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Fans — Performance testing using standardized airways

Ventilateurs — Essais aérauliques sur circuits normalisés

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Foreword

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

The procedures used to develop this document and those intended for its further maintenance are described in the ISO/IEC Directives, Part 1. In particular the different approval criteria needed for the different types of ISO documents should be noted. This document was drafted in accordance with the editorial rules of the ISO/IEC Directives, Part 2 (see www.iso.org/directives).

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. Details of any patent rights identified during the development of the document will be in the Introduction and/or on the ISO list of patent declarations received (see www.iso.org/patents).

Any trade name used in this document is information given for the convenience of users and does not constitute an endorsement.

For an explanation on the voluntary nature of standards, the meaning of ISO specific terms and expressions related to conformity assessment, as well as information about ISO's adherence to the World Trade Organization (WTO) principles in the Technical Barriers to Trade (TBT) see the following URL: www.iso.org/iso/foreword.html.

This document was prepared by Technical Committee ISO/TC 117, *Fans*.

This third edition cancels and replaces the second edition (ISO 5801:2007), which has been technically revised. It also incorporates the Technical Corrigendum ISO 5801:2007/Cor.1:2008.

Introduction

This document is the result of almost 50 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 become possible since 1997 to complete this document by agreement on certain essential points. It is to be borne in mind that the test equipment, especially for large fans, is very expensive and it was necessary to include in this document many setups from various national codes in order to authorize their future use. This explains the sheer volume of the first edition (ISO 5801:1997).

The second edition (ISO 5801:2007) of this document was the result of a survey of ISO members, deleting those methods that were the least frequently used. A significant reduction in the number of pages had been achieved.

For the third edition, the contents were reorganized to define and allow all possible configurations of defined component parts as standardized test setups. A further significant reduction of volume has been achieved by streamlining the content.

Essential features of this document are as follows.

— **Installation categories and test configurations** (see Clause 5 and Clause 6).

Since the connections of a duct to a fan inlet and/or outlet affect the fan's performance, a number of installation categories and test configurations need to be recognized.

— **Common segments** (see Clause 8).

It is essential that all standardized test airways to be used with fans need to 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.

— **Flow rate measurement** (see 12.5 and Annex A).

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

— **Test results** (see Clause 15).

Methods of measurement and calculation for the flow rate, for the fan pressure and for the fan efficiencies are established taking into account all compressibility effects of the air. For fan pressure less than 2 000 Pa, the change of density between fan inlet and fan outlet is allowed to be neglected. Other compressibility effects are allowed to be neglected for reference velocity values not higher than 65 m/s (see Clause 13).

Fans — Performance testing using standardized airways

1 Scope

This document specifies procedures for the determination of the performance of fans of all types except those designed solely for air circulation, e.g. ceiling fans and table fans. Testing of jet fans is described in ISO 13350.

This document provides estimates of uncertainty of measurement 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 documents are referred to in the text in such a way that some or all of their content constitutes requirements 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 5136, *Acoustics — Determination of sound power radiated into a duct by fans and other air-moving devices — In-duct method*

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 5802, *Industrial fans — Performance testing in situ*

ISO 13347 (all parts), *Industrial fans — Determination of fan sound power levels under standardized laboratory conditions*

ISO 13348, *Industrial fans — Tolerances, methods of conversion and technical data presentation*

ISO 13349, *Fans — Vocabulary and definitions of categories*

ISO 13350, *Fans — Performance testing of jet fans*

IEC 60034-1:2010, *Rotating electrical machines — Part 1: Rating and performance*

IEC 60034-2-1:2014, *Rotating electrical machines — Part 2-1: Standard methods for determining losses and efficiency 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*

3 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 13349 and the following apply.

ISO and IEC maintain terminological databases for use in standardization at the following addresses:

— ISO Online browsing platform: available at http://www.iso.org/obp

— IEC Electropedia: available at http://www.electropedia.org/

3.1
air
working fluid for tests with standardized airways shall be atmospheric air

3.2
standard air
air (3.1) with a density of 1,2 kg/m^3

Note 1 to entry: See 12.9.4.

3.3
upstream
direction from where the air flow comes

3.4
downstream
direction to where the air flow discharges

3.5
cross sectional area
A
area contained in a boundary

3.6
fan inlet area
A_1
cross sectional area (3.5) of fan inlet as defined in ISO 13349

3.7
fan outlet area
A_2
cross sectional area (3.5) of fan outlet as defined in ISO 13349

3.8
hydraulic diameter
D_h
four times the cross sectional area divided by the perimeter which encloses the area

$$D_h = \frac{4 \cdot A}{P_{\square}}$$

3.9
hydraulic mean depth
H_h
cross sectional area divided by the perimeter which encloses the area

$$H_h = \frac{A}{P_{\square}}$$

3.10
absolute temperature
θ
temperature expressed in Kelvin and calculated from the relative gas temperature in degree Celsius plus the thermodynamic temperature according to ISO 13349 of absolute zero

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

3.11
absolute temperature of moving air
θ
absolute temperature (3.10) theoretically registered by a thermal sensor moving at the air velocity calculated from the formula

$$\theta = \theta_{sg} \Big/ \left(1 + \frac{\kappa - 1}{2} \cdot Ma^2\right)$$

or

$$\theta = \theta_{sg} - \frac{v^2}{2 \cdot c_p}$$

3.12
stagnation temperature
θ_{sg}
absolute temperature (3.10) measured at an isentropic stagnation point

Note 1 to entry: The stagnation temperature is constant along an airway without exchange of energy or heat.

3.13
specific heat at constant pressure
c_p
amount of heat energy required to raise the temperature of air per unit of mass at constant pressure

3.14
specific heat at constant volume
c_V
amount of heat energy required to raise the temperature of air per unit of mass at constant volume

3.15
isentropic exponent
k
ratio of the specific heat at constant pressure to the specific heat at constant volume

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

3.16
specific gas constant
R
difference between the specific heat at constant pressure versus the specific heat at constant volume

$$R = c_p - c_V$$

use

$$R = \frac{p}{\rho \cdot \theta}$$

where

ρ is the *air density* (3.18) (kg/m³)

p is the *absolute pressure* (3.27) (Pa)

θ is the *absolute temperature* (3.10) (K)

For dry air, R_{dry} = 287.058 J/(kg·K)

3.17
specific gas constant for humid air
R_{wet}
atmospheric pressure divided by the density of air multiplied by the absolute ambient temperature

$$R_{\text{wet}} = \frac{p_a}{\rho \cdot \theta_a}$$

3.18
air density
ρ
air density calculated from the *absolute pressure*, p, (3.27) and the air temperature, θ

$$\rho = \frac{p}{R_{\text{wet}} \cdot \theta}$$

3.19
stagnation density
ρ_{sg}
air density (3.18) calculated from the *stagnation pressure*, p_{sg}, (3.29) and the *stagnation temperature*, θ_{sg} (3.12)

$$\rho_{\text{sg}} = \frac{p_{\text{sg}}}{R_{\text{wet}} \cdot \theta_{\text{sg}}}$$

3.20
mean density
ρ_{m}
mean value of the air densities of the fan

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

3.21
mean stagnation density
ρ_{sgm}
mean value of the stagnation densities of the fan

$$\rho_{\text{sgm}} = \frac{\rho_{\text{sg1}} + \rho_{\text{sg2}}}{2}$$

3.22
mass flow rate
q_m
mean value, over time, of the mass of air which passes through the airway per unit of time

3.23
volume flow rate
q_{V1}
mass flow rate (3.22) divided by the density at fan inlet

$$q_{V1} = \frac{q_m}{\rho_1}$$

3.24
volume flow rate at stagnation conditions
q_{Vsg1}
mass flow rate (3.22) divided by the *stagnation density* (3.19) at fan inlet

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

3.25
average velocity
v
mass volume flow rate divided by the cross sectional area multiplied by the air density

$$v = \frac{q_m}{\rho \cdot A}$$

3.26
reference velocity
$v_{2.\mathrm{ref}}$
velocity calculated at fan outlet, A_2, for the maximum *mass flow rate* (3.22) of the fan, $q_{m,\mathrm{max}}$, and for the reference density of *standard air* (3.2), ρ_{ref} = 1,200 kg/m^3

$$v_{2.\mathrm{ref}} = \frac{q_{m.\mathrm{max}}}{\rho_{\mathrm{ref}} \cdot A_2}$$

Note 1 to entry: See 12.9.3.

3.27
absolute pressure
p
pressure, measured with respect to absolute zero pressure

3.28
mean gauge pressure
differential pressure, measured with respect to ambient pressure $p_e = p - p_a$

3.29
stagnation pressure
p_{sg}
absolute pressure (3.27), if the air were brought to rest via an isentropic process

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

3.30
dynamic pressure
p_d

pressure calculated from the velocity, v, and the density, ρ

$$p_{\mathrm{d}} = \rho \cdot \frac{v^2}{2}$$

3.31
total pressure
p_{tot}
pressure calculated from the *absolute pressure* (3.27) and the *dynamic pressure* (3.30)

$$p_{\mathrm{tot}} = p + p_{\mathrm{d}}$$

3.32
fan dynamic pressure
p_{fd}
dynamic pressure (3.30) of the fan, defined at the fan outlet with the *average velocity* (3.25)

$$p_{\mathrm{fd}} = p_{d2} = \rho_2 \cdot \frac{v_2^2}{2}$$

3.33
fan pressure
p_{f}
difference between the *stagnation pressures* (3.29) at the fan outlet and the fan inlet

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

or difference between the *total pressures* (3.31) at the fan outlet and the fan inlet

$$p_{\mathrm{f}} = p_{\mathrm{tot2}} - p_{\mathrm{tot1}} \text{ allowed if } v_{2.\mathrm{ref}} \leq 65 \text{ m/s}$$

Note 1 to entry: For the establishment of the value 65 m/s, see Clause 13.

3.34
fan static pressure
p_{fs}
difference between the static pressure at the fan outlet and the *stagnation pressure* (3.29) at the fan inlet

$$p_{fs} = p_2 - p_{\mathrm{sg1}} = p_{\mathrm{sg2}} - p_{\mathrm{d2}} \cdot f_{\mathrm{M2}} - p_{\mathrm{sg1}} = p_{\mathrm{f}} - p_{\mathrm{fd}} \cdot f_{\mathrm{M2}}$$

or difference between the static pressure at the fan outlet and the *total pressure* (3.31) at the fan inlet

$$p_{\mathrm{fs}} = p_2 - p_{\mathrm{tot1}} = p_{\mathrm{tot2}} - p_{\mathrm{d2}} - p_{\mathrm{tot1}} = p_{\mathrm{f}} - p_{\mathrm{fd}} \text{ allowed if } v_{2.\mathrm{ref}} \leq 65 \text{ m/s}$$

Note 1 to entry: For the establishment of the value 65 m/s, see Clause 13.

3.35
fan pressure ratio
r
ratio of the average absolute *stagnation pressure* (3.29) at the outlet section of a fan to that at its inlet section as given by the following formula

$$r = \frac{p_{\mathrm{sg2}}}{p_{\mathrm{sg1}}}$$

Note 1 to entry: The fan pressure ratio is dimensionless.

3.36
rotational frequency of the impeller
n
number of revolutions of the fan impeller per second

3.37
tip speed of the impeller
u
peripheral speed of the impeller blade tips

$$u = \pi \cdot n \cdot D_r$$

3.38
velocity of sound
c
distance travelled per unit time by a sound wave at it propagates through air $c = \sqrt{\kappa \cdot R_{wet} \cdot \theta}$

3.39
Mach number
Ma
ratio of the air velocity to the *velocity of sound* (3.38)

$$Ma = \frac{v}{c}$$

3.40
peripheral Mach number
Ma_u
ratio of tip speed to the *velocity of sound* (3.38) at the fan inlet stagnation conditions

$$Ma_u = \frac{u}{\sqrt{\kappa \cdot R_{wet} \cdot \theta_{sg1}}}$$

3.41
Mach factor
f_M
correction factor applied to the *dynamic pressure* (3.30)

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

Note 1 to entry: The Mach factor f_M may be calculated by:

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

3.42
fan work per unit mass
y_f
increase in energy of air per unit mass passing through the fan

$$y_f = \frac{p_2 - p_1}{\rho_m} + \frac{1}{2} \cdot f_{M2} \cdot v_2^2 - \frac{1}{2} \cdot f_{M1} \cdot v_1^2$$

3.43
fan air power

P_u
mass flow multiplied by the fan work per unit mass $P_u = q_m \cdot y_f = q_{Vsg1} \cdot p_f \cdot k_P$

3.44
compressibility coefficient
k_P
ratio of the compressible to the incompressible determination of the fan output power, of the air moved, for the same fan inlet air density, pressure ratio and *mass flow rate* (3.22)

Note 1 to entry: The fan output power is derived on the assumption of polytropic compression with no heat transfer through the fan casing.

3.45
impeller power
P_r
mechanical power supplied to the fan impeller

3.46
fan shaft power
P_a
mechanical power supplied to the fan shaft

Note 1 to entry: Fan shaft power includes bearing losses, while fan *impeller power* (3.45) does not.

3.47
motor output power
P_o
shaft power output of the motor or other prime mover

3.48
motor input power
P_e
electrical input power supplied at the terminals of an electric motor drive without a variable speed drive

3.49
drive control electrical input power
P_{ed}
electrical input power measured at the input terminals to the variable speed drive of a motor

3.50
fan impeller efficiency
η_r
fan air power (3.43) divided by the *impeller power* (3.45)

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

3.51
fan shaft efficiency
η_a
fan air power (3.43) divided by the *fan shaft power* (3.46)

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

3.52
fan motor shaft efficiency
η_o
fan air power (3.43) divided by the *motor output power* (3.47)

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

Note 1 to entry: Differs from *fan shaft power* (3.46) by the amount of transmission losses because of the transmission losses.

3.53
overall efficiency for a fan without variable speed drive
η_e
fan air power (3.43) divided by the *motor input power* (3.48) for the fan without variable speed drive

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

3.54
overall efficiency for a fan with variable speed drive
η_{ed}
fan air power (3.43) divided by the *drive control electrical input power* (3.49) for the fan with variable speed drive

$$\eta_{ed} = \frac{P_u}{P_{ed}}$$

3.55
motor efficiency
η_{mot}
motor output power (3.47) divided by the *motor input power* (3.48)

$$\eta_{mot} = \frac{P_o}{P_e}$$

Note 1 to entry: See Annex F.

3.56
variable speed drive efficiency
η_c
motor input power (3.48) divided by the *drive control electrical input power* (3.49)

$$\eta_c = \frac{P_e}{P_{ed}}$$

3.57
kinetic energy factor at section *x*
α_{Ax}
dimensionless coefficient equal to the time-averaged flux of kinetic energy through the considered cross section, A_x, divided by the kinetic energy corresponding to the mean air velocity through this cross section

$$\alpha_{Ax} = \frac{\iint_{Ax} \left(\rho \cdot v_{nx} \cdot v^2\right) \mathrm{d}A}{q_m \cdot v_x{}^2}$$

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

Note 1 to entry: Throughout this document, it shall be assumed, by convention, $\alpha_{Ax} = 1$, accepting some physical distortion of the results, for the sake of simplicity.

3.58
Reynolds number based on the exit diameter
Re_d
ratio of inertial to viscous forces

$$Re_d = 0{,}95 \cdot \varepsilon \cdot d \cdot \frac{\sqrt{2 \cdot \rho_{up} \cdot \Delta p}}{\mu_{up}}$$

4 Symbols, abbreviated terms and subscripts

4.1 Symbols and abbreviated terms

Symbol	Represented quantity	Definition reference	SI Unit
iCS	Common segment at the inlet side of the fan under test	8.1	—
iDS	Duct simulation at the inlet side of the fan under test	8.2	—
iLD	Long duct at the inlet side of the fan under test	8.5	—
iTC	Test chamber at the inlet side of the fan under test	9.3	—
iTS	Free test space at the inlet side of the fan under test	6.9	—
oCS	Common segment at the outlet side of the fan under test	8.3	—
oDS	Duct simulation at the outlet side of the fan under test	8.4	—
oLD	Long duct at the outlet side of the fan under test	8.5	—
oTC	Test chamber at the outlet side of the fan under test	9.4	—
oTS	Free test space at the outlet side of the fan under test	6.9	—
F, Fan	Fan under test	11	—
VSS	Variable supply system	10.1	—
iNZ	Inlet nozzle	A.5	—
dNZ	In duct nozzle	ISO 5167–1	—
mNZ	Multi nozzle	A.4	—
cOR	Orifice in chamber	A.6.4.5	—
dOR	In duct orifice	ISO 5167–1	—
iOR	Inlet orifice	A.6.4.4	—
oOR	Outlet orifice	A.6.4.3	—
b	Width of the rectangular section of a duct	—	M
c	Velocity of sound at a point	3.38	m/s
c_p	Specific heat capacity at constant pressure	3.13	J/kg/K
c_V	Specific heat capacity at constant volume	3.14	J/kg/K
C	Discharge coefficient	Annex A	—
d	Diameter, e.g. diameter of orifice or nozzle throat	Annex A	m
D	Diameter	—	m
D_h	Hydraulic diameter	3.8	m
f_M	Mach factor for correction of dynamic pressure	3.41	—
f_θ	Recovery factor for correction of temperature measurement	12.6.2	—
g	Acceleration due to gravity	12.8.1	m/s^2

Symbol	Represented quantity	Definition reference	SI Unit
h	Height of the rectangular section of a duct	12.2.2.2	M
h_{rel}	Relative humidity	12.9.1	—
H_h	Hydraulic mean depth	3.9	M
K_P	Compressibility coefficient for the calculation of fan air power P_u	15.1.8	—
L	Length of a conduit	—	M
Ma	Mach number	3.39	—
Ma_u	Peripheral Mach number	3.40	—
m	Area ratio of an orifice plate $(d/D)^2$	Annex A	—
n	Rotational frequency of the impeller	3.36	r/s
n_n	Polytropic exponent	Annex J	—
N	Impeller speed	E.2.4	r/min
p	Absolute pressure	3.27	Pa
p_e	Mean gauge pressure	3.28	Pa
p_f	Fan pressure	3.33	Pa
p_{fd}	Fan dynamic pressure	3.32	Pa
p_{fs}	Fan static pressure	3.34	Pa
P_a	Mechanical power supplied to the fan shaft	3.46	W
P_e	Motor input power	3.48	W
P_{ed}	Drive control electric input power	3.49	W
P_o	Motor output power	3.47	W
P_r	Mechanical power supplied to the impeller of the fan	3.45	W
P_u	Fan air power	3.43	W
$P_{\square}$	Perimeter	3.8 and 3.9	m
q_m	Mass flow rate	3.22	kg/s
q_{Vsg1}	Volume flow rate of the fan at stagnation conditions	3.24	m³/s
q_{V1}	Volume flow rate of the fan	3.23	m³/s
r_d	Pressure ratio for a flow meter $r_d = p_{do}/p_{up}$	Annex A	—
$r_{\Delta p}$	Pressure ratio for a flow meter $r\Delta p = \Delta p/p_{do}$	Annex A	—
R	Specific gas constant	3.16	J/kg/K
Re_d	Reynolds number with the internal diameter of an in-line flow meter	3.58	—
Re_D	Reynolds number	8.6	—
Re_u	peripheral Reynolds number	14.2.3	—
t	Time	Annex C	s
t_m	Motor output torque	E.4	Nm
T	Temperature	—	°C
v	Average velocity of air	3.25	m/s
u	Tip speed of the impeller	3.37	m/s
y_f	Fan work per unit mass	3.42	J/kg
y_{fs}	Fan static work per unit mass	15.1.9	J/kg
z	Altitude	—	m
Z_k and Z_p	Auxiliary quantities	15.1.8	—
α	Flow rate coefficient of an in-line flow meter	Annex A	—
α_{Ax}	Kinetic energy factor in the section x of area A_x	3.57	—

Symbol	Represented quantity	Definition reference	SI Unit
β	Ratio of the internal diameter of an orifice or nozzle to the upstream diameter of the duct d/D	Annex A	—
β'	Ratio of the internal diameter of an orifice or nozzle to the downstream diameter of the duct	Annex A	—
Δp	Differential pressure	Annex A	Pa
ε	Expansibility factor	Annex A	—
ξ	Loss coefficient	—	—
$(\xi_n - x)_x$	Loss coefficient between planes n and x calculated for section x	—	—
η	Efficiency	15.2	—
θ	Absolute temperature	3.10	K
θ_{ind}	Measured temperature indicated by the probe ($\theta < \theta_{ind} < \theta_{sg}$)	12.6.2	K
κ	Isentropic exponent	3.15	—
Λ	Coefficient of friction loss for a straight duct	—	—
μ	Dynamic viscosity	12.9.2	Pa·s
ρ	Density of air	3.18	kg/m³

4.2 Subscripts

1	Test fan inlet
2	Test fan outlet
3	Pressure measurement section in an inlet-side airway
4	Pressure measurement section in an outlet-side airway
5	Throat or downstream tapping for Δp for an inlet-side flow rate measurement
6	Upstream tapping for Δp and tapping for p_{up} for an outlet-side flow rate measurement
7	Upstream tapping for Δp and tapping for p_{up} for an inlet-side flow rate measurement
8	Throat or downstream tapping for Δp for an outlet-side flow rate measurement
a	Ambient atmosphere in the test space/regarding efficiency related to shaft
aux	auxiliary
b	barometer
C	converted
d	dynamic
do	downstream of a flow rate measurement device
dry	dry bulb
eq	equivalent
f	fan

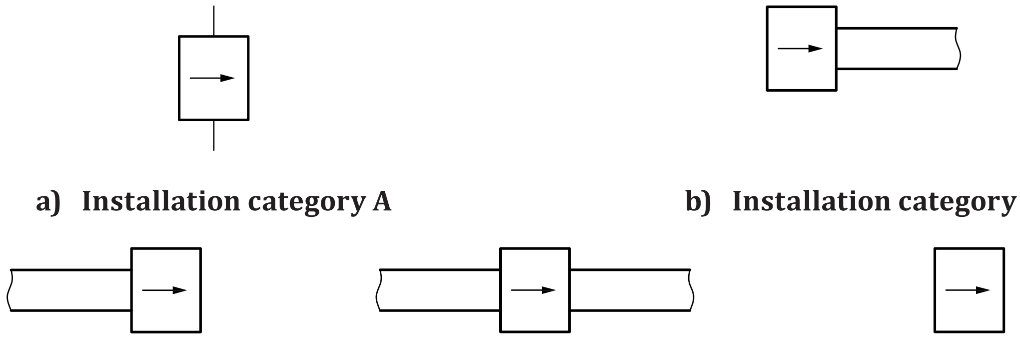
h	hydraulic
L	large
m	arithmetic mean value between inlet and outlet cross-sections of the fan under test
mot	motor
max	maximum
min	minimum
n	reference plane of the fan; n = 1 for inlet, n = 2 for outlet (see 15.1.2)
r	rotor/impeller
ref	reference
s	static conditions
S	small
sat	saturation conditions
sg	stagnation conditions
T	transition
Te	tested
tot	total
up	upstream of a flow rate measurement device
v	vapour
wet	wet bulb
x	section and the mean value, over time, averaged over the area of the airway cross-section

5 General

The upper limit of fan air 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 shall be within the normal atmospheric range.

In ISO 13349, five fan installation categories are defined; see Figure 1.



a) Installation category A

b) Installation category B

c) Installation category C

d) Installation category D

e) Installation category E

Key

A free inlet, free outlet

B free inlet, ducted outlet

C ducted inlet, free outlet

D ducted inlet, ducted outlet

E free inlet and free outlet without a partition

Figure 1 — Installation categories

In the above classification, the terms 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.

