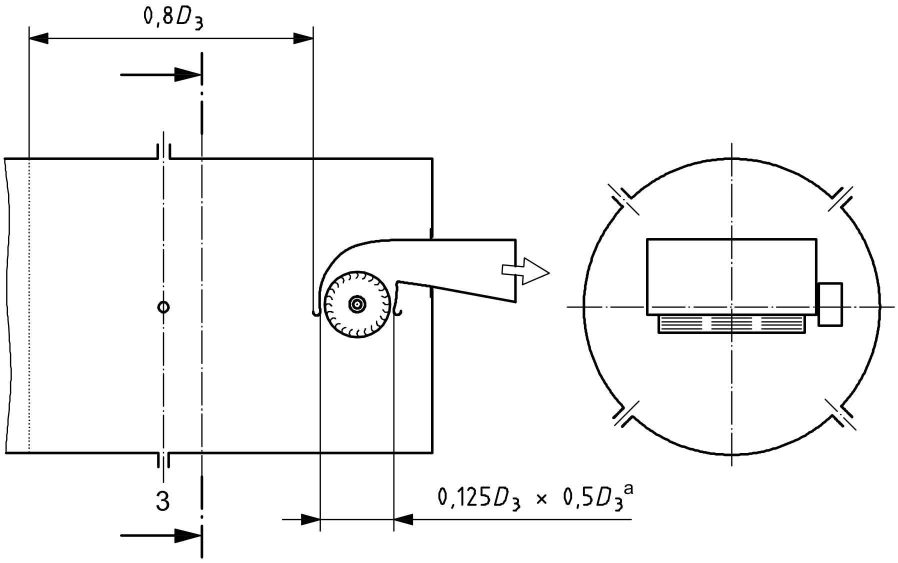
b) Example of a propeller fan **c) Example of an axial fan**

Figure 36 — Examples of inlet-side test chambers type 1



NOTE The fan illustrated has the maximum permissible dimensions.

d) Example of a double-inlet centrifugal fan



NOTE The fan illustrated has the maximum permissible dimensions.

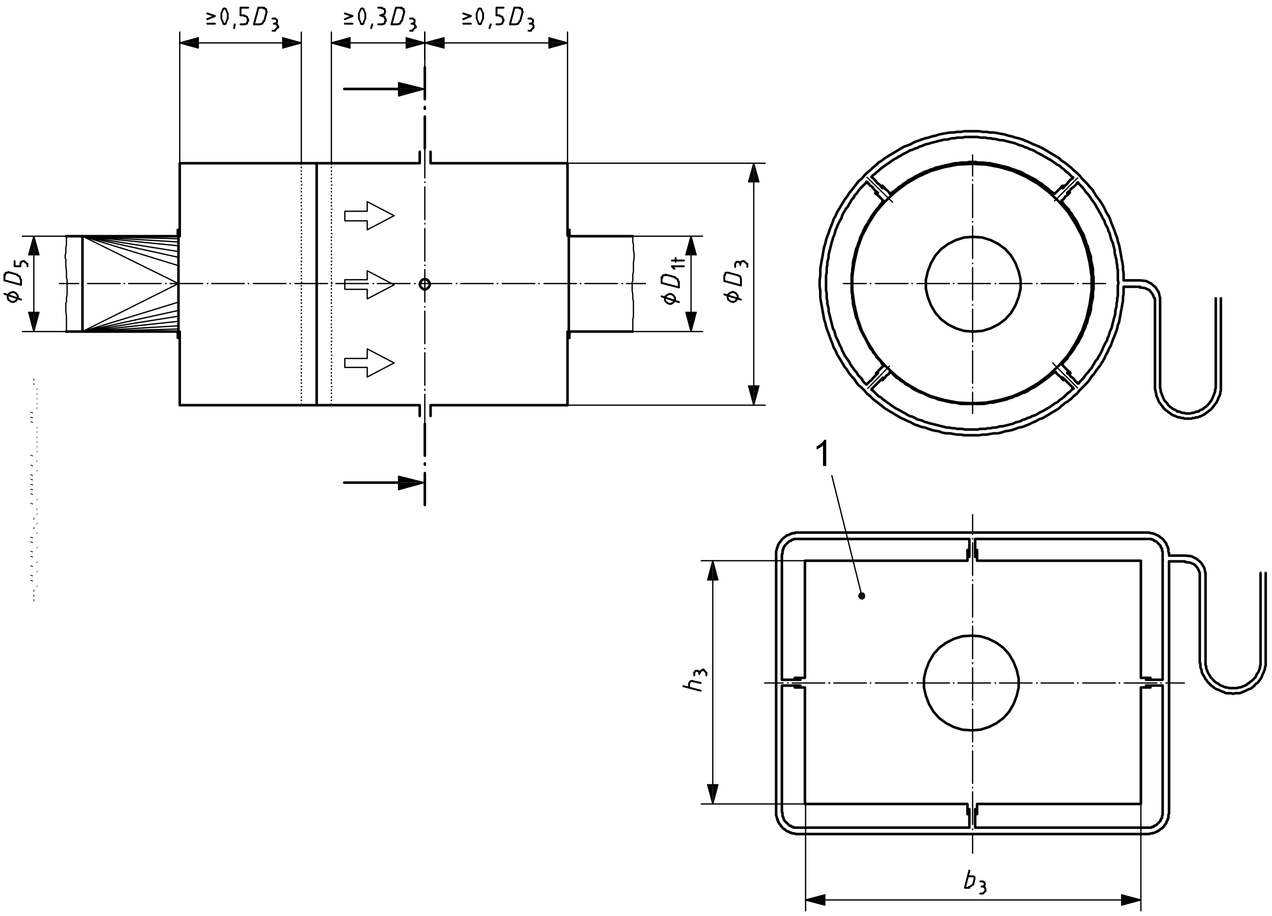
e) Example of a cross-flow fan

Key

1 test fan inlet
2 test fan outlet
3 pressure-measurement section
4 fan on test

[a] Inlet.

Figure 36 (*continued*)



Key

1 this figure shows a rectangular chamber

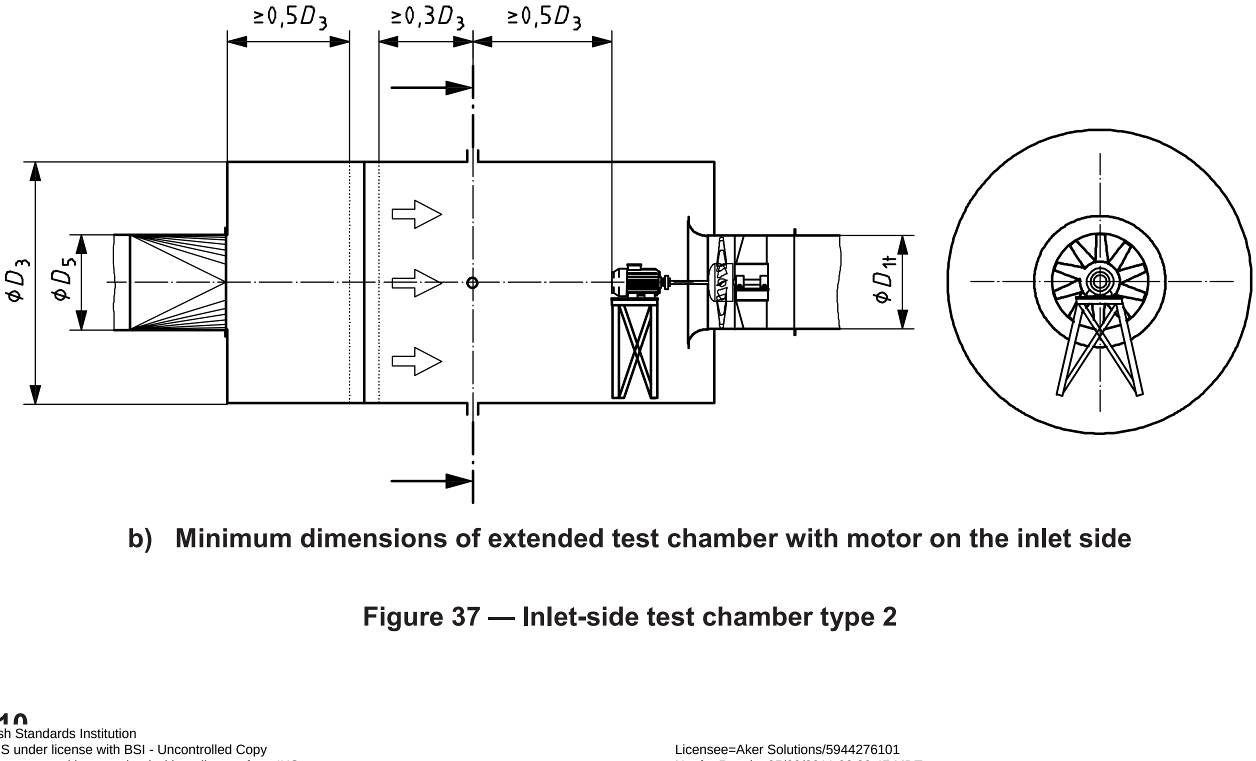
NOTE 1 Chamber diameter $D_3 \geqslant 2{,}5D_{1t}$.

NOTE 2 Equivalent chamber diameter $D_3 = \sqrt{h_3 b_3} \geqslant 2{,}5D_{1t}$.

NOTE 3 h_3 min. $\geqslant 2D_{1t}$.

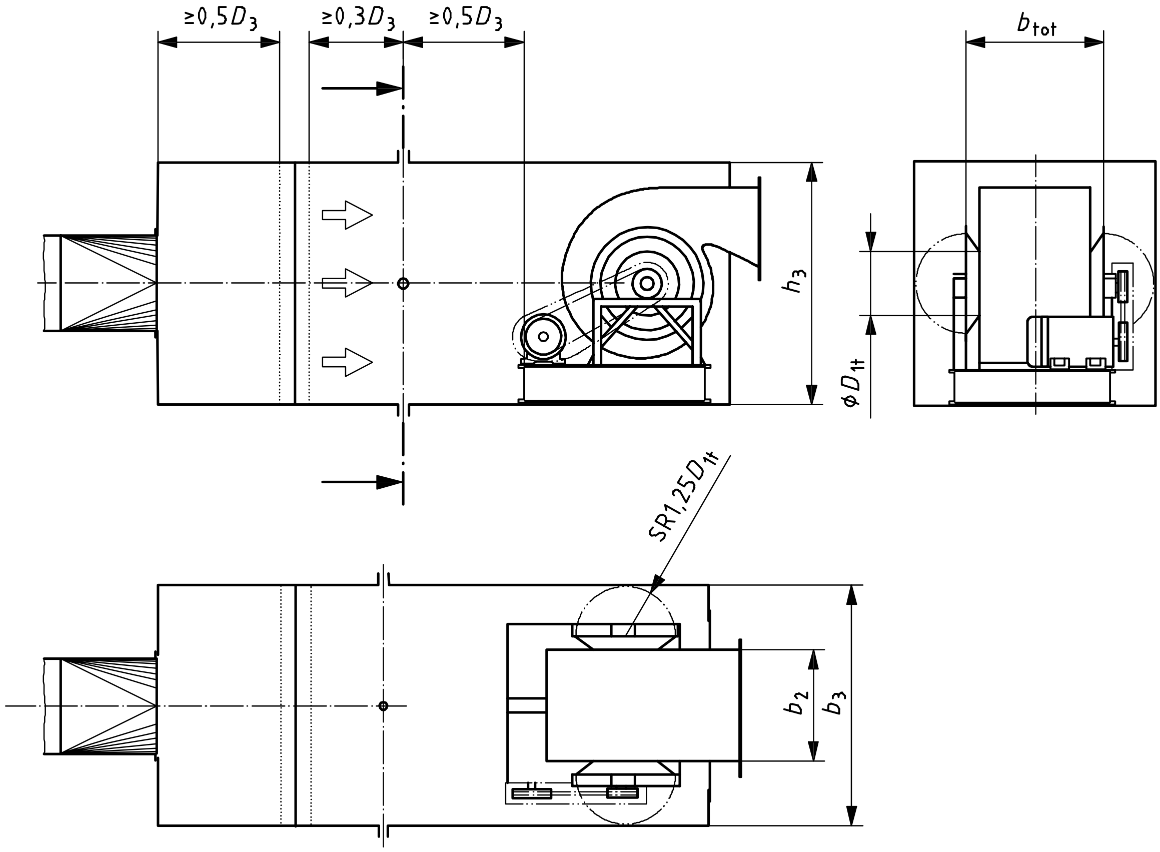
NOTE 4 b_3 max. $\geqslant 1{,}5h_3$.

a) Dimensions of inlet test chamber



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

Figure 37 — Inlet-side test chamber type 2



NOTE Example of double-inlet centrifugal fan.

c) Minimum dimensions of extended test chamber for the installation of double-inlet fans

Figure 37 (*continued*)

29.1.2 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.

29.1.3 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 $0,1D_h$ downstream of the screen shall not exceed the average velocity by more than 25 % unless the maximum velocity is less than 2 m/s.

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 areas 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.

29.1.4 Multiple nozzles

Multiple nozzles shall be located as symmetrically as possible. The centreline of each nozzle shall be at least $1{,}5d$, where d is the nozzle throat diameter, from the chamber wall. The minimum distance between centres of any two nozzles in simultaneous use shall be $3d$, with the d measurement taken from the larger nozzle.

The distance from the exit face of the largest nozzle to the downstream settling means shall be a minimum of $2{,}5d$, with the d measurement taken from the largest nozzle.

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

29.1.5 Orifice plate in chamber

The orifice shall be coaxial within the chamber to within ± 1° and $\pm\, 0{,}005D_h$ (see 24.2). 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 24.2.

29.2 Variable supply and exhaust systems

A means of varying the point of operation shall be provided in a laboratory setup.

29.2.1 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.

29.2.2 Auxiliary fans

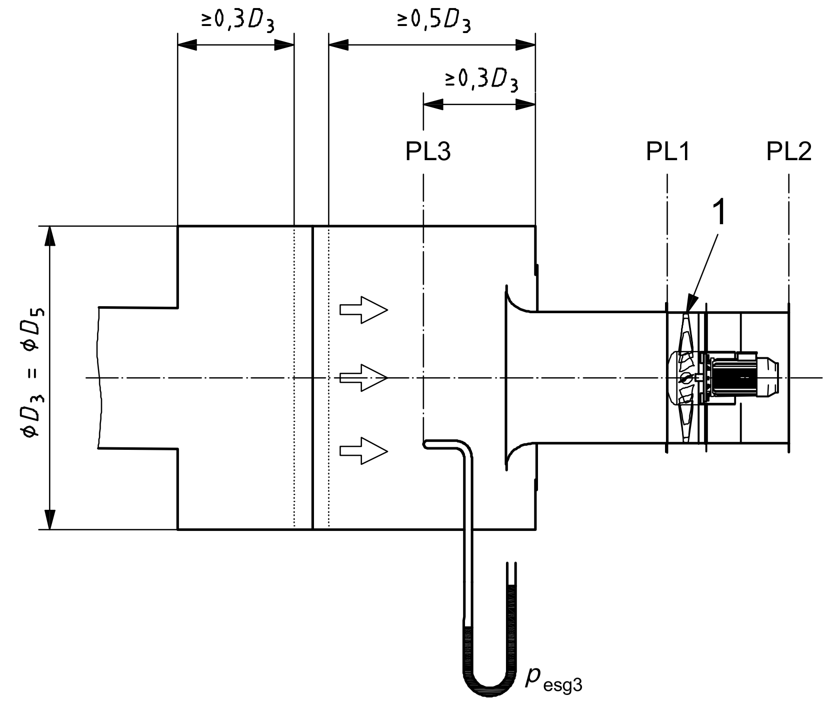
Auxiliary fans may be used to control the point of operation of the test fan. They shall be designed to produce sufficient pressure at the desired flow rate to overcome losses through the test setup. Flow-adjustment means, such as dampers, pitch control or speed control may be required. Auxiliary fans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

29.3 Standardized inlet test chambers

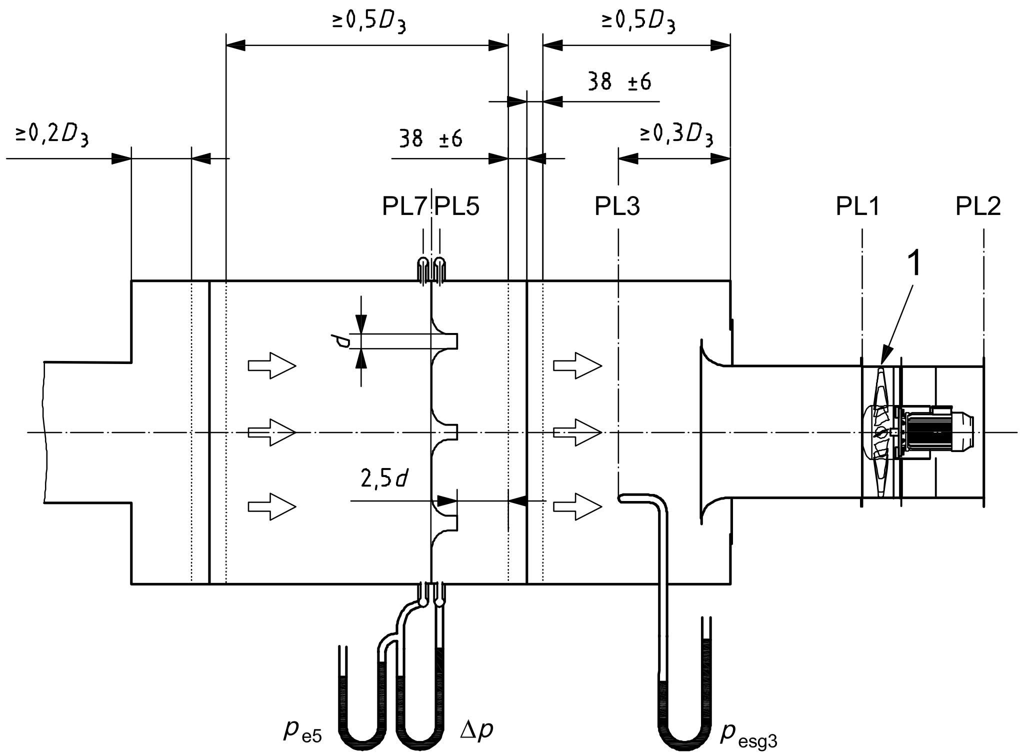
29.3.1 Test chamber

Three types of inlet test chamber are described in this International Standard (see Figures 36, 37 and 38).

Dimensions in millimetres



a)



b)

Key

1 fan

Figure 38 — Inlet-side test chamber type 3

29.3.1.1 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 being 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 36.

29.3.1.2 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 $b_3 h_3$ with $b_3 < 1{,}5 h_3$ and the equivalent chamber diameter:

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

For fans with an inlet-side drive 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 37.

29.3.1.3 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

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

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 flow rate measurement (see Figure 38).

29.3.2 Fan under test

29.3.2.1 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 36 and 37.

29.3.2.2 Inlet chamber type 2

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

or

$$A_{1t} < A_3/6{,}25$$

or

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

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 37.

29.3.2.3 Inlet chamber type 3

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

$$A_3 > 5A_1$$

They may optionally be fitted with multinozzles for flow rate measurement (see Figure 38).

29.4 Standardized outlet test chambers

29.4.1 Test chamber (see Figure 39)

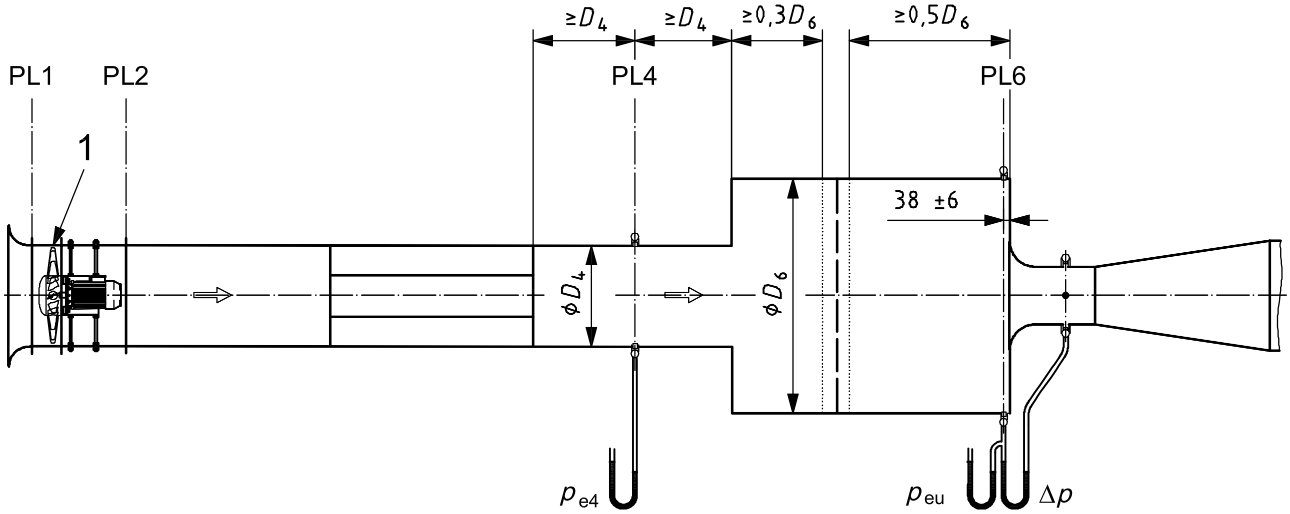
The test chamber cross-section may be circular with inside diameter D_6, square $D_6 \times D_6$ or rectangular $h_6 b_6$.

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

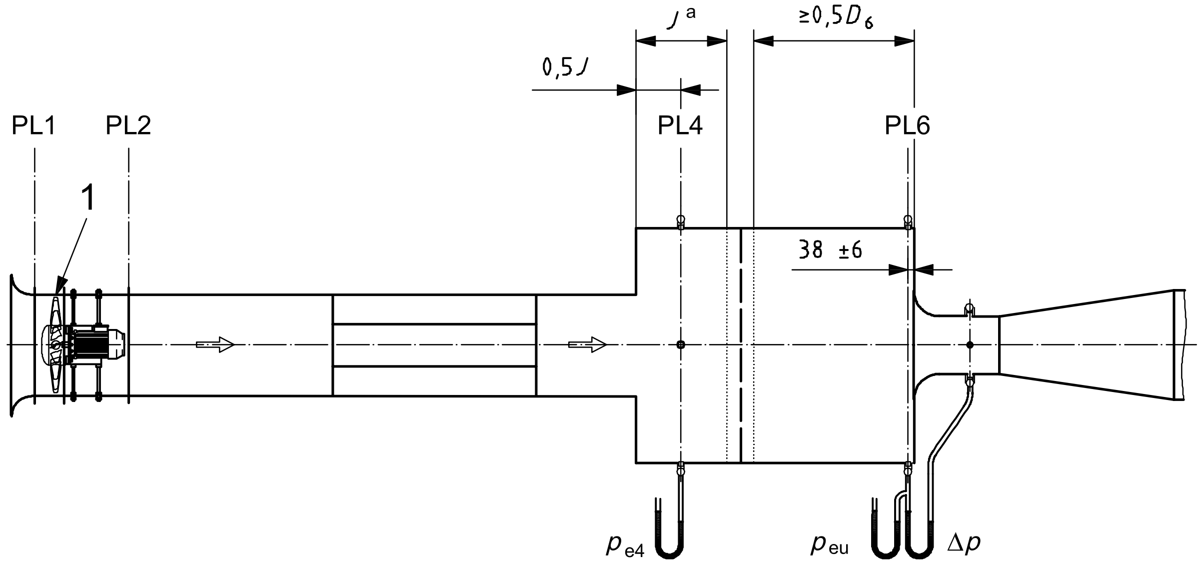
$$D_6 = \sqrt{\frac{4 h_6\, b_6}{\pi}}$$

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 single or multiple nozzles for flow rate measurement (see Figure 39).

Dimensions in millimetres



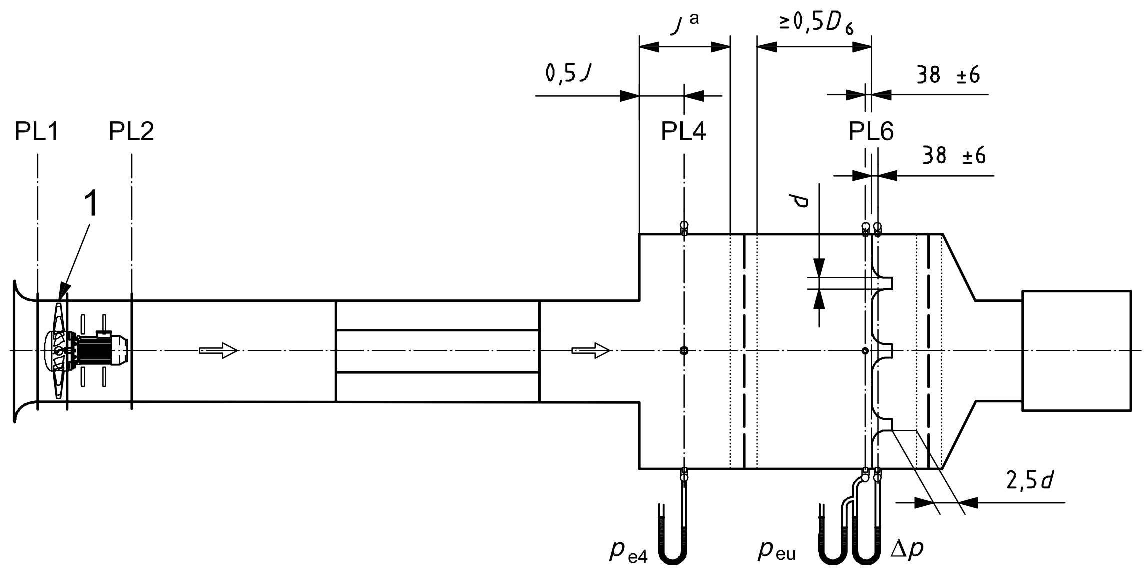
a)



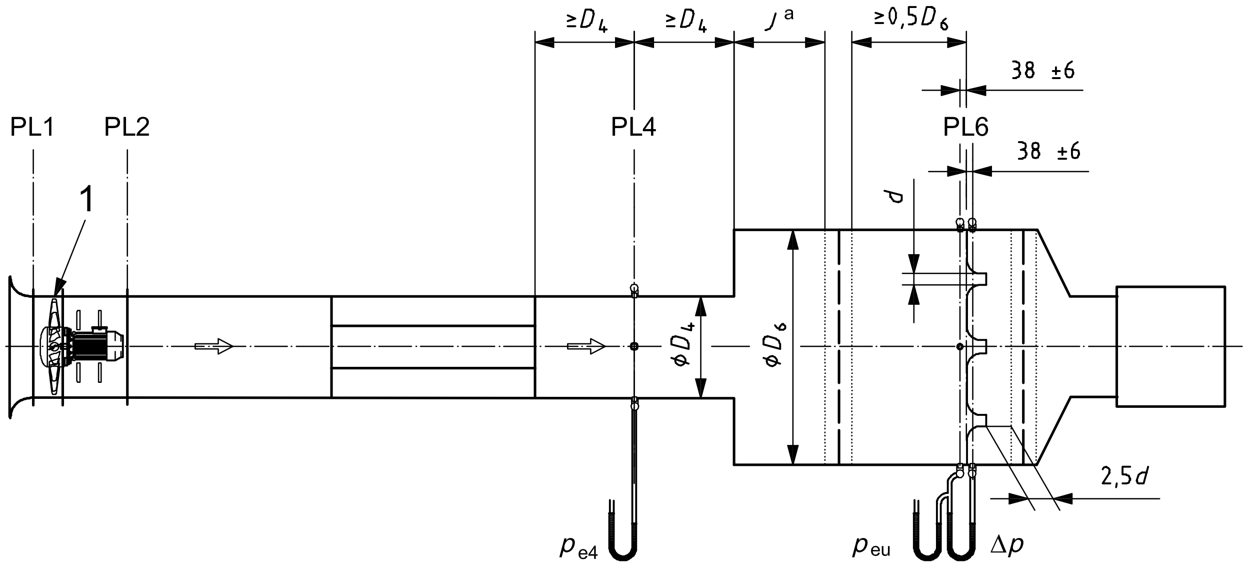
b)

Figure 39 — Outlet-side test chambers

Dimensions in millimetres



c)



d)

Key

1 fan

[a] 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 39 (*continued*)

29.4.2 Fan under test

An outlet test chamber (see Figure 39) 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 ($A_6 > 9A_2$) 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 ($A_6 > 16A_2$).

30 Standard methods with test chambers — Category A installations

30.1 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.

A number of methods of controlling and measuring the flow rate using inlet or outlet chambers are shown. The method of flow rate measurement is specified in each case, together with the clauses and figures giving details of the flow-measurement procedure.

A common procedure, comprising measurements to be taken and quantities to be calculated, allowing the determination of fan performance in category A installations, together with a number of methods for determining flow rate in the case of inlet chamber setup and two methods in the case of outlet chamber setup are given in 30.2 and 30.3.

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.

The simplified procedure may be followed when the reference Mach number Ma_{2ref} is less than 0,15 and the pressure ratio is less than 1,02.

The procedures for these cases are given in 30.2.4.

30.2 Inlet-side test chambers

30.2.1 Flow rate determination

The flow rate is determined by:

— conical or bellmouth inlet, see Figure 40 a);

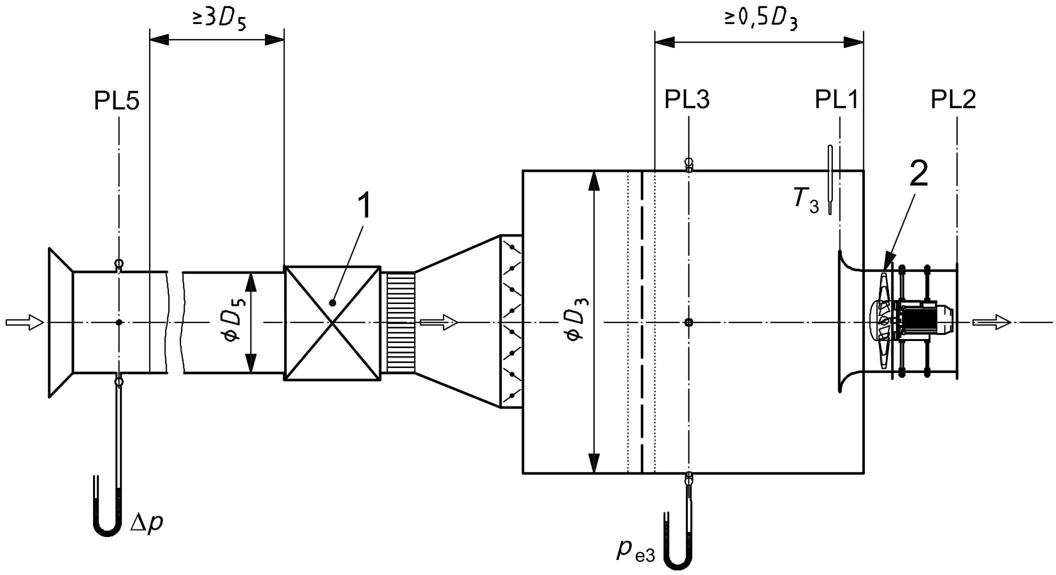
— inlet orifice with wall taps, see Figure 40 b);

— in-duct orifice with D and $D/2$ taps, see Figure 40 c);

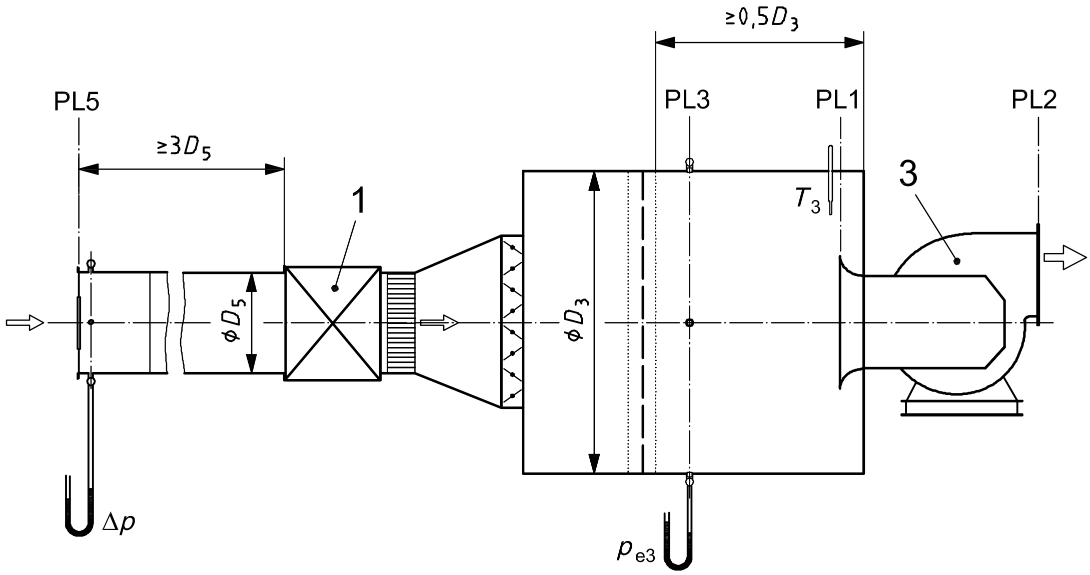
— Pitot-static tube traverse, see Figure 40 d);

— multiple nozzles in chamber, see Figure 40 e);

— orifice plate in chamber, see Figures 40 e) and 20 e).

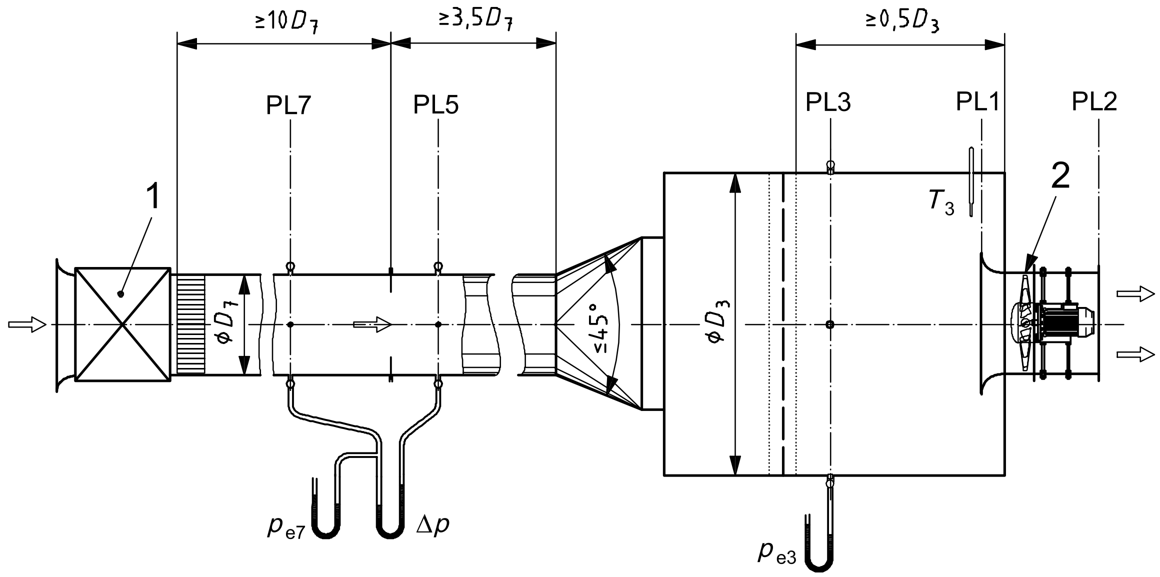


a) Flow rate determination using ISO conical or bellmouth inlet

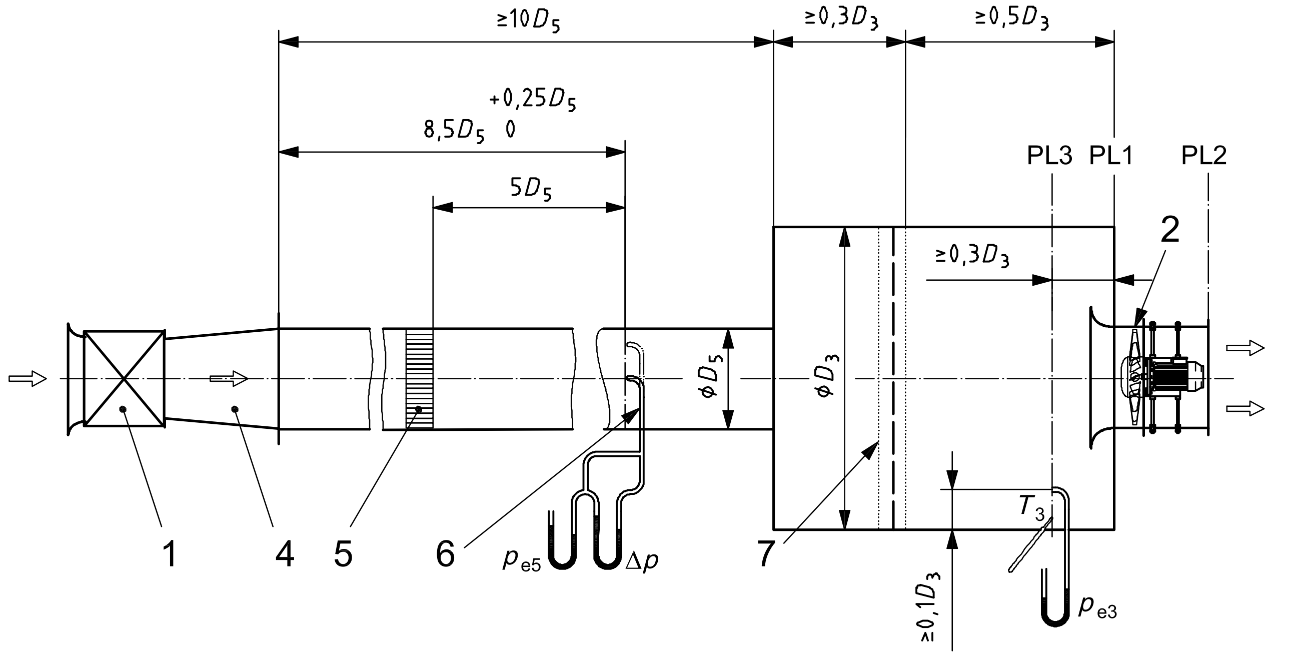


b) Flow rate determination using inlet orifice with wall taps

Figure 40 — Category A test installations (inlet-side test chamber)



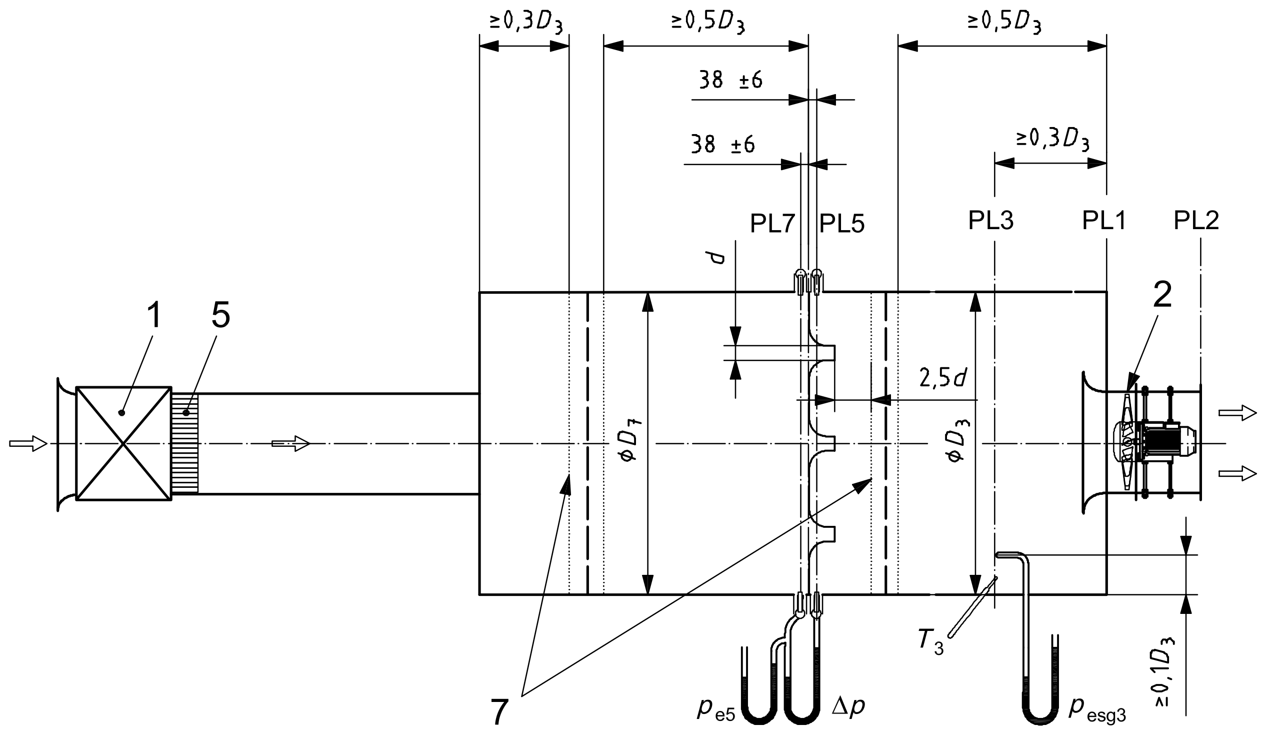
c) Flow rate determination using in-duct orifice with wall taps at D and 0,5D



d) Flow rate determination using Pitot-static tube traverse

Figure 40 (*continued*)

Dimensions in millimetres



e) Flow rate determination using multi-nozzle chamber

Key

1 auxiliary fan
2 test fan (centrifugal, shown with an integral inlet box)
3 test fan (tube-axial type shown)
4 transition section
5 flow straightener
6 Pitot-static tube traverse
7 flow-settling means

NOTE See 4.2 for plane descriptions.

Figure 40 (*continued*)

30.2.2 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 40 and chamber stagnation pressure for Figure 40 d) and e);

— chamber temperature, T_3.

In the test enclosure, measure

— atmospheric pressure, p_a, at the mean altitude of the fan;

— ambient temperature near the 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).

30.2.3 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).

30.2.3.1 Calculation of flow rate

30.2.3.1.1 The flow rate is determined by

— conical or bellmouth inlet, see Clause 23 and Figure 40 a);

— inlet orifice with wall taps, see Clause 24 and Figure 40 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 23.4 and Figure 19 for a conical or bellmouth inlet or in accordance with 24.5 for an inlet orifice with wall taps.

The mass flow rate is given by the following equation:

$$q_m = \alpha \varepsilon \pi \frac{d_5^2}{4} \sqrt{2 \rho_7 \Delta p}$$

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 equation above.

30.2.3.1.2 The flow rate is determined using an in-duct orifice with taps at D and D/2, see 24.7 and Figure 40 c).

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 flow rate is determined by the following equation:

$$q_m = \alpha\varepsilon \frac{\pi d_5^2}{4} \sqrt{2\rho_7 \Delta p}$$

The expansibility coefficient is determined in accordance with 24.7 and 24.8.

After an estimation of the flowmeter Reynolds number:

$$Re_{d5} = \frac{\alpha\varepsilon d_5 \sqrt{2\rho_7 \Delta p}}{\left(17{,}1 + 0{,}048 T_7\right)} \times 10^6$$

or

$$Re_{D7} = Re_{d5} \beta$$

the flow coefficient, α, or the compound coefficient, $\alpha\varepsilon$, are determined in accordance with 24.7 and Figure 21 for an in-duct orifice with taps at D and D/2.

A first approximation of q_m may be obtained with $\Theta_7 = \Theta_{sg7}$; Θ_7 may be determined and new values of α and q_m calculated.

Two iterations are sufficient for a calculation accuracy of three places of decimals.

30.2.3.1.3 The flow rate is determined using a Pitot-static tube traverse [see Clause 25 and Figure 40 d)].

A control device or an auxiliary fan with a control device is set upstream of the duct for flow rate 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 25.5.

The mass flow is determined by the following equation:

$$q_m = \alpha \varepsilon A_5 \sqrt{2 \rho_5 \Delta p_{\mathrm{m}}}$$

where

$$\rho_5 = \frac{p_5}{R_{\mathrm{w}} \Theta_5}$$

$$\varepsilon = \left[1 - \frac{1}{2\kappa} \frac{\Delta p_{\mathrm{m}}}{p_5} + \frac{\kappa + 1}{6\kappa^2} \left(\frac{\Delta p_{\mathrm{m}}}{p_5} \right)^2 \right]^{1/2}$$

α is a function of Reynolds number Re_{D5} very close to 0,99 (see 25.6).

A first approximation of q_m is calculated with $\alpha = 0{,}99$ and corrected for α variation.

30.2.3.1.4 The flow rate is determined using multiple nozzles in chamber [see Clause 22 and Figure 40 e)].

A control device or an auxiliary fan with a control device is set upstream of the chamber.

Assuming that

$$\Theta_3 = \Theta_{\mathrm{sg3}} = \Theta_{\mathrm{sg7}} = \Theta_7 = T_3 + 273{,}15$$

$$p_7 = p_{\mathrm{e7}} + p_{\mathrm{a}}$$

$$\rho_7 = \frac{p_7}{R_{\mathrm{w}} \Theta_7}$$

$$\beta = \frac{d_5}{D_7} \approx 0$$

The mass flow rate is given by the following equation in accordance with 22.4:

$$q_m = \varepsilon \pi \sum_{j=1}^{n} \left(C_j \frac{d_{5j}^2}{4} \right) \sqrt{2 \rho_7 \Delta p}$$

where

- ε is the expansibility coefficient in accordance with 22.4.3 and Table 5;
- 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 22.4.2 and Table 4;
- n is the number of nozzles.

For each nozzle, the throat Reynolds number Re_{d5} is estimated by the following equation:

$$Re_{d5j} = \frac{\varepsilon C_j d_{5j} \sqrt{2 \rho_7 \Delta p}}{17{,}1 + 0{,}048\, T_7} \times 10^6$$

with $C_j = 0{,}95$.

After a first estimation of the mass flow rate, the discharge coefficients C_j are corrected.

30.2.3.1.5 The flow rate is determined using an orifice plate in the test chamber with wall tappings [see 24.8.1 and Figures 40 e) and 20 e)].

An orifice plate is fitted instead of the multiple Venturi nozzles.

Assuming that

$$\Theta_3 = \Theta_{sg3} = \Theta_{sg7} = \Theta_7 = T_3 + 273{,}15$$

$$p_7 = p_{e7} + p_a$$

$$\rho_7 = \frac{p_7}{R_w \Theta_7}$$

$$\beta = \frac{d_5}{D_7} < 0{,}25$$

The mass flow rate is given by the following equation in accordance with 24.5:

$$q_m = \alpha \varepsilon \pi \frac{d_5^2}{4} \sqrt{2 \rho_7 \Delta p}$$

where $\alpha\varepsilon$ is determined in accordance with 24.5.

30.2.3.2 Calculation of fan pressure

30.2.3.2.1 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 40 a) to d). For the setup in Figure 40 e), the stagnation pressure p_{esg3} is measured by a Pitot-static tube and

$$p_{sg1} = p_{esg3} + p_a = p_{sg3}$$

$$p_{esg1} = p_{esg3}$$

$$p_{e3} < 0 \text{ and } p_{esg1} < 0$$

In accordance with 14.4.3.2 and 14.5.2, Ma_1, Θ_1/Θ_{sg1}, and p_1 may be determined.

The inlet static pressure p_1 is given by the following equation:

$$p_1 = p_{sg1} - p_{d1} f_{M1} = p_{sg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2 f_{M1}$$

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.

30.2.3.2.2 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.

This may also be written

$$p_{esg2} = \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 f_{M2}$$

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

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

30.2.3.2.3 Fan pressure

The fan static pressure p_{sfA} is given by the following equation:

$$p_{sfA} = p_2 - p_{sg1} = p_a - p_{sg1} = -p_{esg1}$$

and the fan pressure p_{fA} by

$$p_{\mathrm{fA}} = p_{\mathrm{sg2}} - p_{\mathrm{sg1}} = p_{\mathrm{a}} + \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 f_{\mathrm{M2}} - p_{\mathrm{sg1}}$$

$$p_{\mathrm{fA}} = p_{\mathrm{esg2}} - p_{\mathrm{esg1}} = \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 f_{\mathrm{M2}} - p_{\mathrm{esg1}}$$

30.2.3.3 Calculation of volume flow rate

In the test conditions, the volume flow rate is calculated by the following equation:

$$q_{V\mathrm{sg1}} = \frac{q_m}{\rho_{\mathrm{sg1}}} = \frac{q_m}{\left(\dfrac{p_{\mathrm{sg1}}}{R_{\mathrm{w}}\Theta_{\mathrm{sg1}}}\right)}$$

30.2.3.4 Calculation of fan air power

30.2.3.4.1 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 equation:

$$W_{m\mathrm{sA}} = \frac{p_2 - p_1}{\rho_{\mathrm{m}}} - \frac{v_{\mathrm{m1}}^2}{2} = \frac{p_2 - p_1}{\rho_{\mathrm{m}}} - \frac{1}{2}\left(\frac{q_m}{A_1\rho_1}\right)^2$$

and the fan work per unit mass by

$$W_{m\mathrm{A}} = \frac{p_2 - p_1}{\rho_{\mathrm{m}}} + \frac{v_{\mathrm{m2}}^2}{2} - \frac{v_{\mathrm{m1}}^2}{2}$$

$$= \frac{p_2 - p_1}{\rho_{\mathrm{m}}} + \frac{1}{2}\left[\left(\frac{q_m}{A_2\rho_2}\right)^2 - \left(\frac{q_m}{A_1\rho_1}\right)^2\right]$$

The fan static power and the fan air power P_{usA} and P_{uA} are given by the following equations:

$$P_{\mathrm{usA}} = q_m \cdot W_{m\mathrm{sA}}$$

$$P_{\mathrm{uA}} = q_m \cdot W_{m\mathrm{A}}$$

30.2.3.4.2 Calculation of fan air power and compressibility coefficient

In accordance with 14.8.2:

$$P_{\mathrm{usA}} = q_{V\mathrm{sg1}} \cdot p_{\mathrm{sfA}} \cdot k_{\mathrm{ps}}$$

$$P_{\mathrm{uA}} = q_{V\mathrm{sg1}} \cdot p_{\mathrm{fA}} \cdot k_{\mathrm{p}}$$

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

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

30.2.3.5 Calculation of efficiencies

In accordance with 14.8.1, the efficiencies are given by the following equations:

— fan static efficiency:

$$\eta_{srA} = \frac{P_{usA}}{P_r}$$

— fan efficiency:

$$\eta_{rA} = \frac{P_{uA}}{P_r}$$

— fan static shaft efficiency:

$$\eta_{\text{saA}} = \frac{P_{\text{usA}}}{P_{\text{a}}}$$

— fan shaft efficiency:

$$\eta_{\text{aA}} = \frac{P_{\text{uA}}}{P_{\text{a}}}$$

30.2.4 Simplified method

The reference Mach number $Ma_{2\text{ref}}$ is less than 0,15 and pressure ratio less than 1,02 (see 14.8.5).

The air flow through the fan may be considered as incompressible.

$$\Theta_1 = \Theta_{\text{sg1}} = \Theta_3 = \Theta_{\text{sg3}} = \Theta_2 = \Theta_{\text{sg2}}$$

$$\rho_1 = \rho_2$$

$$f_{\text{M1}} = f_{\text{M2}} = 1$$

$$k_{\text{p}} = 1$$

30.2.4.1 Calculation of mass flow rate

The mass flow rate is determined according to 31.2.3.1.

30.2.4.2 Calculation of fan pressure

30.2.4.2.1 Fan inlet pressure

$$\rho_1 = \rho_{\text{sg1}} = \rho_{\text{sg3}} = \frac{p_3}{R_{\text{w}}\Theta_3}$$

$$p_{\text{sg1}} = p_3 + \frac{1}{2\rho_{\text{sg1}}}\left(\frac{q_m}{A_3}\right)^2$$

$$p_{\text{esg1}} = p_{\text{e3}} + \frac{1}{2\rho_1}\left(\frac{q_m}{A_3}\right)^2$$

$$p_1 = p_{\text{sg1}} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2$$

or

$$p_{\text{e1}} = p_{\text{esg1}} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2$$

except in the case of Figure 40 e), for which the stagnation pressure p_{esg3} is measured and $p_{\text{esg1}} = p_{\text{esg3}}$ or $p_{\text{sg1}} = p_{\text{sg3}}$.

30.2.4.2.2 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

$$p_{esg2} = \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2$$

30.2.4.2.3 Fan pressure

The fan pressures are given by the following equations:

$$p_{sfA} = p_2 - p_{sg1} = p_a - p_{sg1} = -p_{esg1}$$

$$p_{fA} = p_{sg2} - p_{sg1} = p_2 + \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2 - \left[p_3 + \frac{1}{2\rho_1}\left(\frac{q_m}{A_3}\right)^2\right]$$

$$= \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2 - \left[p_{e3} + \frac{1}{2\rho_1}\left(\frac{q_m}{A_3}\right)^2\right]$$

30.2.4.3 Calculation of volume flow rate

The volume flow rate at inlet conditions is determined by the following equation:

$$q_{Vsg1} = \frac{q_m}{\rho_{sg1}} = \frac{q_m}{\left(\dfrac{p_{sg1}}{R_w \Theta_{sg1}}\right)}$$

30.2.4.4 Calculation of fan air power

The fan air powers are determined by the following equations:

$$P_{usA} = q_{Vsg1} \cdot p_{sfA}$$

$$P_{uA} = q_{Vsg1} \cdot p_{fA}$$

Fan efficiencies are calculated in accordance with 14.8.1.

30.2.5 Fan performances under test conditions

Under test conditions, the fan performances are:

- inlet volume flow, $q_{V\text{sg1}}$;
- fan static pressure, p_{sfA};
- fan pressure, p_{fA};
- fan efficiency, η_{srA} or η_{rA}.

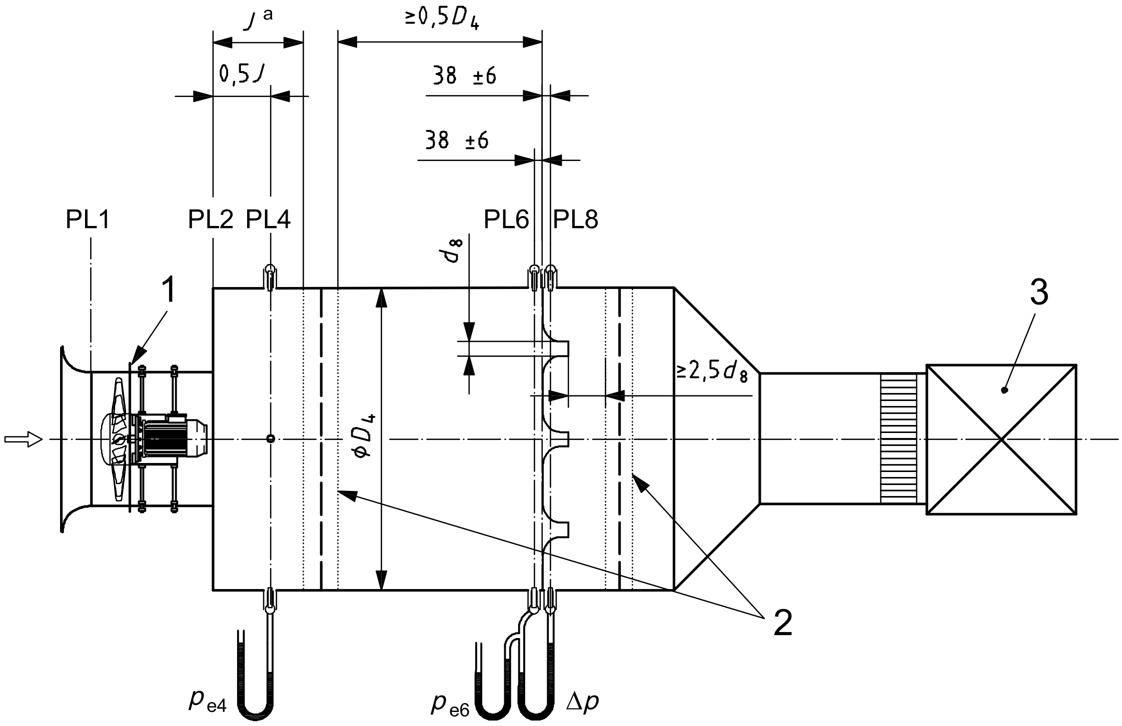
30.3 Outlet-side test chambers

30.3.1 Flow rate determination

The flow rate is determined using:

- multiple nozzles in chamber (see Clause 22, Figure 41);
- an orifice plate in chamber (see 24.8.1, Figure 41).

Dimensions in millimetres



Key

1 test fan (vane-axial type shown)

2 flow-settling means

3 variable exhaust system

[a] 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 41 —Category A test installations (outlet-side multiple nozzle test chamber)

30.3.2 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.

30.3.3 General procedure for compressible fluid flow

This procedure should be applied when the reference Mach number Ma_{2ref} is greater than 0,15 and the pressure ratio greater than 1,02.

30.3.3.1 Calculation of mass flow rate

30.3.3.1.1 The mass flow rate is determined using multiple nozzles in the chamber (see Clause 22 and Figure 41).

The chamber is followed by a control device or an auxiliary fan with a control device.

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 flow rate is given by the following equation in accordance with 22.4:

$$q_m = \varepsilon \pi \sum_{j=1}^{n} \left(C_j \frac{d_{8j}^2}{4} \right) \sqrt{2 \rho_6 \Delta p}$$

where

ε is the expansibility coefficient in accordance with 22.4.3 and Table 5;

C_j is the discharge coefficient of the jth nozzle as a function of the nozzle throat Reynolds number Re_{d8j}, see 22.4;

$\beta = 0$ and $C_j = \alpha$;

$C_j = \alpha_j$ is calculated in accordance with 22.4 and Table 4;

n is the number of nozzles, equal to 1 for a nozzle at the end of the chamber.

For each nozzle, the throat Reynolds number Re_{d8} is estimated using the following equation:

$$Re_{d8j} = \frac{\varepsilon C_j d_{8j} \sqrt{2\rho_6 \Delta p}}{17{,}1 + 0{,}048\ T_6} \times 10^6$$

with $C_j = 0{,}95$.

After a first estimation of the mass flow rate, the discharge coefficients, C_j, are corrected for the Reynolds number variations.

30.3.3.1.2 The mass flow rate is determined using an orifice plate in the test chamber with wall tappings, see 24.8.1 and Figures 20 e), f) and g) and 41.

Assuming that

$$p_6 = p_a + p_{e6}$$

$$\Theta_6 = T_6 + 273{,}15 = \Theta_{sg6}$$

$$\frac{d_8}{D_6} = \beta < 0{,}25$$

$$\rho_6 = \frac{p_6}{R_w \Theta_6}$$

The mass flow rate is given by the following equation in accordance with 24.5.

$$q_m = \alpha\varepsilon\pi \frac{d_8^2}{4} \sqrt{2\rho_6 \Delta\rho}$$

where $\alpha\varepsilon$ is determined in accordance with 24.5 and 24.8.1.

30.3.3.2 Calculation of fan pressure

30.3.3.2.1 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 4.

$$\Theta_2 = \Theta_{sg2} \frac{Ma_2}{Ma_{sg2}}$$

$$\rho_2 = \frac{p_2}{R_w \Theta_2} = \frac{p_4}{R_w \Theta_2}$$

$$p_{sg2} = p_2 + \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 f_{M2}$$

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.

30.3.3.2.2 Fan inlet pressure

$$p_{sg1} = p_a;\ p_{esg1} = 0$$

$$\Theta_{sg1} = T_a + 273{,}15$$

The Mach number Ma_1, the ratio ρ_{sg1}/ρ_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

$$p_{e1} = -\frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2 f_{M1}$$

30.3.3.2.3 Fan pressure

The fan static pressure p_{sfA} and the fan pressure p_{fA} are given by the following equations:

$$p_{sfA} = p_2 - p_{sg1} = p_2 - p_a = p_{e2}$$

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

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

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

30.3.3.3 Calculation of the volume flow rate

Under the test conditions the volume flow rate is determined by the following equation:

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

where

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

30.3.3.4 Reference Mach number less than 0,15 and pressure ratio less than 1,02 (see 14.8.5)

The flow through the fan and the test airway may be considered as incompressible.

$$\Theta_1 = \Theta_{sg1} = \Theta_2 = \Theta_{sg2} = \Theta_4 = \Theta_{sg4} = \Theta_u = \Theta_a = T_a + 273{,}15$$

$$\rho_1 = \rho_2$$

$$f_{M1} = f_{M2} = 1$$

$$k_p = 1$$

30.3.3.4.1 Calculation of mass flow rate

The mass flow rate is determined in accordance with 31.3.3.1.

30.3.3.4.2 Calculation of fan pressure

30.3.3.4.2.1 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

$$p_{esg2} = p_{e4} + \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2$$

30.3.3.4.2.2 Fan inlet pressure

$$p_1 = p_{sg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2$$

$$p_{e1} = -\frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2$$

30.3.3.4.2.3 Fan pressure

The fan pressures are given by the following equations:

$$p_{sfA} = p_2 - p_{sg1} = p_4 - p_a = p_{e4}$$

$$p_{fA} = p_{sg2} - p_{sg1} = p_4 + \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2 - p_a$$

$$= p_{e4} + \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2 = p_{esg4}$$

30.3.3.4.3 Calculation of volume flow rate

$$q_{Vsg1} = \frac{q_m}{\rho_{sg1}} = \frac{q_m}{\left(\dfrac{p_{sg1}}{R_w \Theta_{sg1}}\right)}$$

30.3.3.4.4 Calculation of fan air power

The fan air powers are determined by the following equations:

$$P_{usA} = q_{Vsg1} \cdot p_{sfA}$$

$$P_{uA} = q_{Vsg1} \cdot p_{fA}$$

30.3.3.4.5 Calculation of fan efficiencies

Fan efficiencies are calculated in accordance with 14.8.1.

30.3.4 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}.

31 Standard test methods with outlet-side test ducts — Category B installations

31.1 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; $2D$ or $3D$ 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 flow rate in the test duct are shown in the first case, and two methods in the second case. The method of flow rate measurement is specified in each case, together with the clauses and figures giving details of the flow-measurement procedure.

A common procedure, comprising measurements to be taken and quantities to be calculated, allowing the determination of fan performance in category B installations, is given in 31.2.3 to 31.2.3.5. It is generally valid for all fans in accordance with this International Standard.

A simplified procedure may be followed when the reference Mach number $Ma_{2\text{ref}}$ is less than 0,15 and the pressure ratio less than 1,02. In these circumstances, the procedure which is given in 31.2.4 may be followed.