In this document, a number of test configurations are defined (see Clause 6) corresponding to the installation categories A, B, C and D defined in ISO 13349. The installation category E is tested according to ISO 13350 (jet fans) or other relevant standards (e.g. ISO 27327-2 for air curtains).

The fan to be tested shall be classified according to one of the categories A, B, C or D specified above. The test report shall state the category of installation for which the fan was tested. A fan adaptable to more than one installation category and test configuration will have more than one performance.

Furthermore the performance of the fan may be modified by the upstream air flow, e.g. if the velocity profile is distorted or if there is swirl.

The downstream flow generally cannot act on the flow through the impeller, but a duct at the outlet of the fan may modify its performance, and the losses in the downstream duct may be modified by the air flow at the fan outlet.

Methods of measurement and calculation for the flow rates, fan pressures and fan efficiencies are established in this document, taking into account all compressibility effects of the air. For fan pressure less than 2 000 Pa, the change of density between fan inlet and fan outlet can be neglected. Other compressibility effects can be neglected for reference velocity values not higher than 65 m/s (see Clause 13).

6 Test configurations

6.1 General

The laboratory shall test the fan as delivered; it is the responsibility of the organization supplying the fan to the laboratory to include all ancillaries are included for their intended use. The test report shall include all information to the measurement setup, i.e. the shape and position of the used ancillaries.

6.2 Category A configuration

In order to qualify for installation category A, the fan shall be tested without any inlet or outlet duct, but may be tested with the ancillaries supplied with the fan, i.e. protection grid, inlet bell, etc.

An inlet or outlet chamber is used in this case as defined in 9.4 and 9.5.

6.3 Category B configuration

An outlet duct, simulating ducted conditions, shall be used and the fan may be tested with the ancillaries supplied with the fan.

When the outlet pressure of the fan is measured in the outlet duct, a common segment at the fan outlet shall be used (see 8.4). See Annex G for recommendations for using this configuration for fans with significant swirl.

6.4 Category C configuration

An inlet duct, simulating ducted conditions, shall be used and the fan may be tested with the ancillaries supplied with the fan.

When the inlet pressure of the fan is measured in the inlet duct, a common segment at the fan inlet shall be used (see 8.2).

6.5 Category D configuration

An inlet duct and an outlet duct, simulating ducted conditions, shall be used and the fan may be tested with the ancillaries supplied with the fan.

When the outlet pressure of the fan is measured in the outlet duct, a common segment at the fan outlet shall be used (see 8.4). See Annex G for recommendations for using this configuration for fans with significant swirl.

When the inlet pressure of the fan is measured in the inlet duct, a common segment at the fan inlet shall be used (see 8.2).

6.6 Inlets and outlets

Fan inlet and outlet areas are defined in ISO 13349.

6.7 Fans with significant swirl

The performance of a fan for a defined installation category will be determined correctly with the standardized test setups of this document within a given uncertainty (see Clause 17).Fans with significant swirl (as for example, axial fans without outlet guide vanes) in installation categories B and D may have additional uncertainty. This additional uncertainty depends on fan design, operation point and test setup (see Annex G). The test report shall clearly state the geometry of duct used at the outlet of the fan.

6.8 Airways

All test airways shall be straight. Joints between airway sections shall be in good alignment and free from internal protrusions. Where provision is made for the insertion and manipulation of measuring instruments, special care shall be taken to minimize leakage and obstruction of the airway.

6.9 Test space

The assembly of the fan with its test airways shall 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 shall be taken to avoid the presence of any obstruction which might significantly modify the air flow at inlet or outlet of the fan under test or the test configuration. For details, see 9.4.1 and 9.5.1.

Other unobstructed space at the inlet and outlet of flow-measurement devices is specified with these devices. Related dimensions of the test chambers are specified in 9.4.1 and 9.5.1.

6.10 Leakage

Leakage of the test installation between the flow rate measurement plane and the fan under test shall be negligible compared to the maximum flow rate of the fan under test.

A leakage test shall be performed prior to initial use and periodically thereafter, with corrective action taken if necessary. See Annex C for recommended leakage test methods.

6.11 Test report

All references to fan performance stated to be in accordance with this document shall also state the test configuration to which they refer. In reporting a test, the test configuration defined in Clause 11 shall be stated.

7 Carrying out the test

7.1 Working fluid

The working fluid for tests with standardized airways shall be atmospheric air.

7.2 Rotational speed

For constant speed characteristics, the fan shall 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 14 and 15.3 shall be applied.

In the case of inherent speed characteristics, as defined in 16.4, the fan motor shall be operated at nominal or rated supply conditions of the motor or prime mover.

7.3 Steady operation

Before taking measurements for any point on the fan flow rating curve, the fan shall be run until steady operation is achieved. Where steady-state is not possible, a measurement may be taken and a note about the unsteadiness shall be made in the test record.

7.4 Ambient conditions

Readings of atmospheric pressure, dry bulb temperature and wet bulb temperature shall be taken within the test space during the series of observations required to determine the fan characteristic curves. If the ambient conditions are varying, sufficient readings shall 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 this document.

7.5 Pressure readings

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

7.6 Test for a specified duty

Tests for a specified duty shall comprise not less than three test points determining a short portion of the fan characteristic curve including the specified duty. Measured points shall be indicated in the plot.

7.7 Test for a fan characteristic curve

Tests for determining fan characteristic curves shall comprise a sufficient number of plotted 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.

7.8 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.

8 Airways for duct simulations

8.1 General

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

8.2 Common segments at fan inlet (iCS)

This comprises the section of the inlet side test airway adjacent to the fan and incorporates a set of wall tappings in accordance with 12.8 as shown in Figure 2.

A transition section may be used to accommodate a difference of area and/or shape within the limits indicated (see Figure 2):

$0{,}95 < A_3/A_1 < 2{,}25$ for circular A_1

$0{,}95 < A_3/A_1$ For rectangular A_1, there is no upper limit on A_3 or on the aspect ratio b/h (where $b > h$), but the included angle of expansion between the short sides shall not exceed 15° and the included angle of contraction between the long sides shall not exceed 30°.

$L_{T1}/D_{h3} = 1$ For rectangular A_1: $L_{T1}/D_{h3} \geq 1$ is possible (see above note to A_3/A_1).

If acoustic in duct measurements are carried out at the same time as air flow performance tests, the transition shall also comply with the requirements of ISO 5136 (cited in 10.4). For other test configurations, the relevant parts of ISO 13347 (all parts) shall be used.

The section adjacent to the fan inlet has the same cross-section as the fan inlet to which it is attached and its length, L_{S1}, is as given in Formula (1):

$$L_{S1} = D_{h1} \tag{1}$$

where

D_{h1} is the hydraulic diameter at the inlet of the fan.

If the air enters the common segment directly from the unobstructed free atmosphere or if the common segment is installed downstream of an inlet test chamber as a duct simulation, it shall be used with a bell mouth entry at its inlet end (see Figure 3).

If a throttling device or an auxiliary fan is installed upstream of the common segment, a straightener (see 10.3) shall be used upstream of the common segment to provide the measuring plane 3 with swirl free conditions for reliable pressure measurement.

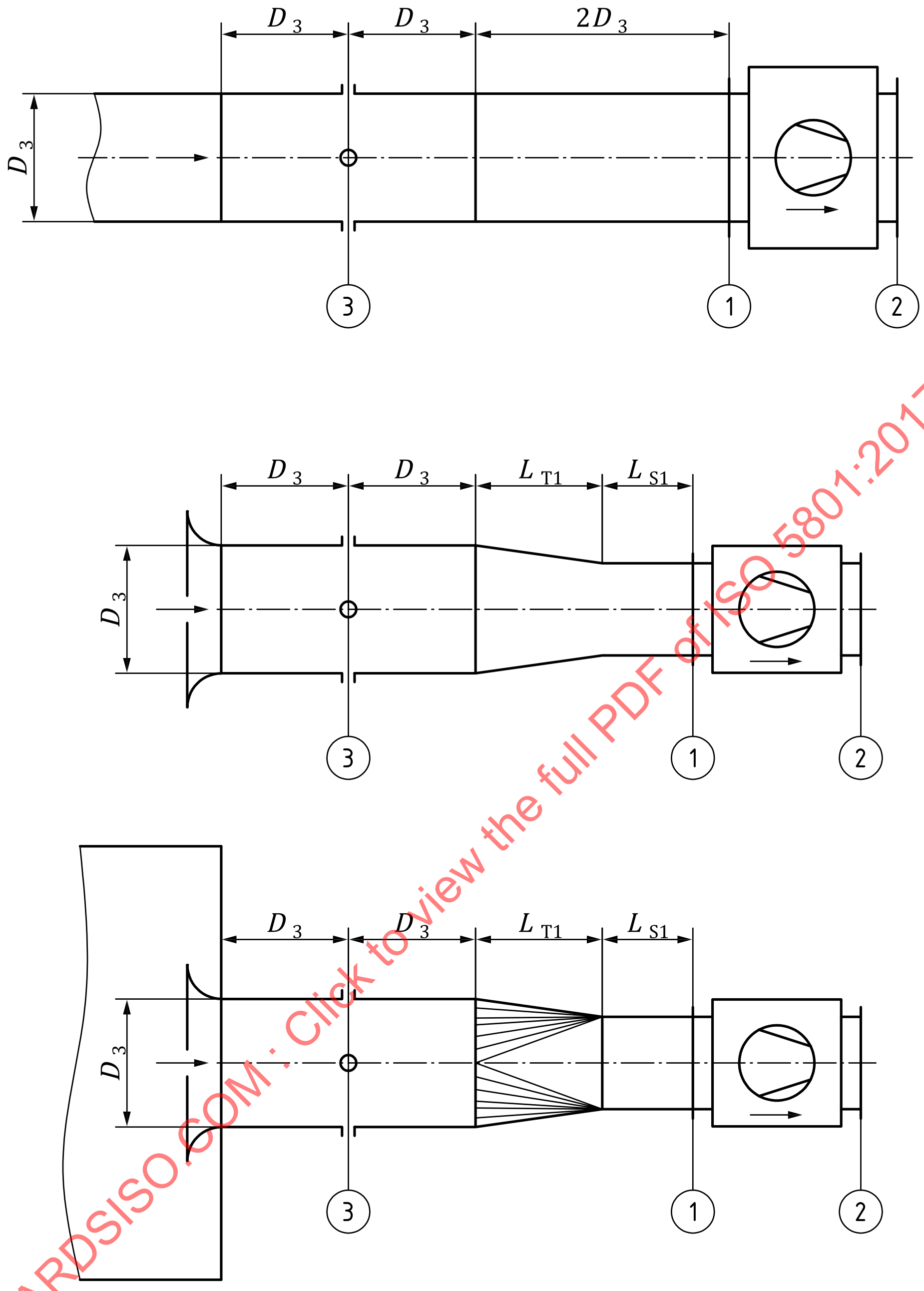


Figure 2 — Common segment at fan inlet

8.3 Inlet duct simulation (iDS)

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

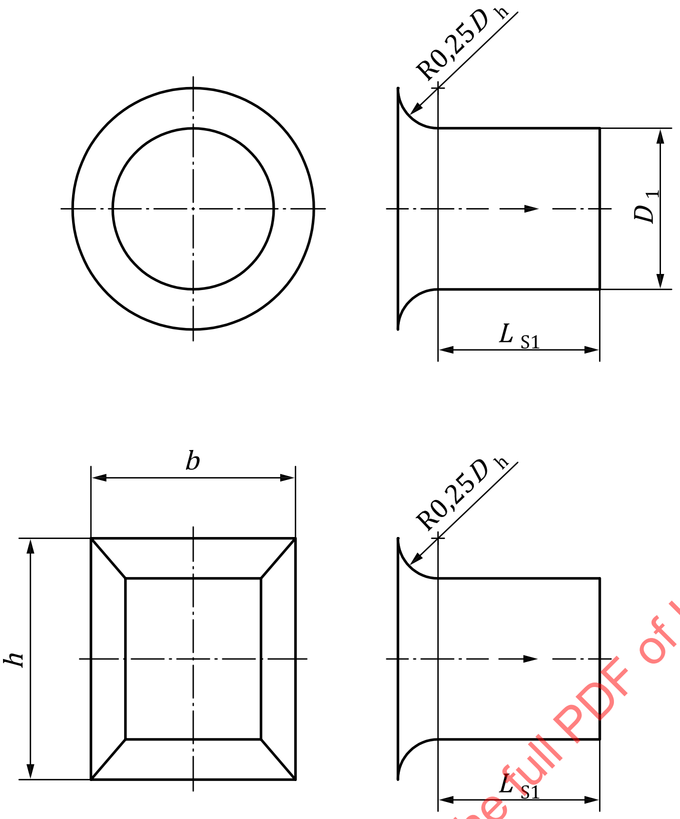


Figure 3 — Inlet duct simulation with bellmouth entry

The simulation section shall have the same cross-section as the fan inlet to which it is attached, with the length L_{S1}, given by Formula (2):

$$L_{S1} \geq D_{h1} \tag{2}$$

A bellmouth entry shall be fitted; geometry shown in Figure 3.

An inlet length equal to L_{S1} 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 10.3.3 with a bellmouth entry at its inlet end.

8.4 Common segment at fan outlet (oCS)

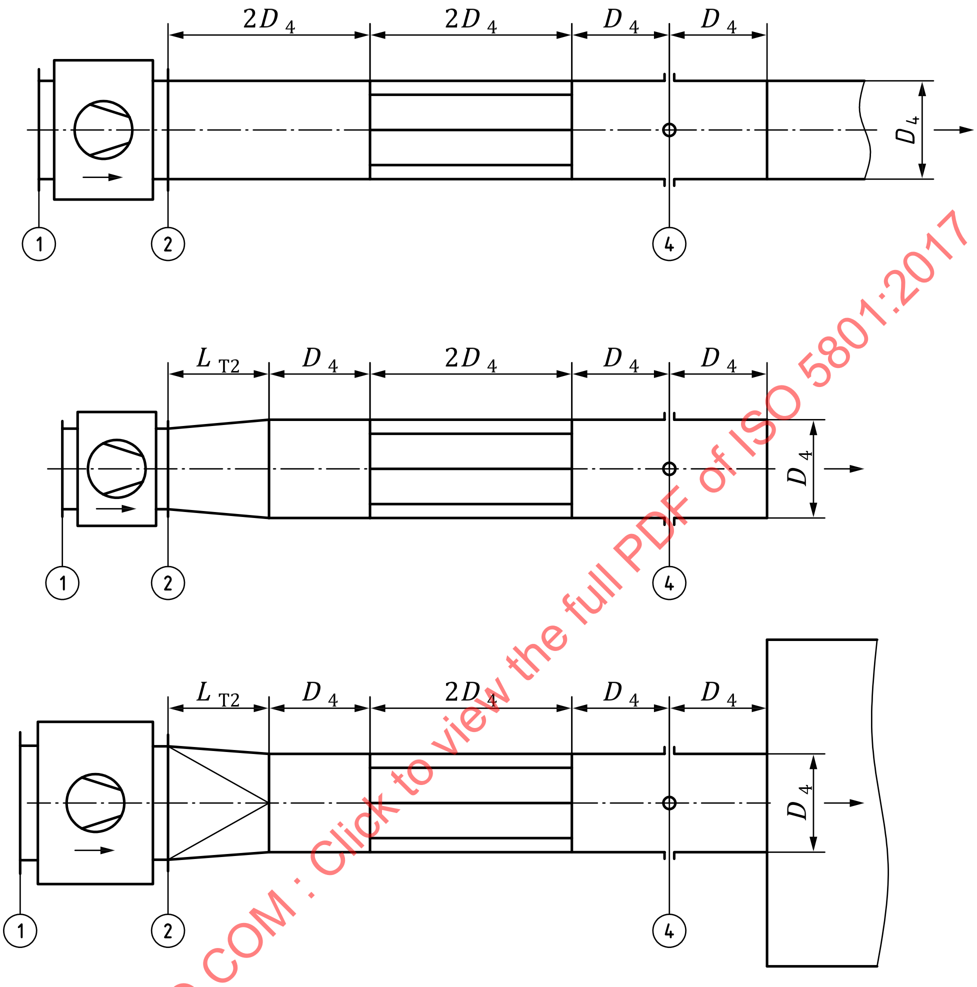
This comprises the section of an outlet side test airway adjacent to the fan. It incorporates a standardized flow straightener of star type in accordance with 10.3.3 in the central cylindrical section, together with a set of wall tappings in accordance with 12.8.

A transition section may be used to accommodate a difference of area and/or shape within the limits indicated (see Figure 4):

$0{,}95 < A_4/A_2 < 1{,}07$	for all A_2-shapes
$L_{T2}/D_4 = 1$	for all A_2–shapes except for rectangular A_2 with
	$b_2/h_2 > 4/3 : L_{T2}D_4 = 3/4 \cdot b_2/h_2$

If acoustic in duct measurements are carried out at the same time as air flow performance tests, the transition shall also comply with the requirements of ISO 5136 (cited in 10.4). For other test configurations, the relevant parts of ISO 13347 (all parts) shall be used.

The friction-loss coefficient of the transition section is that of a duct of diameter D_4 and length L_{T2}.



NOTE Alternatively, the pressure tapping (4) can be placed in the chamber as shown in Figure 10.

Figure 4 — Common segment at fan outlet [with flow straightener (star type shown)]

For centrifugal, cross-flow or vane-axial fans (not having significant swirl), a simplified outlet duct (8.5) may alternatively be fitted to obtain a type B or D configuration when discharging to atmosphere (oTS) or an outlet test chamber (oTC).

For large fans (outlet diameter 800 mm or larger), 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 simulation on the outlet side (8.5). Results obtained in this way may differ to some extent from those obtained using the common segment at fan outlet (8.4), especially if the fan produces a large swirl.

8.5 Outlet duct simulation (oDS)

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 (see Figure 5).

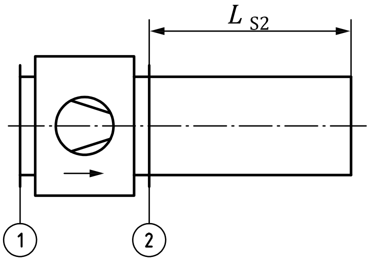


Figure 5 — Duct simulation at fan outlet

The outlet duct simulation shall be of the same cross-section as the fan outlet and the length shall be determined by the condition given in Formula (3):

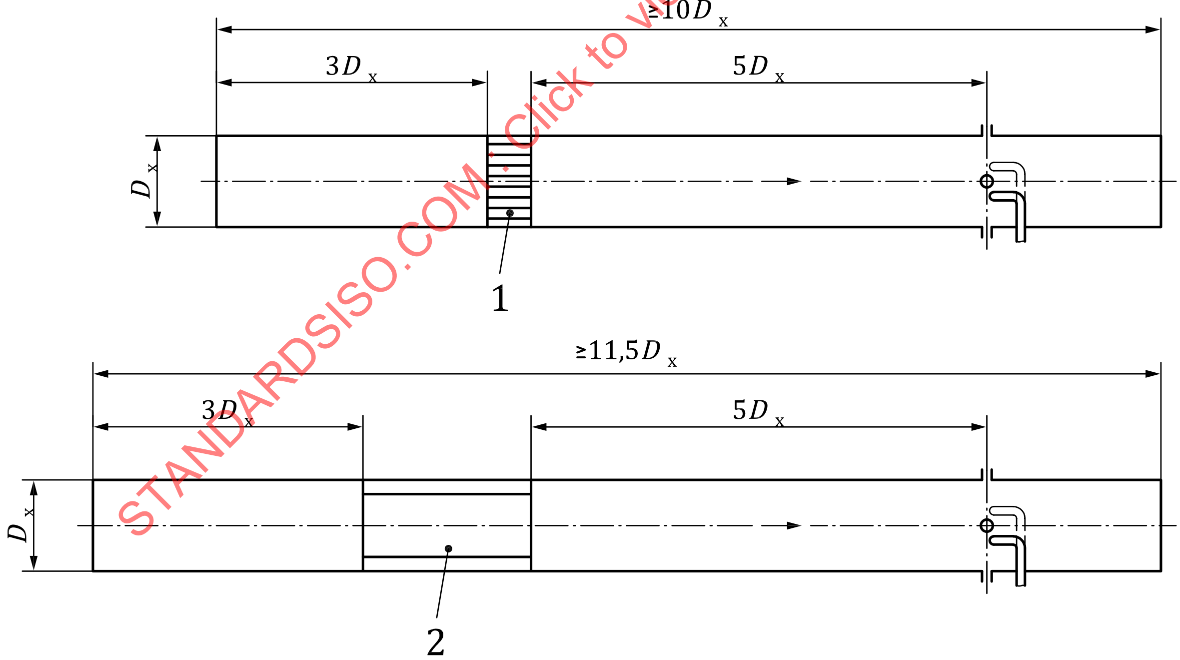
$$2 \leq L_{S2}/D_{h2} \leq 4 \tag{3}$$

The outlet duct simulation can be used with an outlet test chamber (oTC) in accordance with 8.4 or outlet test space (oTS).

The outlet static pressure of the fan under test is not measured in this duct but considered as equal to the static pressure in the outlet test space (oTS) or the outlet test chamber (oTC).

8.6 Long duct (LD)

A long duct shall be straight and have a uniform circular cross-section. Duct length between the straightener and the Pitot static tube shall be not less than $5D_x$.



Key

1 cell straightener

2 star straightener

Figure 6 — Long duct

8.7 Loss allowances for standardized airways

Conventional allowances given in this subclause shall be made for airway friction in tests with standardized airways.

Friction factors for duct losses may be based on either hydraulic diameter or hydraulic mean depth. Care should be taken to ensure the correct value of friction factor is used. Figure 7 shows values for smooth ducts.

The coefficient of friction loss can be determined from Formula (4):

$$\Lambda = 0{,}005 + 0{,}42 (Re_{D3})^{-0{,}3} \tag{4}$$

where Λ is the coefficient of friction loss for a length of one diameter of a straight duct.

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 inlet common segments described in 8.7.1, the outlet common segments described in 8.4 and the long duct described in 8.6 between the fan and the plane of pressure measurement. The same allowances shall be made when transition sections are incorporated.