31.2 Outlet-side test ducts with antiswirl device

31.2.1 Mass flow rate determination

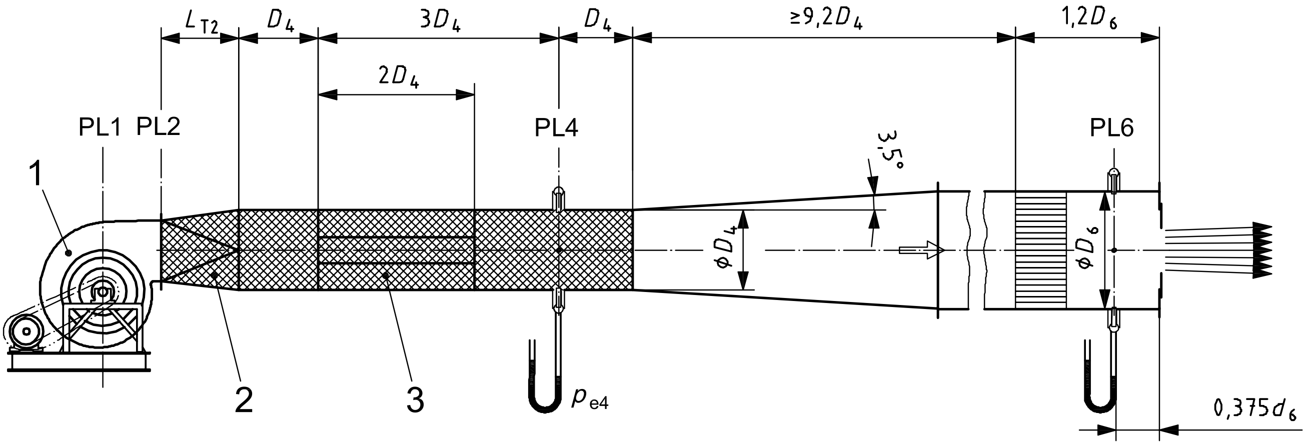
The mass flow rate is determined using:

— outlet orifice with wall taps, see 24.8 and Figure 42 a);

— in-duct orifice with D and D/2 taps, see 24.7 and Figure 42 b);

— Pitot-static tube traverse, see Clause 25 and Figure 42 c);

— multiple nozzles in chamber, see Clause 22 and Figure 42 d).



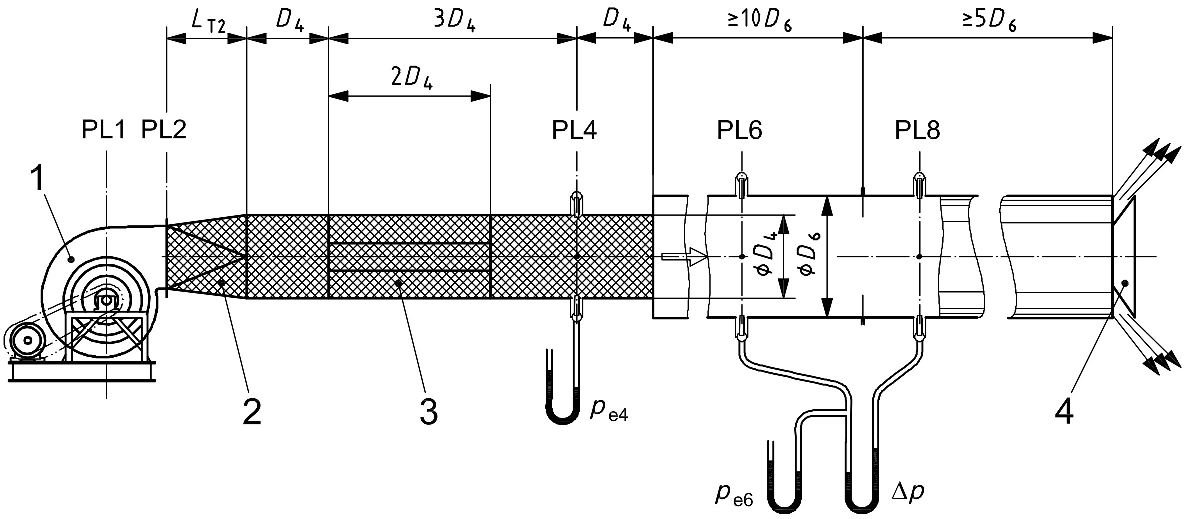
Key

1 test fan (open inlet centrifugal type shown)

2 transition duct rectangular to round

3 flow straightener (only required if swirl present)

a) Flow rate determination using an outlet orifice



Key

1 test fan (open inlet centrifugal type shown)

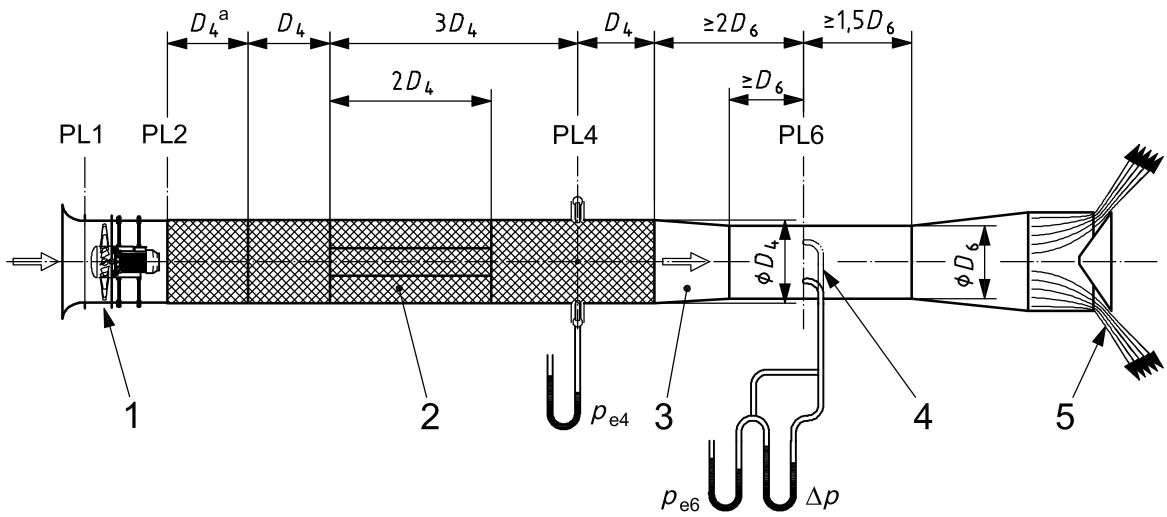
2 transition duct rectangular to round

3 flow straightener (only required if swirl present) (star type shown)

4 flow throttling device

b) Flow rate determination using an in-duct orifice with taps at D and $0,5D$

Figure 42 — Category B test installations (with antiswirl device)

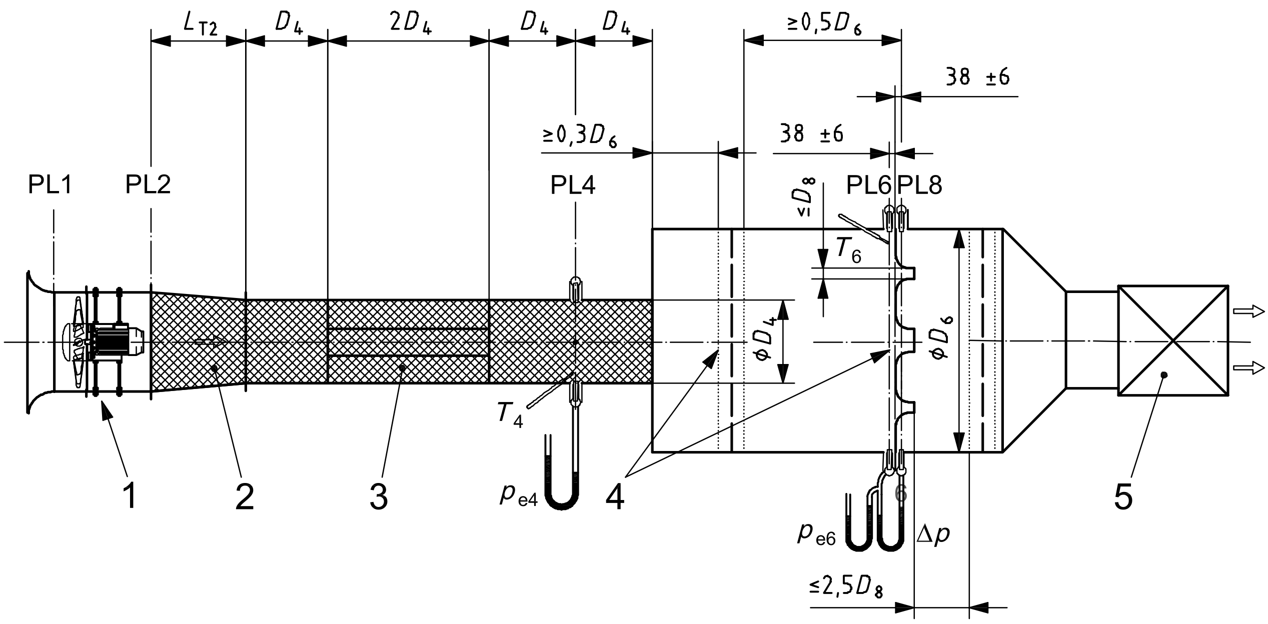


Key

1 test fan (tube-axial type shown)
2 flow straightener (only required if swirl present) (star type shown)
3 transition piece, convergent when $D_6 \neq D_4$; included angle 20°
4 Pitot-static tube traverse
5 flow throttling device

[a] 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.

c) Flow rate determination using a Pitot-static tube traverse



Key

1 test fan (vane-axial type shown)
2 transition section
3 flow straightener (only required if swirl present) (star type shown)
4 flow-settling means
5 auxiliary fan

d) Flow rate determination using a multiple nozzle chamber

Figure 42 (*continued*)

31.2.2 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).

31.2.3 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.

31.2.3.1 Calculation of mass flow rate

31.2.3.1.1 The mass flow rate is determined using:

— outlet orifice with wall taps, see 24.8 and Figure 42 a);

— in-duct orifice with taps at D and D/2, see 24.7 and Figure 42 b).

The outlet test ducts for pressure and flow rate 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} and q_m are unknown.

The mass flow rate is determined by the following equation:

$$q_m = \alpha \varepsilon \pi \frac{d_8^2}{4} \sqrt{2\rho_6 \Delta p}$$

where

ε is the expansibility coefficient determined in accordance with 24.7 and 24.8;

α is the flow coefficient function of Reynolds number Re_{d8} or Re_{D6} estimated by the following equations:

$$Re_{d8} = \frac{\alpha \varepsilon d_8 \sqrt{2\rho_6 \Delta p}}{17{,}1 + 0{,}048 T_6} \times 10^6$$

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 24.7, 24.8 and Figures 21, 22 and 23.

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 three places of decimals.

b) The temperature T_6 is measured. It is considered as a stagnation temperature Θ_{sg6}:

$$\Theta_6 = T_6 + 273{,}15 - \frac{q_m^2}{2 A_2^2 \rho_6^2 c_p}$$

and the above procedure is applied.

31.2.3.1.2 Flow rate is determined using a Pitot-static tube traverse, see Clause 25 and Figure 42 c) and d).

NOTE For the installation in Figure 42 d), plane 4 and plane 6 are identical.

The outlet ducts for pressure and flow rate measurements are followed by a control device or an auxiliary fan with a control device.

Assuming that

$$p_{e6} = \frac{1}{n} \sum_{j=n}^{n} p_{e6j}$$

$$p_6 = p_{e6} + p_a$$

$$\Theta_{\mathrm{sg6}} = \Theta_{\mathrm{sg1}} + \frac{P_{\mathrm{r}} \text{ or } P_{\mathrm{e}}}{q_m c_p}$$

$$\Theta_6 = \Theta_{\mathrm{sg6}} \left(\frac{p_6}{p_6 + \Delta p_{\mathrm{m}}} \right)^{\frac{\kappa - 1}{\kappa}}$$

where

$$\Delta p_{\mathrm{m}} = \left(\frac{1}{n} \sum_{j=1}^{n} \Delta p_j^{0,5} \right)^2$$

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

$$\rho_6 = \frac{p_6}{R_{\mathrm{w}} \Theta_6}$$

The mass flow rate q_m is determined by the following equation:

$$q_m = \alpha \varepsilon A_6 \sqrt{2 \rho_6 \Delta p_{\mathrm{m}}}$$

where α is a flow rate coefficient function of the Reynolds number Re_{D6} very close to 0,99 (see 25.6).

$$Re_{D6} = \alpha \varepsilon D_6 \frac{\sqrt{2 \rho_6 \Delta p_{\mathrm{m}}}}{17,1 + 0,048 T_6} \times 10^6$$

where ε is the expansibility coefficient (see 25.5):

$$\varepsilon = \left[1 - \frac{1}{2\kappa} \frac{\Delta p_{\mathrm{m}}}{p_6} + \frac{\kappa + 1}{6\kappa^2} \left(\frac{\Delta p_{\mathrm{m}}}{p_6} \right)^2 \right]^{1/2}$$

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 flow rate with a calculation accuracy of three places of decimals.

31.2.3.1.3 The mass flow rate is determined using multiple nozzles in chamber, see Clause 22 and Figure 42 d).

The outlet ducts, for pressure and flow rate measurements, are connected to a downstream flow control device or an auxiliary fan combined with a flow control device.

The temperature T_6 in the chamber may be measured:

$$p_6 = p_{\mathrm{e6}} + p_{\mathrm{a}}$$

$$\Theta_6 = \Theta_{sg6} = T_6 + 273{,}15$$

$$\beta = \frac{d_8}{D_6} = 0$$

$$\rho_6 = \frac{p_6}{R_w \Theta_6}$$

The mass flow rate is given by the following equation:

$$q_m = \varepsilon \pi \sum_{j=1}^{n} \left(C_j \frac{d_{8j}^2}{4} \right) \sqrt{2 \rho_6 \Delta p}$$

where

ε is the expansibility coefficient in accordance with 22.4.3 and Table 5;

C_j is the discharge coefficient of the jth nozzle, as a function of the nozzle throat Reynolds number;

Re_{d8j} see 22.4;

$\beta = 0$ and $C_j = \alpha_j$;

$C_j = \alpha_j$ is calculated in accordance with 22.4 and Table 4;

n is the number of nozzles, equal to 1 for a nozzle on the end of the chamber.

For each nozzle, the throat Reynolds number, Re_{d8}, is estimated using the following equation:

$$Re_{d8j} = \frac{\varepsilon C_j d_{8j} \sqrt{2 \rho_6 \Delta p}}{17{,}1 + 0{,}048\, T_6} \times 10^6$$

with $Cj = 0{,}95$.

After a first estimation of the mass flow rate, the discharge coefficients C_j are corrected for the Reynolds number variations.

31.2.3.2 Calculation of fan pressure

31.2.3.2.1 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 Ma_4/Ma_{sg4} are determined in accordance with 14.4.3.1 and Figure 4.

$$\Theta_4 = \Theta_{sg4} \frac{Ma_4}{Ma_{sg4}}$$

$$\rho_4 = \frac{p_4}{R_w \Theta_4}$$

$$f_{M4} = 1 + \frac{Ma_4^2}{4} + \frac{Ma_4^4}{40} + \frac{Ma_4^6}{1600} \text{ (see 14.5.1)}$$

The friction-loss coefficient between sections 2 and 4 $(\xi_{2-4})_4$ is calculated in accordance with 28.6 and Figure 35.

The stagnation pressure at fan outlet p_{sg2} is given by the following equation:

$$p_{sg2} = p_4 + \frac{\rho_4 v_{m4}^2}{2} f_{M4} \left[1 + \left(\xi_{2-4} \right)_4 \right]$$

or

$$p_{esg2} = p_{e4} + \frac{\rho_4 v_{m4}^2}{2} f_{M4} \left[1 + \left(\xi_{2-4} \right)_4 \right]$$

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

$$p_{e2} = p_{esg2} - \frac{1}{2\rho_2} \left(\frac{q_m}{A_2} \right)^2 f_{M2}$$

31.2.3.2.2 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:

$$p_{esg1} = 0$$

$$p_{e1} = -\frac{1}{2\rho_1} \left(\frac{q_m}{A_1} \right)^2 f_{M1}$$

31.2.3.2.3 Fan pressure

The fan pressure p_{fB} and the fan static pressure p_{sfB} may be calculated using the following equation:

$$p_{\text{fB}} = p_{\text{sg2}} - p_{\text{sg1}} = p_{\text{esg2}}$$

$$p_{\text{sfB}} = p_2 - p_{\text{sg1}} = p_{\text{e2}}$$

$$\rho_{\text{m}} = \frac{\rho_2 + \rho_1}{2}$$

and

$$k_\rho = \rho_1/\rho_{\text{m}}$$

31.2.3.3 Calculation of volume flow rate

The volume flow rate is calculated by the following equation:

$$q_{V\text{sg1}} = \frac{q_m}{\rho_{\text{sg1}}} = \frac{q_m}{\left(\dfrac{p_{\text{sg1}}}{R_{\text{w}}\Theta_{\text{sg1}}}\right)}$$

31.2.3.4 Calculation of fan air power

31.2.3.4.1 Fan work per unit mass and fan air power

In accordance with 14.8.1, the fan work per unit mass $W_{m\text{B}}$ and the fan static work per unit mass $W_{m\text{sB}}$ are given by the following equations:

$$\begin{aligned} W_{m\text{B}} &= \frac{p_2 - p_1}{\rho_{\text{m}}} + \frac{v_{\text{m}2}^2}{2} - \frac{v_{\text{m}1}^2}{2} \\ &= \frac{p_2 - p_1}{\rho_{\text{m}}} + \frac{1}{2}\left(\frac{q_m}{A_2\rho_2}\right)^2 - \frac{1}{2}\left(\frac{q_m}{A_1\rho_1}\right)^2 \\ &= \frac{p_{\text{e2}} - p_{\text{e1}}}{\rho_{\text{m}}} + \frac{v_{\text{m}2}^2}{2} - \frac{v_{\text{m}1}^2}{2} \end{aligned}$$

$$\begin{aligned} W_{m\text{sB}} &= \frac{p_2 - p_1}{\rho_{\text{m}}} - \frac{v_{\text{m}1}^2}{2} = \frac{p_2 - p_1}{\rho_{\text{m}}} - \frac{1}{2}\left(\frac{q_m}{A_1\rho_1}\right)^2 \\ &= \frac{p_{\text{e2}} - p_{\text{e1}}}{\rho_{\text{m}}} - \frac{v_{\text{m}1}^2}{2} \end{aligned}$$

The fan air power P_{uB} and the fan static power P_{usB} are given by the following equations:

$$P_{\text{uB}} = q_{\text{m}} W_{m\text{B}}$$

$$P_{\text{usB}} = q_{\text{m}} W_{m\text{sB}}$$

31.2.3.4.2 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:

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

where

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

for k_p or

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

for k_{ps} and

$$Z_k = \frac{\kappa - 1}{\kappa} \frac{P_r}{q_{Vsg1}\, p_{fC}}$$

for k_p or

$$Z_k = \frac{\kappa - 1}{\kappa} \frac{P_r}{q_{Vsg1}\, p_{sfC}}$$

for k_{ps}.

b) Second method:

$$k_{ps} \text{ or } k_p = \frac{\ln\left(1 + x\right)}{x} \frac{Z_p}{\ln\left(1 + Z_p\right)}$$

where

$$x = r - 1 = \frac{p_{fB}}{p_{sg1}} \quad \text{for} \quad k_p$$

or

$$x = \frac{p_{sfB}}{p_{sg1}} \quad \text{for} \quad k_{ps}$$

and

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

31.2.3.5 Calculation of efficiencies

In accordance with 14.8.1, the efficiencies are calculated by the following equations:

— fan efficiency:

$$\eta_{rC} = \frac{P_{uB}}{P_r}$$

— fan static efficiency:

$$\eta_{srC} = \frac{P_{usB}}{P_r}$$

— fan shaft efficiency:

$$\eta_{aC} = \frac{P_{uB}}{P_a}$$

— fan static shaft efficiency:

$$\eta_{saC} = \frac{P_{usB}}{P_a}$$

31.2.4 Simplified method

The reference Mach number Ma_{2ref} is less than 0,15 and pressure ratio less than 1,02 (see 14.8.5).

The air flow through the fan and the test airway may be considered as incompressible.

$$\Theta_1 = \Theta_{sg1} = \Theta_2 = \Theta_{sg2} = \Theta_a = T_a + 273{,}15$$

$$\rho_1 = \rho_2 = \rho_4 = \rho_6 = \rho_a = \frac{p_a}{R_w \Theta_a}$$

$$f_{M1} = f_{M2} = 1$$

$$k_p = 1$$

31.2.4.1 Calculation of mass flow rate

The mass flow rate is determined in accordance with 32.2.3.1

31.2.4.2 Calculation of fan pressure

31.2.4.2.1 Fan outlet pressure

According to the assumption above,

$$\rho_1 = \rho_{sg1} = \rho_2 = \rho_{sg2} = \rho_4 = \rho_6 = \rho_a = \frac{p_a}{R_w \Theta_a}$$

$$p_{sg2} = p_4 + \frac{1}{2\rho_1}\left(\frac{q_m}{A_4}\right)^2\left[1 + \left(\xi_{2-4}\right)_4\right]$$

$$p_{esg2} = p_{e4} + \frac{1}{2\rho_1}\left(\frac{q_m}{A_4}\right)^2\left[1 + \left(\xi_{2-4}\right)_4\right]$$

$$p_2 = p_{sg2} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2$$

$$p_{e2} = p_{esg2} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2$$

31.2.4.2.2 Fan inlet pressure

$$p_{sg1} = p_a$$

$$p_{esg1} = 0$$

$$p_1 = p_{sg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2$$

$$p_{e1} = p_{esg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2$$

31.2.4.2.3 Fan pressure

The fan pressure p_{fB} and the fan static pressure p_{sfB} may be determined by the following equations:

$$p_{fB} = p_{sg2} - p_{sg1} = p_{sg2} - p_a = p_{esg2}$$

$$p_{sfB} = p_2 - p_{sg1} = p_2 - p_a = p_{e2}$$

31.2.4.3 Calculation of volume flow rate

The volume flow rate is given by the following equation:

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

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

31.2.4.4 Calculation of fan air power

In accordance with 14.8.5.6

$$P_{uB} = q_{Vsg1}\, p_{fB}$$

$$P_{usB} = q_{Vsg1}\, p_{sfB}$$

31.2.4.5 Calculation of fan efficiencies

Fan efficiencies are determined from P_{uB} or P_{usB} as in 32.2.3.5.

31.2.5 Fan performance under test conditions

Under test conditions, the fan performances are:

— inlet volume flow rate, $q_{V\mathrm{sg1}}$;

— fan pressure, p_{fB};

— fan static pressure, p_{sfB};

— fan efficiency, η_{rB} or η_{srB}.

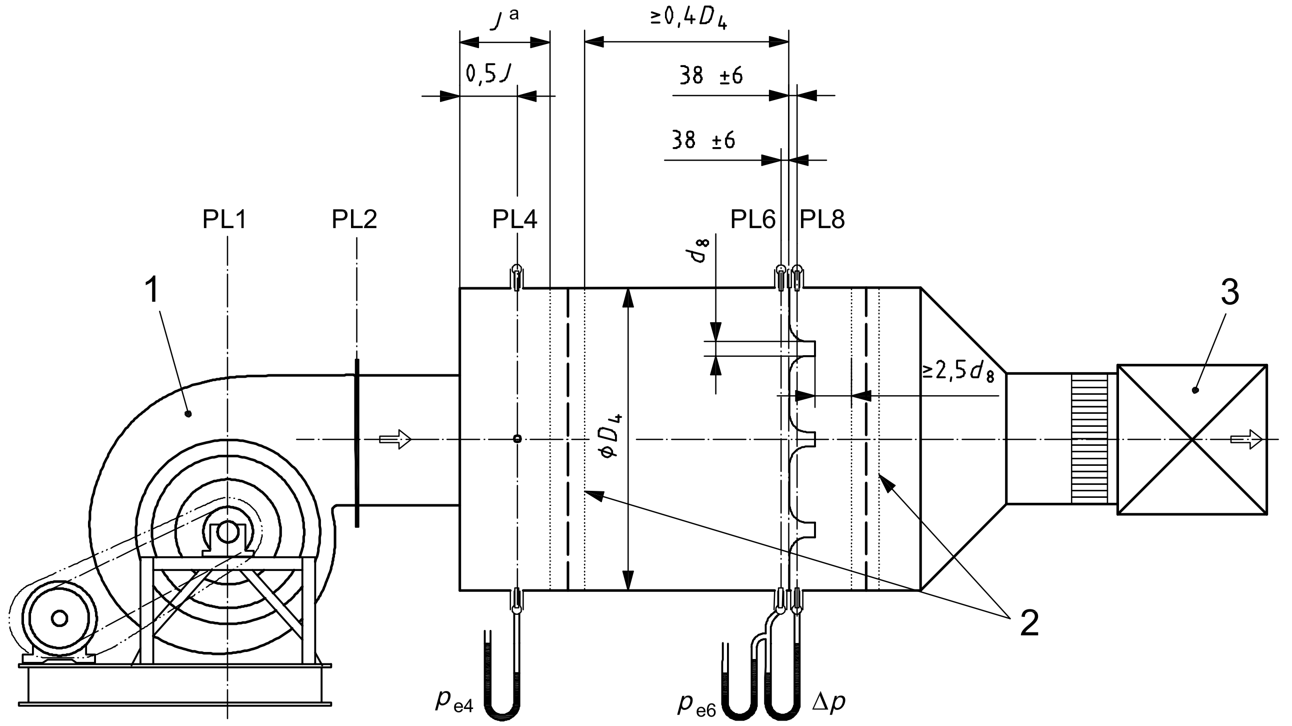
31.3 Outlet chamber test ducts without antiswirl device

It is recommended to use the setups below only for fans without outlet swirling flow.

31.3.1 Mass flow rate determination

The mass flow rate is determined using multiple nozzles in chamber, see Clause 22 and Figure 43.

Dimensions in millimetres



Key

1 test fan

2 flow-settling means

3 auxiliary fan

[a] 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.

NOTE The distance between section PL2 and the test chamber must be in accordance with Figure 36.

Figure 43 — Category B test installations (without antiswirl device) using multi-nozzle chamber

31.3.2 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).

31.3.3 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.

31.3.3.1 Calculation of mass flow rate

The mass flow rate is determined using multiple nozzles in chamber, see Clause 22 and Figure 43).

The outlet ducts for pressure and flow rate measurements are followed by a flow rate control device or an auxiliary fan with a flow rate control device.

The temperature T_6 in the chamber may be measured:

$$p_6 = p_{e6} + p_a$$

$$\Theta_6 = \Theta_{sg6} = T_6 + 273{,}15$$

$$\rho_6 = \frac{p_6}{R_w \Theta_6}$$

The mass flow rate is given by the following equation:

$$q_m = \varepsilon \pi \sum_{j=1}^{n} \left[C_j \frac{d_{8j}^2}{4} \right] \sqrt{2 \rho_6 \Delta p}$$

where

ε is the expansibility coefficient in accordance with 22.4.3 and Table 5;

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 22.4 and Table 4;

n is the number of nozzles.

For each nozzle, the throat Reynolds number, Re_{d8}, is estimated with the following equation:

$$Re_{d8j} = \frac{\varepsilon C_j d_{8j} \sqrt{2\rho_6 \Delta p}}{17{,}1 + 0{,}048 T_6} \times 10^6$$

with $C_j = 0{,}95$.

It is recommended to use these setups only for fans without outlet swirling flow.

After a first estimation of the mass flow rate, the discharge coefficients, C_j, are determined and corrected.

31.3.3.2 Calculation of fan pressure

31.3.3.2.1 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 equation:

$$p_{sg2} = p_4 + \frac{1}{2\rho_{2.4}} \left(\frac{q_m}{A_{2.4}} \right)^2 f_{M2.4}$$

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.

$$\rho_2 = \frac{p_2}{R_\mathrm{w}\Theta_2}$$

$$p_2 = p_\mathrm{sg2} - \rho_2 \frac{v_\mathrm{m2}^2}{2} f_\mathrm{M2} = p_\mathrm{sg2} - \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 f_\mathrm{M2}$$

or

$$p_\mathrm{e2} = p_\mathrm{esg2} - \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 f_\mathrm{M2}$$

31.3.3.2.2 Fan inlet pressure

At the fan inlet

$$\Theta_\mathrm{sg1} = \Theta_\mathrm{a} = T_\mathrm{a} + 273{,}15$$

$$p_\mathrm{sg1} = p_\mathrm{a}$$

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.

$$p_1 = p_\mathrm{sg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2 f_\mathrm{M1}$$

$$p_\mathrm{e1} = -\frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2 f_\mathrm{M1}$$

31.3.3.2.3 Fan pressure

The fan pressure, p_fB, and the fan static pressure p_sfB may be calculated by the following equations:

$$p_\mathrm{fB} = p_\mathrm{sg2} - p_\mathrm{sg1} = p_\mathrm{sg2} - p_\mathrm{a} = p_\mathrm{sg2}$$

$$p_\mathrm{sfB} = p_2 - p_\mathrm{sg1} = p_2 - p_\mathrm{a} = p_\mathrm{e2}$$

$$\rho_\mathrm{m} = \frac{\rho_2 + \rho_1}{2}$$

and

$$k_\rho = \frac{\rho_1}{\rho_\mathrm{m}}$$

31.3.3.3 Calculation of volume flow rate

The volume flow rate is calculated using the following equation:

$$q_{V\mathrm{sg1}} = \frac{q_m}{\rho_\mathrm{sg1}} = \frac{q_m}{\left(\dfrac{p_\mathrm{sg1}}{R_\mathrm{w}\Theta_\mathrm{sg1}}\right)}$$

31.3.3.4 Calculation of fan air power

31.3.3.4.1 Fan work per unit mass and fan air power

In accordance with 14.8.1, the fan work per unit mass $W_{m\mathrm{B}}$ and the fan static work per unit mass $W_{m\mathrm{sB}}$ are given by the following equations:

$$W_{m\mathrm{B}} = \frac{p_2 - p_1}{\rho_\mathrm{m}} + \frac{v_{\mathrm{m}2}^2}{2} - \frac{v_{\mathrm{m}1}^2}{2}$$

$$= \frac{p_2 - p_1}{\rho_\mathrm{m}} + \frac{1}{2}\left(\frac{q_m}{A_2\rho_2}\right)^2 - \frac{1}{2}\left(\frac{q_m}{A_1\rho_1}\right)^2$$

$$= \frac{p_{\mathrm{e}2} - p_{\mathrm{e}1}}{\rho_\mathrm{m}} + \frac{v_{\mathrm{m}2}^2}{2} - \frac{v_{\mathrm{m}1}^2}{2}$$

$$W_{m\mathrm{sB}} = \frac{p_2 - p_1}{\rho_\mathrm{m}} - \frac{v_{\mathrm{m}1}^2}{2} = \frac{p_2 - p_1}{\rho_\mathrm{m}} - \frac{1}{2}\left(\frac{q_m}{A_1\rho_1}\right)^2$$

$$= \frac{p_{\mathrm{e}2} - p_{\mathrm{e}1}}{\rho_\mathrm{m}} - \frac{v_{\mathrm{m}1}^2}{2}$$

The fan air power P_{uB} and the fan static power P_{usB} are given by the following equations:

$$P_{\mathrm{uB}} = q_m W_{m\mathrm{B}}$$

$$P_{\mathrm{usB}} = q_m W_{m\mathrm{sB}}$$

31.3.3.4.2 Calculation of fan air power and compressibility coefficient

In accordance with 14.8.2:

$$P_{\mathrm{uB}} = q_{V\mathrm{sg}1}\, p_{\mathrm{fB}}\, k_\mathrm{p}$$

$$P_{\mathrm{usB}} = q_{V\mathrm{sg}1}\, p_{\mathrm{sfB}}\, k_\mathrm{ps}$$

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:

$$k_\mathrm{ps} \text{ or } k_\mathrm{p} = \frac{Z_\mathrm{k} \log_{10} r}{\log_{10}\left[1 + Z_\mathrm{k}(r-1)\right]}$$

where

$$r = 1 + \frac{p_{\mathrm{fB}}}{p_{\mathrm{sg}1}}$$

for k_p or

$$r = 1 + \frac{p_{\mathrm{sfB}}}{p_{\mathrm{sg}1}}$$

for fan static air power and

$$Z_{\mathrm{k}} = \frac{\kappa - 1}{\kappa} \frac{P_{\mathrm{r}}}{q_{V\mathrm{sg1}}\, p_{\mathrm{fB}}}$$

for fan air power or

$$Z_{\mathrm{k}} = \frac{\kappa - 1}{\kappa} \frac{P_{\mathrm{r}}}{q_{V\mathrm{sg1}}\, p_{\mathrm{sfB}}}$$

for fan static air power.

b) Second method:

$$k_{\mathrm{p}} = \frac{\ln\left(1 + x\right)}{x} \frac{Z_{\mathrm{p}}}{\ln\left(1 + Z_{\mathrm{p}}\right)}$$

where

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

or

$$x = \frac{p_{\mathrm{sfB}}}{p_{\mathrm{sg1}}}$$

and

$$Z_{\mathrm{p}} = \frac{\kappa - 1}{\kappa} \frac{P_{\mathrm{r}}}{q_{V\mathrm{sg1}}\, p_{\mathrm{sg1}}}$$

31.3.3.5 Calculation of efficiencies

In accordance with 14.8.1, the efficiencies are given by the following equations:

— fan efficiency:

$$\eta_{\mathrm{rB}} = \frac{P_{\mathrm{uB}}}{P_{\mathrm{r}}}$$

— fan static efficiency:

$$\eta_{\mathrm{srB}} = \frac{P_{\mathrm{usB}}}{P_{\mathrm{r}}}$$

— fan shaft efficiency:

$$\eta_{\mathrm{aB}} = \frac{P_{\mathrm{uB}}}{P_{\mathrm{a}}}$$

— fan static shaft efficiency:

$$\eta_{\mathrm{saB}} = \frac{P_{\mathrm{usB}}}{P_{\mathrm{a}}}$$

31.3.4 Simplified method

The reference Mach number Ma_{2ref} is less than 0,15 and the pressure ratio less than 1,02 (see 14.8.5).