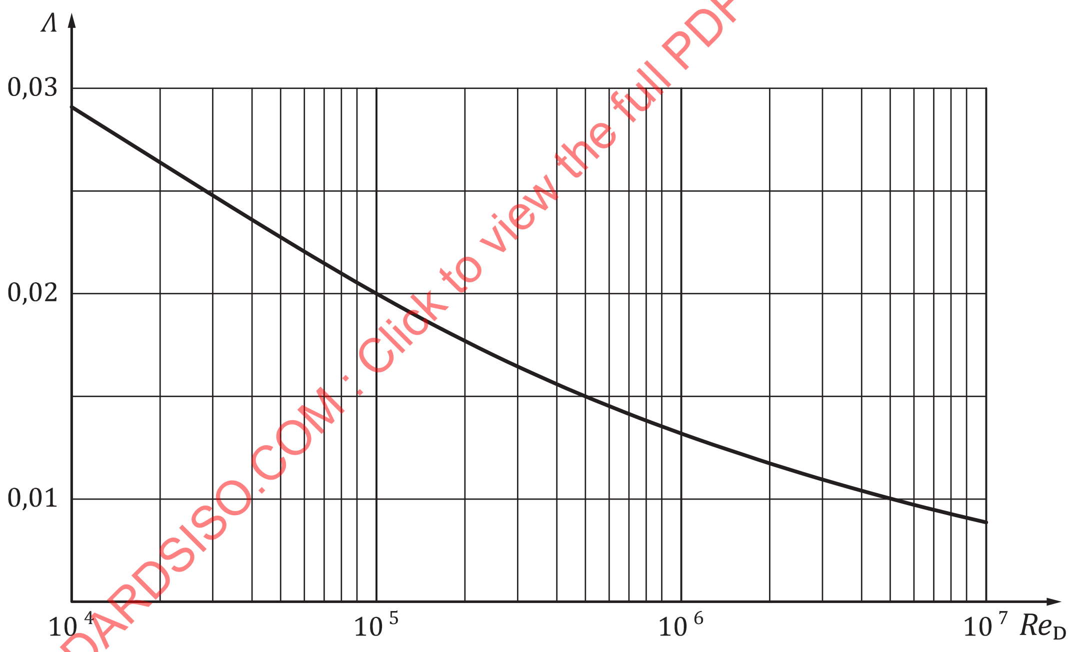


Figure 7 — Coefficient of loss for a length of one diameter of a straight duct

8.7.1 Loss allowances for an inlet common segment (iCS)

The conventional loss coefficient $(\xi_{3-1})_3$ between plane 3 and the inlet is given by Formula (5):

$$(\xi_{3-1})_3 = 0{,}015 + 1{,}26 (Re_{D3})^{-0{,}3} \tag{5}$$

where

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

in standard air.

The energy losses between planes 3 and 1 are given by Formula (6):

$$\Delta p_{3-1} = \left(\xi_{3-1}\right)_3 \frac{\rho 3 v_{\mathrm{m}3}^2}{2} f_{\mathrm{M}3} \tag{6}$$

8.7.2 Loss allowances for inlet duct simulation (iDS)

There are no losses allowed for this inlet duct.

8.7.3 Loss allowances for outlet common segments (oCS)

8.7.3.1 With a star straightener $(\mathrm{oCS})_{\mathrm{star}}$

The conventional loss coefficient of the star straightener including the external duct is given by Formula (7):

$$\xi_{\mathrm{star}} = 0{,}95 \cdot \left(Re_{D4}\right)^{-0{,}12} \tag{7}$$

with

$$Re_{D4} = \frac{4 \cdot q_m}{\pi \cdot D_4 \cdot \mu_4}$$

The conventional loss coefficient $(\xi_{2-4})_4$ between the fan outlet 2 and the measuring plane 4 is given by Formula (8):

$$\left(\xi_{2-4}\right)_{4\cdot\mathrm{star}} = \Lambda \cdot \frac{L_{2-4}}{D_{\mathrm{h}4}} + \xi_{\mathrm{star}} \tag{8}$$

where

$L_{2\text{-}4}$ is the length of the duct between the fan outlet 2 and the measurement section 4;

Λ is the coefficient of friction loss for a length of one diameter of a straight duct given by $\Lambda = 0{,}005 + 0{,}42 \cdot \left(Re_{D4}\right)^{-0{,}3}$ and is plotted against the Reynolds number in Figure 7.

8.7.3.2 With a cell straightener $(\mathrm{oCS})_{\mathrm{cell}}$

The coefficient of friction loss, Λ, for a length of one diameter of a straight duct is given in 8.7.3.1.

The ratio of equivalent length L_{eq} of a cell-straightener to hydraulic diameter D_{h} of the duct is given by Formula (9):

$$\frac{L_{\mathrm{eq}}}{D_{\mathrm{h}}} = \frac{15{,}04}{\left[1 - 26{,}65 \cdot \frac{e}{D_{\mathrm{h}}} + 184{,}6 \cdot \left(\frac{e}{D_{\mathrm{h}}}\right)^2\right]^{1{,}83}} \tag{9}$$

where

e is the vane thickness.

The conventional loss coefficient $\left(\xi_{2-4}\right)_{4.\mathrm{cell}}$ is given by Formula (10):

$$\left(\xi_{2-4}\right)_{4.\mathrm{cell}} = \Lambda \cdot \left[\frac{L_{2-4}}{D_{h4}} + \frac{L_{\mathrm{eq}}}{D_{\mathrm{h4}}}\right] \tag{10}$$

where

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

8.7.4 Loss allowances for duct simulation at outlet (oDS)

There are no losses allowed for this outlet duct.

8.7.5 Loss allowances for long duct (LD)

8.7.5.1 With a star straightener $(\mathrm{LD})_{\mathrm{star}}$

The calculations in 8.7.3.1 shall apply.

8.7.5.2 With a cell straightener $(\mathrm{LD})_{\mathrm{cell}}$

The calculations in 8.7.3.2 shall apply.

9 Standardized test chambers

9.1 General

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.

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

9.2 Pressure tappings

The wall tappings in the measuring planes shall be in accordance with the requirements of 12.8 and be equally spaced around a cylindrical chamber or on each side of a rectangular chamber.

9.3 Flow-settling means

9.3.1 General

Flow-settling means shall be installed in chambers to provide the required flow patterns when necessary.

The effectiveness of the airflow settling means in all chambers shall be verified by tests as described in 9.3.2, 9.3.3, and 9.3.4.

Some validation tests require that the flow and the pressure be determined prior to the settling means having proved their effectiveness. It can be assumed that the tests taken in this condition (with the non-verified settling means) are sufficiently accurate to be used to establish acceptance criteria.

Once the flow settling means have demonstrated that all applicable acceptance criteria have been met, the chamber may be used for all future testing within the limits defined by the acceptance criteria.

9.3.2 Piezometer ring check

Individual pressure readings for each pressure tap of the piezometer ring are to be measured. When the mean of these readings is less than or equal to 1 000 Pa (4 inch water gauge), all of the individual readings shall be within 5 % of the mean. When the mean of these readings is greater than 1 000 Pa (4 inch water gauge) all of the individual readings shall be within 2 % of the mean (see also 12.8.10).

9.3.3 Blow through verification test

This test evaluates the ability of the airflow settling means to provide a substantially uniform airflow ahead of a measurement plane. For this test, equally spaced measurement points are located in a plane $0{,}1D_{\mathrm{h}}$ downstream of the settling means. The number of measurement points shall be in accordance with ISO 5802.

For tests of settling means upstream of the nozzle wall, the auxiliary fan should be set at its maximum flow rate, the entire nozzle shall be open, the inlet of the chamber shall be open and the inlet area shall be equal to the largest area allowed by the chamber cross sectional area.

For tests of settling means upstream of the test fan, the auxiliary fan shall be set at its maximum flow rate, half of the nozzles shall be open, the outlet of the chamber shall be open and the outlet area shall be equal to the largest area allowed by the chamber cross sectional area.

The flow velocities shall be measured and the average determined. If the maximum velocity is less than 2 m/s (400 ft/min) or if the maximum velocity value does not exceed 125 % of the average, the settling screens are acceptable.

9.3.4 Outlet chamber reverse flow verification test

One purpose of the settling means is to absorb the kinetic energy of an 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. If the settling means are too restrictive, excessive backflow will result.

A series of tests shall be run to verify that reverse flow is not excessive at fan static pressure measurement plane. Each test shall be run with a varying opening in the chamber entrance starting from 11 % of the chamber area and proceeding to lower percentage openings. Each test shall be run with all of the nozzles open and the auxiliary at its maximum flow rate. At each test, it shall be verified that the static pressure downstream of the settling means is less than the fan static pressure measurement plane. The series of tests may be stopped at the first set of conditions that verify the above requirement.

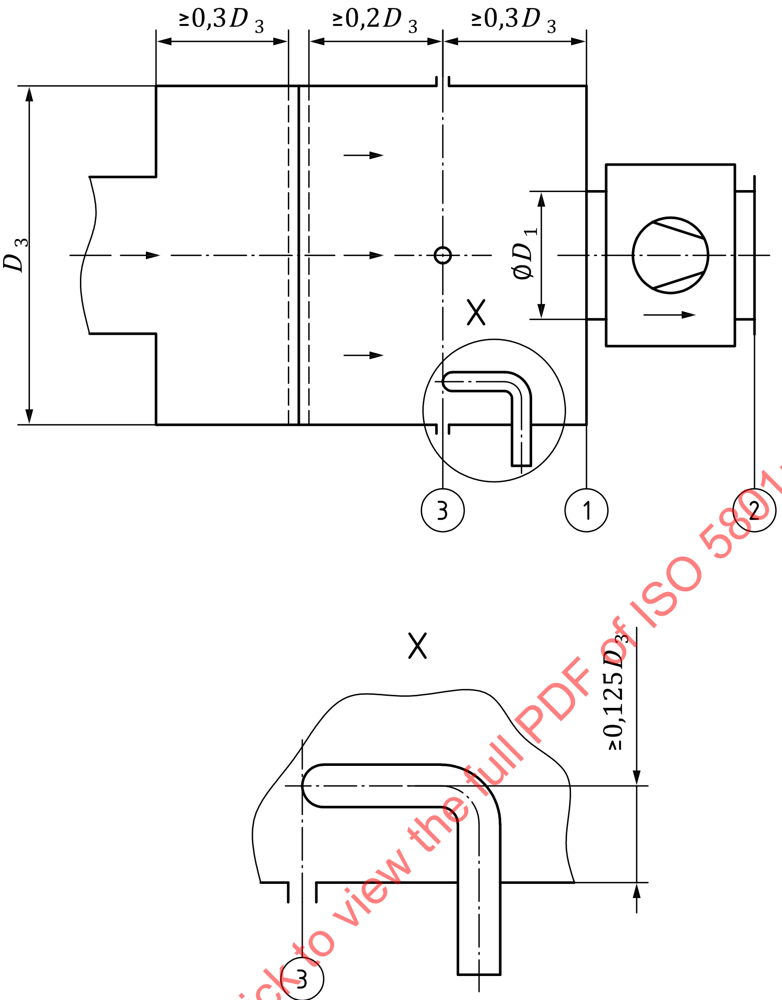
9.4 Standardized inlet test chambers (iTC)

9.4.1 Test chambers

The test chamber cross-section may be circular with inside diameter D_3 or rectangular $b_3 \cdot h_3$ with $2/3 < b_3/h_3 < 3/2$.

The dimension D_3 in the following figures corresponds to the hydraulic diameter of the cross-section given in Formula (11):

$$D_3 = D_{\mathrm{h}3} \tag{11}$$



NOTE Wall tappings and Pitot tube can be used alternatively or together.

Figure 8 — Inlet side test chamber

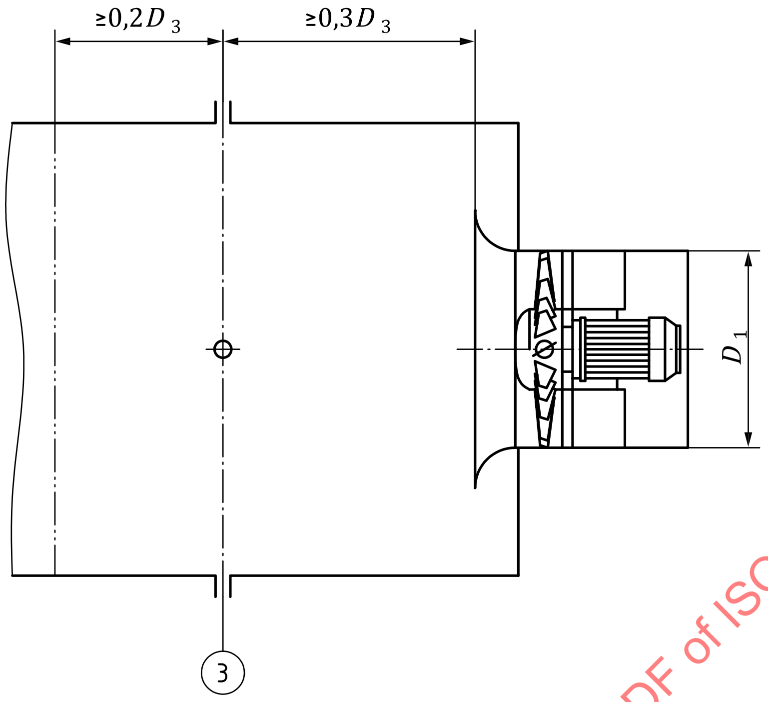
The length of the chamber shall be sufficient to accommodate any fan to be tested without infringing on the minimum distance shown in Figure 8 and Figure 9.

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 9.

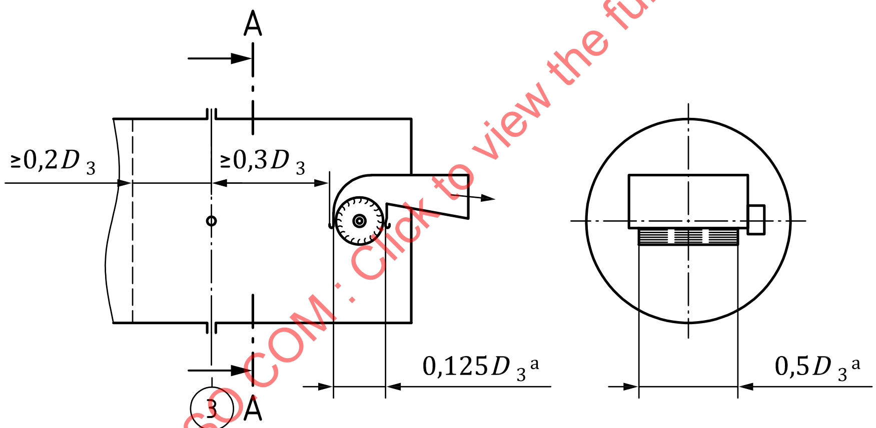
The pressure-measuring plane 3 is (see Figure 8 and Figure 9)

— at least $0,3 \cdot D_3$ upstream of the first downstream distortion, and

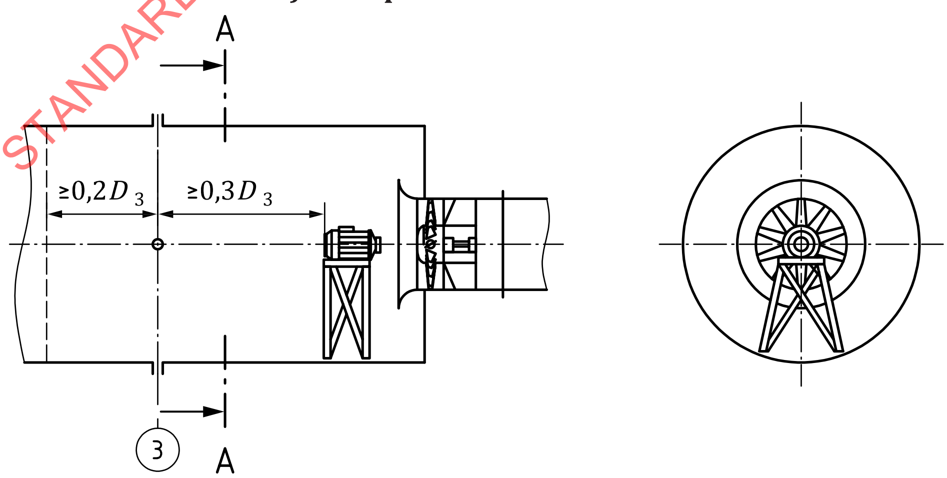
— at least $0,2 \cdot D_3$ downstream of the flow-settling means.



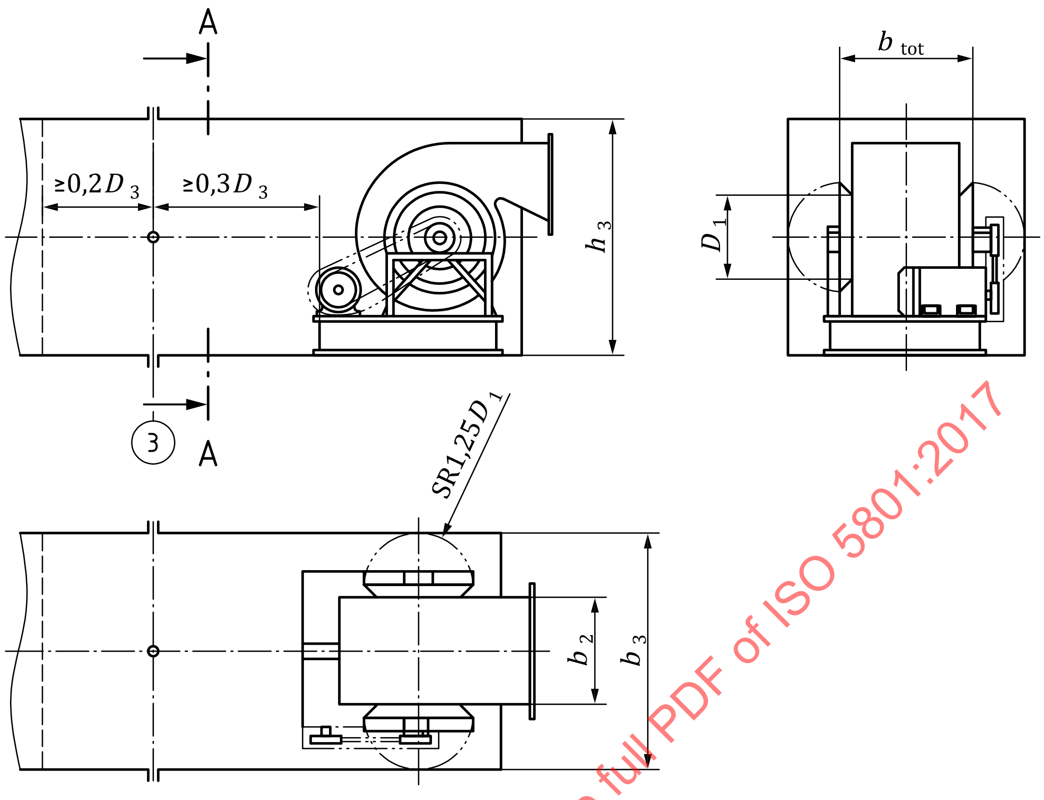
a) Example of an axial fan



b) Example of a cross-flow fan



c) Example of a fan with motor on the inlet side



d) Example of a double-inlet centrifugal fan (SR indicates spherical radius)

Key

a Inlet (maximum dimensions).

NOTE The fan illustrated has the maximum permissible dimensions.

Figure 9 — Examples for fan installations at inlet side test chambers

9.4.2 Fan under test

Inlet chambers shall have a cross-sectional area greater than five times the fan inlet area, as given in Formula (12):

$$A_3 \geq 5 \cdot A_1 \tag{12}$$

If there is more than one inlet, they shall be so located that the flow remains as symmetrical about the chamber axis as possible.

In a chamber of rectangular cross-section, any side shall be greater than twice the inlet diameter, as given in Formula (13).

$$b_3 \geq 2D_1 \tag{13}$$

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 rectangular cross-section, where the chamber width, b_3, is the sum of the fan width, b_{tot}, and an open space surrounding the two intake openings corresponding to a hemisphere of radius equal to 1,25·D_1 as shown in Figure 9 d).

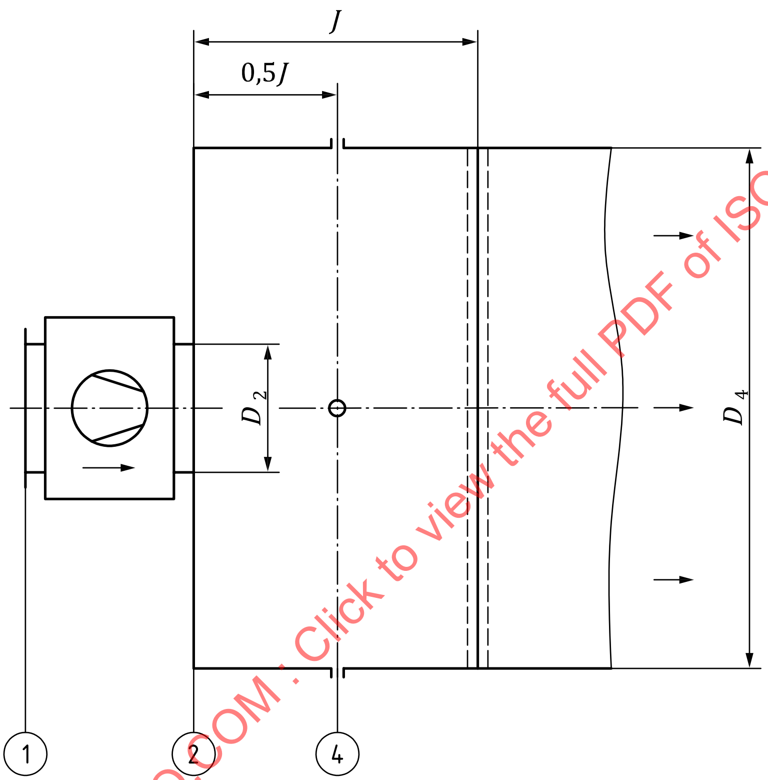
9.5 Standardized outlet test chambers (oTC)

9.5.1 Test chambers

The test chamber cross-section may be circular with inside diameter D_4, rectangular $b_4 \cdot h_4$ with $2/3 < b_4/h_4 < 3/2$.

The dimensions D_2 and D_4 in Figure 10 correspond to the hydraulic diameter of the cross-section given in Formula (14):

$$D_4 = D_{h4} \tag{14}$$



NOTE 1 The distance J is equal to at least the diameter of the outlet duct for fans with axis of rotation perpendicular to the discharge flow ($J \geq D_{h2}$), and to at least twice the diameter of the outlet duct for fans with axis of rotation parallel to the discharge flow ($J \geq 2 \cdot D_{h2}$).

NOTE 2 Pressure tapping (4) can be placed in the chamber as shown in Figure 10 for fans with uniform flow without swirl.

Figure 10 — Outlet side test chambers

9.5.2 Fan under test

The size of the outlet chamber is very large relative to the fan size (equivalent to a large open space).

An outlet test chamber (see Figure 10) 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 perpendicular to the discharge flow, as given in Formula (15):

$$A_4 \geq 9 \cdot A_2 \tag{15}$$

The cross-sectional area of the outlet test chamber for fans with an axis of rotation parallel to the discharge flow shall be at least 16 times the area of the fan outlet or outlet duct, as given in Formula (16):

$$A_4 \geq 16 \cdot A_2 \tag{16}$$

10 Various component parts for a laboratory setup

10.1 General

A laboratory setup is a combination of test chambers, ducts, flow rate or pressure measuring units, straighteners, transition parts and a variable supply system.

10.2 Variable supply system

10.2.1 General

The variable supply system is installed preferably at the outlet of the test configuration or separated from the fan under test and from the flow rate or pressure measuring units by means of a test chamber or a duct of at least $2D$. It is used to vary the point of operation in the laboratory setup and may include a throttling device, an auxiliary booster fan or a combination of both.

10.2.2 Throttling device

Throttling devices may be used to control the point of operation of the fan under test. On the downstream side of the throttling device in a duct, a straightener shall be used to prevent flow disturbances at sections of flow or pressure measurements or at the inlet of the fan under test.

10.2.3 Auxiliary fan

Auxiliary fans may be used to control the point of operation of the fan under test. 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 test. A cell straightener shall be interposed between the auxiliary fan and any test airway to which it is connected to prevent flow disturbances at sections of flow or pressure measurements or at the inlet or the outlet of the fan under test.

10.3 Straightener

10.3.1 General

In the presence of swirl, simple measurements of pressure or volume flow are impossible. Therefore, a straightener must be included when tests are to be taken in a duct on the outlet side of the fan. Even in cases of axial fans with guide vanes, a straightener must always be used upstream of a static or dynamic pressure measuring plane located within a test duct at the fan outlet.