The air flow through the fan and the test airway may be considered incompressible.

$$\Theta_1 = \Theta_{sg1} = \Theta_2 = \Theta_{sg2} = \Theta_4 = \Theta_{sg4} = \Theta_6 = \Theta_{sg6} = \Theta_a = T_a + 273{,}15$$

$$\rho_1 = \rho_2 = \rho_4 = \rho_6 = \rho_a = \frac{p_a}{R_w \Theta_a}$$

$$f_{M1} = f_{M2} = 1$$

$$k_p = 1$$

31.3.4.1 Calculation of mass flow rate

The mass flow rate is calculated in accordance with section 32.2.3.1

31.3.4.2 Calculation of fan pressure

31.3.4.2.1 Fan outlet pressure

In accordance with the assumptions above,

$$\rho_1 = \rho_{sg1} = \rho_2 = \rho_{sg2} = \rho_a$$

$$p_{sg2} = p_4 + \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2$$

$$p_{esg2} = p_{e4} + \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2$$

$$p_2 = p_{sg2} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2 = p_4$$

$$p_{e2} = p_{esg2} \quad \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2 = p_{e4}$$

31.3.4.2.2 Fan inlet pressure

$$p_{sg1} = p_a$$

$$p_{esg1} = 0$$

$$p_1 = p_{sg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2$$

$$p_{e1} = -\frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2$$

31.3.4.2.3 Fan pressure

The fan pressure p_{fB} and the fan static pressure p_{sfB} may be determined by the following equations:

$$p_{fB} = p_{sg2} - p_{sg1} = p_{sg2} - p_a = p_{esg2}$$

$$p_{sfB} = p_2 - p_{sg1} = p_2 - p_a = p_{e2}$$

31.3.4.3 Calculation of volume flow rate

The volume flow rate is given by the following equation:

$$q_{Vsg1} = \frac{q_m}{\rho_{sg1}} = \frac{q_m}{\left(\dfrac{p_a}{R_w \Theta_{sg1}}\right)}$$

31.3.4.4 Calculation of fan air power

In accordance with 14.8.5.6:

$$P_{uB} = q_{Vsg1}\, p_{fB}$$

$$P_{usB} = q_{Vsg1}\, p_{sfB}$$

31.3.4.5 Calculation of fan efficiencies

Fan efficiencies are determined from P_{uB} or P_{usB}.

31.3.5 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}.

32 Standard test methods with inlet-side test ducts or chambers — Category C installations

32.1 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.

Six methods of controlling and measuring the flow rate in the test duct are shown. The method of flow rate measurement is specified in each case, together with the clauses and figures detailing the flow rate measurement procedure.

A common procedure, comprising measurements to be taken and quantities to be calculated, allowing the determination of fan performance in category C installations with nine methods for determining flow rate, is given in 32.2.3.1.1 to 32.2.3.1.3 and 32.3.3.1.1 to 32.3.3.1.2. This procedure is generally valid for all fans conforming to this International Standard.

However, a simplified procedure may be followed when the reference Mach number $Ma_{2\text{ref}}$ is less than 0,15 and the pressure ratio is less than 1,02. In these circumstances, the procedure which is given in 32.2.4 may be followed.

32.2 Inlet-side test ducts

32.2.1 Mass flow rate determination

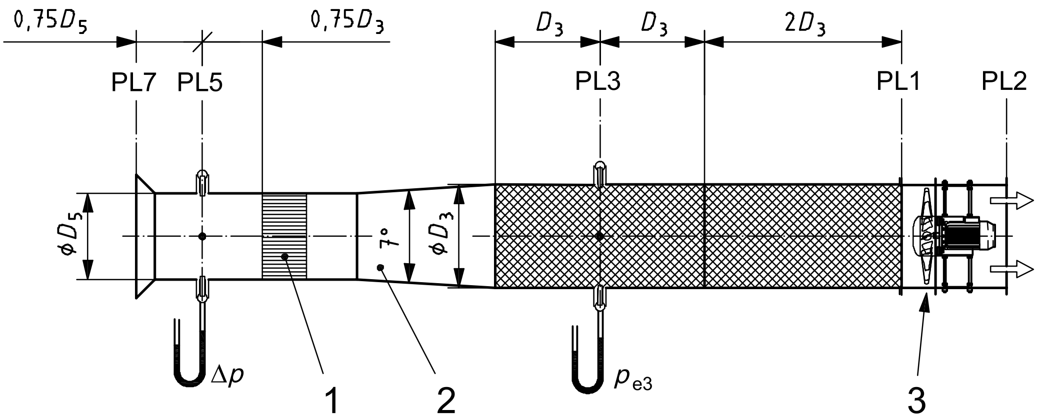
The mass flow rate is determined by

— conical or bellmouth inlet, see Figure 44 a);

— in-duct orifice with D and $D/2$, see Figure 44 d);

— Pitot-static tube traverse, see Figure 44 e);

— Pitot-static tube traverse, see Figure 44 f).



Key

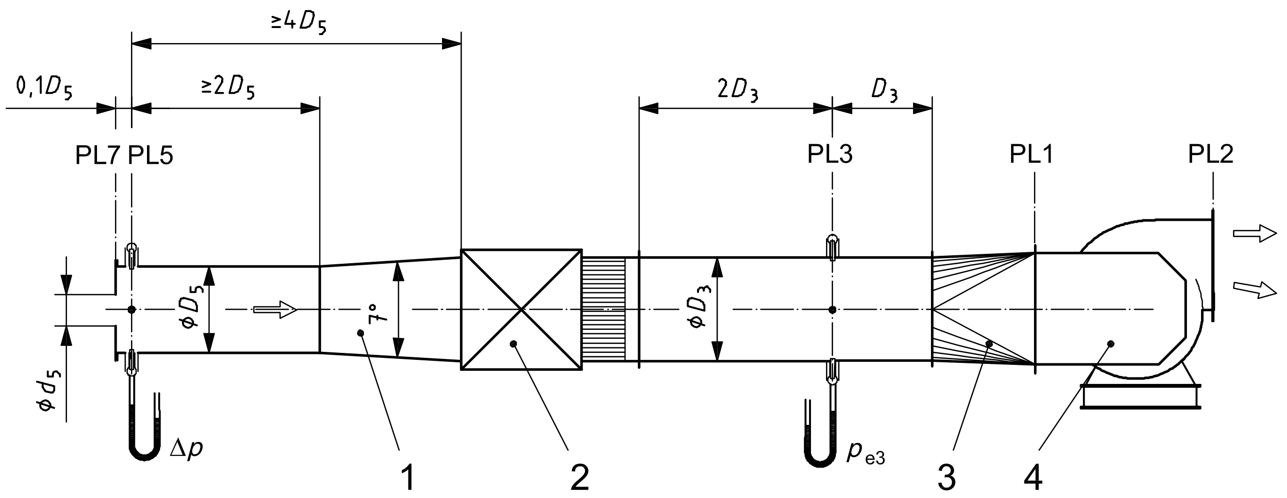
1 flow straightener (cell type shown)

2 transition section — the straight section with $\alpha = 0$ and length $\geqslant 3D$ can also be used, i.e. $D_5 = D_3$ test installation

3 test fan (tube-axial type shown)

a) Flow rate determination using a conical or bellmouth inlet

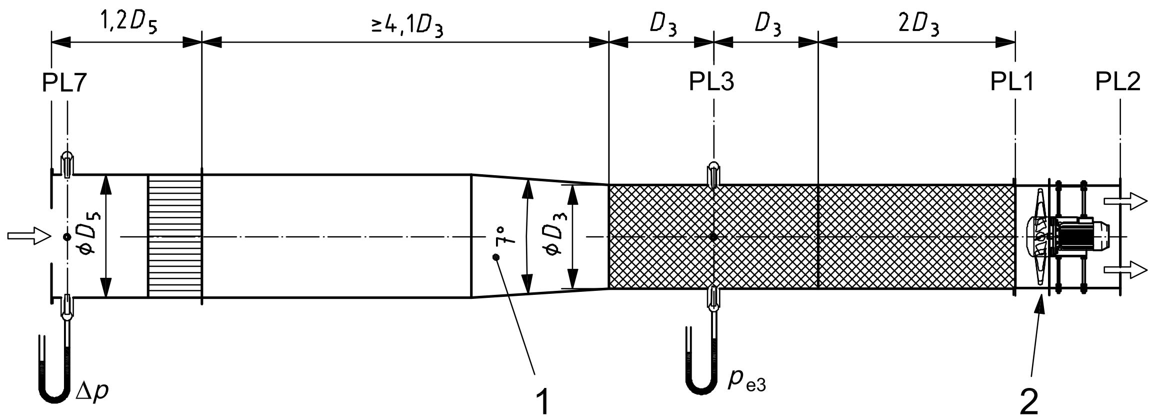
Figure 44 — Category C test installations (inlet-side test duct)



Key

1 transition section
2 auxiliary fan
3 transition section, round to rectangular, in accordance with Clause 31
4 test fan, shown with an integral inlet-box

b) Flow rate determination using an inlet orifice with wall taps



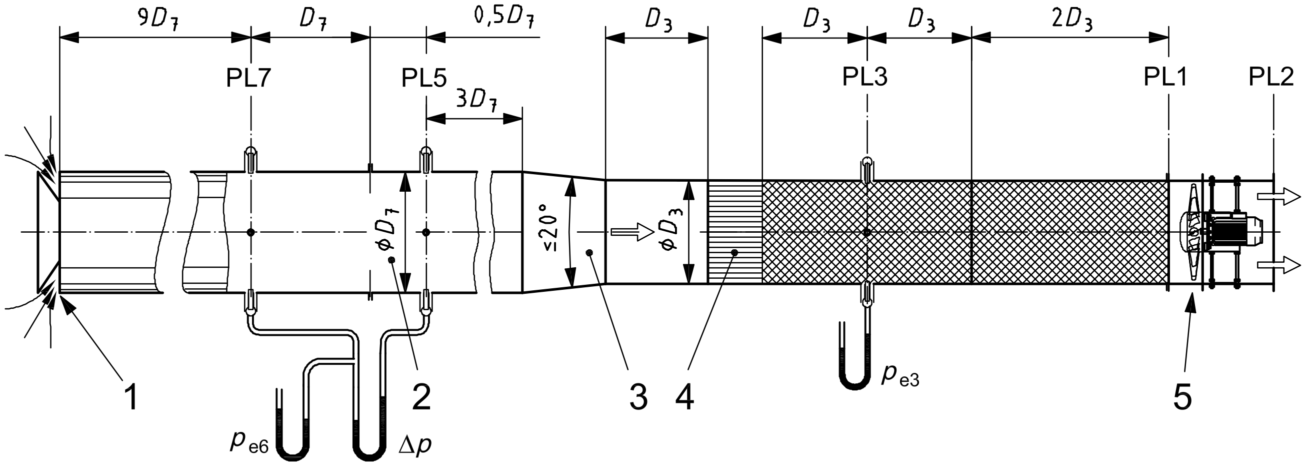
1,5 ⩽ D_5/D_6 ⩽ 6

Key

1 transition section
2 test fan (vane-axial type shown)

c) Flow rate determination using an inlet orifice with wall taps

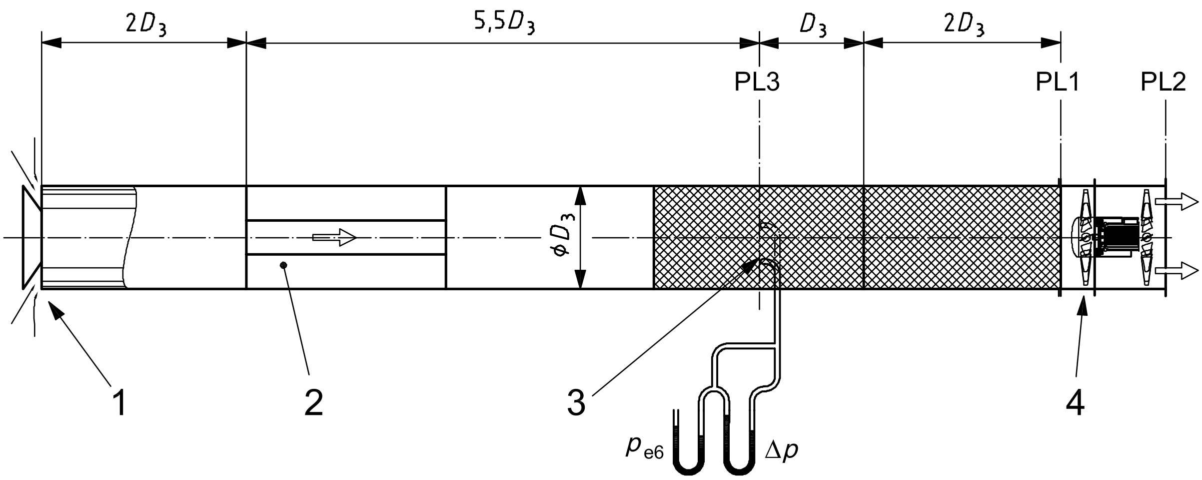
Figure 44 (*continued*)



Key

1 inlet throttling device
2 in-duct fixed orifice
3 transition piece
4 flow straightener (cell type shown)
5 test fan (tube-axial type shown)

d) Flow rate determination using an in-duct orifice with D and $0{,}5D$ wall taps

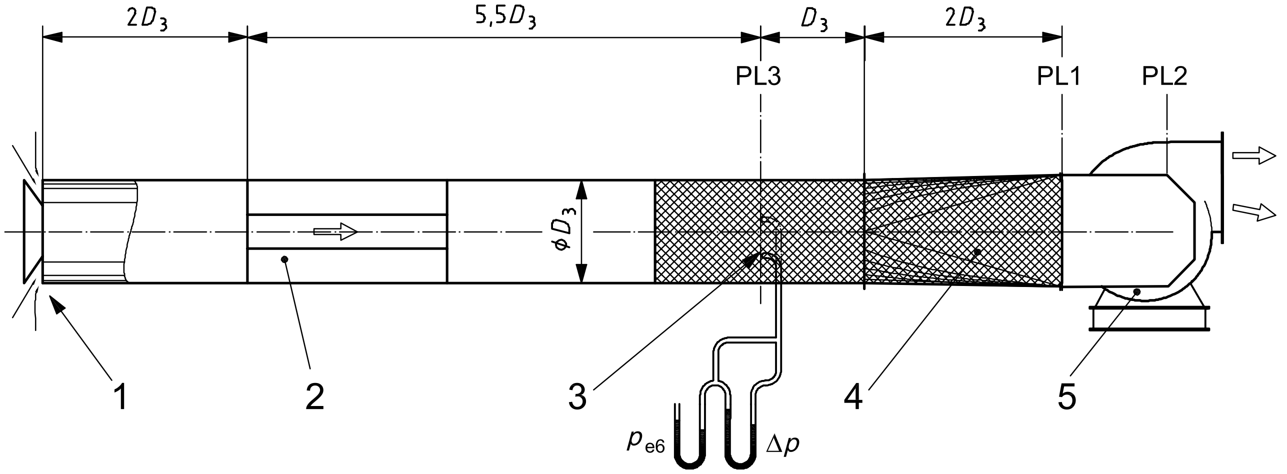


Key

1 inlet throttling device
2 flow straightener (star type shown)
3 Pitot-static tube traverse plane
4 test fan (two-stage axial type shown)

e) Flow rate determination using a Pitot-static tube traverse

Figure 44 (*continued*)



Key

1 inlet throttling device
2 flow straightener (star type shown)
3 Pitot-static tube traverse plane
4 round to square transition duct, in accordance with Clause 30
5 test fan, shown with an integral inlet-box

f) Flow rate determination using a Pitot-static tube traverse

Figure 44 (*continued*)

32.2.2 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 the 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).

32.2.3 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.

32.2.3.1 Calculation of mass flow rate

32.2.3.1.1 The mass flow rate is determined using

- conical or bellmouth inlet, see Clause 23 and Figure 44 a);
- inlet orifice with wall tappings, see 24.8.2 and Figure 44 b) and c);

The flow rate is controlled by an adjustable screen loading [see Figure 44 a) and 23.2], by the orifice plate [see Figure 42 c)], or by an auxiliary fan with a control device [Figure 44 b)].

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 flow rate is given by the following equation:

$$q_m = \alpha \varepsilon \pi \frac{d_5^2}{4} \sqrt{2 \rho_7 \Delta p}$$

where

α is the flow rate coefficient function of the Reynolds number Re_{d5} estimated by the following equation, 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 23.4, 24.8.2 and Figures 19, 22, and 23 after estimation of Re_{d5}.

32.2.3.1.2 The mass flow rate is determined using an in-duct orifice with taps at D and D/2 [see 24.7, and Figure 44 d)]

Assuming that

$$p_7 = p_{e7} + p_a$$

$$\Theta_{sg7} = \Theta_a = T_a + 273{,}15$$

$$\Theta_7 = \Theta_{sg7} - \frac{v_{m7}^2}{2 c_p} = \Theta_{sg7} - \frac{1}{2 c_p} \left(\frac{q_m}{A_7 \rho_7} \right)^2$$

$$\rho_7 = \frac{p_7}{R_w \Theta_7}$$

The mass flow rate is given by the following equation:

$$q_m = \alpha \varepsilon \pi \frac{d_5^2}{4} \sqrt{2 \rho_7 \Delta p}$$

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 flow rate 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 to three places of decimals.

32.2.3.1.3 The mass flow rate is determined using a Pitot-static tube traverse [see Clause 25 and Figures 44 e) and f)]

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 25.4 and Figure 25.

The mass flow rate q_m is given by the following equation (see 25.5):

$$q_m = \alpha \varepsilon \pi \frac{D_3^2}{4} \sqrt{2 \rho_3 \Delta p_\mathrm{m}}$$

where

ε is the expansibility factor (see 25.5);

α is the correction factor or flow coefficient (see 25.6) depending upon the Reynolds number Re_{D3}:

$$Re_{D3} = \frac{4\, q_m}{\pi D_3 \left(17{,}1 + 0{,}048 T_3\right)} \times 10^6$$

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 25.6).

32.2.3.2 Determination of fan pressure

32.2.3.2.1 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.

$$p_3 = p_{\mathrm{e}3} + p_\mathrm{a}$$

$$\Theta_{\mathrm{sg}3} = \Theta_{\mathrm{sg}5} = \Theta_{\mathrm{sg}7} = \Theta_\mathrm{a} = \Theta_{\mathrm{sg}1} = T_\mathrm{a} + 273{,}15$$

The Mach number Ma_3 and the ratio $Ma_3/Ma_{\mathrm{sg}3}$ are calculated in accordance with 14.4.3.1.

$$\Theta_3 = \Theta_{\mathrm{sg}3} \frac{Ma_3}{Ma_{\mathrm{sg}3}}$$

$$\rho_3 = \frac{p_3}{R_\mathrm{w} \Theta_3}$$

The inlet stagnation pressure $p_{\mathrm{sg}1}$ is given by the following equation (see 14.6.1):

$$p_{\mathrm{sg}1} = p_3 + \frac{1}{2} \rho_3 v_{\mathrm{m}3}^2 f_{\mathrm{M}3} \left[1 + \left(\xi_{3-1}\right)_3\right]$$

$$= p_3 + \frac{1}{2\rho_3} \left(\frac{q_m}{A_3}\right)^2 f_{\mathrm{M}3} \left[1 + \left(\xi_{3-1}\right)_3\right]$$

where

$(\xi_{3-1})_3 < 0$ is the conventional coefficient calculated in accordance with 28.6.3 and 28.6.4;

f_{M3} is the Mach factor determined in accordance with 14.5.1;

p_{e3} is always negative.

$$p_{esg1} = p_{e3} + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 f_{M3}\left[1 + \left(\xi_{3-1}\right)_3\right]$$

b) There is an auxiliary fan between planes 5 and 3 [see Figure 45 b)].

In this case, $(\xi_{3-1})_3 < 0$ 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, then:

$$\Theta_{sg3} = \Theta_{sg7} + \frac{P_{rx} \text{ or } P_{ex}}{q_m c_p} = \Theta_a + \frac{P_{rx} \text{ or } P_{ex}}{q_m c_p} = \Theta_{sg1}$$

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 Θ_1/Θ_{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 equation (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

$$p_{e1} = p_{esg1} - \frac{1}{2}\rho_1 v_{m1}^2 f_{M1} = p_{esg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2 f_{M1}$$

32.2.3.2.2 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 Ma_2/Ma_{sg2} are determined in accordance with 14.4.3.1.

$$\Theta_2 = \Theta_{sg2} \frac{Ma_2}{Ma_{sg2}}$$

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

and p_{sg2} is given by the following equation (see 14.5.1):

$$p_{sg2} = p_2 + \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 f_{M2} = p_a + \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 f_{M2}$$

$$p_{esg2} = \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 f_{M2}$$

32.2.3.2.3 Fan pressure

The fan pressure, p_{fC}, is given by the following equation:

$$p_{fC} = p_{sg2} - p_{sg1} = p_a + \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 f_{M2} - \left\{ p_3 + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 f_{M3}\left[1 + \left(\xi_{3-1}\right)_3\right]\right\}$$

$$= \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 f_{M2} - \left\{ p_{e3} + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 f_{M3}\left[1 + \left(\xi_{3-1}\right)_3\right]\right\}$$

The fan static pressure, p_{sfC}, is given by the following equation:

$$p_{sfC} = p_2 - p_{sg1} = p_a - \left\{ p_3 + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 f_{M3}\left[1 + \left(\xi_{3-1}\right)_3\right]\right\}$$

$$= -\left\{ p_{e3} + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 f_{M3}\left[1 + \left(\xi_{3-1}\right)_3\right]\right\}$$

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

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

32.2.3.3 Determination of volume flow rate

The volume flow rate at stagnation inlet conditions is given by the following equation:

$$q_{Vsg1} = \frac{q_m}{\rho_{sg1}} = \frac{q_m}{\left(\dfrac{p_{sg1}}{R_w \Theta_{sg1}}\right)}$$

32.2.3.4 Determination of fan air power

32.2.3.4.1 Fan work per unit mass and fan air power

According to 14.8.1, the fan work per unit mass, $W_{m\mathrm{C}}$, and the fan static work per unit mass, $W_{m\mathrm{sC}}$, are given by the following equations:

$$W_{m\mathrm{C}} = \frac{p_2 - p_1}{\rho_\mathrm{m}} + \frac{v_{\mathrm{m}2}^2}{2} - \frac{v_{\mathrm{m}1}^2}{2}$$

$$= \frac{p_2 - p_1}{\rho_\mathrm{m}} + \frac{1}{2}\left(\frac{q_m}{A_2\rho_2}\right)^2 - \frac{1}{2}\left(\frac{q_m}{A_1\rho_1}\right)^2$$

$$= \frac{p_{\mathrm{e}2} - p_{\mathrm{e}1}}{\rho_\mathrm{m}} + \frac{1}{2}\left(\frac{q_m}{A_2\rho_2}\right)^2 - \frac{1}{2}\left(\frac{q_m}{A_1\rho_1}\right)^2$$

$$W_{m\mathrm{sC}} = \frac{p_2 - p_1}{\rho_\mathrm{m}} - \frac{v_{\mathrm{m}1}^2}{2} = \frac{p_2 - p_1}{\rho_\mathrm{m}} - \frac{1}{2}\left(\frac{q_m}{A_1\rho_1}\right)^2$$

$$= \frac{p_{\mathrm{e}2} - p_{\mathrm{e}1}}{\rho_\mathrm{m}} - \frac{1}{2}\left(\frac{q_m}{A_1\rho_1}\right)^2$$

The fan power, P_{uC}, and the fan static power, P_{usC}, are given by the following equations:

$$P_{\mathrm{uC}} = q_m W_{m\mathrm{C}}$$

$$P_{\mathrm{usC}} = q_m W_{m\mathrm{sC}}$$

32.2.3.4.2 Calculation of fan air power and compressibility coefficients

In accordance with 14.8.2

$$P_{\mathrm{uC}} = q_{V\mathrm{sg}1}\, p_{\mathrm{fC}}\, k_\mathrm{p}$$

$$P_{\mathrm{usC}} = q_{V\mathrm{sg}1}\, p_{\mathrm{sfC}}\, k_\mathrm{ps}$$

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:

$$k_\mathrm{ps} \text{ or } k_\mathrm{p} = \frac{Z_\mathrm{k} \log_{10} r}{\log_{10}\left[1 + Z_\mathrm{k}\left(r - 1\right)\right]}$$

where

$$r = 1 + \frac{p_{\mathrm{fC}}}{p_{\mathrm{sg}1}}$$

for k_p or

$$r = 1 + \frac{p_{\mathrm{sfC}}}{p_{\mathrm{sg}1}}$$

for k_{ps} and

$$Z_k = \frac{\kappa - 1}{\kappa} \frac{P_r}{q_{V sg1}\, p_{fC}}$$

for k_p or

$$Z_k = \frac{\kappa - 1}{\kappa} \frac{P_r}{q_{V sg1}\, p_{sfC}}$$

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

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

32.2.3.5 Calculation of efficiencies

In accordance with 14.8.1 and 14.8.2, the efficiencies are calculated using the following equations:

— 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:

$$\eta_{saC} = \frac{P_{usC}}{P_a}$$

32.2.4 Simplified method

The reference Mach number, Ma_{2ref}, is less than 0,15 and the 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.

32.2.4.1 Calculation of mass flow rate

The mass flow rate is determined in accordance with section 32.2.3.1 with $\rho_u = \rho_a$.

32.2.4.2 Determination of fan pressure

32.2.4.2.1 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 14.8.5

$$p_{sg1} = p_3 + \frac{1}{2}\rho_3 v_{m3}^2 \left[1 + \left(\xi_{3-1}\right)_3\right]$$

$$= p_3 + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 \left[1 + \left(\xi_{3-1}\right)_3\right]$$

$$p_{esg1} = p_{e3} + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 \left[1 + \left(\xi_{3-1}\right)_3\right]$$

where p_{e3} and $(\xi_{3-1})_3 < 0$ [see 33.2.3.2.1 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 equation:

$$p_1 = p_{sg1} - \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 \left(\frac{A_3}{A_1}\right)^2 = p_{sg1} - \frac{1}{2\rho_3}\left(\frac{q_m}{A_1}\right)^2$$

$$p_{e1} = p_{esg1} - \frac{1}{2\rho_3}\left(\frac{q_m}{A_1}\right)^2$$

32.2.4.2.2 Fan outlet pressure

At the fan outlet

$$p_2 = p_a$$

$$p_{e2} = 0$$

and the stagnation pressure p_{sg2} is given by

$$p_{sg2} = p_a + \frac{1}{2}\rho_3 v_{m2}^2 = p_a + \frac{1}{2\rho_3}\left(\frac{q_m}{A_2}\right)^2$$

$$p_{esg2} = \frac{1}{2\rho_3}\left(\frac{q_m}{A_2}\right)^2$$

32.2.4.2.3 Fan pressure

The fan pressure, p_{fC}, and the fan static pressure, p_{sfC}, are given by the following equations:

$$p_{fC} = p_{sg2} - p_{sg1} = p_a + \frac{1}{2\rho_3}\left(\frac{q_m}{A_2}\right)^2 - \left\{p_3 + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 \left[1 + \left(\xi_{3-1}\right)_3\right]\right\}$$

$$= \frac{1}{2\rho_3}\left(\frac{q_m}{A_2}\right)^2 - \left\{p_{e3} + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 \left[1 + \left(\xi_{3-1}\right)_3\right]\right\}$$

$$p_{sfC} = p_2 - p_{sg1} = p_a - p_{sg1} = -p_{esg1}$$

$$= -\left\{p_{e3} + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 \left[1 + \left(\xi_{3-1}\right)_3\right]\right\}$$

32.2.4.3 Determination of volume flow rate

The volume flow rate in the inlet stagnation conditions is given by

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

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

32.2.4.4 Calculation of fan air power

In accordance with 14.8.5.6

$$P_{uC} = q_{Vsg1}\, p_{fC}$$

$$P_{usC} = q_{Vsg1}\, p_{sfC}$$

Efficiencies are determined in accordance with 14.8.1 and 32.2.3.5.

32.2.5 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 flow rate, q_{Vsg1}

— fan efficiency, η_{rC}

— fan static efficiency, η_{srC}

32.3 Inlet-side test chambers

32.3.1 Mass flow rate determination

The mass flow rate is determined by

— Pitot-static tube traverse, see Clause 25 and Figure 45 a);

— multiple nozzles in chamber, see Clause 22 and Figure 45 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) 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

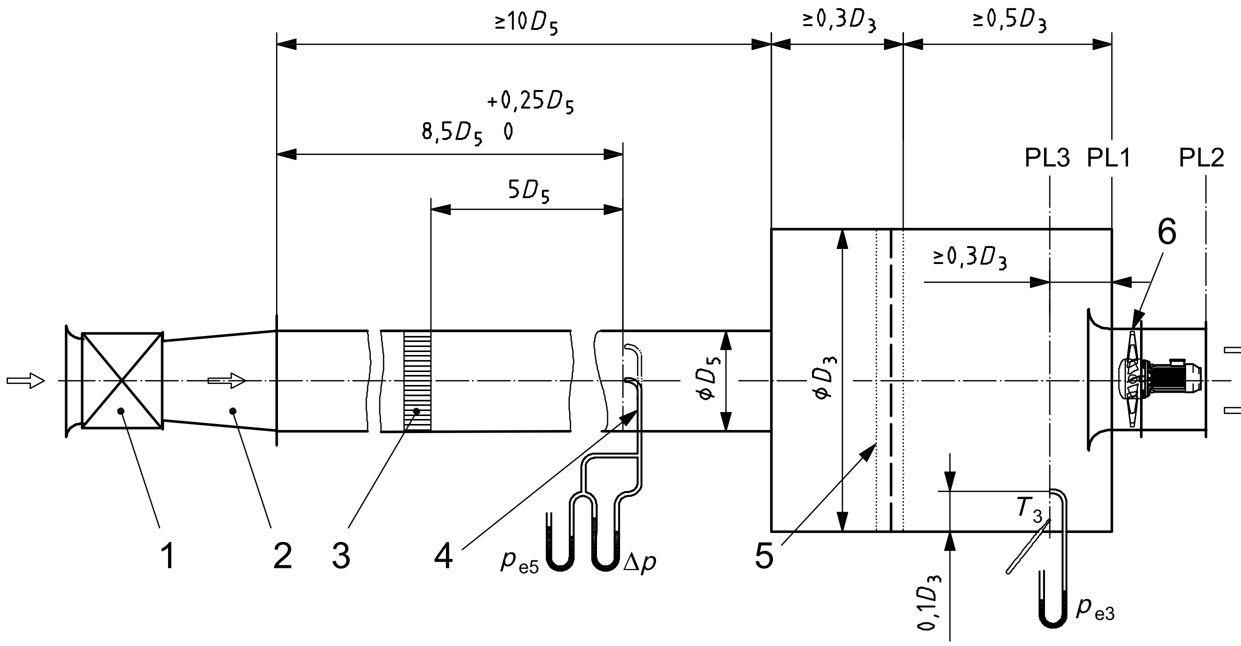
— atmospheric pressure, p_a, at the mean fan altitude;

— 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).

Dimensions in millimetres



Key

1 auxilliary fan

2 transition section

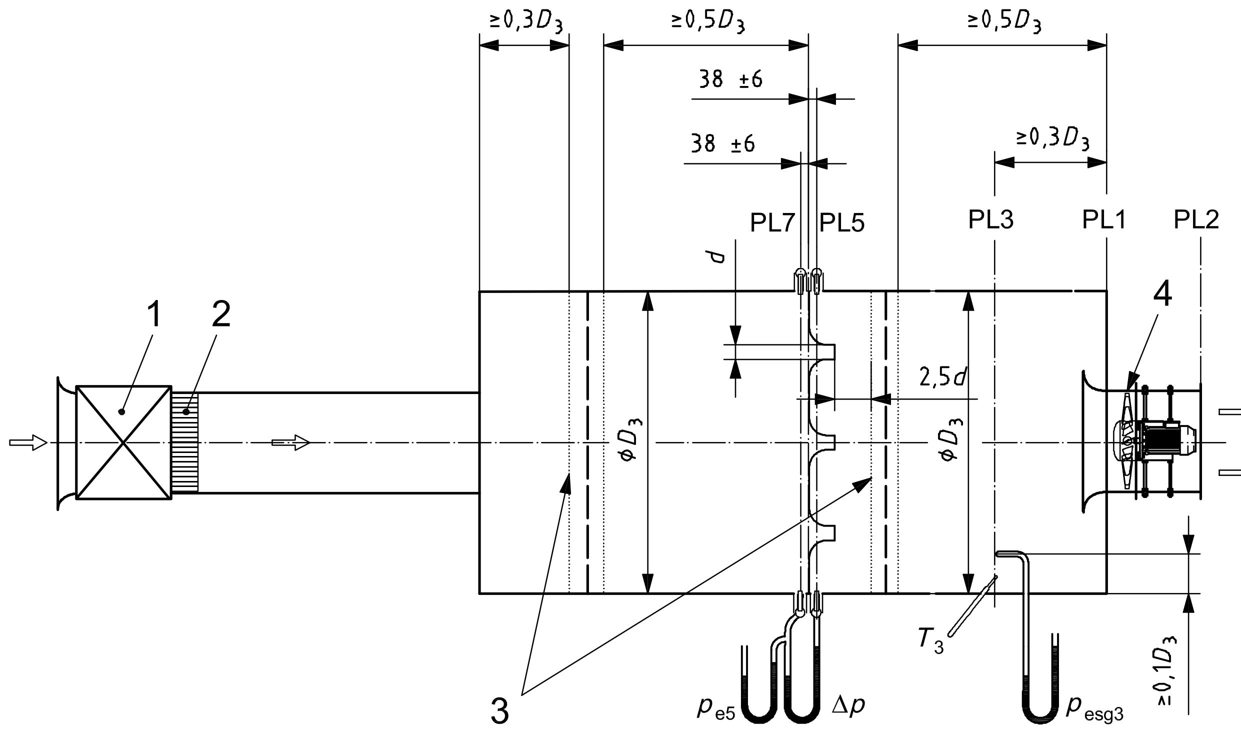
3 flow straightener (cell type shown)

4 Pitot-static tube traverse

5 flow-settling means

6 test fan (tube-axial type shown)

a) Flow rate determination using Pitot-static tube traverse



Key

1 auxilliary fan

2 flow straightener (cell type shown)

3 flow-settling means

4 test fan (vane-axial type shown)

b) Flow rate determination using multi-nozzle chamber

Figure 45 — Category C test installations (inlet-side test chamber)

32.3.3 General procedure for compressible fluid flow

This procedure should be applied when both the reference Mach number $Ma_{2\mathrm{ref}}$ is more than 0,15 and the fan pressure ratio is more than 1,02.

32.3.3.1 Determination of mass flow rate

32.3.3.1.1 The mass flow rate is determined using a Pitot-static tube traverse [see Clause 25 and Figure 45 a)].

Assuming that

$$p_{\mathrm{e}5} = \frac{1}{n} \sum_{j=1}^{n} p_{\mathrm{e}5\,j}$$

$$p_5 = p_{\mathrm{e}5} + p_{\mathrm{a}}$$

$$\Theta_{\mathrm{sg}5} = T_3 + 273{,}15 = \Theta_{\mathrm{a}} + \frac{P_{\mathrm{r}x} \text{ or } P_{\mathrm{e}x}}{q_m c_p}$$

$$\Delta p_{\mathrm{m}} = \left(\frac{1}{n} \sum_{j=1}^{n} \Delta p_j^{0{,}5} \right)^2$$

$$\Theta_5 = \Theta_{\mathrm{sg}5} \left(\frac{p_5}{p_5 + \Delta p_{\mathrm{m}}} \right)^{\frac{\kappa-1}{\kappa}}$$

$$\rho_5 = \frac{p_5}{R_{\mathrm{w}} \Theta_5}$$

The location of the measuring points, j, is given in 25.4 and Figure 25.

The mass flow rate is given by the following equation (see 25.5):

$$q_m = \alpha \varepsilon \pi \frac{D_5^2}{4} \sqrt{2 \rho_5 \Delta p_{\mathrm{m}}}$$

where

- ε is the expansibility coefficient in accordance with 25.5;
- α is the correction factor or flow coefficient (see 25.6), depending upon the Reynolds number, Re_{D5}:

$$Re_{D5} = \frac{4 q_m}{\pi D_5 \left(17{,}1 + 0{,}048 T_5\right)} \times 10^6$$

- 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 25.6).

32.3.3.1.2 The mass flow rate is determined using multiple nozzles in the chamber, see Clause 22 and Figure 45 b).

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 flow rate is given by the following equation:

$$q_m = \varepsilon \pi \sum_{j=1}^{n} \left(C_j \frac{d_{5j}^2}{4} \right) \sqrt{2 \rho_7 \Delta p}$$

where

ε is the expansibility coefficient in accordance with 22.4.3 and Table 5;

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 22.4 and Table 4;

n is the number of nozzles.

For each nozzle, the throat Reynolds number, Re_{d5}, is estimated by the following equation:

$$Re_{d5j} = \frac{\varepsilon C_j d_{5j} \sqrt{2 \rho_7 \Delta p}}{17,1 + 0,048\ T_7} \times 10^6$$

with $C_j = 0,95$.

After a first estimation of the mass flow rate, the discharge coefficients C_j are corrected.

32.3.3.2 Determination of fan pressure

32.3.3.2.1 Fan inlet pressure

Figure 45 a) and b) show 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 equation:

$$p_{sg3} = p_3 + \frac{1}{2}\rho_3 v_{m3}^2 = p_3 + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2$$

$$p_{esg3} = p_{e3} + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2$$

b) The chamber pressure is an absolute stagnation pressure, p_{esg3}

Under 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 D_1 or $2D_1$, and

$$p_{sg1} = p_{sg3}$$

$$p_{esg1} = p_{esg3}$$

When an inlet simulation duct longer than D_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 equations:

$$\rho_{3.1} = \rho_{sg3}\frac{\rho_{3.1}}{\rho_{sg3}}$$

$$p_{sg1} = p_{sg3} + \frac{1}{2\rho_{3.1}}\left(\frac{q_m}{A_1}\right)^2 f_{M3.1}\left(\xi_{3-1}\right)_1$$

$$p_{esg1} = p_{esg3} + \frac{1}{2\rho_{3.1}}\left(\frac{q_m}{A_1}\right)^2 f_{M3.1}\left(\xi_{3-1}\right)_1$$

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;

$(\xi_{3-1})_1 < 0$ is the conventional friction-loss coefficient for the inlet simulation duct of diameter D_1 and length L in accordance with 28.6;

$$\left(\xi_{3-1}\right)_1 = -\Lambda \frac{L}{D_1}$$

The static pressure, p_1, is determined by the following equation:

$$p_1 = p_{sg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2 f_{M1}$$

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.

32.3.3.2.2 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

$$p_{esg2} = \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 f_{M2}$$

32.3.3.2.3 Fan pressure

The fan pressure, p_{fC}, and the fan static pressure, p_{sfC}, are given by the following equations:

$$p_{FC} = p_{sg2} - p_{sg1} = p_a + \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 f_{M2} - p_{sg1}$$

$$= \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 f_{M2} - p_{esg1}$$

$$p_{sfC} = p_2 - p_{sg1} = p_a - p_{sg1} = p_{esg1}$$

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

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

32.3.3.3 Determination of volume flow rate

The volume flow rate under inlet stagnation conditions is given by the following equation:

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

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

32.3.3.4 Determination of fan air power

32.3.3.4.1 Fan work per unit mass and fan air power

The fan work per unit mass, W_{mC}, and the fan static work per unit mass, W_{msC}, are given by the following equations:

$$W_{mC} = \frac{p_2 - p_1}{\rho_m} + \frac{v_{m2}^2}{2} - \frac{v_{m1}^2}{2}$$

$$= \frac{p_2 - p_1}{\rho_m} + \frac{1}{2}\left(\frac{q_m}{A_2\rho_2}\right)^2 - \frac{1}{2}\left(\frac{q_m}{A_1\rho_1}\right)^2$$

$$W_{msC} = \frac{p_2 - p_1}{\rho_m} - \frac{v_{m1}^2}{2} = \frac{p_2 - p_1}{\rho_m} - \frac{1}{2}\left(\frac{q_m}{A_1\rho_1}\right)^2$$

$$= \frac{p_{e2} - p_{e1}}{\rho_m} - \frac{1}{2}\left(\frac{q_m}{A_1\rho_1}\right)^2$$

The corresponding fan air power, P_{uC}, and fan static air power, P_{usC}, are given by the following equations:

$$P_{uC} = q_m W_{mC}$$

$$P_{usC} = q_m W_{msC}$$

32.3.3.4.2 Calculation of fan air power and compressibility coefficient

In accordance with 14.8.2

$$P_{\mathrm{uC}} = q_{V\mathrm{sg1}}\, p_{\mathrm{fC}}\, k_{\mathrm{p}}$$

$$P_{\mathrm{usC}} = q_{V\mathrm{sg1}}\, p_{\mathrm{sfC}}\, k_{\mathrm{ps}}$$

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:

$$k_{\mathrm{ps}} \text{ or } k_{\mathrm{p}} = \frac{Z_{\mathrm{k}} \log_{10} r}{\log_{10}\left[1 + Z_{\mathrm{k}}\left(r - 1\right)\right]}$$

where

$$r = 1 + \frac{p_{\mathrm{fC}}}{p_{\mathrm{sg1}}}$$

for k_{p} or

$$r = 1 + \frac{p_{\mathrm{sfC}}}{p_{\mathrm{sg1}}}$$

for k_{ps} and

$$Z_{\mathrm{k}} = \frac{\kappa - 1}{\kappa}\, \frac{P_{\mathrm{r}}}{q_{V\mathrm{sg1}}\, p_{\mathrm{fC}}}$$

for k_{p} or

$$Z_{\mathrm{k}} = \frac{\kappa - 1}{\kappa}\, \frac{P_{\mathrm{r}}}{q_{V\mathrm{sg1}}\, p_{\mathrm{sfC}}}$$

for k_{ps}.

b) Second method:

$$k_{\mathrm{ps}} \text{ or } k_{\mathrm{p}} = \frac{\ln\left(1 + x\right)}{x}\, \frac{Z_{\mathrm{p}}}{\ln\left(1 + Z_{\mathrm{p}}\right)}$$

where

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

or

$$x = \frac{p_{\mathrm{sfC}}}{p_{\mathrm{sg1}}}$$

and

$$Z_{\mathrm{p}} = \frac{\kappa - 1}{\kappa}\, \frac{P_{\mathrm{r}}}{q_{V\mathrm{sg1}}\, p_{\mathrm{sg1}}}$$

32.3.3.5 Calculation of efficiencies

In accordance with 14.8.1, the efficiencies are calculated using the following equations:

— fan efficiency:

$$\eta_{\mathrm{rC}} = \frac{P_{\mathrm{uC}}}{P_{\mathrm{r}}}$$

— fan static efficiency:

$$\eta_{\mathrm{srC}} = \frac{P_{\mathrm{usC}}}{P_{\mathrm{r}}}$$

— fan shaft efficiency:

$$\eta_{\mathrm{sC}} = \frac{P_{\mathrm{uC}}}{P_{\mathrm{a}}}$$

— fan shaft static efficiency:

$$\eta_{\mathrm{saC}} = \frac{P_{\mathrm{usC}}}{P_{\mathrm{a}}}$$

32.3.4 Simplified method

The reference Mach number, $Ma_{2\mathrm{ref}}$, is less than 0,15 and the pressure ratio less than 1,02.

$$\Theta_3 = \Theta_{\mathrm{sg3}} = \Theta_1 = \Theta_{\mathrm{sg1}} = \Theta_2 = \Theta_{\mathrm{sg2}} = T_3 + 273{,}15$$

The air flow through the fan may be considered as incompressible.

$$\rho_3 = \rho_1 = \rho_2$$
$$k_{\mathrm{p}} = 1$$

32.3.4.1 Determination of mass flow rate

The mass flow rate is determined in accordance with 32.3.3.1.

The temperature upstream of the flowmeter may be measured.

32.3.4.2 Determination of fan pressure

32.3.4.2.1 Fan inlet pressure

— When the measured pressure is a static pressure, p_{e3}:

$$p_3 = p_{\mathrm{e3}} + p_2$$

$$p_{\mathrm{sg3}} = p_3 + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^3$$

$$p_{esg3} = p_{e3} + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2$$

where

$$\rho_3 = \frac{p_3}{R_w \Theta_3} = \rho_{sg1}$$

— When the measured pressure is a stagnation pressure:

$$p_{sg3} = p_{esg3} + p_a$$

In the two cases:

$$p_{sg1} = p_{sg3} + \frac{1}{2\rho_3}\left(\frac{q_m}{A_1}\right)^2 \left(\xi_{3-1}\right)_1$$

or

$$p_{esg1} = p_{esg3} + \frac{1}{2\rho_3}\left(\frac{q_m}{A_1}\right)^2 \left(\xi_{3-1}\right)_1$$

where $(\xi_{3-1})_1 < 0$ is determined in accordance with 32.3.3.2.1, 28.6.4 and 28.6.5.

$$p_1 = p_{sg1} - \frac{1}{2\rho_3}\left(\frac{q_m}{A_1}\right)^2$$

32.3.4.2.2 Fan outlet pressure

At the fan outlet

$$p_2 = p_a$$

$$p_{sg2} = p_2 + \frac{1}{2\rho_3}\left(\frac{q_m}{A_2}\right)^2$$

$$p_{esg2} = \frac{1}{2\rho_3}\left(\frac{q_m}{A_2}\right)^2$$

32.3.4.2.3 Fan pressure

The fan pressure, p_{fC}, and the fan static pressure, p_{sfC}, are given by the following equations:

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

$$= \frac{1}{2\rho_3}\left(\frac{q_m}{A_2}\right)^2 - p_{esg1}$$

$$p_{sfC} = p_2 - p_{sg1} = -p_{esg1}$$

32.3.4.3 Determination of volume flow rate

The volume flow rate under inlet stagnation conditions is given by the following equations:

$$q_{V\text{sg1}} = \frac{q_m}{\rho_{\text{sg3}}}$$

$$\rho_{\text{sg1}} = \rho_{\text{sg3}} = \frac{p_{\text{sg3}}}{R_{\text{w}} \Theta_{\text{sg3}}}$$

32.3.4.4 Determination of fan air power

Fan air powers are given by the following equations:

$$P_{\text{uC}} = q_{V\text{sg1}}\, p_{\text{fC}}$$

$$P_{\text{usC}} = q_{V\text{sg1}}\, p_{\text{sfC}}$$

32.3.4.5 Determination of fan efficiencies

Fan efficiencies are determined in accordance with 14.8 and 32.3.3.5 from P_{uC} and P_{usC}.

32.3.5 Fan performance under test conditions

Under the test conditions, the fan performances are the following:

— fan pressure, p_{fC};

— fan static pressure, p_{sfC};

— inlet volume flow rate, $q_{V\text{sg1}}$;

— fan efficiency, η_{rC};

— fan static efficiency, η_{srC}.

33 Standard methods with inlet- and outlet-side test ducts — Category D installations

33.1 Types of fan setup

Generally, a test installation of category D is obtained from:

a) a category B test installation with an additional inlet-duct simulation in accordance with 28.3 and 28.5;

b) a category C test installation with an additional outlet-duct simulation in accordance with 28.2 and 28.4.

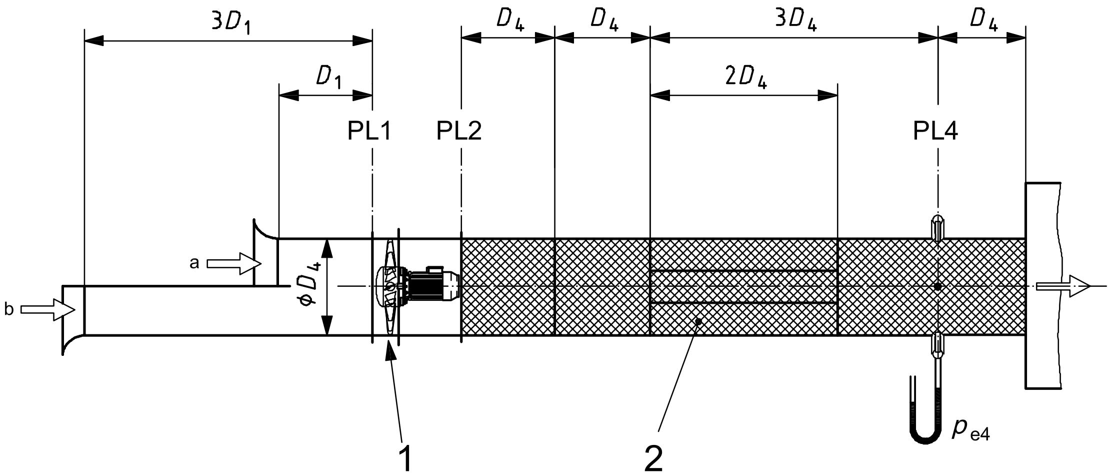
Consequently, four category D installations are described in this clause.

1) Category B installation with common segment outlet duct and outlet antiswirl device and with inlet-duct simulation in accordance with 28.2, 28.3 and 28.5 [see Figure 46 a) and b)].

2) Category B installation without outlet antiswirl device, without common segment outlet duct and with inlet-duct simulation in accordance with 28.2.5, 28.3 and 28.5 [see Figure 46 d)].

3) Category C installation with common segment outlet duct and outlet antiswirl device and common segment inlet duct in accordance with 28.2.1, 28.2.2, 28.2.3 or 28.2.4, 28.3, 28.4 and 28.5 [see Figure 46 c)].

4) Category C installation with outlet-duct simulation without antiswirl device in accordance with 28.2.5, 28.3 and 28.5 [see Figure 46 e), f) and g)].



Key

1 test fan (axial)

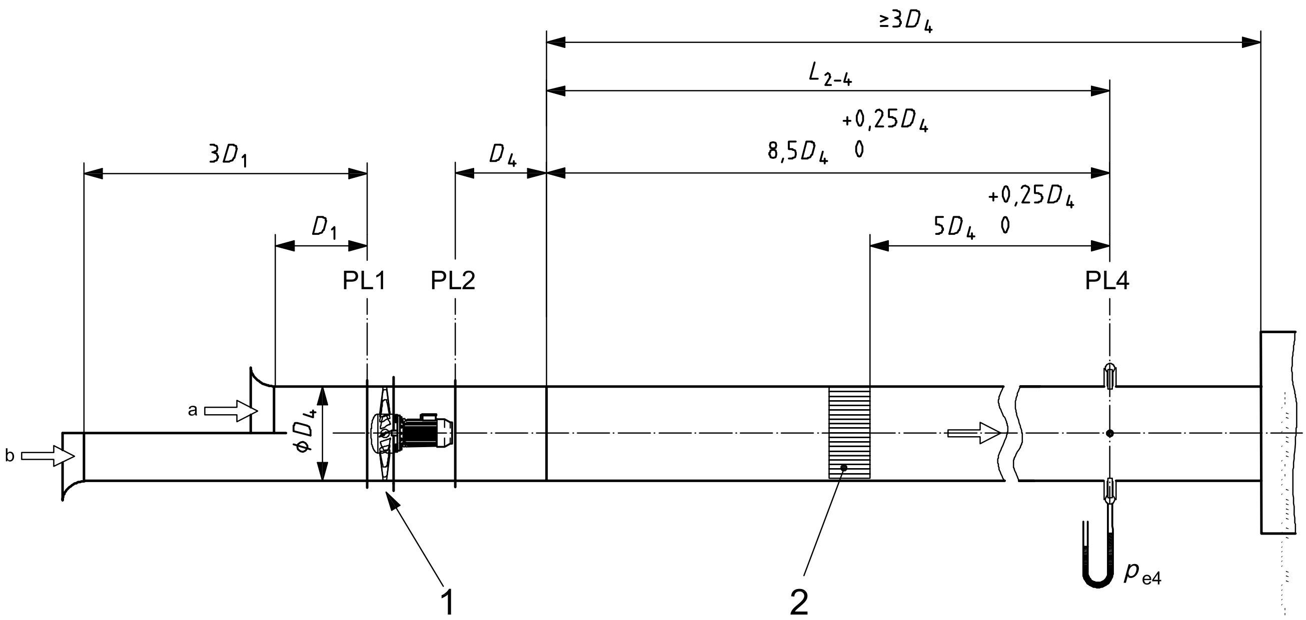
2 flow straightener (star type shown)

[a] See 29.3.

[b] See 29.3.1.

NOTE Flow rate measurement and control in accordance with 30.2.3.1.

a) Category B (outlet-side test duct) with common segment and antiswirl device and inlet-duct simulation



Key

1 test fan (axial)

2 flow straightener (cell type shown)

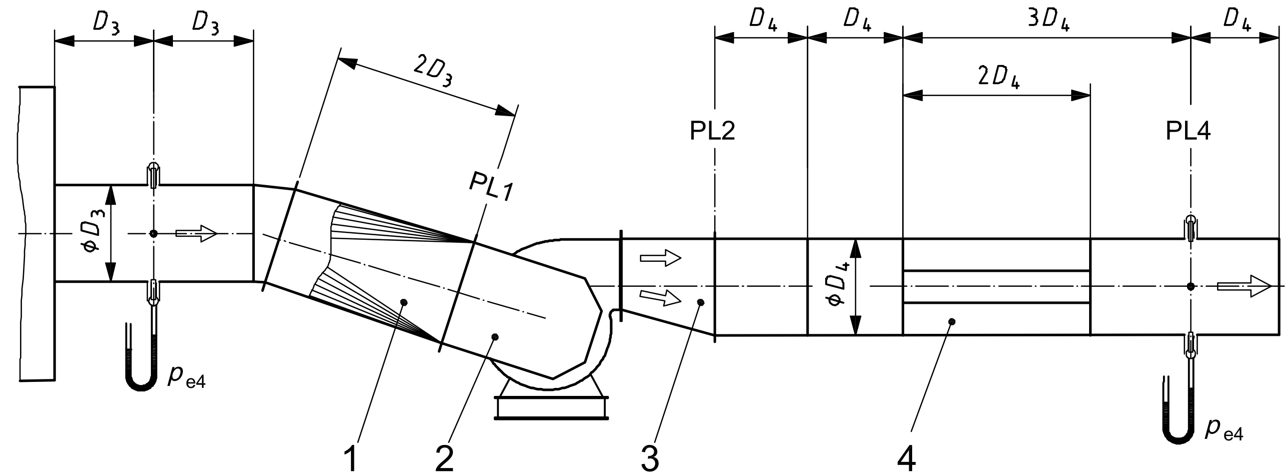
[a] See 28.5.

[b] See 28.3.1.

NOTE Flow rate measurement and control in accordance with 30.2.3.1.

b) Category B (outlet-side test duct) with common segment and antiswirl device and inlet-duct simulation

Figure 46 — Category D test installations

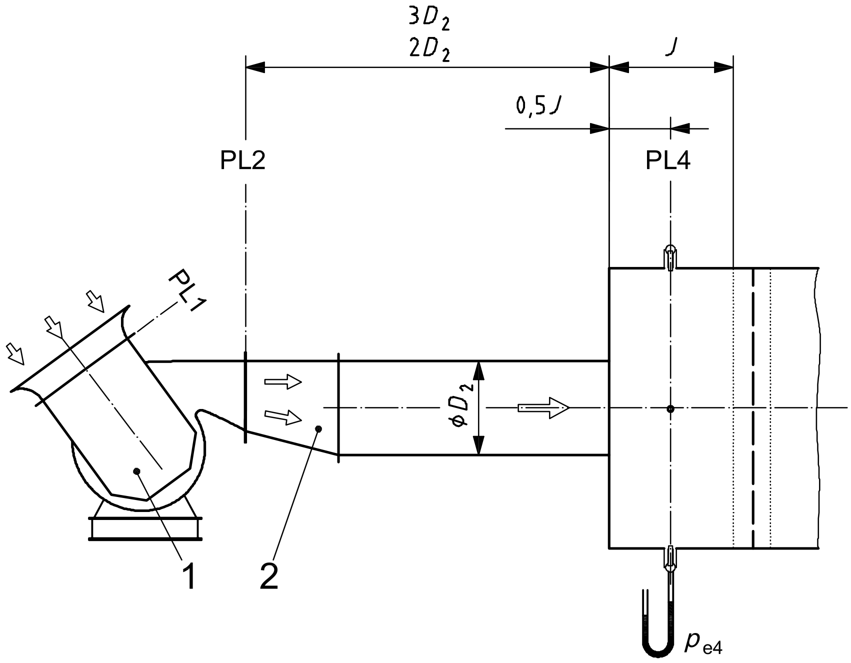


Key

1 transition duct round to rectangular
2 test fan shown with integral inlet box
3 transition duct rectangular to round
4 flow straightener (star type shown)

NOTE Flow rate measurement and control in accordance with 31.2.

c) Category C (inlet-side test duct) with antiswirl device and common inlet segment



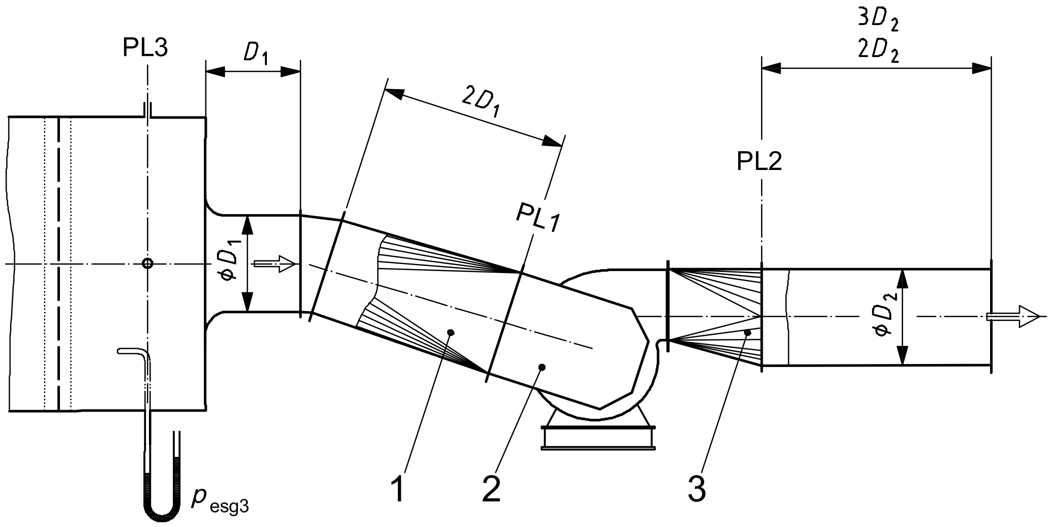
Key

1 test fan shown with an integral inlet box with an inlet flair
2 transition duct round to rectangular

NOTE Flow rate measurement and control in accordance with 30.3.3.1.1 and Figure 43 (only for fans without outlet swirling flow).

d) Category B (outlet-side test duct) without antiswirl device, without common segment and with inlet-duct simulation

Figure 46 (*continued*)



Key

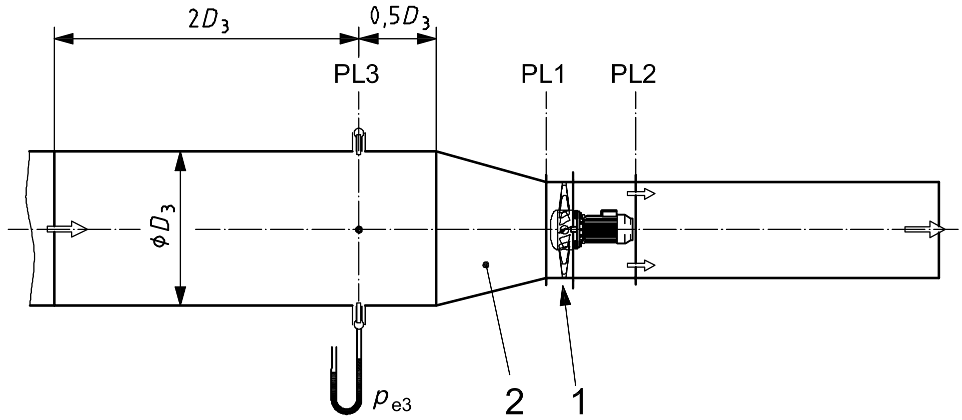
1 transition duct round to rectangular

2 test fan shown with integral inlet box

3 transition duct rectangular to round

NOTE Flow rate measurement and control in accordance with 32.3.3.1.1, 32.3.3.1.2, 32.3.3.1.3 and 32.3.3.1.4 and Figure 45 a) and b) (only for fans without outlet swirling flow).

e) Category C (inlet-side test duct) with outlet-duct simulation and without antiswirl device



Key

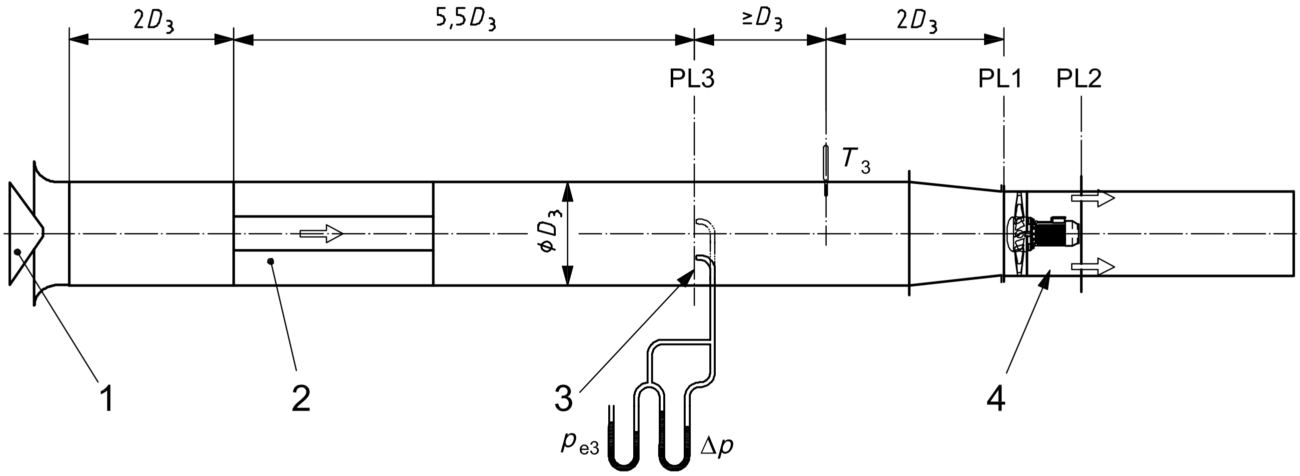
1 test fan

2 duct transition

NOTE Flow rate measurement and control in accordance with 32.2.3.1.1 and Figure 44 b).

f) Category C (inlet-side test duct) with outlet-duct simulation and without antiswirl device (may be used for large fans only by mutual agreement between the parties concerned)

Figure 46 (*continued*)



Key

1 inlet throttling device

2 flow straightener (star type shown)

3 Pitot-static tube traverse plane

4 test fan (vane-axial type shown)

NOTE Flow rate measurement and control in accordance with 32.2 and Figure 44 f).

g) Category C (inlet-side test duct) with outlet-duct simulation and without antiswirl device (may be used for large fans only by mutual agreement between the parties concerned)

Figure 46 (*continued*)

Installations 1 and 3 are recommended.