10.3.2 Cell straightener

The cell straightener is used to reduce swirl. It does not improve asymmetric velocity distributions.

The cell straightener 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 $0{,}005D$ (see Figure 11).

$w = 0{,}075D$

$L = 0{,}45D$

$e \leq 0{,}005D$

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

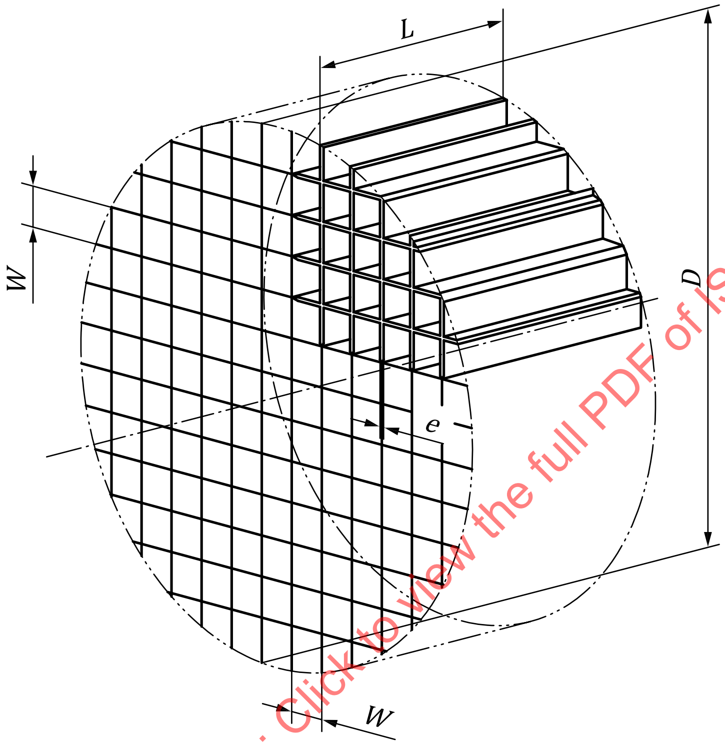


Figure 11 — Cell type flow straightener

10.3.3 Star straightener

The star straightener is designed to eliminate swirl but is of little use in equalization of asymmetric velocity distributions. It allows, unlike the cell type, the static pressure to equalize radially as the air flows through it.

The star straightener, as shown in Figure 12, is constructed of eight radial blades of length $2D$ (with a ±1 % tolerance) and of thickness not greater than $0{,}007D$ for pressure loss considerations. The blades will be arranged to be equidistant on the circumference with the angular deviation being no greater than 5° between adjacent plates.

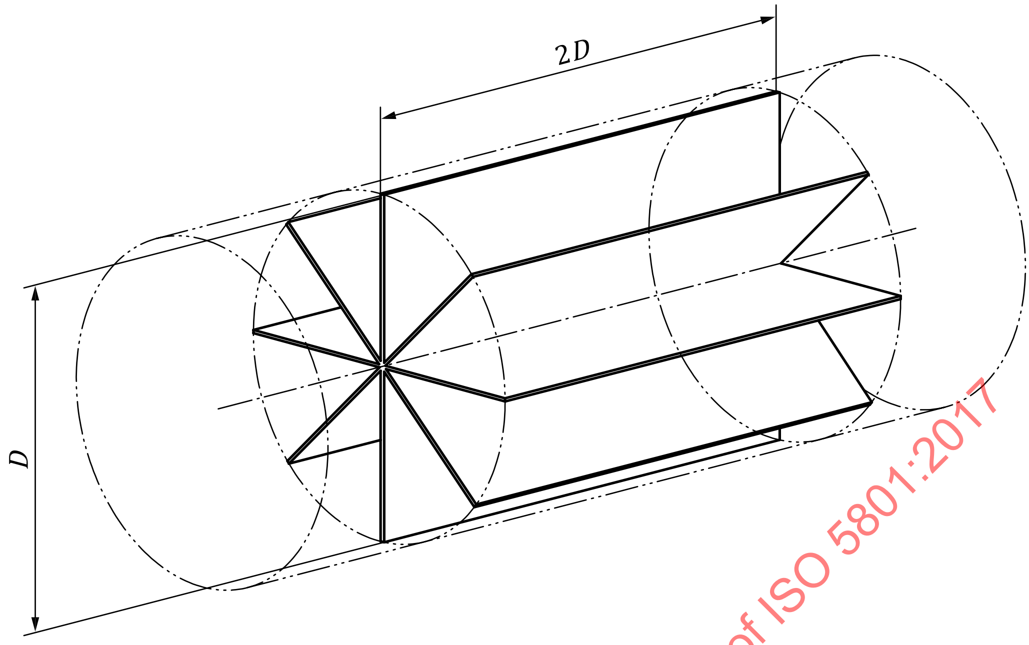


Figure 12 — Star type flow straightener

10.4 Transition parts

10.4.1 General

In 8.2 and 8.4, transition parts are defined for the connections between the fan under test and the common segments. Further transition parts will be needed to connect setup-components of different cross-sectional shape or size, which can be used in both flow directions and shall be in accordance with the condition given in Formula (17):

$$L \geq 4 \cdot \left[\sqrt{\frac{4 \cdot A_L}{\pi}} - \sqrt{\frac{4 \cdot A_S}{\pi}} \right] \tag{17}$$

where

A_L is the larger area of the transition part;

A_S is the smaller area of the transition part.

If acoustic in duct measurements are carried out at the same time as air flow performance tests, the transition shall also comply with the requirements of ISO 5136 (cited in 10.3). For other test configurations, the relevant parts of ISO 13347 (all parts) shall be used.

10.4.2 Rectangular/circular transition

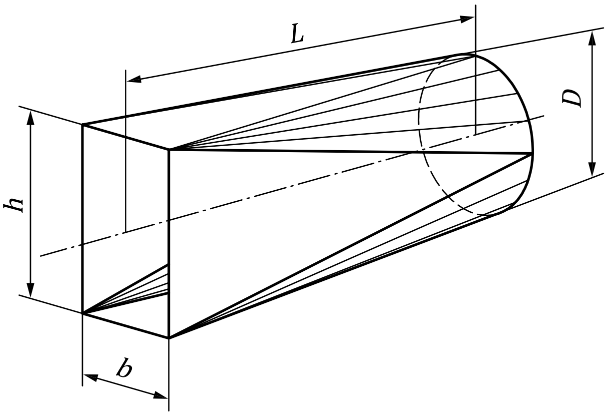


Figure 13 — Rectangular/circular transition

10.4.3 Circular/circular transition

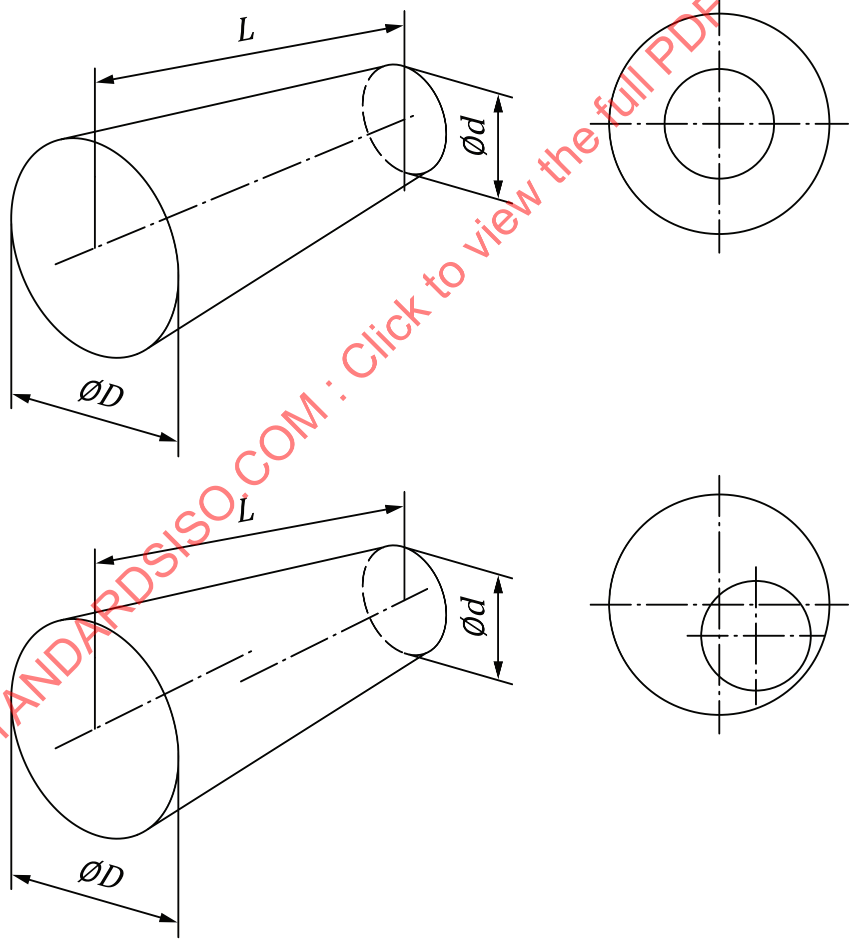
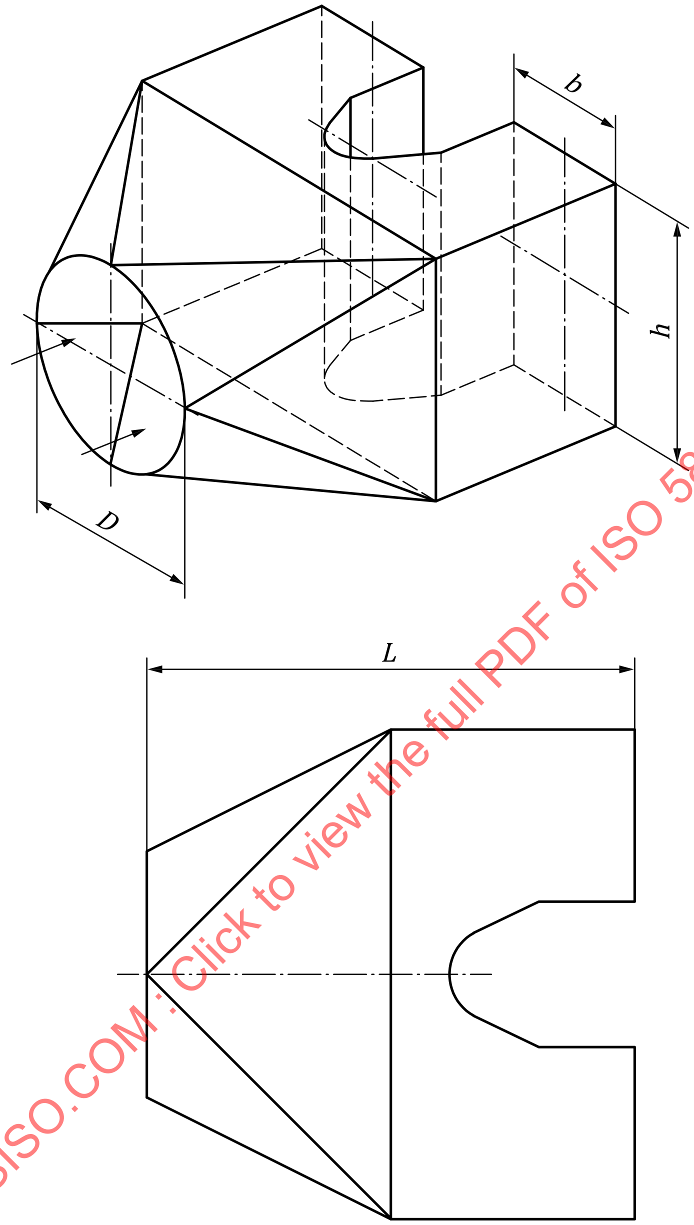


Figure 14 — Circular/circular transition

10.4.4 Connection for double-inlet fans



NOTE A_L is the total area of $2b{\cdot}h$, $A_L = 2b{\cdot}h$. Outlet cross-section of $2b{\cdot}h$ is be smaller or equal to circular inlet cross-section.

Figure 15 — Connection for double inlet fans

11 Standard test configurations

11.1 Units

Following defined units are to combine for setup configurations of type A, B, C and D. Restrictions of their use are indicated. Some examples are shown in Annex K.

Fan under test: **Fan**

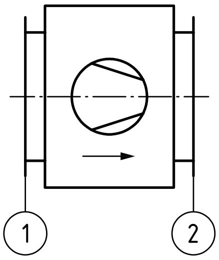


Figure 16 — Fan under test

Variable supply system (including auxiliary fan and/or throttling device: vss (10.2.2 and 10.2.3)

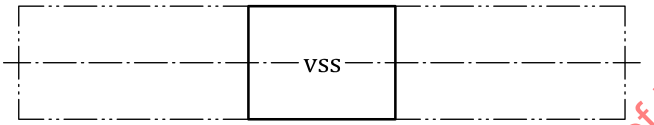


Figure 17 — Variable supply system

A flow straightener of cell type and a duct of 2*D*-length shall be interposed between an auxiliary fan and any test airway to which it is connected. On the downstream side of the throttling device in a duct a straightener and a duct of 2*D* length shall be used.

Flow rate measurement: ***qV*** **(12.5)**

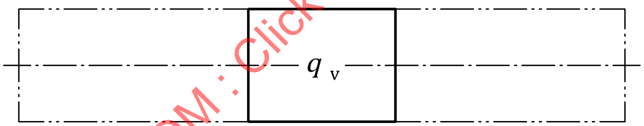


Figure 18 — Flow rate measurement

Clearances at the inlet and the outlet of flow rate measurement devices are defined with them in Annex A.

Inlet and outlet text chamber:

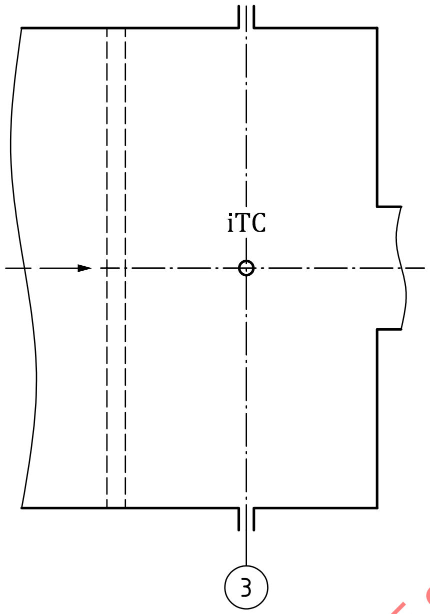


Figure 19 — Inlet test chamber

iTC and oTC (9.3 and 9.4)

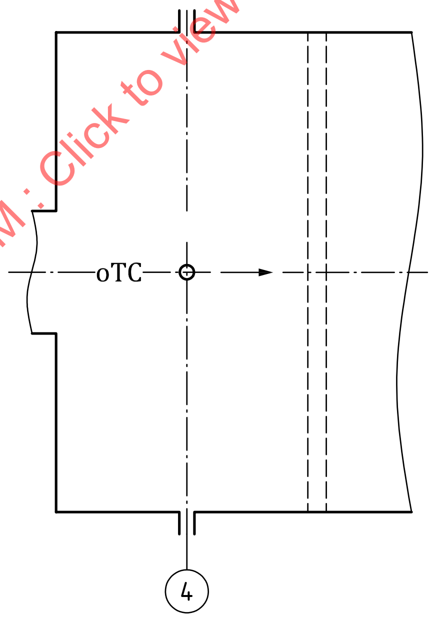


Figure 20 — Outlet test chamber

iTC and oTC may contain a flow rate measurement unit (see Annex A).

Inlet (iCS) and outlet common segment (oCS) (8.2 and 8.4):

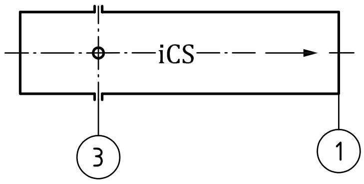


Figure 21 — Inlet common segment

Inlet (iCS) and outlet common segment (oCS) (8.2 and 8.4):

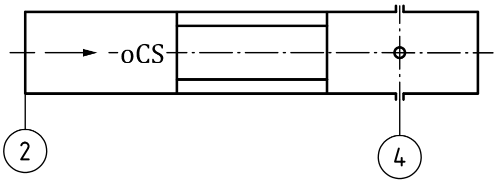


Figure 22 — Outlet common segment

Long duct: **LD (8.5)**

Figure 23 — Long duct

LD is used at the inlet or the outlet side of the fan under test. LD may contain Pitot-travers, which may be used simultaneously for flow rate and for p_3 or p_4 measurements.

Inlet (iDS) and outlet duct simulation (oDS) (8.3 and 8.5):

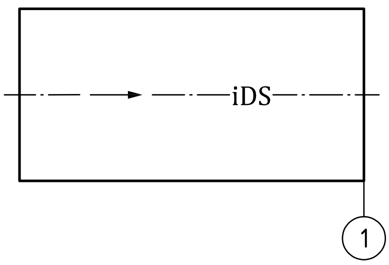


Figure 24 — Inlet duct simulation

Inlet (iDS) and outlet duct simulation (oDS) (8.3 and 8.5):

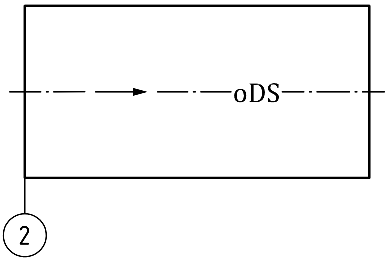


Figure 25 — Outlet duct simulation

iDS usually intakes air directly from iTS but it may also be combined with iTC. iDS is between 1 $D1$ and 3 $D1$ long; no transitions are allowed.

oDS usually discharges air directly to oTS but it may also be combined with oTC in accordance with 8.4. oDS is between 2 $D2$ and 4 $D2$ long; no transitions are allowed.

Inlet and outlet test space: iTS and oTS (6.8)

Inlet side pressure is measured at ❸: iTS, iTC, iCS or LD

Table 1 — Test configurations A and B

iTS	— Free and unobstructed inlet — Only a bellmouth inlet is allowed — ❸ is the static pressure measured in free test space where the air velocity is negligible (atmospheric pressure)
iTC	

Table 2 — Test configurations C and D

iDS	— $1D \leq L_{iDS} \leq 3D$; $\xi_{iDS} = 0$ — ❸ is the static pressure measured in free test space where the air velocity is negligible (atmospheric pressure) NO transition or other duct allowed
iCS	— iDS in or on iTC is allowed iCS– $L_{iCS} = 4D$; $\xi_{iCS} = 0$ — Transition and other duct is allowed — iCS in or on iTC is allowed LD
LD	— $L_{LD} = 11{,}5D$; $\xi_{LD} \neq 0$ — Transition and other duct is allowed — LD in or on iTC is allowed

NO other pressure measuring unit at the inlet of the fan under test is allowed. Temperature measurement is recommended in iTS or iTC.

Outlet side pressure is measured at “4”: oTS, oTC, oCS or LD

Table 3 — Test configurations A and C

oTS	— Free and unobstructed outlet — NO duct is allowed — ❹ is the static pressure measured in free test space where the air velocity is negligible (atmospheric pressure)
oTC	

Table 4 — Test configurations B and D

oDS	— $2D \leq L_{oDS} \leq 4D$; $\xi_{oDS} = 0$ — ❹ is the static pressure measured in free test space where the air velocity is negligible (atmospheric pressure) — NO transition or other duct allowed — oDS in or on oTC is allowed oCS
oCS	— $L_{oCS} = 6D$; $\xi_{oCS} \neq 0$ — Transition and other duct allowed — oCS in or on oTC is allowed LD
LD	— $L_{LD} = 11{,}5D$; $\xi_{LD} \neq 0$ — Transition and other duct is allowed — LD in or on oTC is allowed

NO other pressure measuring unit at the outlet of the fan under test is allowed.

11.2 Measuring flow rate

Clearances and straight ducts at the inlet and the outlet of flow rate measuring devices (see Table 5) are defined with these units in Annex A or in other International Standards.

Table 5 — Flow rate measuring devices

Flow rate measuring device	Definitions
inlet nozzle (“iNZ”)	A.5
inlet orifice (“iOR”)	A.6.4.4
PITOT traversing	ISO 3966
multi nozzle in chamber (“mNZ”)	A.4
orifice in chamber (“cOR”)	A.6.4.5
in duct nozzle (“dNZ”)	ISO 5167–1
in duct orifice (“dOR”)	ISO 5167–1
outlet orifice (“oOR”)	A.6.4.3

In this document, the pressure tappings are also indicated with following numbers:

— for an inlet-side flow rate measurement:

  — 7 Upstream tapping for the pressure difference and tapping for p_{up} (Index up);

  — 5 Throat or downstream tapping for the pressure difference (Index do);

— for an outlet-side flow rate measurement:

  — 6 Upstream tapping for the pressure difference and tapping for p_{up} (Index up);

  — 8 Throat or downstream tapping for the pressure difference (Index do).

11.3 Standard test configurations A

Table 6 — Test configurations A

	inlet side				outlet side		
	vss and/or q_V	p	iDS	fan	oDS	p	vss and/or q_V
A-01		iTS		x		oTC	x
A-02	x	iTC		x		oTS	
A-03	x	iTC		x		oTC	
A-04	x	iTC		x		oTC	x
A-05		iTC		x		oTC	x
	x Component is installed.						

The conditions at the inlet and the outlet of the fan are calculated as shown in Clause 15 according to the components used.

Examples for test configurations A are shown in Figure 26, Figure 27 and Figure 28.

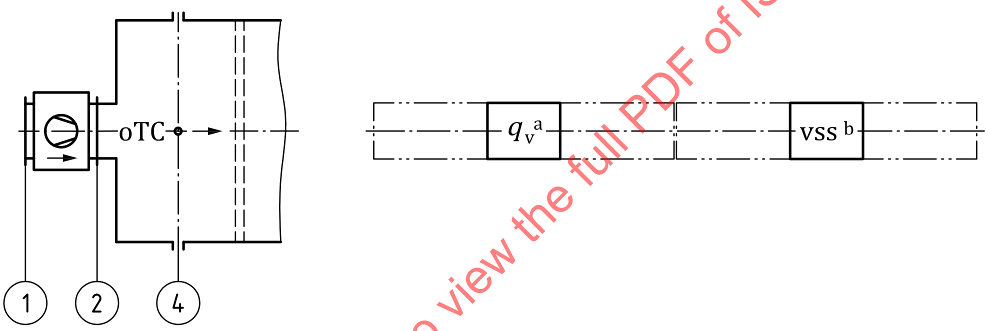


Figure 26 — Test configuration A-01

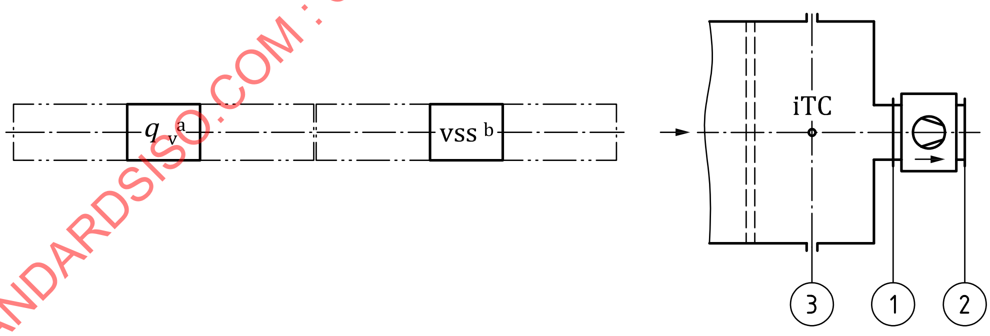


Figure 27 — Test configuration A-02

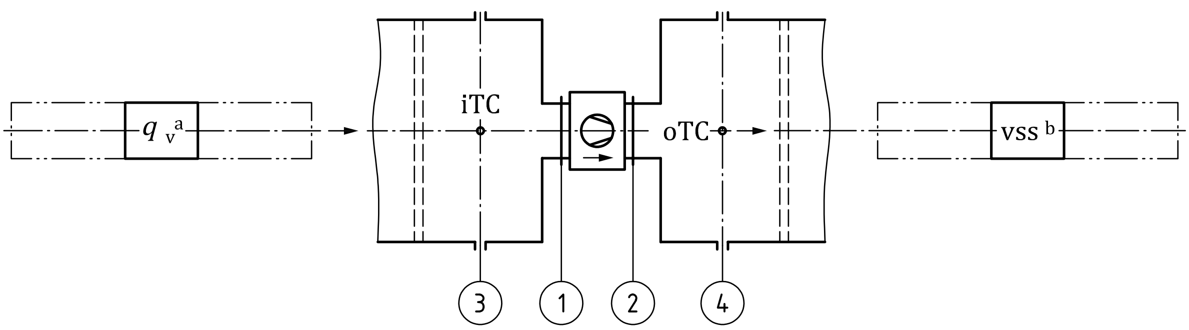


Figure 28 — Test configuration A-04

11.4 Standard test configurations B

Table 7 — Test configurations B

	inlet side				outlet side			
	vss and/or q_V	*p*	iDS	fan	oDS	*p*	vss and/or q_V	
B-01		iTS		x		oCS, LD	x	
B-02		iTS		x	x	oTC	x	
B-03	x	iTC		x		oCS, LD		
B-04	x	iTC		x		oCS, LD	x	
B-05		iTC		x		oCS, LD	x	
B-06	x	iTC		x	x	oTS		
B-07	x	iTC		x	x	oTC		
B-08	x	iTC		x	x	oTC	x	
B-09		iTC		x	x	oTC	x	
	x Component is installed.							

The conditions at the inlet and the outlet of the fan are calculated as shown in Clause 15 according to the components used.

11.5 Standard test configurations C

Table 8 — Test configurations C

	inlet side				outlet side			
	vss and/or q_V	*p*	iDS	fan	oDS	*p*	vss and/or q_V	
C-01	x	iCS, LD		x		oTS		
C-02	x	iTC	x	x		oTS		
C-03	x	iCS, LD		x		oTC		
C-04	x	iCS, LD		x		oTC	x	
C-05		iCS, LD		x		oTC	x	
C-06		iTS	x	x		oTC	x	
C-07	x	iTC	x	x		oTC		
C-08	x	iTC	x	x		oTC	x	
C-09		iTC	x	x		oTC	x	
	x Component is installed							

The conditions at the inlet and the outlet of the fan are calculated as shown in Clause 15 according to the components used.

11.6 Standard test configurations D

Table 9 — Test configurations D

	inlet side				outlet side			
	vss and/or q_V	*p*	iDS	fan	oDS	*p*	vss and/or q_V	
D-01	x	iCS, LD		x		oCS, LD		
D-02	x	iCS, LD		x		oCS, LD	x	
D-03		iCS, LD		x		oCS, LD	x	
D-04	x	iCS, LD		x	x	oTC		

Table 9 *(continued)*

	inlet side				outlet side			
	vss and/or q_V	p	iDS	fan	oDS	p	vss and/or q_V	
D-05	x	iCS, LD		x	x	oTC	x	
D-06		iCS, LD		x	x	oTC	x	
D-07	x	iCS, LD		x	x	oTS		
D-08	x	iTC	x	x		oCS, LD		
D-09	x	iTC	x	x		oCS, LD	x	
D-10		iTC	x	x		oCS, LD	x	
D-11		iTS	x	x		oCS, LD	x	
D-12	x	iTC	x	x	x	oTC		
D-13	x	iTC	x	x	x	oTC	x	
D-14		iTC	x	x	x	oTC	x	
D-15	x	iTC	x	x	x	oTS		
D-16		iTS	x	x	x	oTC	x	
	x Component is installed.							

The conditions at the inlet and the outlet of the fan are calculated as shown in Clause 15 according to the components used.

12 Measurements

12.1 Calibration

All instrumentation shall be calibrated on an annual basis with certificates traceable to International Standards.

12.2 Dimensions and cross-sectional areas

12.2.1 Tolerance on dimensions

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.

12.2.2 Cross-sectional area

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.

12.2.2.1 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 is as given in Formula (18):

$$\frac{\pi \cdot D^2}{4} \qquad (18)$$

where

D is the arithmetic mean of the measured diameters.

12.2.2.2 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 area of the rectangular section is given in Formula (19):

$$b \cdot h \tag{19}$$

where

b is the average width;

h is the average height.

12.3 Rotational speed

The fan shaft speed shall be measured for each test point, with an uncertainty not exceeding ±0,5 %.

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

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

12.4 Power input

12.4.1 General

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 %.

12.4.2 Motor input power

12.4.2.1 General

When the power to be determined is the input to motor, IEC 60034-2-1 and the following principles shall be applied.

The equipment used for measuring voltage, current and electrical power 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.

12.4.2.2 Motor directly fed by the grid

The electrical power input to the motor during the fan tests shall be measured by one of the following methods:

— for ac motors, by the two-wattmeter method or by an integrating wattmeter;

— for direct current (dc) motors, by measurement of the input voltage and current.

12.4.2.3 Motor fed from a variable frequency speed device

Useful information is provided in Annex F for motors fed from a variable frequency device.

12.4.3 Fan shaft power

12.4.3.1 General

When the power to be determined is the input to the fan shaft, one of the following methods shall be applied.

12.4.3.2 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.

12.4.3.3 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.

12.4.3.4 Determination by electrical measurement

12.4.3.4.1 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 60034-2-1. For this purpose, measurements of voltage, current, speed and, in the case of alternating current (ac) 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.

12.4.3.4.2 Calibrated motor

The power output of an electric motor for direct drive is determined from the motor input power and a motor efficiency calibration, carried out before the fan test using any one of the methods listed in the preceding clauses.

The efficiency of the motor in the relevant power range has to be determined.

The voltage and frequency of the motor supply may differ, during the test, from the motor nameplate data, but the available motor calibration record shall apply to the actual supply conditions, during the test, within the following tolerance:

— voltage: ±2 %;

— frequency: ±1 %.

The efficiency of a motor is sensitive to the temperature of the motor bearings and windings.

As a normal procedure, the motor shall be run on charge for a time sufficient to ensure that it is running at its stable working temperature. This shall be achieved when either the motor temperature rise, as measured in accordance with IEC 60034-1:2010, Clause 8, does not increase by more than 2 K in 10 min, or when the power input to the motor does not increase by more than 1 % in 10 min.

With motors having a nominal power equal to or larger than 7,5 kW, the time required to actually achieve a stable winding temperature may be impractically long. In this case, the fan test can be carried out when the bearings only shall be thermally stable, and this shall be deemed to be achieved when the motor power input shall not increase by more than 2 % in 10 min. When the test shall be carried out under these conditions, the motor efficiency calibration shall also be carried out under the same temperature conditions, and the use of this procedure shall be noted in the fan test report.

If a variable speed device is used, the motor calibration shall be carried out including the variable speed device; alternatively, a calibration record of the efficiency characteristics of the variable speed device shall be used or the real power at the output of the variable speed device has to be measured with a calibrated power analyser, suitable for such a measurement.

12.4.4 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.

In the absence of the above, the method specified in Annex E shall be used.

12.4.5 Transmission systems

For tests with standardized airways, the interposition of a transmission system between the fan and the point of power measurement shall 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.

In the absence of the above, the method specified in Annex E shall be used.

12.5 Mass flow rate

The methods of flow measurement referred to in this document lead to a determination of the mass flow rate q_m.

The mass flow rate will be constant along the test setup, if requirements in 6.10 are respected.

Any flow rate measurement obtained in accordance with ISO 5167–1 or ISO 3966 conforms to the requirements of this document.

This document specifies in Annex A some further flow-metering methods which are appropriate for fan-testing purposes and in each case, the associated uncertainty of measurement is given.

Any other flow metering method shall be accepted for the purposes of this document, if the uncertainties of the measurement are documented by calibrating the installation against an improved or calibrated standard device in accordance with an ISO Standard, which regulates the flow rate measurement and if the uncertainties are within the range of uncertainties of this document.

12.6 Temperature

12.6.1 General

In order to measure the mean temperature, one or several probes shall 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 or 0,33 *D* from the wall, whichever is less.

12.6.2 Accuracy of temperature measurement

Instruments for the measurement of temperature shall have an accuracy of ±1 °C after the application of any calibration correction.

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

The measured temperature indicated by the probe θ_{ind}, which is neither the stagnation temperature θ_{sg} nor the air temperature θ, is a value lying between them, usually slightly closer to the stagnation value.

If the reference velocity (see Clause 13) is less than 65 m/s, in this document ,the measured temperature is assumed equal to both stagnation and air temperatures.

12.6.3 Correction for high velocities

A distinction between the stagnation and the static values of temperature shall be made for reference velocities > 65 m/s ($Ma_{\text{x}} \approx 0{,}2$) (see Clause 13 and Figure 31).

Considering a recovery factor f_θ it is possible to calculate θ_{sgx} and θ_{x} from the indicated value θ_{ind}:

If p_{x} is known, approximating θ_{x}, as given in Formula (20):

$$\theta_x = \frac{\theta_{\text{ind}}}{1 + f_\theta \cdot \frac{(\kappa - 1)}{2} \cdot \left(\frac{q_{\text{m}}}{A_x \cdot p_x} \right)^2 \cdot \frac{R_{\text{wet}}}{\kappa} \cdot \theta_{\text{ind}}} \tag{20}$$

and

$$\theta_{\text{sg}x} = \theta_x + \frac{1}{f_\theta} (\theta_{\text{ind}} - \theta_x)$$

If p_{sgx} is known, approximating Ma_{sgx}, as given in Formula (21):

$$Ma_{\text{sg}x}^2 \approx \left(\frac{q_m}{A_x \cdot p_{\text{sg}x}} \right)^2 \cdot \frac{R_{\text{wet}}}{\kappa} \cdot \theta_{\text{ind}} \tag{21}$$

and Formula (22):

$$Ma_x^2 = \frac{Ma_{\text{sg}x}^2}{1 - \frac{(\kappa - 1)}{2} \cdot Ma_{\text{sg}x}^2}$$

$$\theta_x = \theta_{ind} \cdot \left(1 - f_\theta \cdot \frac{\kappa - 1}{2} \cdot Ma_x^2 \right) \tag{22}$$

and

$$\theta_{sgx} = \theta_x + \frac{1}{f_\theta} \cdot \left(\theta_{ind} - \theta_x \right)$$

In the absence of any more specific values for f_θ, the following may be used (values from ISO 5389):

a) thermometers and thermocouples in wells: 0,65;

b) bare thermocouples: 0,80;

c) bare thermocouples with insulation shields: 0,97.

12.7 Humidity

Relative humidity may be measured directly provided the apparatus used has an accuracy of ±2 %RH or determined by measuring dry and wet bulb temperatures.

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

The wet bulb thermometer shall be located in an air stream of velocity at least 3 m/s. The sleeving shall be clean, in good contact with the bulb, and kept wetted with water.

12.8 Pressure

12.8.1 Barometers

The atmospheric pressure in the test space shall be determined with an accuracy not exceeding ±0,2 %. Barometers of the direct-reading mercury column type shall be read to the nearest 100 Pa (1 mbar) or to the nearest 1 mmHg. They shall 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 present 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 shall be located in the test space at an altitude within 2 m of the mean altitude between the centre of the fan inlet and outlet sections. A correction shall be added for any difference in altitude exceeding 10 m,

$$\rho_a \cdot g \cdot \left(z_b - z_m \right) \tag{23}$$

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;

r_a is the ambient air density.

12.8.2 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 shall 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.

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

The reference instrument shall be a precision manometer or micro manometer capable of being read to an accuracy of ±0,25 % or 0,5 Pa, whichever is greater.

12.8.3 Damping of manometers

Rapid fluctuations of manometer readings shall 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 shall be linear, and of a type which ensures equal resistance to movement in either direction. The damping shall not be so heavy that it prevents the proper indication of slower changes. If these occur, a sufficient number of readings shall be taken to determine an average within ±1 % of the significant pressure.

12.8.4 Checking of manometers

Liquid column manometers shall be checked in their test location to confirm their calibration near the significant pressure. Inclined tube instruments shall be frequently checked for level and rechecked for calibration if disturbed. Before starting measurements, the zero-offset of the manometers shall be checked and set to zero, if necessary. The zero reading of all manometers shall be checked after each series of readings without disturbing the instrument to ensure valid measurement.

12.8.5 Position of manometers

The altitude of zero level of manometers or of pressure transducers shall be the mean altitude of the section for pressure measurement (see Figure 29).

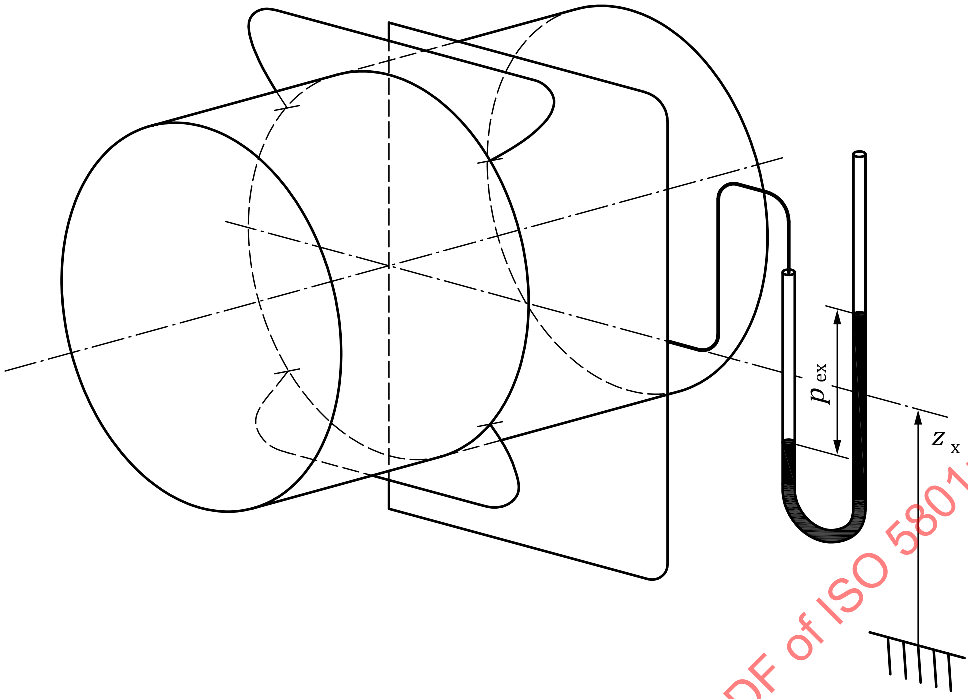


Figure 29 — Trapping connections to obtain average static pressure and altitude of manometer

12.8.6 Average pressure in an airway

At each of the sections for pressure measurement in the standardized airways, the average static pressure shall be taken to be the average of the static pressures at least at four wall tappings constructed in accordance with 12.8.7.

12.8.7 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 30. The axis of each tap shall intersect the duct axis at right angles. 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 $0{,}1 \cdot a$ is permissible.

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

Special care is required when the velocity in the airway is comparable with that at the fan inlet and outlet. In these cases, the tapping shall 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 12.8.11 may be used.

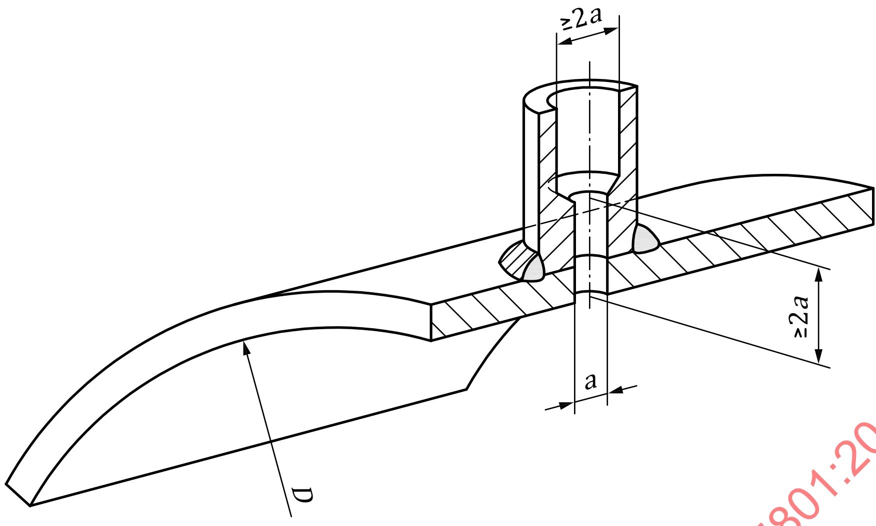


Figure 30 — Construction of wall pressure tappings

12.8.8 Position and connections

In the case of a cylindrical airway, the four tappings shall be equally spaced around the circumference. In the case of a rectangular airway, they shall be at the centres of the four sides. More than four tappings are permissible, if they are symmetrically positioned in the section. The similar tappings may be connected to a single manometer. The lengths of the tapping connections to the manometer for each hole shall be equal. They shall be connected as shown in Figure 29.

12.8.9 Methods of measurement

A differential manometer complying with the specifications of 12.8.2 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 space.

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 Figure 29.

12.8.10 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 shall 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}, p_{ex} being the mean gauge pressure, the tappings and manometer connections shall be examined for defects. If none are found, eight equally spaced pressure tappings shall 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.

12.8.11 Use of Pitot-static tube

At the appropriate pressure measurement plane in a circular airway, a minimum of four points shall be selected, equally and symmetrically spaced around the axis at approximately 0,125*D* from the wall or, in the case of a rectangular airway, 0,125 times the duct width from the centre of each wall. Under steady flow conditions, a static pressure reading shall be taken at each point and the average calculated.

Alternatively, if desired, the static pressure connections of the Pitot-static tubes may be connected together to give a single average reading in the manner described in 12.8.6 and Figure 29.

12.9 Air properties

12.9.1 General

Properties of dry air are as follows:

$$\kappa = 1{,}4$$

$$R = R_{\mathrm{dry}} = 287\ \mathrm{J \cdot kg^{-1} \cdot K^{-1}}$$

$$c_p = 1\ 004{,}5\ \mathrm{J \cdot kg^{-1} \cdot K^{-1}}$$

$$c_V = 717{,}5\ \mathrm{J \cdot kg^{-1} \cdot K^{-1}}$$

12.9.2 Density of air at section *x*

The average density of the air in an airway section x may be obtained by Formula (24):

$$\rho_x = \frac{p_x}{R_{\mathrm{wet}} \cdot \theta_x} \tag{24}$$

The gas constant of humid air, R_{wet}, is given by Formula (25):

$$R_{\mathrm{wet}} = \frac{R}{1 - 0{,}378 \cdot \frac{p_{\mathrm{v}}}{p_{\mathrm{a}}}} \tag{25}$$

$$R = R_{\mathrm{dry}}$$

with

$$0{,}378 = \frac{R_{\mathrm{v}} - R}{R_{\mathrm{v}}}$$

$R_{\mathrm{v}} = 461{,}5\ \mathrm{J \cdot kg^{-1} \cdot K^{-1}}$ being the gas constant of water vapour.

The air relative humidity, h_{rel}, can be directly measured in order to obtain

$$p_{\mathrm{v}} = h_{\mathrm{rel}} \cdot p_{\mathrm{sat} \cdot (T_{\mathrm{dry}})}$$

where

$p_{\mathrm{sat} \cdot (T_{\mathrm{dry}})}$ is the saturation vapour pressure at the dry bulb temperature T_{dry}.

p_{sat} shall be calculated by Formula (26) between 0 °C and 100 °C:

$$p_{sat \cdot \left(T_{dry}\right)} = 610{,}8 + 44{,}442 \cdot T_{dry} + 1{,}4133 \cdot T_{dry}^2 + 0{,}02768 \cdot T_{dry}^3 + 2{,}55667 \cdot 10^{-4} \cdot T_{dry}^4 + 2{,}89166 \cdot 10^{-6} \cdot T_{dry}^5 \tag{26}$$

The air humidity can be also determined by means of a psychrometer at the fan inlet. This is shown in Annex H.

12.9.3 Air viscosity

Formula (27) can be used in the range −20 °C to +100 °C to obtain the dynamic viscosity at a section, in Pascal seconds:

$$\mu_x = \left(17{,}1 + 0{,}048 \cdot T_x\right) \cdot 10^{-6} \tag{27}$$

12.9.4 Standard air

For reference in this document, a reference air is defined in Table 10.

Table 10 — Air properties

Reference conditions	Properties of "standard air in ISO 5801"	
θ = 293,15K	κ = 1,4	ρ = 1,200 kg·m^{-3}
P = 101 325 Pa	c_p = 1 008 J·kg^{-1}·K^{-1}	μ = 18,06·10^{-6} Pa·s
h_{rel} = 0,40	c_V = 720 J·kg^{-1}·K^{-1}	c = 343,8 m·s^{-1}
	R_{wet} = 288 J·kg^{-1}·K^{-1}	

NOTE This is not the "standard atmosphere" defined in ISO 2533.

13 Reference conditions

When carrying out low-pressure or medium-pressure fan tests using standardized airways, it can be assumed that the air velocity is sufficiently low and its influence on temperature may be neglected. But the change of the density with static pressure and temperature shall be taken into account for fan pressure greater than 2 000 Pa.

For high-pressure fan tests, a distinction shall be made between the stagnation and the static values of temperature, pressure and density.

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 velocity, $v_{2.ref}$, is defined as Formula (28):

$$v_{2 \cdot ref} = \frac{q_{m,max}}{\rho_{ref} \cdot A_2} \tag{28}$$

The air reference conditions are those of standard air (12.9.4), the reference section is the outlet of the fan and the reference mass flow rate is the maximum mass flow rate of the fan.

The reference velocity limit above which a distinction between the stagnation and the static values of temperature, pressure and density shall be made is regarded as equal to 65 m/s. This value corresponds $Ma_x \approx 0{,}2$ with $\theta_{sgx} / \theta_x < 1{,}01$, and $f_{Mx} < 1{,}01$ (see Figure 31).

Specific instructions for the calculation of temperatures, pressures and densities at the fan inlet and the fan outlet (indices 1 and 2) from measurements at measuring sections (indices 3 and 4) are given in Clause 15 depending on the test configuration used.

This document details methods of testing fans using standardized airways. Due to geometrical complexity a number of duct variations are required. Examples of these are given in Figure 13 to 25. Details of test configuration are also given in Table 1 to 4.

An example of a test configuration is given in Figure N.2. An example of the presentation of test results is given in Figure N.1.

14 General rules for conversion of test results

14.1 General

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.

During a laboratory test, the air density and rotational speed may vary slightly from one determination point to another. It may be desirable to convert all test points to a nominal density, a constant speed or both.

If the nominal air density, ρ_C, is within 10 % of the fan air density, ρ_{Te}, and the constant rotational speed, N_C, is within 5 % of the actual rotational speed, N_{Te}, then the air can be treated as if it were incompressible and the conversion rules can be used as detailed in 15.3.

Conversion to other air densities or rotational speeds is given in ISO 13348.