Installations 2 and 4 are accepted, but the results obtained in this way may differ to some extent from those obtained using common airways on both the inlet and the outlet sides.

The methods of flow rate measurements described in Clauses 22 to 25 and 31.2.3.1, 31.3.3.1, 32.2.3.1 and 32.3.3.1 may be used.

The procedures comprising measurements to be taken and quantities to be calculated, allowing the determination of fan performance in D category installations, are given in 31.2.3.1, 33.3.3.1, 33.4.3.1 and 33.5.3.1.

It is generally valid for all fans conforming to this International Standard.

33.2 Installation category B with outlet antiswirl device and with an additional inlet duct or inlet-duct simulation

33.2.1 Determination of mass flow rate

The mass flow rate is determined using:

— outlet orifice with wall tappings, see Figures 42 a) and 46 a);

— in-duct orifice with taps at D and D/2, see Figures 42 b) and 46 a);

— Pitot-static tube traverse, see Figures 42 c) and 46 a);

— multiple nozzles in chamber, see Figures 42 d) and 46 a) or b).

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_e or P_o, and estimate impeller power (see 10.4);
- inlet pressure, p_{e3};
- 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 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).

33.2.3 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 more than 1,02.

33.2.3.1 Calculation of mass flow rate

33.2.3.1.1 The mass flow rate is determined using:

- outlet orifice with wall tappings, see 24.8 and Figure 42 a);
- in-duct orifice with taps at D and $D/2$, see 24.7 and Figure 42 b).

The procedure described in 31.2.3.1.1 is followed.

33.2.3.1.2 The mass flow rate is determined using a Pitot-static tube traverse, see Clause 25 and Figure 42 c).

The procedure described in 31.2.3.1.2 is followed.

33.2.3.1.3 The mass flow rate is determined using multiple nozzles in the chamber, see Clause 22 and Figure 42 d).

The procedure described in 31.2.3.1.1 is followed.

33.2.3.2 Determination of fan pressure

33.2.3.2.1 Fan outlet pressure

Assuming that (see 31.2.3.2.1)

$$p_4 = p_{e4} + p_a$$

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

The Mach number, Ma_4, and the ratio, Ma_4/Ma_{sg4}, are determined in accordance with 14.4.3.1.

$$\Theta_4 = \Theta_{sg4} \frac{Ma_4}{Ma_{sg4}}$$

$$\rho_4 = \frac{p_4}{R_w \Theta_4}$$

$$f_{M4} = 1 + \frac{Ma_4^2}{4} + \frac{Ma_4^4}{40} + \frac{Ma_4^6}{1\,600} \quad \text{(see 14.5.1)}$$

The stagnation pressure at fan outlet, p_{sg2}, is given by the following equation:

$$p_{sg2} = p_4 + \frac{\rho_4 v_{m4}^2}{2} f_{M4} \left[1 + \left(\xi_{2-4}\right)_4\right]$$

$$= p_4 + \frac{1}{2\rho_4} \left(\frac{q_m}{A_4}\right)^2 f_{M4} \left[1 + \left(\xi_{2-4}\right)_4\right]$$

or

$$p_{esg2} = p_{e4} + \frac{1}{2\rho_4} \left(\frac{q_m}{A_4}\right)^2 f_{M4} \left[1 + \left(\xi_{2-4}\right)_4\right]$$

$(\xi_{2-4})_4$ being calculated in accordance with 28.6.1 or 28.6.2 and Figure 35.

The pressure, p_2, is given by the following equation:

$$p_2 = p_{sg2} - \rho_2 \frac{v_{m2}^2}{2} f_{M2}$$

$$= p_{sg2} - \frac{1}{2\rho_2} \left(\frac{q_m}{A_2}\right)^2 f_{M2}$$

or

$$p_{e2} = p_{esg2} - \frac{1}{2\rho_2} \left(\frac{q_m}{A_2}\right)^2 f_{M2}$$

where Ma_2, ρ_2, f_{M2} are determined in accordance with 14.4.3.2 and 14.5.

33.2.3.2.2 Fan inlet pressure

a) Inlet duct in accordance with 28.3 (see 32.2.3.2.1).

The inlet pressure, p_{e3}, is measured.

$$p_3 = p_a + p_{e3}$$

$$\Theta_{sg3} = \Theta_{sg1} = \Theta_a = T_a + 273{,}15$$

The Mach number, Ma_3, and the fluid temperature, Θ_3, are determined in accordance with 14.4.3.1.

$$\rho_3 = \frac{p_3}{R_w \Theta_3}$$

$$p_{sg1} = p_3 + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 f_{M3}\left[1 + \left(\xi_{3-1}\right)_3\right]$$

or

$$p_{esg1} = p_{e3} + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 f_{M3}\left[1 + \left(\xi_{3-1}\right)_3\right]$$

$(\xi_{3-1})_3 < 0$ being determined in accordance with 28.6.4, 28.6.5 and Figure 35 a) and b).

b) Inlet-duct simulation in accordance with 28.5 or 28.3.

The inlet pressure, p_{e3}, is not measured.

The stagnation pressure upstream of the duct inlet is the atmospheric pressure, p_a, and the fan stagnation inlet temperature, Θ_{sg1}, is equal to the ambient temperature:

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

The fan stagnation inlet pressure, p_{sg1}, is given by the following equation:

$$p_{sg1} = p_a + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 f_{M3}\left(\xi_{3-1}\right)_3$$

or

$$p_{esg1} = \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 f_{M3}\left(\xi_{3-1}\right)_3$$

where

ρ_3 is the air density at the duct inlet downstream of the bell mouth, determined in accordance with 14.4.4 after the determination of Ma_3 in accordance with 14.4.3.2;

$(\xi_{3-1})_3 < 0$ is the conventional friction-loss coefficient between section 3 (throat section of the inlet bell mouth) and section 1 at the fan inlet, determined in accordance with 28.6.4 and 28.6.5:

$$\left(\xi_{3-1}\right)_3 = -\Lambda \frac{L}{D_3}$$

L is the length of the duct (see 28.3 and 28.5);

D_3 is the duct diameter;

Λ depends upon the Reynolds number, Re_{D3}.

The pressure, p_1, is determined in accordance with 14.5.2:

$$p_1 = p_{sg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2 f_{M1}$$

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 and f_{M1} are determined in accordance with 14.4.3.2, 14.4.4 and 14.5.2.

33.2.3.2.3 Fan pressure

The fan pressure, p_{fD}, and the fan static pressure, p_{sfD}, are determined using the following equations:

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

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

The mean density is equal to:

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

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

33.2.3.3 Determination of volume flow rate

The volume flow rate is given by the following equation:

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

where

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

33.2.3.4 Calculation of fan air power

The fan work per unit mass and the fan air power are determined in accordance with 14.8.1 and 14.8.2.

33.2.3.5 Determination of fan efficiencies

The procedure described in 31.2.3.5 is followed.

33.2.4 Simplified method

The Mach number, $Ma_{2\text{ref}}$, is less than 0,15 and the pressure ratio less than 1,02.

At a section of the test ducts, the stagnation and static temperatures may be considered as equal:

$$\Theta_{\text{sg}x} = \Theta_x$$

$$f_{\text{M1}} = f_{\text{M2}} = 1$$

$$\rho_1 = \rho_2 = \rho_3 = \rho_{\text{sg1}} = \rho_{\text{sg2}} = \rho_{\text{sg3}} = \rho_{\text{a}} = \frac{p_{\text{a}}}{R_{\text{w}}\,\Theta_{\text{a}}}$$

$$\Theta_1 = \Theta_{\text{sg1}} = \Theta_{\text{sg2}} = \Theta_2 = \Theta_{\text{sg3}} = \Theta_3 = T_{\text{a}} + 273{,}15$$

and the procedure based upon the assumption of incompressible air flow through the test airway is applied.

33.2.4.1 Determination of mass flow rate

The mass flow rate is determined by the procedure described in 31.2.3.1 and 31.2.4.2.1 with $\rho_{\text{u}} = \rho_{\text{sg1}}$. However, the Reynolds number effect on the flow rate coefficient, α, should be taken in account.

33.2.4.2 Determination of fan pressure

33.2.4.2.1 Fan outlet pressure

The stagnation pressure, p_{sg2}, is given by the following equation (see 31.2.4.2.1):

$$p_{\text{sg2}} = p_4 + \frac{1}{2\rho_1}\left(\frac{q_m}{A_4}\right)^2 \left[1 + \left(\xi_{2-4}\right)_4\right]$$

where $(\xi_{2-4})_4$ is determined in accordance with 28.6.1 and 28.6.2 and Figure 35 a) and b).

$$p_{\text{esg2}} = p_{\text{e4}} + \frac{1}{2\rho_1}\left(\frac{q_m}{A_4}\right)^2 \left[1 + \left(\xi_{2-4}\right)_4\right]$$

$$p_2 = p_{\text{sg2}} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2$$

$$p_{\text{e2}} = p_{\text{esg2}} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2$$

33.2.4.2.2 Fan inlet pressure (see 32.2.4.2.1)

$$p_{\text{sg1}} = p_3 + \frac{1}{2\rho_1}\left(\frac{q_m}{A_3}\right)^2 \left[1 + \left(\xi_{3-1}\right)_3\right] \left(p_{\text{e3}} \text{ being measured}\right)$$

or, if p_{e3} is not measured

$$p_{sg1} = p_a + \frac{1}{2\rho_1}\left(\frac{q_m}{A_3}\right)^2 \left(\xi_{3-1}\right)_3$$

$$p_{esg1} = \frac{1}{2\rho_1}\left(\frac{q_m}{A_3}\right)^2 \left(\xi_{3-1}\right)_3$$

where $(\xi_{3-1})_3 < 0$ is determined in accordance with 28.6.4 and 28.6.5, 33.2.3.2.1 or 33.2.3.2.2.

The pressure, p_1, is given by the following equation:

$$p_1 = p_{sg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2$$

33.2.4.2.3 Fan pressure

The fan pressure, p_{fD}, and the fan static pressure, p_{sfD}, are given by the following equations:

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

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

33.2.4.3 Determination of volume flow rate

The volume flow rate is given by

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

33.2.4.4 Determination of fan air power

The fan air power, P_{uD}, and the fan static power, P_{usD}, are given by the following equations:

$$P_{uD} = p_{fD} q_{Vsg1}$$

$$P_{usD} = p_{sfD} q_{Vsg1}$$

33.2.4.5 Determination of fan efficiencies

Fan efficiencies are determined by the following equations:

$$\eta_{rD} = \frac{P_{uD}}{P_r}$$

$$\eta_{srD} = \frac{P_{usD}}{P_r}$$

33.3 Installation category B without outlet antiswirl device nor common segment, modified with addition of an inlet duct or inlet-duct simulation

This setup may be used for fans with low outlet swirling flow.

33.3.1 Flow rate determination

The flow rate is determined using multiple nozzles in the chamber, see Clause 22 and Figures 43 and 45 b).

33.3.2 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);
- outlet pressure, p_{e4};
- pressure, p_{e6}, upstream of the flowmeter;
- differential pressure, Δp;
- chamber temperature, T_6.

In the test enclosure, measure

- atmospheric pressure, p_a, at the mean altitude of the fan;
- 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 gas constant of humid air, R_w (see Clause 12).

33.3.3 General procedure for compressible fluid

This procedure should be applied when both the reference Mach number, Ma_{2ref}, is more than 0,15 and the pressure ratio more than 1,02.

33.3.3.1 Determination of mass flow rate

The mass flow rate is determined using multiple nozzles in the chamber, see Clause 22 and Figure 43.

The procedure described in 31.3.3.1.1 is followed.

33.3.3.2 Calculation of fan pressure

33.3.3.2.1 Fan outlet pressure

The procedure described in 31.3.3.2.1 is followed but $(\xi_{2-4})_4 = 0$ and section $A_{2.4}$ is the area of the test duct at the discharge in the chamber.

33.3.3.2.2 Fan inlet pressure

The procedure described in 33.2.3.2.2 is applied.

33.3.3.2.3 Fan pressure

The fan pressure, p_{fD}, and the fan static pressure, p_{sfD}, are determined as in 33.2.3.2.3.

33.3.3.3 Determination of volume flow rate

The volume flow rate is given by

$$q_{V\mathrm{sg1}} = \frac{q_m}{\rho_{\mathrm{sg1}}}$$

33.3.3.4 Calculation of fan air power

The fan work per unit mass and the fan air power are determined in accordance with 14.8.1, 14.8.2.1 and 14.8.2.2 and as in 31.2.3.4.

33.3.3.5 Calculation of fan efficiencies

The procedure described in 31.2.3.5 is followed.

33.3.4 Simplified method

This method is applicable to cases where the reference Mach number, $Ma_{2\mathrm{ref}}$, is less than 0,15 and pressure ratio less than 1,02.

At a section of the test duct, the stagnation and static temperatures are considered as equal:

$$\Theta_x = \Theta_{\mathrm{sg}x}$$

$$\rho_1 = \rho_{\mathrm{sg1}} = \rho_2 = \rho_{\mathrm{sg2}} = \rho_3 = \rho_4 = \rho_6 = \frac{p_\mathrm{a}}{R_\mathrm{w}\Theta_\mathrm{a}}$$

$$f_{\mathrm{M1}} = f_{\mathrm{M2}} = 1$$

The temperature in the test duct may be measured and the air flow through the test airway is considered as incompressible.

33.3.4.1 Determination of mass flow rate

The mass flow rate is determined in accordance with the procedure described in 31.3.3.1.1 with $\rho_6 = \rho_\mathrm{a}$. However, the Reynolds number effect on the flow rate coefficient α should be taken into account.

33.3.4.2 Determination of fan pressure

33.3.4.2.1 Fan outlet pressure

The stagnation pressure, p_{sg2}, and the pressure, p_2, are determined in accordance with the procedure used in 33.3.2.4.2.1, where $(\xi_{2-4})_4 = 0$ and $A_{2.4}$ is the area of the test duct at discharge into the chamber.

33.3.4.2.2 Fan inlet pressure

The stagnation pressure, p_{sg1}, and the pressure, p_1, are determined in accordance with the procedure used in 33.2.4.2.2.

33.3.4.2.3 Fan pressure

The fan pressure, p_{fD}, and the fan static pressure, p_{sfD}, are determined as in 33.2.4.2.3.

33.3.4.3 Determination of volume flow rate

The volume flow rate is given by the following equation:

$$q_{V\text{sg1}} = \frac{q_m}{\rho_{\text{sg1}}}$$

33.3.4.4 Determination of fan air power

The procedure described in 33.2.4.2.1 is followed.

33.3.4.5 Determination of fan efficiencies

Fan efficiencies are determined as in 33.2.4.5.

33.4 Installation category C with common inlet duct, modified with the addition of an outlet common segment with antiswirl device

33.4.1 Mass flow rate determination

The mass flow rate is determined using

— conical or bellmouth inlet, see Figure 44 a);

— inlet orifice with wall taps, see Figures 44 b) and 46 c);

— inlet orifice with wall taps, see Figures 44 c) and 46 c);

— in-duct orifice with taps at D and $D/2$, see Figures 44 d) and 46 c);

— Pitot-static tube traverse, see Figures 44 e) and 46 c).

33.4.2 Measurements to be taken during tests, (see Clause 20)

Measure:

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

— power input, P_a, P_o, P_e, and estimate impeller power (see 10.4);

— flowmeter differential pressure, Δp;

— flowmeter upstream pressure, p_{e7} or p_{e3};

— inlet static pressure, p_{e3};

— outlet static pressure, p_{e4}.

In the test enclosure, measure:

— atmospheric pressure, p_a, at the mean fan altitude;

— ambient temperature near duct 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).

33.4.3 General procedure for compressible fluid flow

This procedure should be applied when both the reference Mach number, $Ma_{2\text{ref}}$ (see 14.4.2), is more than 0,15 and the pressure ratio more than 1,02.

33.4.3.1 Calculation of mass flow rate

33.4.3.1.1 The flow rate is determined using:

— conical or bellmouth inlet, see Clause 23 and Figure 44 a);

— inlet orifice with wall taps, see 24.2.1 and Figure 44 c).

The procedure described in 32.2.3.1.1 is applied.

33.4.3.1.2 The flow rate is determined using an in-duct orifice with taps at D and $D/2$, see 24.7, 24.8 and Figure 44 d).

The procedure described in 32.2.3.1.2 is applied.

33.4.3.1.3 The flow rate is determined using a Pitot-static tube traverse, see Clause 25 and Figure 44 e) and f).

The procedure described in 32.2.3.1.3 is applied.

33.4.3.2 Determination of fan pressure

33.4.3.2.1 Fan inlet pressure

The stagnation pressure, p_{sg1}, and the pressure, p_1, are determined in accordance with the procedure described in 31.2.3.2.1.

33.4.3.2.2 Fan outlet pressure

The stagnation pressure, p_{sg2}, and the pressure, p_2, are determined in accordance with the following procedure (see 30.2.3.2.1).

Assuming that

$$p_4 = p_{\text{e4}} + p_{\text{a}}$$

$$\Theta_{\text{sg4}} = \Theta_{\text{sg2}} = \Theta_{\text{sg1}} + \frac{P_{\text{r}} \text{ or } P_{\text{e}}}{q_m c_p}$$

The Mach number at section 4 and the temperature, Θ_4, are determined in accordance with 14.4.3.1.

$$\Theta_4 = \Theta_{\text{sg4}} \frac{Ma_4}{Ma_{\text{sg4}}}$$

$$\rho_4 = \frac{p_4}{R_{\text{w}} \Theta_4}$$

$$f_{\text{M4}} = 1 + \frac{Ma_4^2}{4} + \frac{Ma_4^4}{40} + \frac{Ma_4^6}{1\,600} \quad \text{(see 14.5.1)}$$

The conventional friction-loss coefficient between sections 2 and 4, $(\xi_{2-4})_4$, is calculated in accordance with 28.6 and Figure 35.

The stagnation pressure, p_{sg2}, is given by the following equation:

$$p_{sg2} = p_4 + \rho_4 \frac{v_{m4}^2}{2} f_{M4} \left[1 + \left(\xi_{2-4}\right)_4\right]$$

$$= p_4 + \frac{1}{2\rho_4}\left(\frac{q_m}{A_4}\right)^2 f_{M4} \left[1 + \left(\xi_{2-4}\right)_4\right]$$

or

$$p_{esg2} = p_{e4} + \frac{1}{2\rho_4}\left(\frac{q_m}{A_4}\right)^2 f_{M4} \left[1 + \left(\xi_{2-4}\right)_4\right]$$

The pressure, p_2, and the density, ρ_2, are calculated in accordance with 14.5.2, Ma_2 being determined in accordance with 14.4.3.2:

$$p_2 = p_{sg2} - \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 f_{M2}$$

or

$$p_{e2} = p_{esg2} - \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2 f_{M2}$$

33.4.3.2.3 Fan pressure

The fan pressure, p_{fD}, and the fan static pressure, p_{sfD}, are given by the following equations:

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

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

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

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

33.4.3.3 Calculation of volume flow rate

The volume flow rate is given by the following equation:

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

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

33.4.3.4 Calculation of fan air power

The fan work per unit mass and the fan air powers are calculated in accordance with 32.2.3.4.

33.4.3.5 Calculation of fan efficiencies

The fan efficiencies are determined in accordance with 32.2.3.5.

33.4.4 Simplified method

The reference Mach number, Ma_{2ref}, is less than 0,15 and the pressure ratio less than 1,02.

The air flow through the test airway may be considered as incompressible, except with an auxiliary fan between planes 3 and 5 [see Figure 44 b)].

$$\Theta_3 = \Theta_{sg3} = \Theta_1 = \Theta_{sg1} = \Theta_2 = \Theta_{sg2} = T_3 + 273{,}15$$

$$\rho_1 = \rho_2 = \rho_3$$

$$f_{M1} = f_{M2} = 1$$

$$k_p = 1$$

33.4.4.1 Determination of mass flow rate

The mass flow rate is determined in accordance with 33.4.1 and the assumptions of 32.2.4.1.

33.4.4.2 Determination of fan pressure

33.4.4.2.1 Fan inlet pressure

The stagnation pressure, p_{sg1}, and the pressure, p_1, are determined by the following procedure (see 32.2.4.2.1).

Assuming that

$$p_3 = p_{e3} + p_a$$

$$\rho_1 = \frac{p_3}{R_w \Theta_1}$$

$$p_{sg1} = p_3 + \frac{1}{2\rho_1}\left(\frac{q_m}{A_3}\right)^2 \left[1 + \left(\xi_{3-1}\right)_3\right] \quad \text{(see 31.2.3.2.1)}$$

$$p_{esg1} = p_{e3} + \frac{1}{2\rho_1}\left(\frac{q_m}{A_3}\right)^2 \left[1 + \left(\xi_{3-1}\right)_3\right]$$

$$p_1 = p_{sg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2$$

$$p_{e1} = p_{esg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2$$

33.4.4.2.2 Fan outlet pressure

The stagnation pressure, p_{sg2}, and the pressure, p_2, are determined by the following equations (see 31.2.4.2.2):

Assuming that

$$p_4 = p_{e4} + p_a$$

$$p_{sg2} = p_4 + \frac{1}{2\rho_1}\left(\frac{q_m}{A_4}\right)^2 \left[1 + \left(\xi_{2-4}\right)_4\right]$$

or

$$p_{esg2} = p_{e4} + \frac{1}{2\rho_1}\left(\frac{q_m}{A_4}\right)^2 \left[1 + \left(\xi_{2-4}\right)_4\right]$$

$$p_2 = p_{sg2} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2$$

$$p_{e2} = p_{esg2} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_2}\right)^2$$

33.4.4.2.3 Fan pressure

The fan pressure, p_{fD}, and the fan static pressure, p_{sfD}, are given by the following equations:

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

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

33.4.4.3 Determination of volume flow rate

The volume flow rate is given by

$$q_{V\text{sg1}} = \frac{q_m}{\rho_{sg1}}$$

33.4.4.4 Determination of fan air power

In accordance with 14.8.5.6

$$P_{uD} = q_{V\text{sg1}}\, p_{fD}$$

$$P_{usD} = q_{V\text{sg1}}\, p_{sfD}$$

33.4.4.5 Determination of fan efficiencies

Fan efficiencies are determined in accordance with 14.8.1 and 14.8.2.

33.5 Installation category C, modified with the addition of an outlet-duct simulation without antiswirl device

This setup may be used for fans without outlet swirling flow or for large fans.

In this case, by mutual agreement between the parties concerned, the fan performance may be measured using this installation.

Results obtained in this way may differ to some extent from those obtained using common segments on both the inlet and outlet sides, especially if the flow produces a large swirl at the outlet.

33.5.1 Mass flow rate determination

The mass flow rate is determined using

- Pitot-static tube traverse upstream of the chamber, see Figures 45 a) and 46 e);
- multiple nozzles in chamber, see Figures 45 b) and 46 e);
- Pitot-static tube traverse, see Figures 44 f) and 46 g);

33.5.2 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);
- differential pressure, Δp;
- pressure, p_{e7} or p_{e5}, upstream of flowmeter;
- chamber stagnation or static pressure, p_{e3} or p_{esg3};
- chamber temperature, T_3;
- power input, P_{ex}, of auxiliary fan (optional).

In the test enclosure, measure

- atmospheric pressure, p_a, at the mean fan altitude;
- ambient temperature near the fan inlet, T_a;
- dry and wet bulb temperatures, T_d and T_w.

Determine the ambient density, ρ_a, and the gas constant of humid air, R_w (see Clause 12).

33.5.3 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 more than 1,02.

33.5.3.1 Determination of mass flow rate

33.5.3.1.1 The mass flow rate is determined using a Pitot-static tube traverse, see Clause 25 and Figures 46 e) and g), 44 f).

The procedure described in 32.3.3.1.2 is applied.

33.5.3.1.2 The mass flow rate is determined using multiple nozzles in the chamber, see Clause 22 and Figures 46 e) and 45 b).

The procedure developed in 34.3.3.1.4 is applied.

33.5.3.2 Determination of fan pressure

33.5.3.2.1 Fan inlet pressure

33.5.3.2.1.1 The inlet pressure, p_{e3}, is measured in the inlet duct, see Figures 44 a) to d) and 46 f).

See 31.2.3.2.2.

a) There is no auxiliary fan between planes 5 and 3.

$$p_3 = p_{e3} + p_a$$

$$\Theta_{sg3} = \Theta_{sg7} = \Theta_a = \Theta_{sg1} = T_a + 273{,}15$$

The Mach number, Ma_3, and the ratio, Ma_3/Ma_{sg3}, are determined in accordance with 14.4.3.1.

$$\Theta_3 = \Theta_{sg3} \frac{Ma_3}{Ma_{sg3}}$$

$$\rho_3 = \frac{p_3}{R_w \Theta_3}$$

The stagnation pressure, p_{sg1}, is given by the following equations (see 14.5):

$$p_{sg1} = p_3 + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 f_{M3}\left[1 + \left(\xi_{3-1}\right)_3\right]$$

$$p_{esg1} = p_{e3} + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 f_{M3}\left[1 + \left(\xi_{3-1}\right)_3\right]$$

where

$(\xi_{3-1})_3 < 0$ is the conventional friction-loss coefficient between sections 1 and 3 calculated in accordance with 28.6.4 and 28.6.5 [see 32.2.3.2.1 a)];

f_{M3} is the Mach factor (see 14.5.1).

b) There is an auxiliary fan between planes 5 and 3 [see 32.2.3.2.1 b)].

When the impeller power, P_{rx}, or the electrical power, P_{ex}, in the case of an immersed motor of the auxiliary fan, is measured:

$$\Theta_{sg3} = \Theta_{sg7} + \frac{P_{rx} \text{ or } P_{ex}}{q_m c_p} = \Theta_{sg1}$$

In the other cases, the temperature in the inlet duct is measured and assumed as a stagnation temperature Θ_{sg3}. The temperature, Θ_3, is determined in accordance with 14.4.3.1 and the stagnation pressure, p_{sg1}, calculated as in the first case.

The pressure, p_1, is determined after calculation of the Mach number, Ma_1, and the ratio Θ_1/Θ_{sg1} in accordance with 14.4.3.2 with $\Theta_{sg1} = \Theta_{sg3}$.

The density ρ_1 is calculated in accordance with 14.4.4 and the static pressure p_1 is given by the following equation:

$$p_1 = p_{sg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2 f_{M1}$$

or

$$p_{e1} = p_{esg1} - \frac{1}{2\rho_1}\left(\frac{q_m}{A_1}\right)^2 f_{M1}$$

33.5.3.2.1.2 The inlet pressure, p_{e3}, is measured in the test chamber (see 32.3.3.2.1).

a) The pressure measured in the chamber is a gauge pressure, p_{e3}.

Assuming that

$$f_{M3} = 1$$

$$p_3 = p_{e3} + p_a$$

$$\Theta_{sg1} = \Theta_{sg3} = \Theta_3 = T_3 + 273{,}15$$

$$\rho_3 = \frac{p_3}{R_w \Theta_3}$$

The stagnation pressure at section 3, p_{sg3}, is given by the following equation:

$$p_{sg3} = p_3 + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2$$

b) The pressure measured in the chamber is a gauge stagnation pressure, p_{esg3}.

$$p_{sg3} = p_{esg3} + p_a$$

$$\Theta_{sg3} = \Theta_3 = \Theta_{sg1} = T_3 + 273{,}15$$

There is no friction-loss allowance for the inlet-duct simulation of length D_1 or $2D_1$, and

$$p_{sg1} = p_{sg3}$$

$$p_{esg1} = p_{esg3}$$

When an inlet-duct simulation longer than D_1 or $2D_1$ is required, friction losses in this duct may be taken into account.

The stagnation pressure, p_{sg1}, is given by the following equations:

$$p_{sg1} = p_{sg3} + \frac{1}{2\rho_{3.1}}\left(\frac{q_m}{A_1}\right)^2 f_{M3.1}\left(\xi_{3-1}\right)_1$$

or

$$p_{esg1} = p_{esg3} + \frac{1}{2\rho_{3.1}} \left(\frac{q_m}{A_1}\right)^2 f_{M3.1} \left(\xi_{3-1}\right)_1$$

where

$\rho_{3.1}$ is the air density at the throat section of the bell mouth calculated in accordance with 14.4.3.2 and 14.4.4;

$f_{M3.1}$ is the Mach factor corresponding to the Mach number $Ma_{3.1}$;

$(\xi_{3-1})_1$ is the conventional friction-loss coefficient calculated in accordance with 28.6.4 and 28.6.5

$$\left(\xi_{3-1}\right)_1 = -\Lambda \frac{L}{D_1} < 0$$

L is the length of the inlet-duct simulation of diameter D_1.