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.

Scaling test results to other sizes may also be necessary in the development of the ratings of extensive ranges, including many different similar sizes, from a limited number of tests.

The conversion of test results to another fan size is given in ISO 13348.

14.2 Laws on fan similarity

14.2.1 General

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.

14.2.2 Geometrical similarity

Two different fans are geometrically similar when all the corresponding angles are the same in the two fans, and when all the ratios between corresponding dimensions, in the two fans, are also the same.

Complete geometrical similarity requires that equal dimensional ratios are achieved also for thickness values, clearances and roughness, exactly as the other linear dimensions for the flow passages.

14.2.3 Reynolds number similarity

Reynolds number similarity is necessary in order to keep relative thicknesses of boundary layer, velocity profiles and friction losses equal, as given in Formula (29):

$$Re_{u} = \frac{u \cdot D_{r} \cdot \rho_{1}}{\mu_{1}} \tag{29}$$

When the peripheral Reynolds number increases, the friction loss coefficients decrease.

Therefore, efficiency and possibly performance may increase, compared to what is predicted according to the conversion rules provided in 15.3. Conversion rules for these Re_{u}-effects are not defined in this document. They are discussed in ISO 13348.

14.2.4 Mach number and similarity of the velocity triangles

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

For fans, the peripheral Mach number is given by Formula (30):

$$Ma_{u} = \frac{u}{\sqrt{\kappa \cdot R_{wet} \cdot \theta_{1}}} \tag{30}$$

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

When the fan pressure increases, the mean density of the fan increases, while the ratio of the inlet density to the mean density of the fan decrease. The velocity triangle similarity is no longer respected and losses increase.

This is why, when Ma_{u} 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.

The ratio of the inlet density to the mean density of the fan can be used to represent the density variation through the fan and to characterize the deviation of the fan performance, compared to what is predicted according to the conversion rules which are provided in 15.3.

15 Calculations

15.1 Test results

15.1.1 General

The measurements for flow rate are described in Annex A. Further measurements to predict the fan performance are carried out at positions 3 and 4 of the airways and are used to determine the data at positions 1 and 2, the fan inlet and the fan outlet.

For velocities $v_{2.ref} < 65$ m/s, simplified calculations in 15.1.5 may be used.

15.1.2 Temperature

For high velocities, a distinction between the stagnation and static values of temperature, θ_{sgx} and θ_x, shall be made. Usually, an expression with the Mach number at a section, Ma_x or Ma_{sgx}, is used, as given in Formula (31) (see Figure 31):

$$\frac{\theta_{sgx}}{\theta_x} = 1 + \frac{\kappa - 1}{2} \cdot Ma_x^2 = \frac{1}{1 - \frac{\kappa - 1}{2} \cdot Ma_{sgx}^2} \tag{31}$$

with

$$Ma_x = \frac{v_x}{c_x} = \frac{q_m}{A_x \cdot \rho_x \cdot \sqrt{\kappa \cdot R_{wet} \cdot \theta_x}} = \left(\frac{q_m}{A_x \cdot p_x} \right) \cdot \sqrt{\frac{R_{wet}}{\kappa} \cdot \theta_x}$$

and

$$Ma_{sgx} = \frac{v_x}{c_{sgx}} \text{ with } c_{sgx}^2 = \kappa \cdot R_{wet} \cdot \theta_{sgx} \times Ma_{sgx} = \frac{v_x}{c_{sgx}}$$

at stagnation conditions.

$$Ma_x^2 = \frac{Ma_{sgx}^2}{1 - \frac{\kappa - 1}{1} \cdot Ma_{sgx}^2} \tag{32}$$

The temperature, θ_{sgx} and/or θ_x, shall be measured in accordance with 12.6.

If no measurement is possible, the stagnation temperature θ_{sgx} shall be calculated.

The behaviour of air in the test airways, for the provisions of this document, is considered as almost adiabatic, because of the absence of heat transfer. Also, there is no exchange of mechanical energy, except in the test fan and the auxiliary fan.

Consequently, the stagnation temperature θ_{sgx}, in the test airways, shall be considered constant, except across the fan under test and the auxiliary fan.

The temperature at the fan inlet θ_{sg1} is equal to θ_{sg3}, as shown in Formula (33).

$$\theta_{sg1} = \theta_{sg3} \tag{33}$$

The temperature θ_{sg3} is to calculate with

$$\theta_{sg3} = \theta_a + \frac{P_{r \cdot aux} \text{ or } P_{e \cdot aux}}{q_m \cdot c_p}$$

If no auxiliary fan is used upstream of the upstream measuring section 3 of the fan under test, set $P_{r \cdot aux} = P_{e \cdot aux} = 0$.

In the above formula, for $\theta_{sg3} \cdot P_{r \cdot aux}$ shall be replaced by the electric input power $P_{e \cdot aux}$ when the motor is wholly immersed in the airstream.

The temperatures at the fan outlet, θ_{sg2}, and in the downstream airways, θ_{sg4}, are equal to the temperature at the fan inlet θ_{sg1}, increased by the temperature rise through the fan which is dependent upon the impeller power, as shown in Formula (34):

$$\theta_{sg4} = \theta_{g2} = \theta_{sg1} + \frac{P_r \text{ or } P_e}{q_m \cdot c_p} \tag{34}$$

In Formula (34), for θ_{sg4}, P_r shall be replaced by the electric input power P_e when the motor is wholly immersed in the airstream.

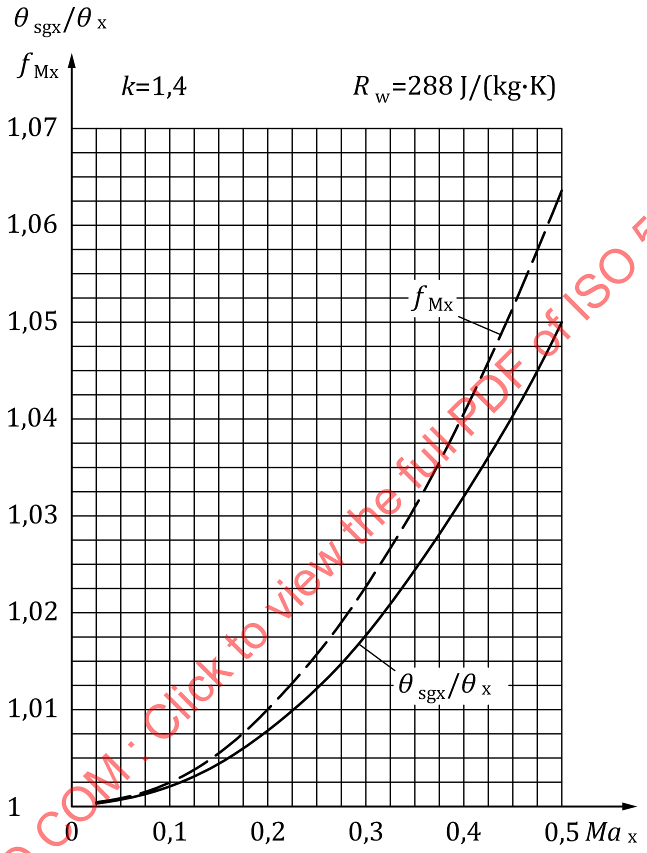


Figure 31 — Changes in f_{Mx} and θ_{sgx}/θ_x as functions of Ma_x

15.1.3 Pressure

The mean stagnation pressure, p_{sgx}, in any duct or chamber section x is given by Formula (35):

$$\frac{p_{sgx}}{p_x} = \left(\frac{\theta_{sgx}}{\theta_x} \right)^{\frac{\kappa}{\kappa - 1}} \tag{35}$$

or Formula (36):

$$p_{sgx} = p_x + p_{dx} \cdot f_{Mx} \tag{36}$$

The Mach factor, f_{Mx}, (see Figure 31) is given for k = 1,4 as a function of Ma_x by Formula (37):

$$f_{Mx} = \frac{p_{sgx} - p_x}{p_{dx}} = 1 + \frac{\left(Ma_x^2\right)}{4} + \frac{\left(Ma_x^2\right)^2}{40} + \frac{\left(Ma_x^2\right)^3}{1600} \tag{37}$$

The stagnation pressure at reference section n is given by Formula (38):

$$p_{sgn} = p_{sgx} + p_{dx} \cdot f_{Mx} \cdot \left(\xi_{n-x}\right)_x \tag{38}$$

with

$\left(\xi_{n-x}\right)_x$ energy loss coefficient between section n and section x calculated for section x;

$\left(\xi_{n-x}\right)_x$ >0 for an outlet test duct n = 2 and x = 4;

$\left(\xi_{n-x}\right)_x$ <0 for an inlet test duct n = 1 and x = 3.

15.1.4 Set of formulae

15.1.4.1 General

After the mass flow q_m (12.5 and Annex A), the temperatures θ_{sgx} and/or θ_x (12.6 and 15.1.2) and the geometries (12.2) are determined conform to this document. The pressure measurements at positions 3 and 4 are used to determine all needed data (ρ, p, p_d, p_{sg}, ...) at positions 1 and 2, the fan inlet and the fan outlet.

Two different sets of formulae can be evaluated, depending if the available pressure at the positions 1, 2, 3 and 4 is p_x or p_{sgx} ("set a" at known p_x and "set b" at known p_{sgx}), which shall be used as indicated in Table 11.

Table 11 — Alliances between components and sets of formulae

Component		Position x	Pressure available	Formulae
Test Space at inlet side	iTS	3 (free test space)	p_x	set a
Test Space at inlet side	iTS	1	p_{sgx}	set b
Duct Simulation at inlet side	iDS	3 (free test space)	p_x	set a
Duct Simulation at inlet side	iDS	1	p_{sgx}	set b
Test Chamber at inlet side	iTC	3	p_x or p_{sgx}	set a or set b
Test Chamber at inlet side	iTC	1	p_{sgx}	set b

Table 11 *(continued)*

Common Segment at inlet side	iCS	3	p_x or p_{sgx}	set a or set b
Common Segment at inlet side	iCS	1	p_{sgx}	set b
Long Duct at inlet side	iLD	3	p_x or p_{sgx}	set a or set b
Long Duct at inlet side	iLD	1	p_{sgx}	set b
Test Space at outlet side	oTS	4 (free test space)	p_x	set a
Test Space at outlet side	oTS	2	p_x	set a
Duct Simulation at outlet side	oDS	4 (free test space)	p_x	set a
Duct Simulation at outlet side	oDS	2	p_x	set a
Test Chamber at outlet side	oTC	4	p_x	set a
Test Chamber at outlet side	oTC	2	p_x	set a
Common Segment at outlet side	oCS	4	p_x or p_{sgx}	set a or set b
Common Segment at outlet side	oCS	2	p_{sgx}	set b
Long Duct at outlet side	oLD	4	p_x or p_{sgx}	set a or set b
Long Duct at outlet side	oLD	2	p_{sgx}	set b

15.1.4.2 Formula set a

If p_x is known, and if θ_x is not determined to conform to 12.6, use Formula (39):

$$\theta_x = \frac{-1+\sqrt{1+4\cdot\frac{(\kappa-1)}{2}\cdot\left(\frac{q_m}{A_x\cdot p_x}\right)^2\cdot\frac{R_{\text{wet}}}{\kappa}\cdot\theta_{\text{sgx}}}}{2\cdot\frac{(\kappa-1)}{2}\cdot\left(\frac{q_m}{A_x\cdot q_x}\right)^2\cdot\frac{R_{\text{wet}}}{\kappa}} \tag{39}$$

with

$$Ma_x^2 = \left(\frac{q_m}{A_x\cdot p_x}\right)^2\frac{R_{\text{wet}}}{\kappa}\cdot\theta_x$$

and

$$f_{\text{M}x} = 1+\frac{\left(Ma_x^2\right)}{4}+\frac{(2-\kappa)\cdot\left(Ma_x^2\right)^2}{24}+\frac{(2-\kappa)\cdot(3-2\kappa)\cdot\left(Ma_x^2\right)^3}{192}+\ldots$$

$$\rho_x = \frac{p_x}{R_{\text{wet}}\cdot\theta_x} \text{ and } p_{\text{d}x} = \frac{\kappa}{2}\cdot Ma_x^2\cdot p_x \text{ and } p_{\text{sg}x} = p_x + f_{\text{M}x}\cdot p_{\text{d}x}$$

15.1.4.3 Formula set b

If $p_{\text{sg}x}$ is known,

$$\rho_{\text{sg}x} = \frac{p_{\text{sg}x}}{R_{\text{wet}}\cdot\theta_{\text{sg}x}}$$

$$v_x \approx \frac{q_m}{\left(A_x \cdot \rho_{\text{sg}x}\right)}$$

and approximating v_x by iteration

$$Ma_{\text{sg}x} = \frac{v_x}{\sqrt{\kappa \cdot R_{\text{wet}} \cdot \theta_{\text{sg}x}}}$$

$$\theta_x = \theta_{\text{sg}x} \cdot \left(1 - \frac{\kappa - 1}{2} \cdot Ma_{\text{sg}x}{}^2\right)$$

$$\rho_x = \rho_{\text{sg}x} \cdot \left(\frac{\theta_x}{\theta_{\text{sg}x}}\right)^{\frac{1}{\kappa - 1}}$$

$$v_x = \frac{q_m}{\left(A_x \cdot \rho_x\right)}$$

End of iteration, if changing of v_x is negligible.

$$Ma_x{}^2 = \frac{Ma_{\text{sg}x}{}^2}{\left(1 - \frac{\kappa - 1}{2} \cdot Ma_{\text{sg}x}{}^2\right)}$$

$$f_{\text{M}x} = 1 + \frac{Ma_x{}^2}{4} + \frac{(2-\kappa) \cdot \left(Ma_x{}^2\right)^2}{24} + \frac{(2-\kappa) \cdot (3 - 2\kappa) \cdot \left(Ma_x{}^2\right)^3}{192} +$$

$$p_{\text{d}x} = \frac{1}{2 \cdot \rho_x} \cdot \left(\frac{q_m}{A_x}\right)^2$$

$$p_x = p_{\text{sg}x} - f_{\text{M}x} \cdot p_{\text{d}x}$$

15.1.5 Simplified sets of formulae, which can be used for $v_{2.\text{ref}} \leq 65\text{m/s}$

As for reference air velocities $v_{2.\text{ref}}$ not greater than 65 m/s, the temperature ratio $\frac{\theta_{\text{sg}x}}{\theta_x}$ does not exceed 1,008 and the Mach factor f_{Mx} does not exceed 1,010 (see Figures 32 and 33), simplified formulae can be used.

To get a clear characterization of the simplified formulation in these cases, the total pressure $p_{\text{tot}x}$ is used instead of the stagnation pressure $p_{\text{sg}x}$.

θ_x and $\theta_{\text{sg}x}$ are treated as equal ($\theta_x = \theta_{\text{sg}x}$), Ma_x and $Ma_{\text{sg}x}$ are not used and f_{Mx} is set to 1.

The total pressure at reference section n is given by Formula (40):

$$p_{\text{tot}n} = p_{\text{tot}x} + p_{\text{d}x} \cdot \left(\xi_{n-x}\right)_x \tag{40}$$

The allocations in Table 11 apply.

15.1.5.1 Simplified formula set a

If p_x is known use Formula (41):

$$\rho_x = \frac{p_x}{R_{\text{wet}} \cdot \theta_x} \tag{41}$$

and

$$p_{\text{d}x} = \frac{\rho_x}{2} \cdot \left(\frac{q_m}{\rho_x \cdot A_x} \right)^2$$

and

$$p_{\text{tot}x} = p_x + p_{\text{d}x}$$

15.1.5.2 Simplified formula set b

If $p_{\text{tot}x}$ is known, use Formula (42):

$$\rho_x = \frac{p_{\text{tot}x} + \sqrt{p_{\text{tot}_x}^2 - 2 \cdot R_{\text{wet}} \cdot \theta \cdot \left(\frac{q_m}{A_E} \right)^2}}{2 \cdot R_{\text{wet}} \cdot \theta_x} \tag{42}$$

and

$$p_{\text{d}x} = \frac{\rho_x}{2} \cdot \left(\frac{q_m}{\rho_x \cdot A_x} \right)^2$$

and

$$p_x = p_{\text{tot}x} - p_{\text{d}x}$$

15.1.6 Fan pressure

The fan pressure, p_f, is the difference between the stagnation pressure at the outlet of the fan and the stagnation pressure at the inlet of the fan, as given in Formula (43):

$$p_{\text{f}} = p_{\text{sg2}} - p_{\text{sg1}} \tag{43}$$

or

$$p_{\text{f}} = p_{\text{tot2}} - p_{\text{tot1}} \quad \text{(allowed if } v_{2.\text{ref}} \leq 65 \text{ m/s)}$$

15.1.7 Fan static pressure

The fan static pressure p_{fs} is the difference between the static pressure at the outlet of the fan and the stagnation pressure at the inlet of the fan.

$$p_{fs} = p_2 - p_{sg1} = p_f - p_{fd} \cdot f_{M2}$$

with Formula (44):

$$p_{fd} = p_{d2} \tag{44}$$

or

$$p_{fs} = p_2 - p_{tot1} = p_f - p_{fd} \text{ (allowed if } v_{2.ref} \le 65 \text{ m/s)}$$

15.1.8 Volume flow rate of the fan

The methods of flow measurement in this document 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 volume flow rate of the fan can be expressed as Formula (45):

$$q_{V1} = \frac{q_m}{\rho_1} \tag{45}$$

It is also possible to express the volume flow rate under inlet stagnation conditions by Formula (46):

$$q_{Vsg1} = \frac{q_m}{\rho_{sg1}} \tag{46}$$

15.1.9 Fan air power and efficiency

15.1.9.1 General

The fan air power can be written as Formula (47):

$$P_u = q_m \cdot y_f = q_{V_{sg1}} \cdot p_f \cdot k_p \tag{47}$$

There are three methods to calculate the air power:

— the first (15.1.9.2) derived from the concept of fan work per unit mass;

— the two others (15.1.9.3 and 15.1.9.4) based on the concept of volume flow rate and fan pressure with a correction factor k_p to take into account the influence of air compressibility.

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

If fan impeller power, P_r, is not measured and cannot be determined from known component efficiencies such as a calibrated motor, the method in 15.1.9.2 shall be used.

15.1.9.2 Calculation of fan air power from fan work per unit mass (first method)

$$P_u = q_m \cdot y_f \tag{48}$$

with

$$y_f = \frac{p_2 - p_1}{\rho_m} + \frac{1}{2} \cdot f_{M2} \cdot \left(\frac{q_m}{\rho_2 \cdot A}\right)^2 - \frac{1}{2} \cdot f_{M1} \cdot \left(\frac{q_m}{\rho_1 \cdot A_1}\right)^2$$

where

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

For the simplified calculation for $v_{2.ref} \leq 65$ m/s, Mach factors f_{M1} and f_{M2} are set equal to 1.

For $p_f \leq 2\ 000$ Pa, ρ_2 can be set equal to ρ_1 and so $\rho_m = \rho_1$.

15.1.9.3 Calculation of fan air power from volume flow rate and fan pressure (second method)

$$P_u = q_{V_{sg1}} \cdot p_f \cdot k_P \tag{49}$$

with

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

for

$$r = \frac{p_{sg2}}{p_{sg1}} = 1 + \frac{p_f}{p_{sg1}}$$

and

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

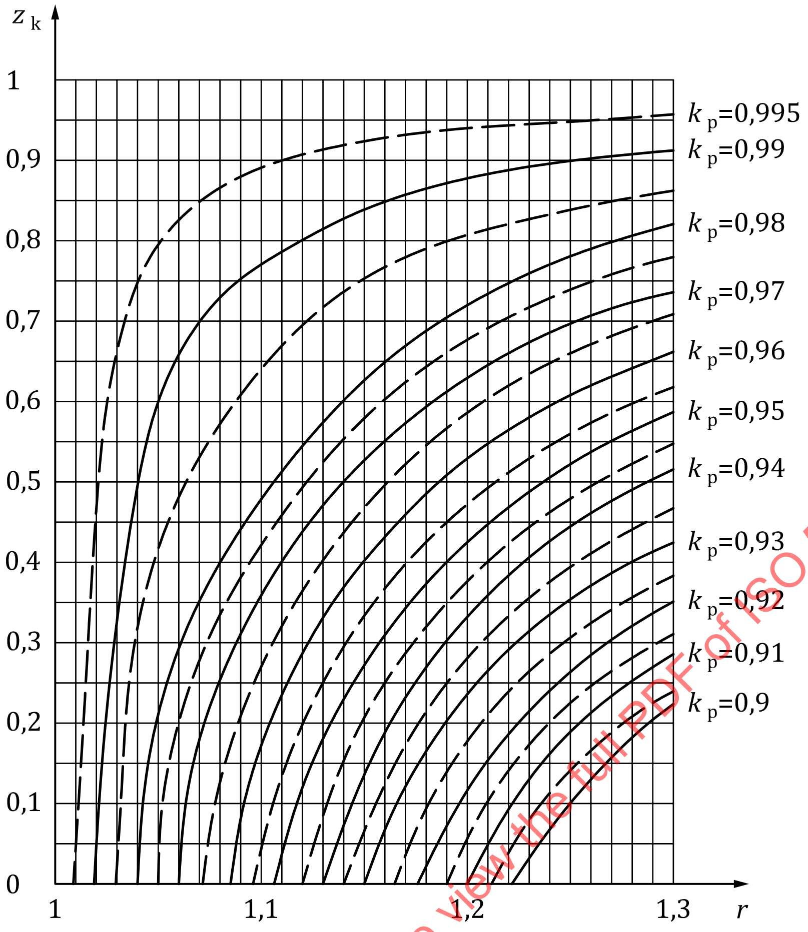


Figure 32 — Compressibility coefficient k_P with r and Z_k

15.1.9.4 Calculation of fan air power from volume flow rate and fan pressure (third method)

$$P_u = q_{V_{sg1}} \cdot p_f \cdot k_P \tag{50}$$

with

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

for

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

and

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

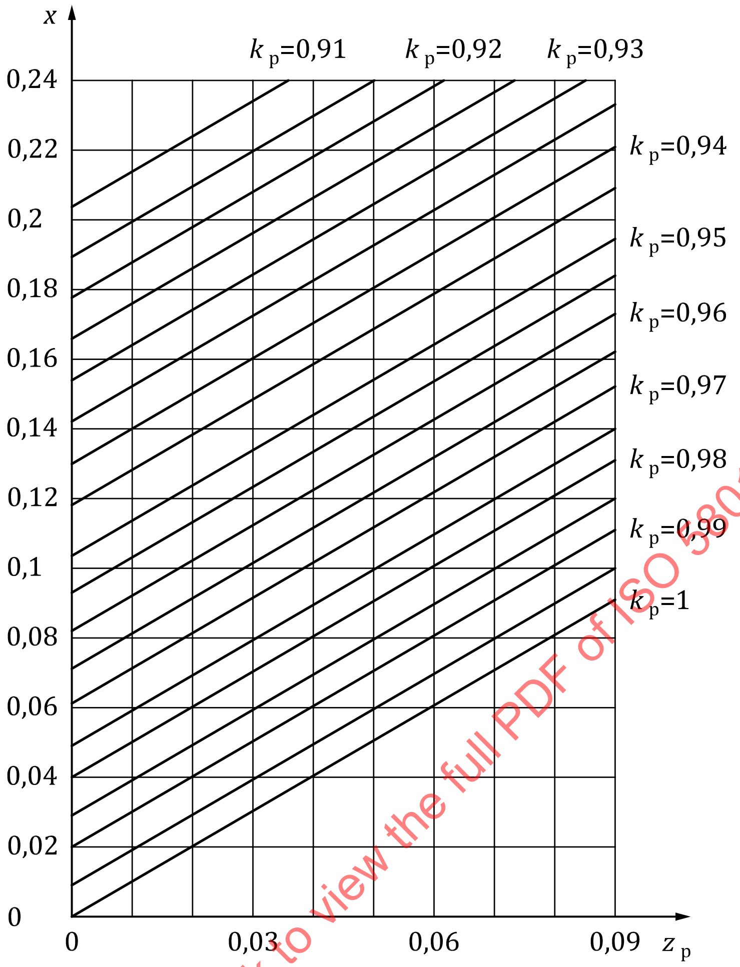


Figure 33 — Compressibility coefficient k_P with x and Z_p

15.2 Efficiencies

15.2.1 General

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, (includes bearing losses, while impeller power, P_r, does not);

— motor output power, P_o;

— motor input power, P_e;

— drive control electrical input power, P_{ed}.

$$\eta = \frac{P_u}{P_r} \tag{51}$$

$$\eta_a = \frac{P_u}{P_a} \tag{52}$$

$$\eta_o = \frac{P_u}{P_o} \tag{53}$$

$$\eta_e = \frac{P_u}{P_e} \tag{54}$$

$$\eta_{ed} = \frac{P_u}{P_{ed}} \tag{55}$$

15.2.2 Fan static air power and static efficiency

The fan static air power can be written as Formula (56):

$$P_{us} = q_m \cdot y_{fs} = q_{V_{sg1}} \cdot p_{fs} \cdot k_P \tag{56}$$

with

$$y_{fs} = \frac{p_2 - p_1}{\rho_m} - \frac{1}{2} \cdot f_{M1} \cdot \left(\frac{q_m}{\rho_1 \cdot A_1} \right)^2 = y_f - \frac{1}{2} \cdot f_{M2} \cdot \left(\frac{q_m}{\rho_2 \cdot A_2} \right)^2$$

where

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

k_P corresponds to the definition in 15.1.9.3 or 15.1.9.4.

For the simplified calculation for $v_{2.ref} \leq 65$ m/s, Mach factors f_{M1} and f_{M2} are set equal to 1.

For $p_f \leq 2\,000$ Pa, ρ_2 can be set equal to ρ_1 and so $\rho_m = \rho_1$.

The various efficiencies are calculated from P_{us} in the same way as in 15.2.

15.3 Conversion rules

15.3.1 General

Under the restrictions provided in 14.2 the laws of fan similarity are valid and if the nominal air density ρ_C is within 10% of the fan air density ρ_{Te} and the constant rotational speed N_C is within 5% of the actual rotational speed N_{Te}, then the air can be treated as if it were incompressible and the conversion rules according to Formula (57) can be used:

$$\frac{q_{V1C}}{q_{v1Te}} = \left(\frac{N_C}{N_{Te}} \right) \left(\frac{D_{rC}}{D_{rTe}} \right)^3$$

$$\frac{p_{fC}}{p_{fTe}} = \left(\frac{N_C}{N_{Te}} \right)^2 \cdot \left(\frac{D_{rC}}{D_{rTe}} \right)^2 \cdot \left(\frac{\rho_{1C}}{\rho_{1Te}} \right) = \frac{p_{fsC}}{p_{fsTe}} \tag{57}$$

$$\frac{P_{rC}}{P_{rTe}} = \left(\frac{N_C}{N_{Te}} \right)^3 \cdot \left(\frac{D_{rC}}{D_{rTe}} \right)^5 \cdot \left(\frac{\rho_{1C}}{\rho_{1Te}} \right)$$

In case of compressible fluids these formula can only be used for the volume flow and the power, for the conversion of the fan pressure, however, the polytropic approach of Annex J shall be used.

Further scaling or upgrading procedures are given in ISO 13348.

15.3.2 Shaft power and impeller power

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

It may be necessary to estimate the bearing losses P_{bTe} at N_{Te} and P_{bC} at N_C and to use the relations given in Formula (58):

$$P_{rTe} = P_{aTe} - P_{bTe} \tag{58}$$

and

$$P_{aC} = P_{rC} + P_{bC}$$

in order to carry out the conversion specified above.

However, the error incurred by assuming Formula (59):

$$\frac{P_{rC}}{P_{rTe}} = \frac{P_{aC}}{P_{aTe}} \tag{59}$$

will not exceed the following, as a percentage, Formula (60):

$$\frac{200 \cdot (N_C - N_{Te}) \cdot P_{bTe}}{N_{te} \cdot P_{aTe}} \tag{60}$$

which is often negligible.

16 Fan characteristic curves

16.1 Methods of plotting

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

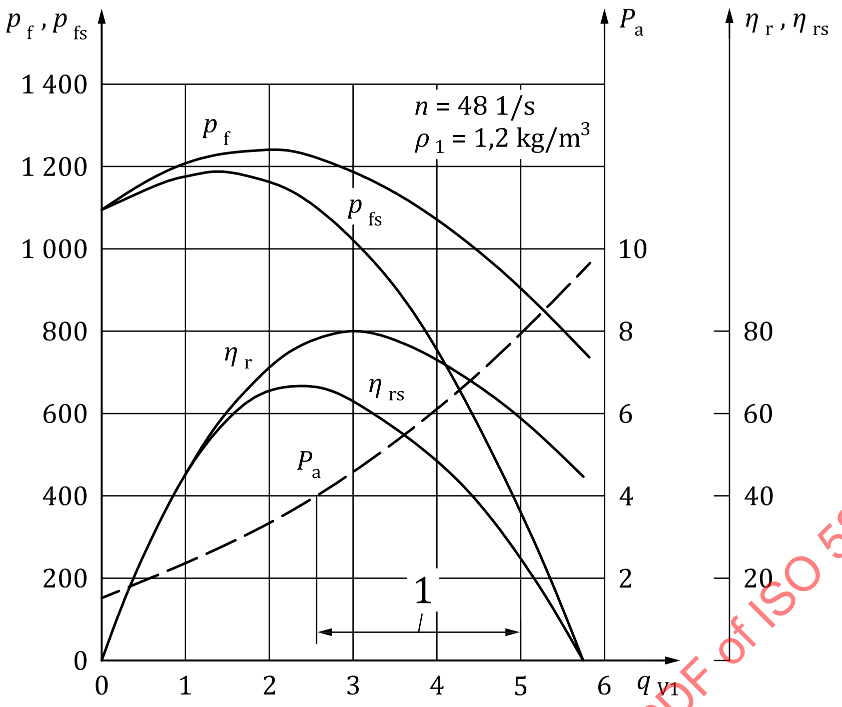
The results of conversion may be used, provided those changes which are outside the conversion limits are clearly indicated on the plotted curves.

For fans for which the design fan pressure is greater than 2 000 Pa, indications of the fan outlet density shall be plotted using the ratio ρ_1/ρ_m for $v_{2.ref} \leq 65$ m/s or the ratio ρ_{sg1}/ρ_{sgm} for $v_{2.ref} > 65$ m/s.

16.2 Characteristic curves at constant speed

Fan characteristic curves at constant rotational speed are obtained from results converted to a constant stated rotational speed and to a constant stated density which shall, unless otherwise agreed, be 1,2 kg/m^3, and to a stated absolute inlet pressure.

The fan pressure, p_f, and the fan static pressure, p_{fs}, or either one of them together with the fan dynamic pressure p_{d2} shall be plotted against the inlet volume flow rate q_{V1} for $v_{2.ref} \leq 65$ m/s or q_{Vsg1} for $v_{2.ref} > 65$ m/s. The fan impeller efficiency, η_r, and/or the fan impeller static efficiency, η_{rs}, or their shaft power equivalents may also be plotted. An example for $v_{2.ref} \leq 65$ m/s is given in Figure 34.



Key

q_{V1} fan inlet volume flow rate, in cubic metres per second

p_f fan pressure, in Pascal

p_{fs} fan static pressure, in Pascal

1 working range

P_a fan shaft power, in kilowatts

η_r fan impeller efficiency, as a percentage

η_{rs} fan impeller static efficiency, as a percentage

Figure 34 — Example of a set of complete, constant-speed, fan characteristic curves ($v_{2.ref}$ ≤ 65 m/s)

16.3 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 shall be operated under fixed and stated conditions, e.g. at the rated voltage and frequency for an electric motor. The rotational speed shall 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 14.2.2 provided the rotational speed is corrected with respect to motor input power by use of performance data on the driving means.

16.4 Complete fan characteristic curve

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

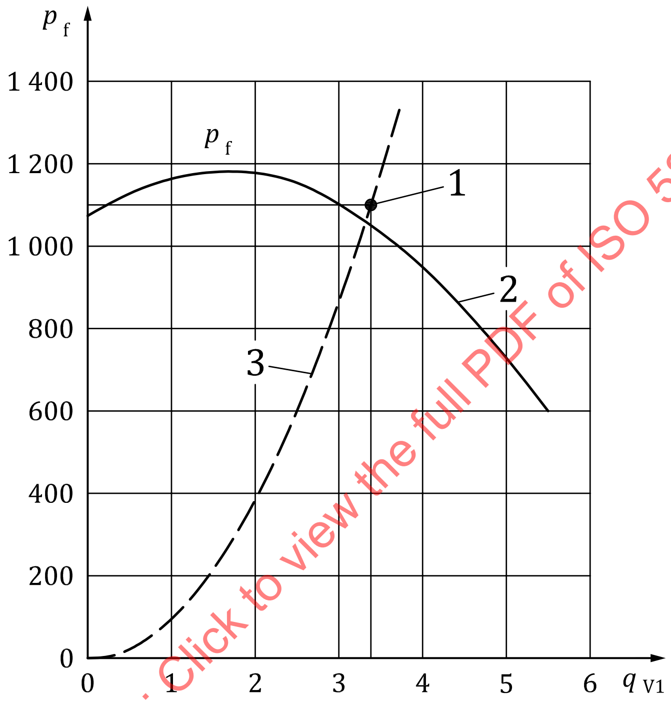
Only a part of this curve is normally used however, and it is recommended that the supplier shall state the range of inlet volume flow rates for which the fan is suitable. The plotted fan characteristic curve may then be limited to this normal operating range. Outside the normal operating range of inlet volume flow rates, the uncertainty of measurement is liable to increase and unsatisfactory flow patterns may develop at inlet or outlet.

16.5 Test for a specified duty

Tests for a specified duty shall 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 pressure or fan static pressure.

A system resistance line shall also be drawn, passing through the specified duty point, and such that the pressure varies with the square of the inlet volume flow rate.

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



Key

q_{V1} fan inlet volume flow rate, in cubic metres per second

p_f fan pressure, in Pascal

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

2 fan pressure volume characteristic curve

3 system resistance curve, $p_f \propto q^2_{V1}$

Figure 35 — Example of test for a specified duty ($v_{2.ref} \leq 65$ m/s)

16.6 Specific fan types

Roof mounted ventilators are a specialist fan application. Details of an appropriate test method are given in Annex B.

For centrifugal fans which do not have a horizontal discharge and the fan characteristic shall be determined by installation category B or D methods information on the method of test is given in Annex D.

Plenum and plug fans shall only be measured in installation category A configuration. Guidance on the measurement method is given in Annex L.

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 air flow measurements. The concepts contained in ISO 5168 provide the basis for the following.

In this document, 95 % confidence level 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 shall be considered in an uncertainty analysis.

The procedure outlined in ISO 5168 includes the following steps:

— list all possible sources of error;

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

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

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

— calculate the uncertainty for each parameter;

— establish the uncertainty interval for each parameter.

NOTE In addition to measurement errors, there can 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 human 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.

NOTE Also refer to Annex N.

17.5 Reporting uncertainties

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

a) the test value of the parameter, e.g. air volume flow rate;

EXAMPLE $R = q_V = 5\ m^3/s$

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, v;

EXAMPLE $b = 0{,}05\ m^3/s$ $v = 5$.

c) the bias limit;

EXAMPLE $B = 0{,}025\ m^3/s$.

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

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

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

then

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

17.6 Maximum allowable uncertainties for measurements

This document 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 shall 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 12 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 should be sufficiently accurate so that when the various error estimates are combined, the resulting uncertainty will not exceed the value given in Table 13.

Table 12 — Maximum allowable uncertainties of measurement of individual parameters

Parameter	Symbol	Relative uncertainty of measurement	Remarks
Atmospheric pressure	p_a	u_{pa} = 0,2 %	Corrected for temperature and altitude
Ambient temperature	θ_a	$u_{\theta a}$ = 0,2 %	Measured near fan inlet or inlet duct, or in a chamber where the velocity is less than 25 m/s
Gauge pressure	p_e	u_{pe} = 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
Differential pressure	Δp	$u_{\Delta p}$ = 1,4 %	As for gauge pressure
Rotational frequency of impeller	n	u_n = 0,5 %	
Power input	P_r	u_{Pr} = 2 %	Measured by torque meter or two-wattmeter method; uncertainty according to class of wattmeter and transformer
Area of a nozzle throat	A_d	u_{Ad} = 0,2 %	
Area of a duct	A_x	u_{Ax} = 0,5 %	
Mass flow rate	q_m	u_{qm}	See Annex A for various flow-measurement methods

17.7 Maximum allowable uncertainty of results

The different parameters comprising the results of a fan test are listed in Clause 13. Also listed is the maximum allowable relative uncertainty for each result, if the test is to qualify as a test conducted under this document. 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 Clause 13 are based on the 95 % confidence level. Precision indices and bias limits are not separately stated. Nevertheless, any test conducted in accordance with this document shall include an uncertainty analysis. The precision indices and bias limits shall be listed separately in such an analysis.

Table 13 — Maximum allowable uncertainty for the results

Parameter	Symbol	Maximum relative uncertainty of result
Ambient density	ρ_a	$u_{\rho a}$ = 0,4 %
Fan temperature rise	$\Delta\theta$	$u_{\Delta\theta}$ = 2,8 %
Outlet stagnation temperature	θ_{sg2}	$u\theta_{sg2}$ = 0,4 %
Outlet stagnation density	ρ_{sg2}	u_{rsg2} = 0,7 %
Fan dynamic pressure	p_{fd}	u_{pfd} = 4,0 %
Fan pressure	p_f	u_{pf} = 1,4 %
Fan air power	P_u	u_{Pu} = 2,5 %
Fan impeller efficiency	η_r	$u_{\eta r}$ = 3,2 %
Fan flow rate	q_m or q_V	u_{qm} or u_{qV} = 2,0%

Annex A
(normative)

Determination of air flow rate

A.1 General

This Annex specifies 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 conditions. i.e. the use of an in-line flowmeter or a traversing method.

A.2 In-line flowmeters (standard primary device)

The flowmeters which may be used are the multi-Venturi nozzles, the conical or bellmouth inlet and the orifice plate. Multi-nozzles are only used within a test chamber. The conical or bellmouth inlet may only be used at the inlet to an airway, drawing air from free space. The orifice plate may be used at the inlet to or outlet from an airway as well as between two sections of an airway.

The requirements for the orifice plate between two sections of an airway and for the simplified installations in which they may be used are given in ISO 5167–1.

The general expression for the mass flow rate through an in-line differential pressure flowmeter is as given in Formula (A.1):

$$q_m = \alpha \cdot \varepsilon \cdot \frac{\pi \cdot d^2}{4} \cdot \sqrt{2 \cdot \rho_{up} \cdot \Delta p} \tag{A.1}$$

where

q_m is the mass flow rate;

d is the throat diameter;

ρ_{up} is the upstream density;

Δp is the pressure difference;

α is the flow coefficient;

ε is the expansibility factor.

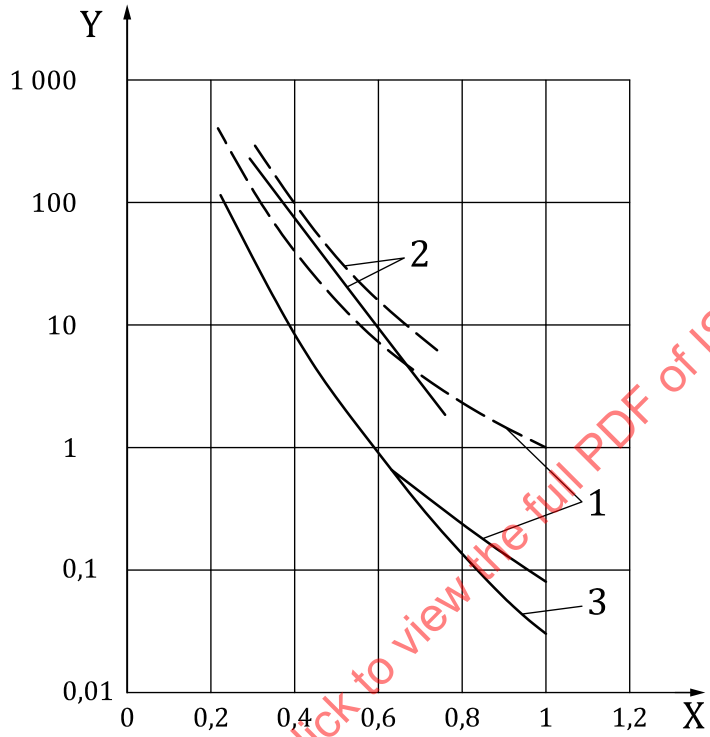
The pressure difference across an in-line flowmeter shall be measured with an uncertainty not exceeding ±1,4 % of the observed value.

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.

In order to facilitate the selection of type and size of flowmeter, the losses associated with each type are given in Figure A.1.

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.

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 present 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.



Key

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

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

1 conical or bellmouth inlet

2 orifice plate

3 conical or bellmouth inlet with 15° angle diffuser included

— pressure loss

---- pressure difference

Figure A.1 — Pressure loss and pressure difference of standard primary systems

A.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.

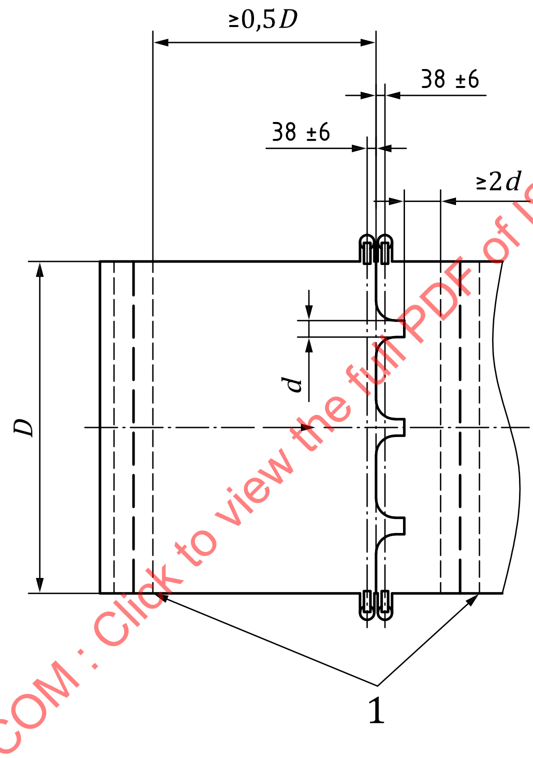
The requirements for the use of a Pitot-static tube in standardized airways are given in ISO 3966.

A.4 Determination of flow rate using multiple nozzles

A.4.1 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. The axes of the nozzle(s) and of the chamber in which they are installed shall be parallel.

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



Key

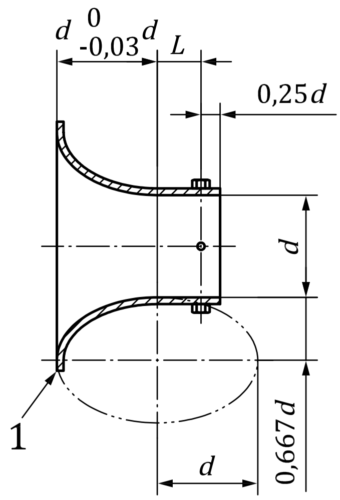
1 flow-settling means

Figure A.2 — Multiple-nozzle unit

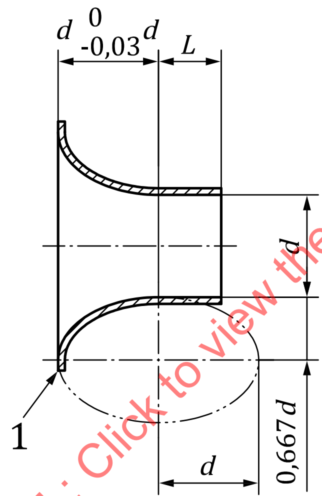
A.4.2 Geometry

Multiple-nozzle dimensions and tolerances are shown in Figure A.2.

The profile shall be axially symmetrical and the outlet edge shall be square, sharp and free from burrs, nicks or rounding's. Nozzle throat length, *L*, shall be either 0,6 *d* ± 0,005 *d* (recommended for new constructions) or 0,5 *d* ± 0,005 *d*.



a) Nozzle with throat-pressure tappings



b) Nozzle without throat-pressure tappings

Key

1 fairing radius approximately 0,05*d*, if necessary

Figure A.3 — Nozzle geometry

The pressure tappings shall conform to the requirements of 12.8.

Nozzles shall have an elliptical form as shown in Figure A.3, but two or three radii approximations that do not differ at any point, in the normal direction, by more than 0,015*d* from the elliptical form may also be used.

The nozzle throat diameter *d* shall be measured to an accuracy of 0,001*d* at the minor axis of the ellipse and the nozzle exit. Four measurements shall be taken at angular spacing of 45° and shall be within ±0,002*d* of the mean.

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

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 0,001*d* peak-to-peak.

A.4.3 Calculation of mass flow rate

The mass flow rate for a multiple nozzle is given by Formula (A.2):

$$q_m = \varepsilon \cdot \sum_{i=1}^{n} \left(\alpha_i \cdot d_i^2 \right) \cdot \frac{\pi}{4} \cdot \sqrt{2 \cdot \rho_{\mathrm{up}} \cdot \Delta p} \tag{A.2}$$

where

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

A.4.4 Multiple-nozzle characteristics

The pressure difference, Δp, shall be measured in accordance with the requirements in A.2. A multiple-nozzle installation manufactured in accordance with the requirements in A.4.2 may be used uncalibrated for pressure ratios $r_{\mathrm{d}} > 0{,}9$ (i.e. $\Delta p < 10$ kPa).