The pressure p_1 is given by the following equation:

$$p_1 = p_{sg1} - \frac{1}{2\rho_1} \left(\frac{q_m}{A_1}\right)^2 f_{M1}$$

where Ma_1, ρ_1 and f_{M1} are calculated in accordance with 14.4.3.2, 14.4.4 and 14.5.

$$p_{e1} = p_{esg1} - \frac{1}{2\rho_1} \left(\frac{q_m}{A_1}\right)^2 f_{M1}$$

33.5.3.2.2 Fan outlet pressure

There is no friction-loss allowance for the outlet duct.

$p_2 = p_a$ or $p_{e2} = 0$

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

The stagnation pressure, p_{sg2}, is given by the following equation:

$$p_{sg2} = p_2 + \frac{1}{2\rho_2} \left(\frac{q_m}{A_2}\right)^2 f_{M2}$$

or

$$p_{esg2} = \frac{1}{2\rho_2} \left(\frac{q_m}{A_2}\right)^2 f_{M2}$$

where ρ_2 is the density at the fan outlet:

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

Ma_2 and Θ_2 are calculated in accordance with 14.4.3.1.

33.5.3.2.3 Fan pressure

The pressure, p_{fD}, and the fan static pressure, p_{sfD}, are determined by the following equations:

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

$$p_{sfD} = p_2 - p_{sg1} = - p_{esg1}$$

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

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

33.5.3.3 Determination of volume flow rate

The volume flow rate is given by the following equation:

$$q_{V\mathrm{sg1}} = \frac{q_m}{\rho_{sg1}}$$

where

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

33.5.3.4 Determination of fan air power

The fan work per unit mass and the fan air power are determined in accordance with the procedure described in 33.2.3.4.

33.5.3.5 Determination of fan efficiencies

The procedure described in 32.2.3.5 is followed.

33.5.4 Simplified method

The reference Mach number, Ma_{2ref}, is less than 0,15 and the pressure ratio less than 1,02.

At a section of the test duct, the stagnation and static temperatures are considered as equal:

$$\Theta_1 = \Theta_{sg1} = \Theta_3 = \Theta_{sg3} = \Theta_2 = \Theta_{sg2} = T_3 + 273{,}15$$

$$f_{M1} = f_{M2} = 1$$

$$k_p = 1$$

and the procedure based on the assumption of incompressible air flow through the test airway is applied, except with an auxiliary fan upstream of the fan tested.

33.5.4.1 Determination of mass flow rate

The mass flow rate is determined in accordance with 32.2.4.2.1 and 32.3.4.1.1.

33.5.4.2 Determination of fan pressure

33.5.4.2.1 Fan inlet pressure

33.5.4.2.1.1 The inlet pressure, p_{e3}, is measured in the test duct (see 32.2.4.2.1).

$$p_3 = p_{e3} + p_a$$

$$\Theta_3 = \Theta_{sg3} = \Theta_{sg1} = T_3 + 273{,}15$$

$$\rho_3 = \frac{p_3}{R_w \Theta_3} = \rho_1 = \rho_{sg1} = \rho_{sg2}$$

The stagnation pressure at the fan inlet is given by the following equations:

$$p_{sg1} = p_3 + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 \left[1 + \left(\xi_{3-1}\right)_3\right]$$

or

$$p_{esg1} = p_{e3} + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2 \left[1 + \left(\xi_{3-1}\right)_3\right]$$

where $(\xi_{3-1})_3 < 0$ is determined in accordance with 33.5.3.2.1.1.

The pressure, p_1, is given by the following equation:

$$p_1 = p_{sg1} - \frac{1}{2\rho_3}\left(\frac{q_m}{A_1}\right)^2$$

or

$$p_{e1} = p_{esg1} - \frac{1}{2\rho_3}\left(\frac{q_m}{A_1}\right)^2$$

33.5.4.2.1.2 The inlet pressure, p_{e3} or p_{esg3}, is measured in the test chamber (see 32.3.4.2.1).

$$p_3 = p_{e3} + p_a$$

$$\Theta_3 = \Theta_{sg3} = \Theta_{sg1} = \Theta_1 = T_3 + 273{,}15$$

$$p_{sg3} = p_3 + \frac{1}{2\rho_3}\left(\frac{q_m}{A_3}\right)^2$$

or

$$p_{sg3} = p_{esg3} + p_a$$

$$\rho_1 = \rho_{sg1} = \rho_3$$

$$p_{sg1} = p_{sg3} + \frac{1}{2\rho_3}\left(\frac{q_m}{A_1}\right)^2 \left(\xi_{3-1}\right)_1$$

$$p_{esg1} = p_{esg3} + \frac{1}{2\rho_3}\left(\frac{q_m}{A_1}\right)^2 \left(\xi_{3-1}\right)_1$$

$$p_1 = p_{sg1} - \frac{1}{2\rho_3}\left(\frac{q_m}{A_1}\right)^2$$

$$p_{e1} = p_{esg1} - \frac{1}{2\rho_3}\left(\frac{q_m}{A_1}\right)^2$$

where $(\xi_{3-1})_1 < 0$ is the conventional friction-loss coefficient.

33.5.4.2.2 Fan outlet pressure

At the fan outlet

$$p_2 = p_a \text{ or } p_{e2} = 0$$

$$p_{sg2} = p_2 + \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2$$

or

$$p_{esg2} = p_{e2} + \frac{1}{2\rho_2}\left(\frac{q_m}{A_2}\right)^2$$

$$\rho_3 = \rho_2 = \rho_1 = \rho_{sg3}$$

33.5.4.2.3 Fan pressure

The fan pressure, p_{fD}, and the fan static pressure, p_{sfD}, are given by the following equations:

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

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

33.5.4.3 Determination of volume flow rate

The volume flow rate is given by the following equation:

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

33.5.4.4 Determination of fan air power

The fan air powers are given by the following equations:

$$P_{uD} = q_{Vsg1}\, p_{fD}$$

$$P_{usD} = q_{Vsg1}\, p_{sfD}$$

33.5.4.5 Determination of fan efficiencies

Fan efficiencies are given by the following equations:

$$\eta_{rD} = \frac{P_{uD}}{P_r}$$

$$\eta_{srD} = \frac{P_{usD}}{P_r}$$

Annex A
(normative)

Fan pressure and fan installation category

A.1 General

In accordance with Clause 18, four fan installation categories are defined:

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

A.2 Fan pressure

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

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

For free outlet fans, the kinetic energy at the fan outlet is considered as lost when the fan is discharging into a test chamber or a partitioned-off room, even though the energy is provided by the fan. The effective fan pressure in this case is therefore the fan static pressure:

$$p_{sf} = p_{sg2} - p_{d2} \cdot f_{M2} - p_{sg1} = p_2 - p_{sg1} = p_{e2} - p_{esg1}$$

In standardized installations, the determination of these quantities depends upon the classification of the fan by category (see Clause 18), and they are defined as follows.

a) Category A: free inlet and free outlet

The effective fan pressure is:

$$p_{sfA} = p_{sg2} - \rho_2 \frac{v_{m2}^2}{2} f_{M2} - p_{sg1}$$

Given that:

$$p_{sg2} - \frac{\rho_2 v_{m2}^2}{2} f_{M2} = p_2$$

the quantity

$$p_2 - p_{sg1}$$

is the fan static pressure

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

Though the fan static pressure, p_{sfA}, is the effective fan pressure for fan installation category A, the fan pressure, p_{fA}, may be calculated by the following equation:

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

b) Category B: free inlet and ducted outlet

The effective fan pressure is the fan pressure given by the following equation:

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

and the fan static pressure is given by

$$p_{sfB} = p_{sg2} - \rho_2 \frac{v_{m2}^2}{2} f_{M2} - p_{sg1} = p_2 - p_{sg1} = p_{e2} - p_{esg1}$$

c) Category C: ducted inlet and free outlet

The effective fan pressure is the fan static pressure, p_{sf}, defined as follows:

$$p_{sfC} = p_{sg2} - \rho_2 \frac{v_{m2}^2}{2} f_{M2} - p_{sg1} = p_2 - p_{sg1} = p_{e2} - p_{esg1}$$

The fan pressure may be calculated as follows:

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

d) Category D: ducted inlet and ducted outlet

The effective fan pressure is the fan pressure defined by the following equation:

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

The fan static pressure may be calculated by

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

A.3 Calculation

A.3.1 Category A

A.3.1.1 Inlet test chamber (see Figures 36, 37 and 38)

The effective fan pressure in this case is the fan static pressure, p_{sfA}, given by the expressions:

$$p_2 = p_a$$

$$p_{sfA} = p_a - p_{sg1} = p_2 - p_{esg1}$$

where

$$p_{sg1} = p_3 + \rho_3 \frac{v_{m3}^2}{2} f_{M3}$$

$$p_{esg1} = p_{e3} + \rho_3 \frac{v_{m3}^2}{2} f_{M3}$$

The Mach number in the test chamber, Ma_3, is normally less than 0,15 and $f_{M3} = 1$.

The fan pressure, p_{fA}, is given by the expression:

$$p_{fA} = p_a + \rho_2 \frac{v_{m2}^2}{2} f_{M2} - \left(p_3 + \rho_3 \frac{v_{m3}^2}{2} \right) = \rho_2 \frac{v_{m2}^2}{2} f_{M2} - \left(p_{e3} + \rho_3 \frac{v_{m3}^2}{2} \right)$$

where ρ_2, v_{m2}, f_{M2}, ρ_3, v_{m3} are calculated in accordance with 14.4, 14.5, 14.6 or 14.8.5.

A.3.1.2 Outlet test chamber

When a fan is drawing air (see Figure 46) from the test enclosure or the free atmosphere, the inlet stagnation pressure is equal to the atmospheric pressure:

$$p_2 = p_4$$

$$p_{sg1} = p_a$$

and the effective fan pressure is the fan static pressure:

$$p_{sfA} = p_4 - p_a = p_{e4}$$

The fan pressure is given by the following equation:

$$p_{fA} = p_4 + \rho_2 \frac{v_{m2}^2}{2} f_{M2} - p_a = p_{e4} + \rho_2 \frac{v_{m2}^2}{2} f_{M2}$$

where ρ_2, v_{m2}, f_{M2} are calculated in accordance with 14.4, 14.5 or 14.6.

NOTE The size of the outlet chamber is very large (equivalent to a large open space) relative to the fan size. The cross-sectional area shall be at least 2 or else 3,2 times the cross-sectional area of the inlet test chamber, depending on the type of chamber, for the same fan.

A.3.2 Category B

The fan draws air from the test enclosure or from the free atmosphere:

$$p_{sg1} = p_a$$

The effective fan pressure is in this case the fan pressure given by the following equation:

$$p_{fB} = p_{sg2} - p_{sg1} = p_{sg2} - p_a = p_{esg2}$$

and

$$p_{sg2} = p_4 + \rho_4 \frac{v_{m4}^2}{2} f_{M4} \left[1 + \left(\xi_{2-4} \right)_4 \right]$$

$$p_{esg2} = p_{e4} + \rho_4 \frac{v_{m4}^2}{2} f_{M4} \left[1 + \left(\xi_{2-4} \right)_4 \right]$$

and the fan static pressure is given by

$$p_{\mathrm{sfB}} = p_{\mathrm{sg2}} - \rho_2 \frac{v_{\mathrm{m2}}^2}{2} f_{\mathrm{M2}} - p_{\mathrm{a}}$$
$$= p_{\mathrm{esg2}} - \rho_2 \frac{v_{\mathrm{m2}}^2}{2} f_{\mathrm{M2}}$$

where ρ_4, v_{m4}, f_{M4}, ρ_2, v_{m2}, f_{M2} are calculated in accordance with 14.4, 14.5 or 14.6.

A.3.3 Category C

The effective fan pressure in this case is the fan static pressure which may be calculated by the following equation:

$$p_2 = p_{\mathrm{a}}$$

$$p_{\mathrm{sfC}} = p_2 - p_{\mathrm{sg1}} = p_{\mathrm{a}} - \left\{ p_3 + \rho_3 \frac{v_{\mathrm{m3}}^2}{2} f_{\mathrm{M3}} \left[1 + \left(\xi_{3-1} \right)_3 \right] \right\}$$
$$= -\left\{ p_{\mathrm{e3}} + \rho_3 \frac{v_{\mathrm{m3}}^2}{2} f_{\mathrm{M3}} \left[1 + \left(\xi_{3-1} \right)_3 \right] \right\}$$

and the fan pressure is given by

$$p_{\mathrm{fC}} = p_{\mathrm{a}} + \rho_2 \frac{v_{\mathrm{m2}}^2}{2} f_{\mathrm{M2}} - \left\{ p_3 + \rho_3 \frac{v_{\mathrm{m3}}^2}{2} f_{\mathrm{M3}} \left[1 + \left(\xi_{3-1} \right)_3 \right] \right\}$$

where ρ_3, v_{m3}, f_{M3}, ρ_2, v_{m2}, f_{M2} are calculated in accordance with 14.4, 14.5, 14.6 or 14.8.5.

A.3.4 Category D

The effective fan pressure in this case is the fan pressure which is given by the following equation:

$$p_{\mathrm{fD}} = p_{\mathrm{sg2}} - p_{\mathrm{sg1}} = p_4 + \rho_4 \frac{v_{\mathrm{m4}}^2}{2} f_{\mathrm{M4}} \left[1 + \left(\xi_{2-4} \right)_4 \right] - \left\{ p_3 + \rho_3 \frac{v_{\mathrm{m3}}^2}{2} f_{\mathrm{M3}} \left[1 + \left(\xi_{3-1} \right)_3 \right] \right\}$$
$$= p_{\mathrm{e4}} + \rho_4 \frac{v_{\mathrm{m4}}^2}{2} f_{\mathrm{M4}} \left[1 + \left(\xi_{2-4} \right)_4 \right] - \left\{ p_{\mathrm{e3}} + \rho_3 \frac{v_{\mathrm{m3}}^2}{2} f_{\mathrm{M3}} \left[1 + \left(\xi_{3-1} \right)_3 \right] \right\}$$

and the fan static pressure is given by the following equation:

$$p_{\mathrm{sfD}} = p_2 - p_{\mathrm{sg1}} = p_{\mathrm{sg2}} - \rho_2 \frac{v_{\mathrm{m2}}^2}{2} f_{\mathrm{M2}} - \left\{ p_3 + \rho_3 \frac{v_{\mathrm{m3}}^2}{2} f_{\mathrm{M3}} \left[1 + \left(\xi_{3-1} \right)_3 \right] \right\}$$
$$= p_{\mathrm{esg2}} - \rho_2 \frac{v_{\mathrm{m2}}^2}{2} f_{\mathrm{M2}} - \left\{ p_{\mathrm{e3}} + \rho_3 \frac{v_{\mathrm{m3}}^2}{2} f_{\mathrm{M3}} \left[1 + \left(\xi_{3-1} \right)_3 \right] \right\}$$

where ρ_4, v_{m4}, f_{M4}, ρ_3, f_{M3}, ρ_2, v_{m2}, f_{M2} are calculated in accordance with 14.4, 14.5, 14.6 or 14.8.5.

NOTE $(\xi_{3-1})_3$ and p_{e3} are usually negative.

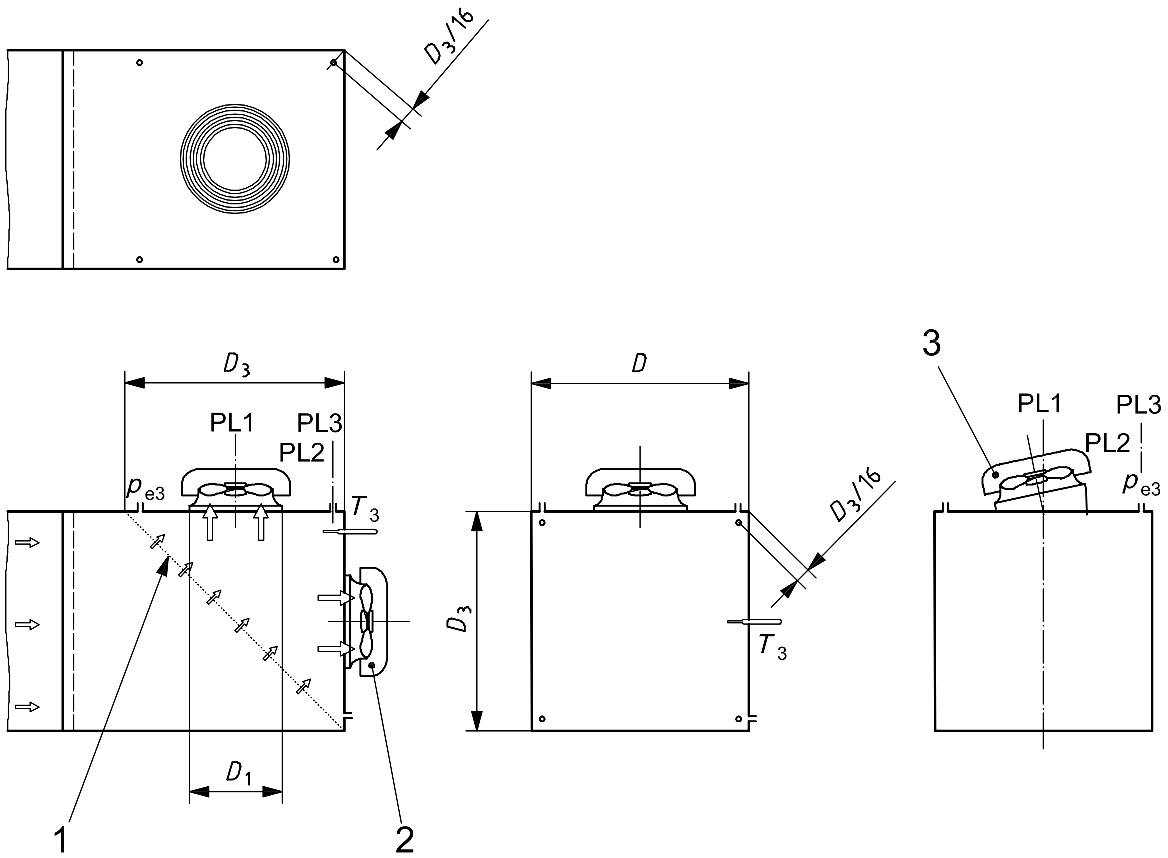
Annex B
(normative)

Fan-powered roof exhaust ventilators

B.1 In order to meet the special installation requirements of fan-powered roof exhaust ventilators where units with gravity-controlled shutters need to be tested in their correct mounting position, it is necessary to make slight departures from the standard configurations.

Figures B.1 and B.2 show two variants which are permitted for this type of unit only.

B.2 Figure B.1 shows one modification which involves the inclination of the final screen of 45 % free area max. and the alternative mounting position of the unit.



Key

1 flow-settling screen

2 alternative mounting arrangement and related pressure taps

3 inclined fan mounting arrangement

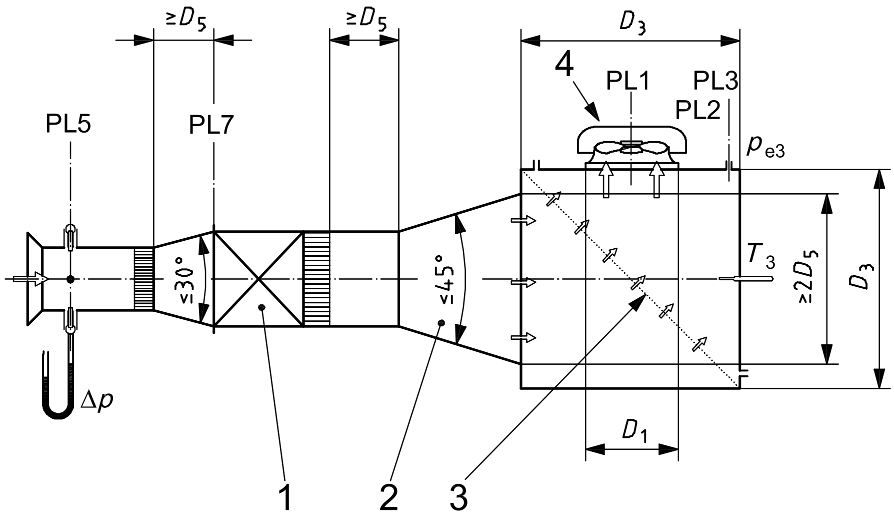
NOTE 1 The remainder of the test circuit is as shown in Figure 40 a) to e).

NOTE 2 $D_1 \leqslant 0{,}5D_3$ where D_1 is the diameter of the opening in the roof or the larger side of the rectangular opening.

NOTE 3 Measurements and calculations shall be made in accordance with 30.2.

Figure B.1 — Setup of a fan-powered roof exhaust ventilator on an inlet chamber

B.3 A further permissible variation is the use of screen loading at the outlet end of the inlet nozzle in accordance with category C tests as shown in Figure B.2.



Key

1 auxiliary fan

2 transition section

3 flow-settling screen

4 test fan (centrifugal impeller type shown)

NOTE p_{e3} is normally $\leqslant 0$.

Figure B.2 — Fan powered roof exhaust ventilator on a category C installation

This change of position of the control device, together with the use of a controlled expansion into the chamber, enables a single diagonal screen of 45 % maximum free area to be used within the chamber.

In the case of a very large inlet chamber, where the use of a diagonal screen becomes impractical, the screen may be omitted provided that it can be demonstrated that over the range of air volume flow rates under consideration, the air flow presented to the fan under test has a substantially uniform velocity profile and is free from swirl.

The test circuit is shown in Figure B.2.

Measure Δp, p_{e3}, T_3 and take $T_u = T_a$

$$p_{e2} = 0$$

$$p_{esg3} = p_{e3} + \rho_3 \frac{v_{m3}^2}{2} = p_{esg1}$$

$$P_{e3} < 0$$

Annex C
(informative)

Chamber leakage test procedure

C.1 General

The volume of interest is the volume between the measurement plane and the air-moving device. For an inlet chamber, the test pressure could be negative and, for outlet chambers, the test pressures could be positive.

Three methods of testing for leakage rate are recommended.

C.2 Pressure decay method

C.2.1 Calculations

Figure C.1 a) and b) show typical test setups where the test chamber is sealed and then pressurized and the valve closed. The initial static pressure, p_o is noted at time, $t = 0$. The pressure is recorded at periodic intervals (at intervals short enough to develop a pressure vs. time curve) until the pressure, p, reaches a steady state value.

Using the ideal gas law:

$$pV = mRT \quad \text{or} \quad p = \rho RT \tag{C.1}$$

where

- p is the static pressure;
- V is the chamber volume;
- m is the mass of air in the chamber;
- R is the gas constant;
- T is the absolute air temperature;
- ρ is the air density.

Differentiating with respect to time,

$$V \frac{\mathrm{d}p}{\mathrm{d}t} = \frac{\mathrm{d}m}{\mathrm{d}t} RT$$

and

$$Q = \frac{1}{\rho} \frac{\mathrm{d}m}{\mathrm{d}t}$$

or

$$Q = \frac{\mathrm{d}m}{\mathrm{d}t}\rho Q$$

Substituting and rearranging gives:

$$\frac{\mathrm{d}p}{\mathrm{d}t} = \frac{\rho QRT}{V}$$

or

$$Q = \frac{V}{\rho RT}\frac{\mathrm{d}p}{\mathrm{d}t}$$

and

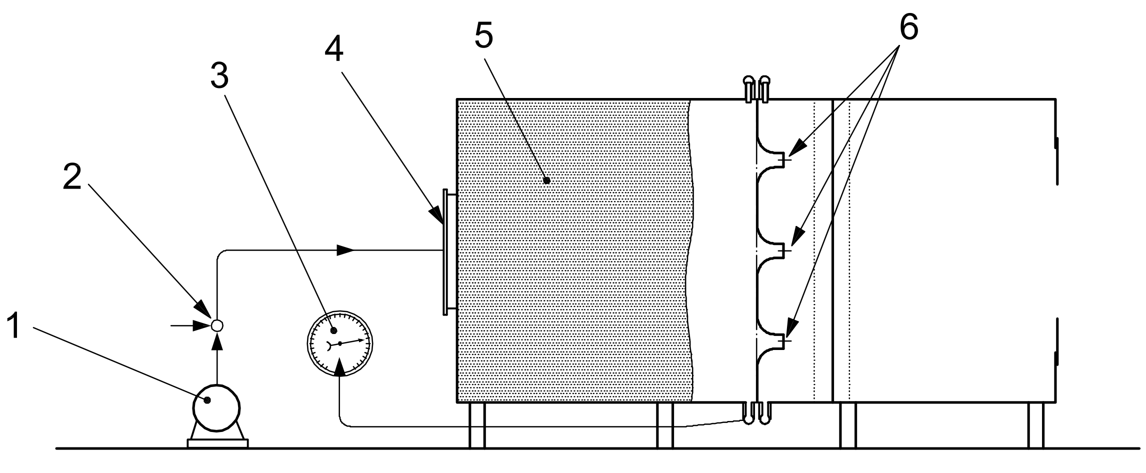
$$Q = \frac{V}{P}\frac{\mathrm{d}p}{\mathrm{d}t}$$

or

$$Q = \frac{V}{P}\frac{\Delta p}{\Delta t} \qquad \text{(C.2)}$$

where Q is the leakage air flow rate

Leakage rate, Q, can be determined from equation C.2 once the pressure decay curve [Figure C.1 c)] is known for the chamber.

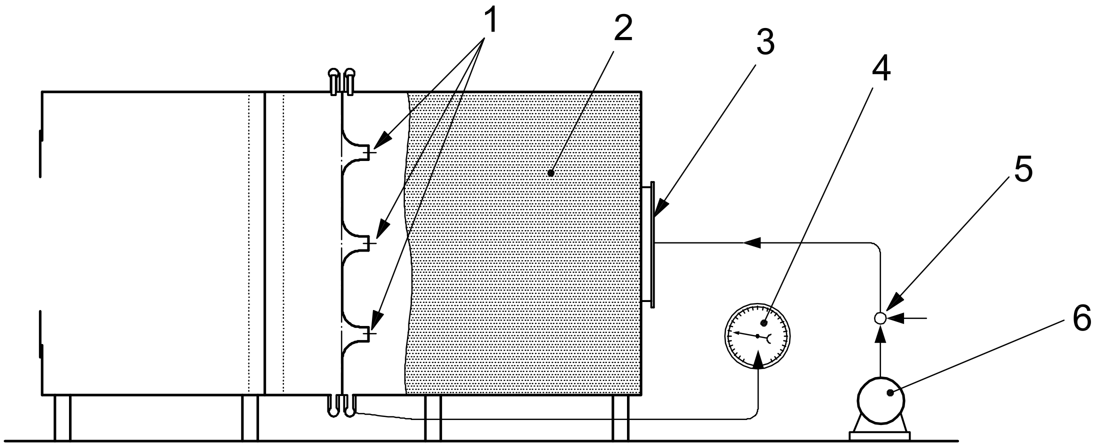


Key

1 fan or air compressor
2 valve
3 pressure gauge
4 test fan location
5 test chamber
6 nozzles plugged

a) Setup for outlet-side chamber

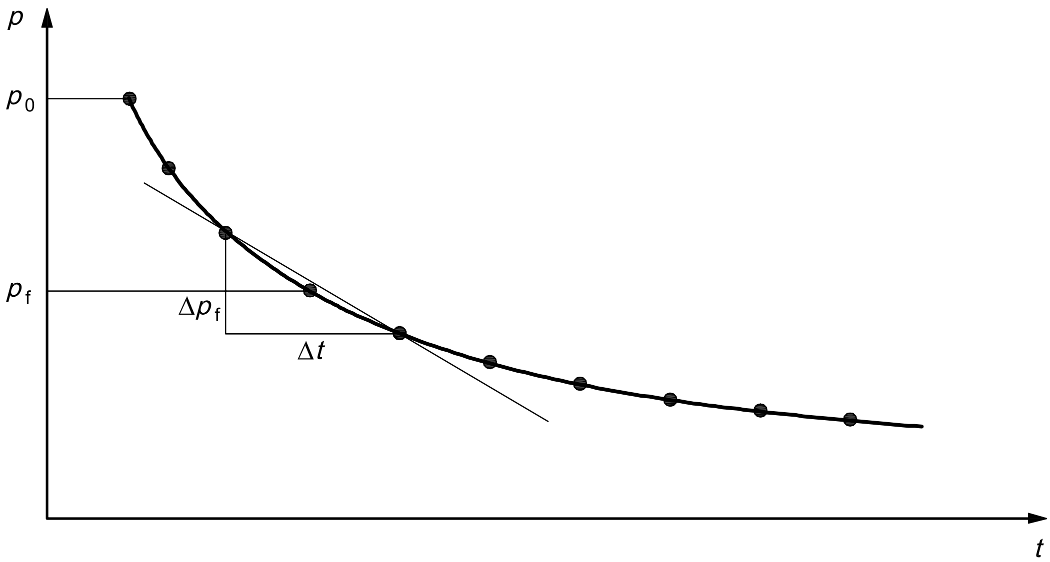
Figure C.1 — Pressure decay method for leakage test



Key

1 nozzles plugged
2 chamber
3 test fan location
4 pressure gauge
5 valve test
6 vacuum pump

b) Setup for inlet-side chamber



$$Q = \frac{V}{p_t}\,\frac{\Delta p_t}{\Delta t}$$

where

p_t = test pressure

$\frac{\Delta p_t}{\Delta t}$ = slope of curve in Figure C.1 a)

Δt_{min} = minimum time difference, 10 s

Key

p pressure, in pascals
t time, in seconds

c) Graph showing the decay in chamber pressure with time

Figure C.1 (*continued*)

C.2.2 Procedure

a) Pressurize or evacuate the test chamber to a test pressure, p_t, greater in magnitude than the pressure at which leakage is to be measured. Close the control valve.

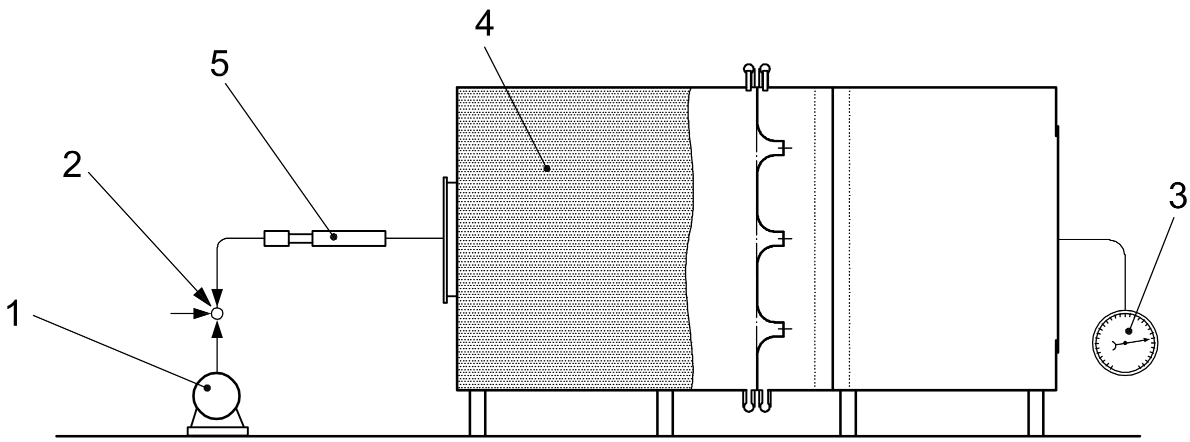
b) At time $t = 0$, start a stopwatch and record the pressure at periodic time intervals (a minimum of 3 readings is recommended) to get a decay curve as in Figure C.1 c). Continue to record until the pressure reaches a state in which the pressure does not change significantly.

c) Quick pressure changes indicate substantial leakage which must be located and attended to.

C.3 Flowmeter method

Figure C.2 below shows the test setup. The procedure is to pressurize or evacuate the test chamber after it is sealed and to use a flowmeter to establish the leakage flow rate. The pressure in the chamber is maintained constant. The flowmeter will give a direct reading of the leakage rate.

The source used to evacuate or pressurize the chamber must be sized to maintain a constant pressure in the chamber.



Key

1 fan or air compressor
2 valve
3 pressure gauge
4 test chamber
5 flowmeter

Figure C.2 — Leakage test setup, flowmeter method

C.4 Two-phase method

For test chambers divided in two parts by a partition, like multiple-nozzle chambers, the single step leakage test methods proposed in Clauses C.2 and C.3 cannot distinguish between leakage through the outer shell of the chamber and that through the nozzle wall.

A two-stage measurement method can provide separate estimates of the two leakages, allowing more information for eliminating the leakage and an estimate of the systematic error produced by the chamber leaks on the accuracy of each volume flow measure.

C.4.1 First phase

C.4.1.1 The connection of the test chamber with the fan or test duct is sealed with a method representative of the typical connection with the fan case or test duct.

C.4.1.2 A small nozzle (e.g. a 25 mm diameter nozzle) having a throat area, A_{tn}, is opened in the nozzle wall, while all the other nozzles are sealed.

C.4.1.3 The auxiliary fan is run, to bring the half-shell of the chamber located between the nozzle wall and the auxiliary fan to a negative pressure (for outlet chambers) or positive pressure (for inlet chambers) of the same order as the typical pressure inside the chamber, relative to the outside pressure. (There is a negative pressure limit for each chamber due to its structural integrity.)

C.4.1.4 The negative pressure inside the upstream half-shell of an outlet chamber, or the positive pressure inside the downstream half-shell of an inlet chamber, $p_{S\text{a}}$, is measured together with the differential pressure across the nozzle wall, Δp_{a}.

C.4.2 Second phase

C.4.2.1 A small nozzle (identical to that used in the first phase), having a throat area, A_{tn}, is mounted in a hole in the panel closing the opening of the test chamber normally connected to the fan case of the test duct.

C.4.2.2 All the nozzles in the nozzle wall are sealed.

C.4.2.3 The auxiliary fan is run to bring the downstream half-shell of the chamber to a negative pressure, or the upstream half-shell of an inlet chamber to a positive pressure of the same order as the typical differential pressure across the nozzle wall. (There is a negative pressure limit for each chamber due to its structural integrity.)

C.4.2.4 The new values for negative pressure inside the upstream shell of an outlet chamber, or the positive pressure inside the downstream half-shell of an inlet chamber is measured, $p_{S\text{b}}$, together with the differential pressure across the nozzle wall, Δp_{a}.

Solving the following system of equations, the equivalent leakage-path areas, through the chamber half-shell located between the connection opening and the nozzle wall, A_{c}, and through the nozzle wall itself, can be estimated, in the same units used for A_{tn}:

$$\sqrt{\mathrm{d}p_{\text{a}}}\,(A_{\text{tn}} + A_{\text{W}}) = \sqrt{p_{S\text{a}}} \cdot A_{\text{C}}$$

$$\sqrt{\mathrm{d}p_{\text{b}}} \cdot A_{\text{W}} = \sqrt{p_{S\text{b}}}\,(A_{\text{tn}} + A_{\text{W}})$$

Different values for test nozzle areas (A_{tna} and A_{tnb}) may be used, if the two nozzles are not identical.

NOTE This calculation is carried out under the simplifying assumptions that the discharge coefficient of the nozzles and also of any leakage path can be assumed to be unity, that a square-law relationship applies between pressure and leakage flow, that the equivalent leakage areas do not depend on the pressure and stress applied to the chamber structure, and particularly that they are not sensitive to the reversal of the static pressure differentials.

The leakage under test conditions, Q_{L}, can finally be estimated, with the following formula, for each measurement point, as a function of the fan static pressure, p_{sf}, of the nozzle wall differential pressure, Δp, and of the air density, ρ, providing an estimate of the measurement error due to chamber leakage.

The volume flow measures shall not be corrected with the calculated measure of the leakage flow, but the estimate of the leakage flow can be compared with the measured volume flow to estimate the relative error and validate the measurement.

$$Q_{\mathrm{L}} = Q_{\mathrm{C}} + Q_{\mathrm{W}} = A_{\mathrm{C}}\sqrt{\frac{2p_{\mathrm{sf}}}{\rho}} + A_{\mathrm{W}}\sqrt{\frac{2\Delta p}{\rho}}$$

Annex D
(informative)

Fan outlet elbow in the case of a non-horizontal discharge axis

In the case of centrifugal fans, installation category B or D, with a non-horizontal discharge axis, it is usually possible to temporarily orientate the casing to provide a horizontal outlet feeding into a horizontal test duct. When this is not possible it is will be necessary to include an elbow between the fan outlet and the common segment with pressure taps, after agreement between manufacturer and purchaser. The losses in the bend can vary according to the expected non-uniform velocity distribution at the fan discharge and as such the method for predicting the losses, given below, is for guidance only.

In addition it should be noted that, especially with larger fans, it may be difficult, for practical reasons, to construct a fully compliant standardized airway and in these cases agreement between the manufacturer and the purchaser on such configurations, tolerances to be used, etc. should be agreed prior to any test work.

An example of an elbow which could be used is shown in Figure D.1. Alternative bend configurations can be used.

The angle between the discharge axis and the axis of the standardized test duct should be the smallest possible.

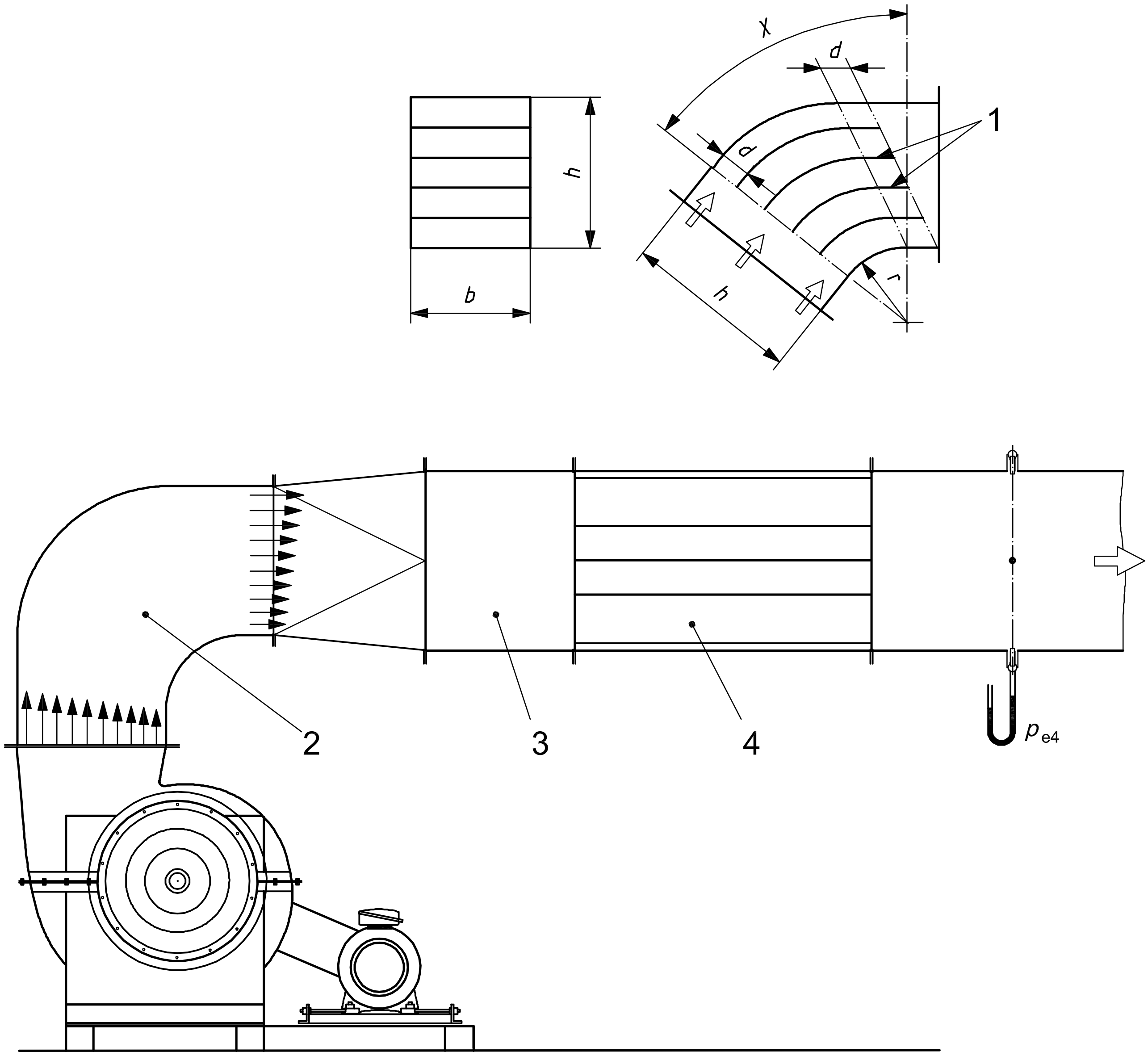
The elbow section should be located between sections A_2 and A_4 and should be of a uniform cross-section with splitter vanes.

A conventional friction-loss coefficient is given by the following equation:

$$(\xi_c)_4 = \left[\frac{\chi}{2\pi} \left(\frac{h}{b} \right)^{\frac{1}{6}} \right] \left(\frac{A_4}{A_c} \right)^2$$

where

A_c	is the area of the inlet and outlet sections of the elbow;
b	is the rectangular width of the duct;
h	is the rectangular height of the duct;
χ	is the angle of the elbow, in radians;
$(\xi_c)_4$	is the conventional friction-loss coefficient of the elbow calculated for section 4;
$(\chi/2\pi)(h/b)^{1/6}$	is plotted in Figure D.2 as a function of h/b and χ.



$d = h/5$
$r = 2{,}5d$

Key

1 turning vanes
2 turning vanes (bend and test-duct wall removed for clarity)
3 rectangular to round transition section
4 star type flow straightener

Figure D.1 — Dimensions of outlet elbow for testing large centrifugal fans

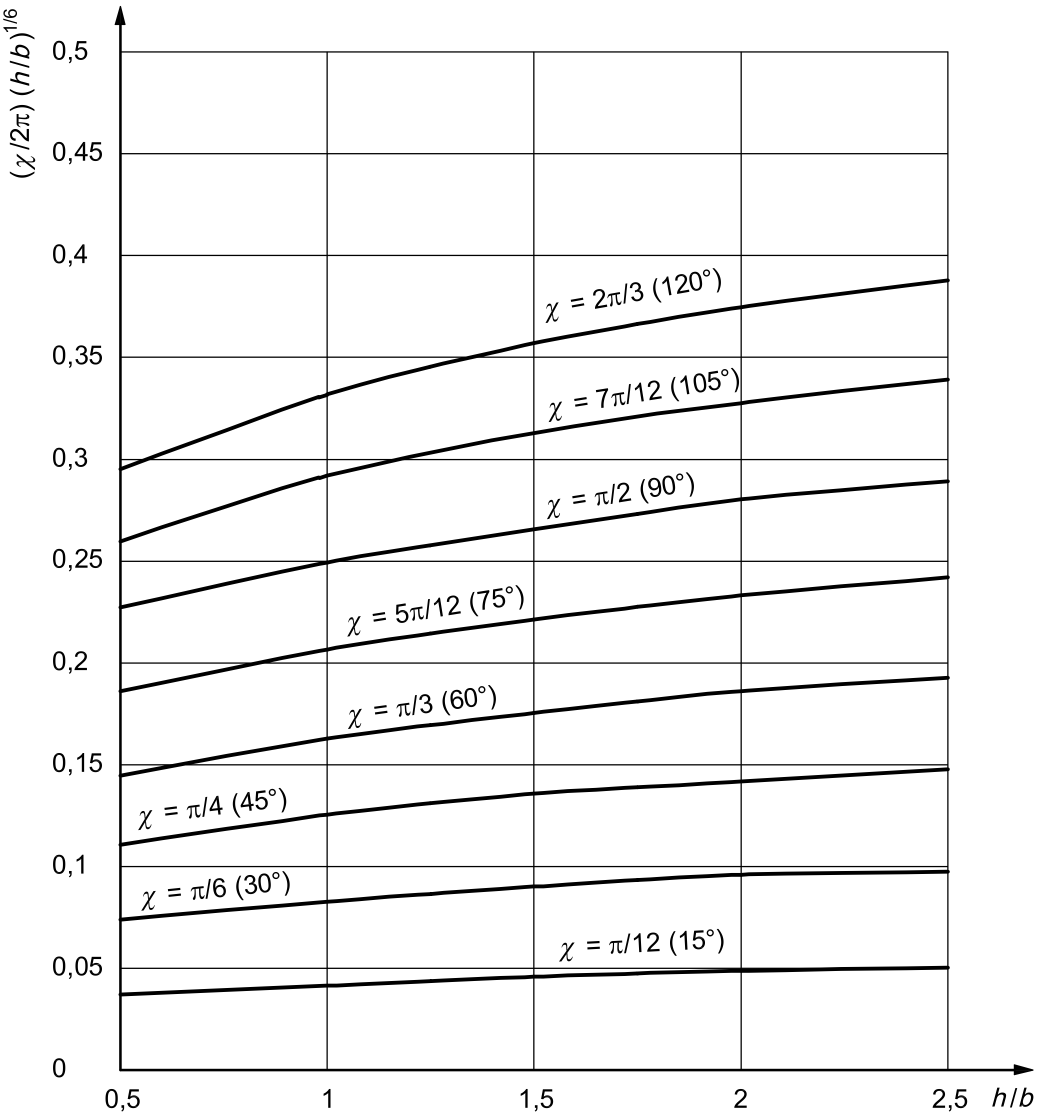


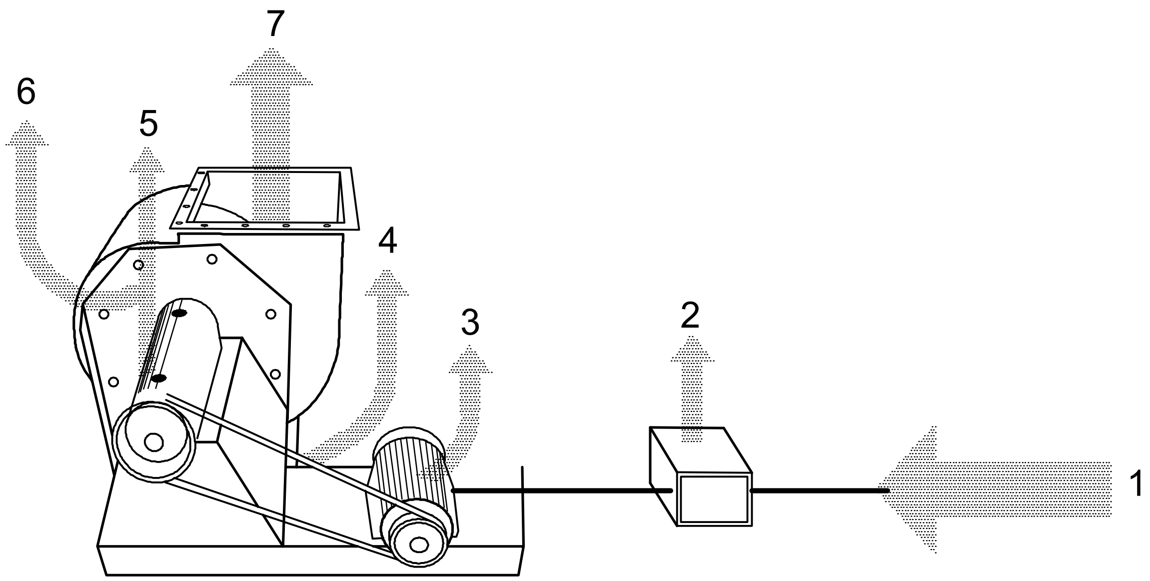
Figure D.2 — Plot of $(\chi/2\pi)(h/b)^{1/6}$ against h/b for calculating the pressure loss in an outlet elbow

Annex E
(informative)

Electrical input power consumed by a fan installation

E.1 Introduction

Economic and/or environmental concerns have resulted in renewed attention being given by many countries to the need for increasing the energy efficiency of all types of fan installations. The need for an agreed approach to the calculation of the electrical input power, P_e, is therefore necessary. Figure E.1 shows a typical V-belt-driven fan and shows the various losses that occur.



Key

1 electrical input power, P_e
2 variable speed device loss (heat)
3 motor losses (heat)
4 belt losses (heat)
5 bearing losses (heat)
6 impeller and casing aerodynamic losses (heat)
7 volume flow and pressure, P_u (air power)

Figure E.1 — Typical belt-driven fan showing power losses

E.2 Power consumption calculations

The electrical input power consumed by a fan installation is made up of a number of elements. These may be summarized as follows.

E.2.1 Impeller power: mechanical power supplied to the fan impeller in a cased fan. This is denoted P_r, and is expressed in watts or kilowatts. P_u is the fan air power (see 3.47).

The fan efficiency, $\eta_r = P_u/P_r$, is expressed as a decimal.

This is directly applicable to fan arrangements 4, 5, 15, and 16 (see ISO 13349:1999).

E.2.2 Fan shaft power: mechanical power supplied to the fan shaft. This is denoted P_a and is expressed in watts or kilowatts. P_u is the fan air power (see 3.47).

The fan efficiency, $\eta_a = P_u/P_a$, is expressed as a decimal.

This is directly applicable to all other fan arrangements, i.e. 1 to 3, 6 to 14, 17 to 19 (see ISO 13349:1999).

It differs from the impeller power by the addition of power losses in the fan bearings as a result of friction.

E.2.3 Bearing frictional power: these losses can be obtained from the formula:

$$P_b = 1{,}05 \times 10^{-4}\, M \cdot N$$

where

P_b is the power loss, in watts, in the bearing;

M is the total frictional moment, in newton millimetres, of the bearing;

N is the impeller/shaft rotational speed.

The frictional moment for a good quality, correctly lubricated bearing can be estimated with sufficient accuracy in most cases taking a coefficient of friction, μ, as constant, and using the following equation:

$$M = 0{,}5\, \mu C_d$$

where

M is the total frictional moment, in newton millimetres, of the bearing;

μ is the coefficient of friction as a constant for the bearings (see Table E.1);

C_d is the equivalent dynamic bearing load, in newtons;

d is the bearing(s) bore diameter(s), in millimetres.

Table E.1 — Approximate constant coefficients of friction for different bearing types (unsealed)

Type of bearing	Coefficient of friction μ
Deep groove ball	0,001 5
Angular contact ball — single row — double row	 0,002 0,002 4
Four-point contact ball	0,002 4
Self-aligning ball	0,001 0
Cylindrical roller — with cage, when $Fa = 0$ — full complement, when $Fa = 0$	 0,001 1 0,002 0
Needle roller	0,002 5
Taper roller	0,001 8
Spherical roller	0,001 8
Thrust ball	0,001 3
Cylindrical roller thrust	0,005 0
Needle roller thrust	0,005 0
Spherical roller thrust	0,001 8
NOTE For all other types of bearing, consult information supplied by the manufacturer.	

The total resistance to rotation of a bearing comprises the rolling and sliding friction in the rolling contacts, in the contact areas between rolling elements and the cage, the guiding surfaces of the rolling elements or the cage, the friction in the lubricant and the sliding friction of contact seals if fitted.

Where bearings are fitted with contact seals, the frictional losses in these may exceed those generated in the bearings. The frictional moment of seals for bearings that have seals on both sides may be estimated from the empirical equation:

$$M_{\text{seal}} = k_1 d_s a + k_2$$

where

M_{seal} is the frictional moment, in newton millimetres, of seals;

k_1 is a constant dependent on bearing type;

k_2 is a constant, in newton millimetres, dependent on bearing type and seal type;

d_s is the shoulder diameter, in millimetres, of the bearing (see Figure E.2);

a is a multiplicand depending on bearing and seal type.

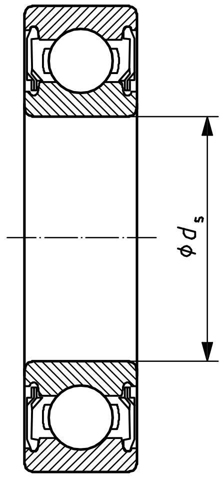


Figure E.2 — Section through a sealed rolling element bearing

Note that a can vary between 0 and 2,3; k_1 can vary between 0 and 0,06; k_2 can vary between 0 and 50. For confirmation of these values, consult information supplied by the bearing manufacturer where necessary, noting that they may use different symbols.

It will be appreciated that as

$$P_\mathrm{b} = P_\mathrm{a} - P_\mathrm{r}$$

then we may define the efficiency as fan bearing efficiency

$$\eta_\mathrm{b} = \frac{P_\mathrm{r}}{P_\mathrm{a}} = 1 - \frac{P_\mathrm{b}}{P_\mathrm{a}}$$

and

$$\eta_\mathrm{r} \times \eta_\mathrm{b} = \eta_\mathrm{a}$$

In all cases, it will be appreciated that it is probably better to test the same fan design in arrangements such as 1 and 4 (see ISO 13349:1999), obtaining the bearing losses by subtraction.

Note that the total moment of the fan bearings is the numerical sum of the individual moments ignoring sign (the direction of the moments is immaterial).

E.2.4 Transmission power: Many fans, especially in the heating, ventilation, air conditioning and refrigeration (HVACR) sector, are driven through pulleys and V-belts. This gives flexibility to fan manufacturers, who can cover a wide duty range with a limited number of models. The system designer can take comfort in the thought that if his system resistance calculations prove wrong then a simple pulley change can rectify the situation, provided there is sufficient motor capacity.

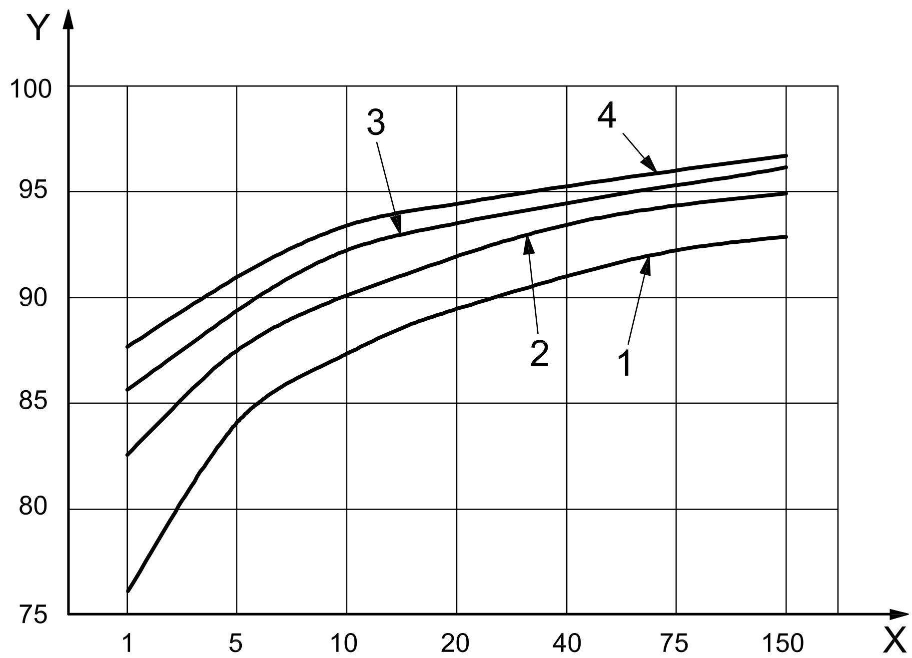
Care should be taken to neither over, nor under, provide in the design of the belt drive. In either case, its efficiency will suffer. While a well designed drive can exceed 95 % in its efficiency, provision of additional belts for a direct-on-line start can often reduce this considerably. A ‘soft’ start can be part of a better solution.

Where fans are driven through flexible couplings (arrangements 7, 8, 9, and 17, see ISO 13349:1999) these are normally assumed to have an efficiency of 97 % unless figures are available from the coupling supplier.

E.2.5 Motor power: Perhaps the most common type of motor used in fan installations (certainly above an output of 1 kW) is the squirrel cage AC induction design. It is robust, reliable, requires minimum maintenance and is relatively inexpensive. Over the last decade there has been a gradual improvement in its efficiency at both full and partial loads. This has been achieved by the inclusion of greater amounts of active material. In many countries three efficiency levels have been recognized (see Figure E.3); however, some countries are additionally specifying “premium” and “super-premium” efficiencies. The efficiency at partial loads (around 75 % of nameplate rating) may be greater than that at full load (see Figure E.4). This is contrary to earlier designs. It is important to use the efficiency at the actual absorbed power, which may be calculated by any of the methods described in 10.3.

Note that IEC 60034-30 [4] will standardize these efficiency grades with only minor modifications.

E.2.6 Control/power loss: This is often ignored, especially with inverters. The efficiency of these at high turn-down ratios may be very much less than 100 %, although, of course, powers absorbed by the fan will also be small. Figure E.4 is a typical example.



Key

X power, in kilowatts

Y motor efficiency at full load, η, as a percentage

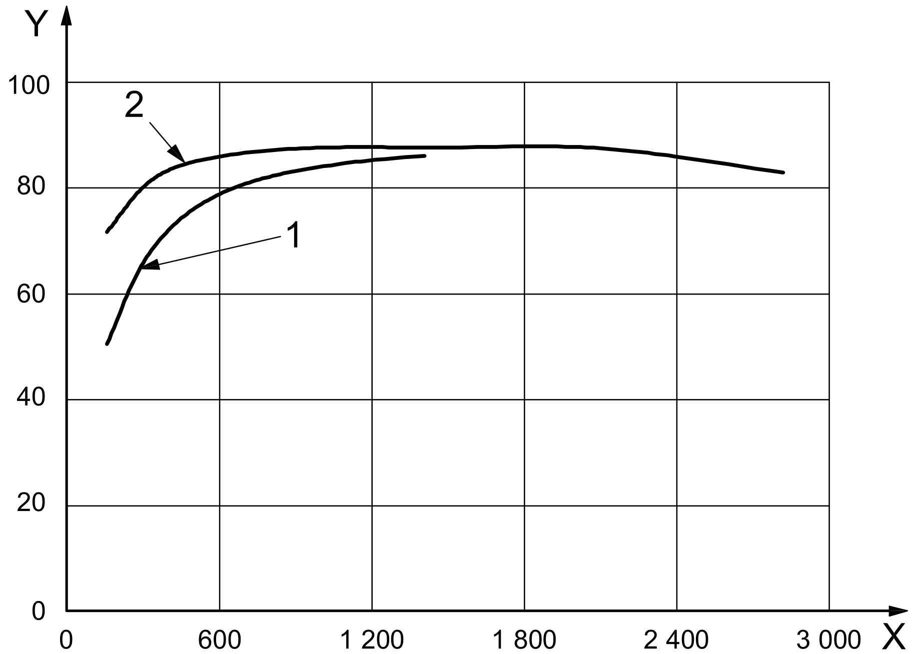
1 standard efficiency

2 high efficiency

3 premium efficiency

4 super-premium efficiency

Figure E.3 — Motor efficiency levels shown against power ratings for 2 and 4 pole machines



Key

X motor rotational frequency, in rotations per minute
Y motor efficiency at full load, η, as a percentage

1 standard efficiency
2 high efficiency

Figure E.4 — Efficiency of a typical motor (2 pole) at 100% (2) and 25% (1) load

E.3 Mains power required

The electrical power input abstracted from the mains may be calculated from the following equation:

$$P_e = \frac{q_{V\text{sg1}} \times p_f}{\eta_r \times \eta_b \times \eta_T \times \eta_m \times \eta_c}$$

where

P_e is the electrical input power, in kilowatts or watts;

$q_{V\text{sg1}}$ is the flow rate, in cubic metres per second or litres per second;

p_f is the fan pressure, in kilopascals or pascals;

η_r is the fan impeller efficiency, expressed as a decimal;

η_b is the fan bearing efficiency, expressed as a decimal;

η_T is the transmission efficiency, expressed as a decimal;

η_m is the motor efficiency, expressed as a decimal;

η_c is the control efficiency, expressed as a decimal.

NOTE 1 If fan pressure is expressed in pascals, then P_e will be in watts; if fan pressure is expressed in kilopascals, then P_e will be in kilowatts.

NOTE 2 $\eta_r \times \eta_b = \eta_a$, where η_a is the fan shaft efficiency.

NOTE 3 Fan pressure can also be defined on a static basis, provided that η_r is also calculated on the same basis. It should be noted that fan static efficiency is not theoretically correct as it could never be 100 % or 1.

NOTE 4 All duties and values should be for the appropriate installation category.

NOTE 5 These calculations are usually conducted at the enquiry stage before an audit can be carried out.

E.4 Specific fan power

This is a value which is being adopted in the legislation of many countries. Target levels are being specified for various types of plant. Figures can vary from around 1 to 2,5 depending on whether it is a new plant or a refurbishment and whether or not heating, cooling and filtration are included.

The specific fan power is expressed in kilowatts per (metre second) or watts per (litre second). As 1 000 W = 1 kW and 1 000 l = 1 m^3, the numerical value is the same in both cases.

Rearranging the previous formula, the specific fan power is given by

$$\frac{p_f}{\eta_r \times \eta_b \times \eta_T \times \eta_m \times \eta_c}$$

It will be seen that reduction of the system resistance is as important, if not more important, than the improvement in individual efficiencies.

These are usually summed for the total input power of all the fans in an HVACR system for the total flow rate (supply or extract, whichever is the greater).

Annex F
(informative)

Preferred methods of performance testing

The first edition, ISO 5801:1997, included methods of measuring flow rate from many existing national standards. These were recognized to be of equal merit, provided that the correct coefficients were established. It attempted to specify standard positions for the measurement of fan pressure. Inevitably this resulted in a very large document with many different ducting arrangements.

The second edition of this International Standard is the result of a survey of ISO members, deleting those methods which were the least popular. A significant reduction in the number of pages has been achieved. It should, however, be recognized that this is a step in the continuous evolution of ISO 5801. When ISO 5801 is next reviewed, we should achieve a further reduction in the number of methods. However, it must be realized that fan companies have to make a considerable investment when manufacturing such test stands. They therefore need advice on what are likely to be the preferred methods in a future edition of this International Standard.

To this end, with the current state of knowledge, the following are preferred for any test stands manufactured in the near future. It should be emphasized that work needs to be carried out to confirm these preferences and that the methods chosen are likely to give equivalent results within the specified tolerances given in ISO 13348.

The following is a list of these preferred arrangements for the future (not in order of preference):

— bellmouth or conical inlet, e.g. Figure 40 a), Clause 23, 30.2, Figure 44 a), 28.2;

— orifice plates, e.g. Figure 40 b), 30.2;

— multi-nozzle chambers (all installation types), e.g. Figures 40 e) and 41, and inlet/outlet simulations as required in accordance with 28.2 to 28.5;

— test duct with Pitot-tube traverse (especially for large or high pressure fans) — installation types B, C and D, e.g. Figures 42 c) and 44 f).

Bibliography

[1] ISO 13348, *Industrial fans — Tolerances, methods of conversion and technical data presentation*

[2] ISO 13349:1999, *Industrial fans — Vocabulary and definitions of categories*

[3] E51-100 (AFNOR), *Ventilateurs industriels — Influence de la compressibilité du fluide* [Industrial fans — Compressibility effect of the fluid]

[4] IEC 60034-30:—[1)], *Rotating electrical machines — Part 30: Efficiency classes of single-speed, three-phase, cage induction motors*

[5] ISO 5802, *Industrial fans — Performance testing* in situ

1) To be published.

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