The nozzle flow rate coefficient, α, may be calculated by the Formula (A.3) and is shown in Figure A.4:

$$\alpha = \left[0{,}9986 - \frac{7{,}006}{\sqrt{Re_d}} + \frac{134{,}6}{Re_d} \right] \left[\frac{1}{\sqrt{1 - \alpha_{\mathrm{Aup}} \cdot \beta^4}} \right] = \frac{C}{\sqrt{1 - \alpha_{\mathrm{Aup}} \cdot \beta^4}} \quad \text{for } L/d = 0{,}6$$

and (A.3)

$$\alpha = \left[0{,}9986 - \frac{6{,}688}{\sqrt{Re_d}} + \frac{131{,}5}{Re_d} \right] \left[\frac{1}{\sqrt{1 - \alpha_{\mathrm{Aup}} \cdot \beta^4}} \right] = \frac{C}{\sqrt{1 - \alpha_{\mathrm{Aup}} \cdot \beta^4}} \quad \text{for L/d 00,5}$$

where

Re_{d} is the Reynolds number based on the exit diameter, which may be estimated by the following formula:

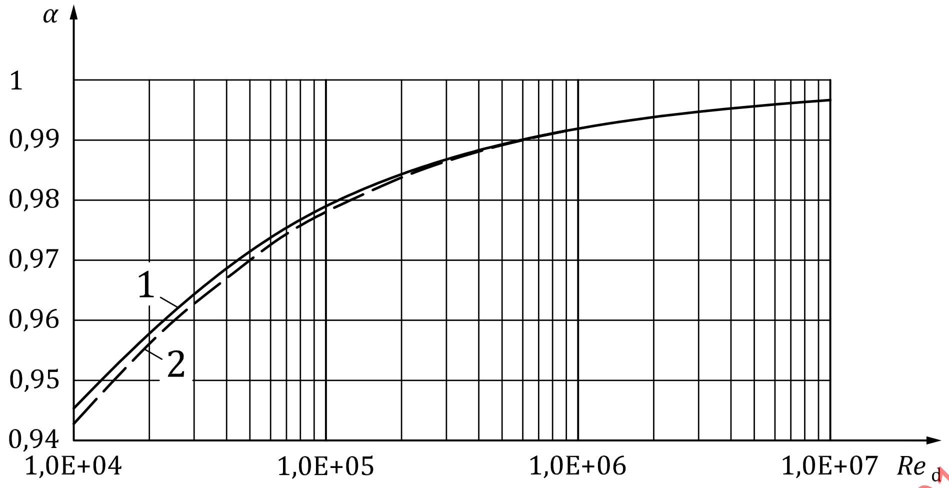
$$Re_d = 0{,}95 \cdot \varepsilon \cdot d \cdot \frac{\sqrt{2 \cdot \rho_{\mathrm{up}} \cdot \Delta p}}{\mu_{\mathrm{up}}}$$

where

α_{Aup} is the kinetic energy factor 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;

$\beta = \frac{d}{D}$ (which may be taken as 0 for a chamber) ($b < 0{,}525$ for an in-duct nozzle);

C is the nozzle discharge coefficient.



Key

1 $L/d = 0,6$

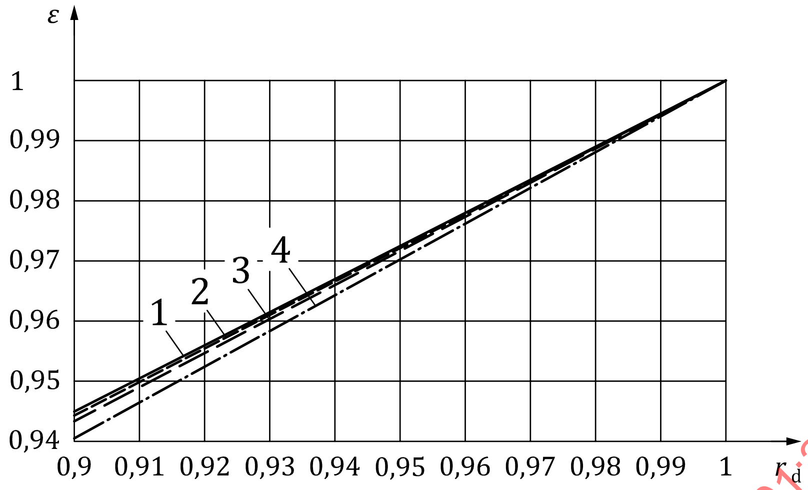
2 $L/d = 0,5$

Figure A.4 — Flow rate coefficient for nozzles used in a chamber (ß = 0)

The expansibility factor may be calculated by Formula (A.4) and is shown in Figure A.5:

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

where $r_{\mathrm{d}} = \dfrac{p_{\mathrm{up}} - \Delta p}{p_{\mathrm{up}}} = 1 - \dfrac{\Delta p}{p_{\mathrm{up}}}$.



Key

1 $\beta = 0$

2 $\beta = 0,3$

3 $\beta = 0,4$

4 $\beta = 0,5$

Figure A.5 — Expansibility factors for nozzles, used in a chamber (ß = 0)

A.4.5 Uncertainty

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

A.5 Determination of flow rate using a conical or bellmouth inlet

A.5.1 Installation

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

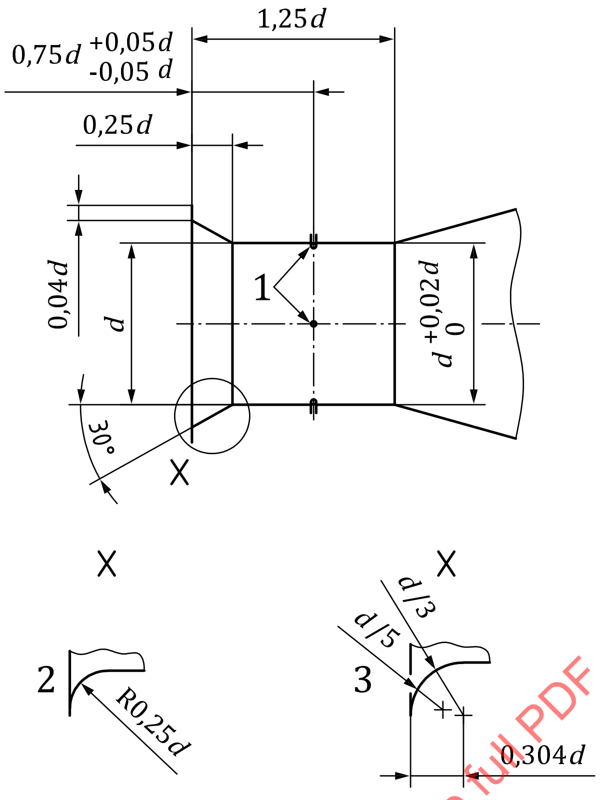
The axis of the inlet and that of the airway shall be coincident. The junctions between the cone and the face and between the cone and the cylindrical throat each are having a sharp edge, free from ridges and projections.

Within the inlet zone defined in Figure A.7, there shall be no external obstruction to the free movement of the air entering the inlet, and the velocity of any cross-currents shall not exceed 5 % of the nozzle throat velocity.

A.5.2 Geometry

The conical or bellmouth inlet dimensions and tolerances are given in Figure A.6.

The profile shall be axially symmetric.



Key

1 four wall pressure tappings

2 and 3 alternative bellmouth inlets (2: one-arc-nozzle and 3: two-arc-nozzle)

Figure A.6 — Geometry of conical or bellmouth inlet

The throat diameter, d, is the arithmetic mean of four measurements, to within an accuracy of 0,003 d, taken at angular spacing's of about 45° in the plane of the throat pressure tappings.

The pressure tappings shall conform to the requirements of 12.8.

The pressure difference, Δp, shall be measured in accordance with the requirements in A.2.

Except where otherwise specified, the included angle of the divergent section may lie anywhere in the range q < 30. The length of the connection piece shall be in accordance with the condition given in Formula (A.5):

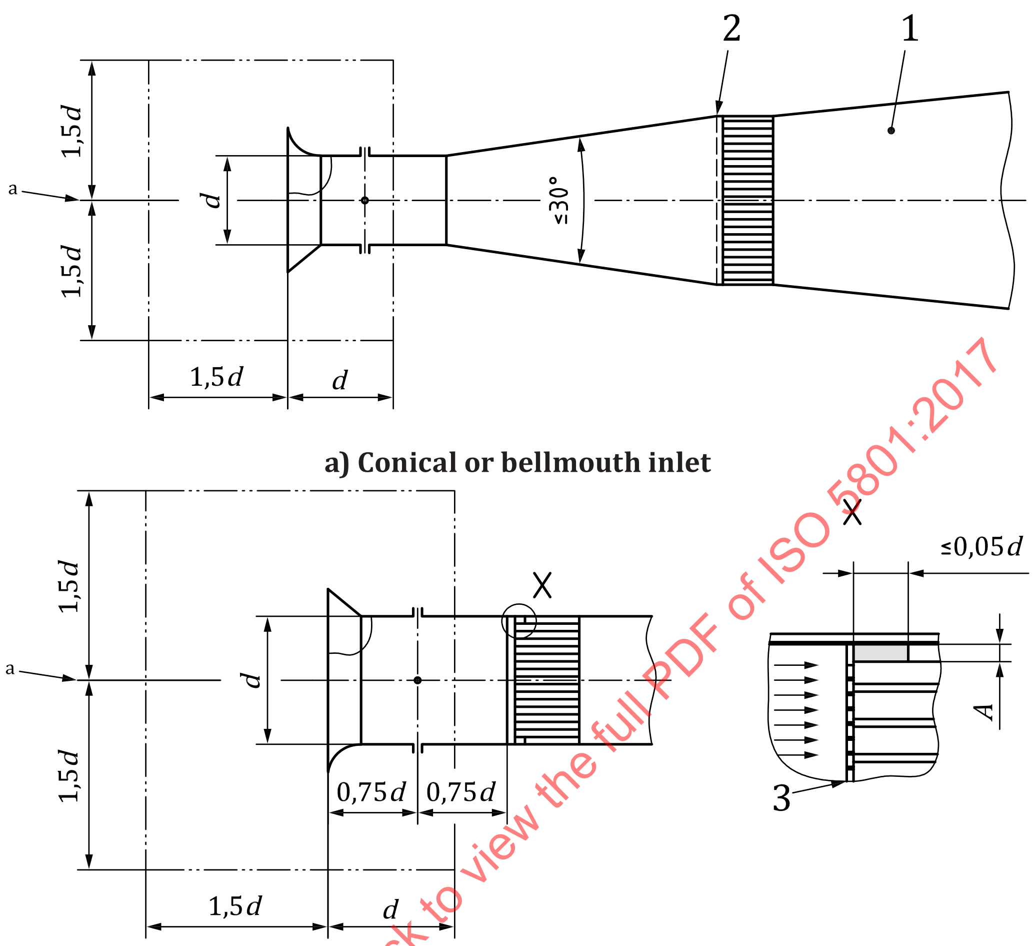
$$L \geq 4 \cdot \left[\sqrt{\frac{4 \cdot A_L}{\pi}} - \sqrt{\frac{4 \cdot A_S}{\pi}} \right] \tag{A.5}$$

where

A_L is the larger and A_S the smaller area of the transition part.

A.5.3 Screen loading

Beside other throttling devices, adjustable screen loading in accordance with Figure A.7 as an example is permissible with the conical or bellmouth inlet, but the uncertainty of the flow rate coefficient α is increased.



b) Conical or bellmouth inlet with adjustable screen loading as an example

Key

1 duct expander, shape transition, sudden expansion

2 resistance screen, if required

3 screen loading and support-ring

A max. 0,012*D* or 6 mm and min. 0,006*D* or 3 mm

X detail for screen loading

a The inlet zone shall be clear from obstruction.

Figure A.7 — Conical or bellmouth inlet flow-metering installations

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.

Supports for screens shall have the minimal frontal area consistent with strength and stiffness for their purpose. For example, no single transverse member shall 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 A.7 b).

Screens shall be accurately cut and a supporting ring with a radial thickness of 0,012 *d* or 6 mm max. and 0,006 *d* or 3 mm min. and a length of 0,05 *d* max. shall be fitted or other means adopted to eliminate leakage at the wall.

A.5.4 Calculation of mass flow rate

A.5.4.1 General

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.

A conical or bellmouth inlet manufactured in accordance with the requirements in A.5.2 may be used uncalibrated for pressure ratios $r_\mathrm{d} > 0{,}96$, i.e. $\Delta p < 4\,000$ Pa.

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

The mass flow rate is given by Formula (A.6):

$$q_m = \alpha \cdot \varepsilon \cdot \frac{\pi \cdot d^2}{4} \cdot \sqrt{2 \cdot \rho_\mathrm{up} \cdot \Delta p} \tag{A.6}$$

A.5.4.2 Conical inlet performance

The compound coefficient $\alpha\varepsilon$ is dependent on the Reynolds number Re_d and is plotted in Figure A.8.

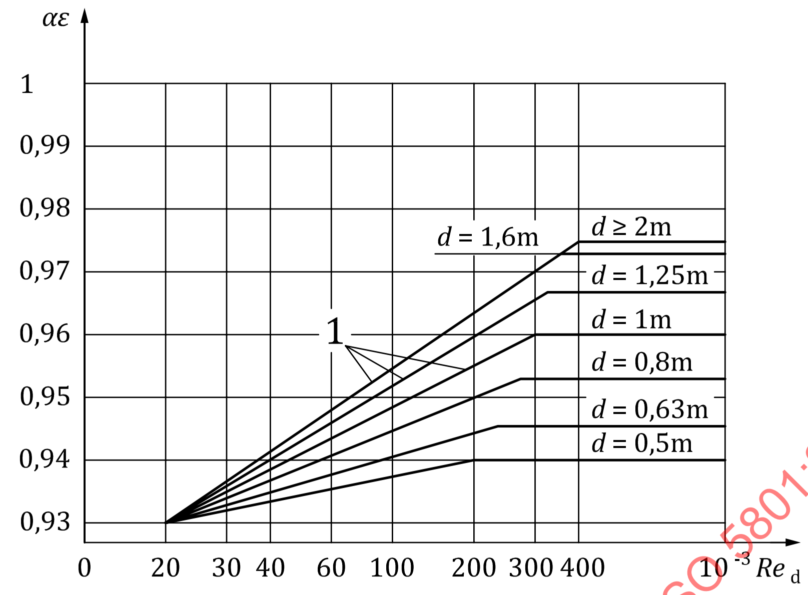
A.5.4.3 Bellmouth inlet performance

For a bellmouth inlet of the shape “one-arc-nozzle”, see Figure A.6.

$$\alpha = 1 - 0{,}004 \cdot \sqrt{\left(\frac{10^6}{Re_d}\right)} \pm 0{,}003 \text{ and } \varepsilon = 1 - 0{,}55 \cdot \frac{\Delta p}{p_1}$$

For a bellmouth inlet of the shape “two-arc-nozzle”, see Figure A.6.

$$\alpha = 0{,}99 \pm 0{,}01 \text{ and } \varepsilon = 1 - 0{,}55 \cdot \frac{\Delta p}{p_1}$$



Key

Re_d Reynolds number

$\alpha\varepsilon$ compound coefficient

1 curves for different duct diameters

NOTE For $d \leq 0{,}5$ m: $m = 0{,}010\ 00$ where m is the gradient

$c = 0{,}887\ 0$ where c is the intercept

$\alpha\varepsilon$ max. = 0,94

For 0,5 m < $d \leq 2$ 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. $= 0{,}913\ 1 + 0{,}062\ 3d - 0{,}015\ 67d^2$

For $d > 2$ m: $m = 0{,}034\ 59$

$c = 0{,}781\ 2$

$\alpha\varepsilon$ max. = 0,975.

Figure A.8 — Compound coefficient $\alpha\varepsilon$ for conical inlets

A.5.5 Uncertainties

For conical and bellmouth inlet, 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 × 105, and when no screen loading is allowed in the connection piece, is ±1,5 %. The next additional uncertainty associated with low Re_d and screen loading shall be arithmetically added to this, if applicable.

The additional uncertainty, as a percentage, due to low Re_d (i.e. 2 × 104 < Re_d < 3 × 105) is given in Formula (A.7):

$$\pm\left(\frac{2\cdot 10^4}{Re_\mathrm{d}}-\frac{1}{15}\right) \tag{A.7}$$

The additional uncertainty due to the presence of a uniform screen complying with A.5.3 is 0,5 % and shall be added arithmetically.

These uncertainties may be reduced if a calibrated value of $\alpha\varepsilon$ is used in place of the values given in A.5.4. 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 A.7 b) may then be taken as ±2 %.

A.6 Determination of flow rate using an orifice plate

A.6.1 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), in test chambers (as far as limits in ISO 5167–1 are respected) 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.

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.

A.6.2 Geometry

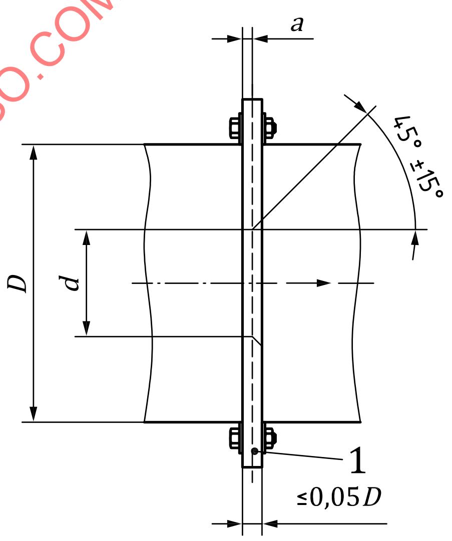
The orifice plate and the associated pressure tapings shall conform to the dimensions shown in Figure A.9.

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

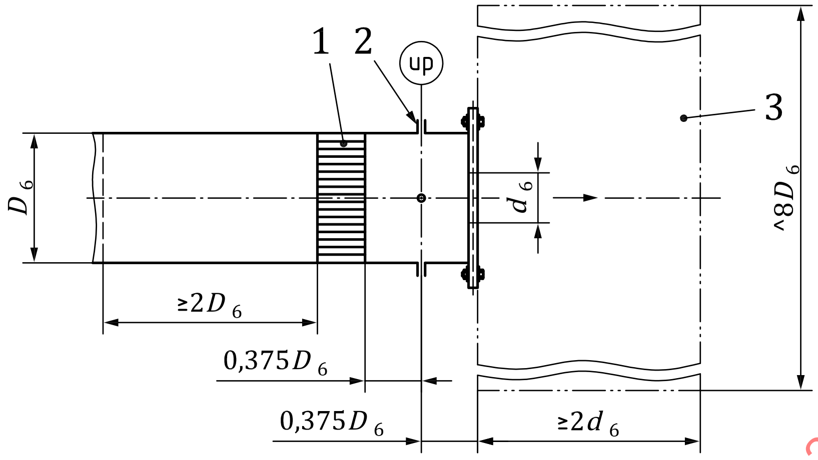
The upstream edge of the orifice shall be sharp and shall not reflect light. Any edge radius shall not exceed 0,000 4 d. These conditions may be met by machining the orifice plate, fine boring the orifice, and then finishing the upstream face by a very fine radial cut from the centre outwards.

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 + 2,3b^4).

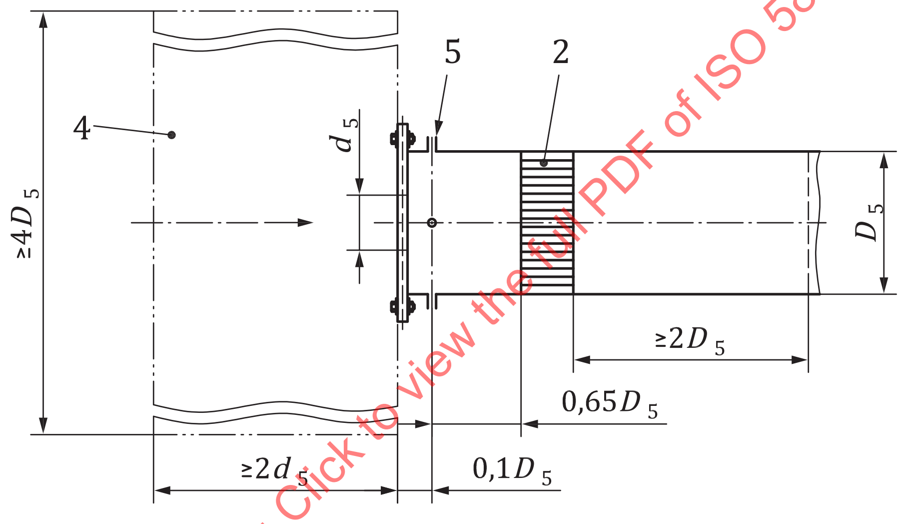
The upstream face of the orifice plate shall be flat to within 1 mm per 100 mm and its roughness, R_a, shall not exceed 0,000 1d. Any gasket sealing of the plate and the duct flange shall not project internally.



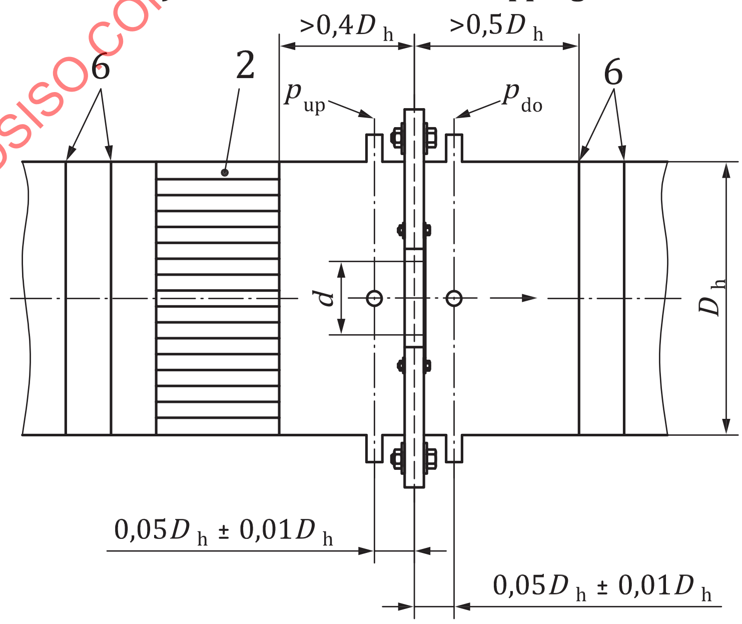
a) Details of orifice plate



b) Outlet orifice with wall tappings



c) Inlet orifice with wall tappings



d) Orifice plate in test chamber (inlet side or outlet side)

Key

1 additional thickness, if required, to stiffen the orifice plate
2 flow straightener (cell-type shown)
3 wall tappings complying with 12.8
4 no obstacles within this space
5 wall tappings
6 flow settling screens
a Dimension e given by: 0,005D < e < 0,02D

NOTE 1 If the orifice plate is held in place by a clip, then the internal diameter is ≥D_5 and the thickness ≤0,01D_5.

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

Figure A.9 — Orifice plates and assemblies

A.6.3 Ducts

The upstream duct diameter D shall be determined, to the nearest 0,003 D, as the average of 12 measurements at about 45 in three cross-sections equally distributed between the upstream tapping and the section at 0,5 D upstream. It is sufficient for the downstream side duct to be nominally cylindrical and of diameter D ± 0,03 D.

A flow straightener shall be fitted in the upstream duct. The length of the upstream and downstream ducts and the installation conditions correspond to ISO 5167–1.

Wall tappings shall be 4 in number, in accordance with 12.8, and in the locations shown in Figure A.9.

The axis of each tap shall intersect the duct axis at right angles.

A.6.4 Calculation of mass flow rate

A.6.4.1 General

$$q_m = \alpha \cdot \varepsilon \cdot \frac{\pi \cdot d^2}{4} \cdot \sqrt{2 \cdot \rho_{up} \cdot \Delta p} \tag{A.8}$$

The definitions and limitations on the quantities on the right-hand side of Formula (A.8) differ slightly according to the orifice installation adopted and are therefore considered separately for each case.

The duct diameter, D, shall be not less than 50 mm.

A.6.4.2 In duct Venturi nozzle

The installation and the use of in duct Venturi nozzle are given in ISO 5167–1.

The orifice diameter, d, shall not be less than 12,5 mm.

A.6.4.3 In duct orifice

The installation and the use of in duct orifice are given in ISO 5167–1.

A.6.4.4 Outlet orifice with wall tappings

Formula (A.9) shall apply:

$$\Delta p = p_{up} - p_a \qquad (A.9)$$

where

p_a is the ambient atmospheric pressure;

$\alpha\varepsilon$ is given by the following formula and plotted in Figure A.10;

as function of β

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

where

$$r_{\Delta p} = \frac{\Delta p}{p_a}$$

with

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

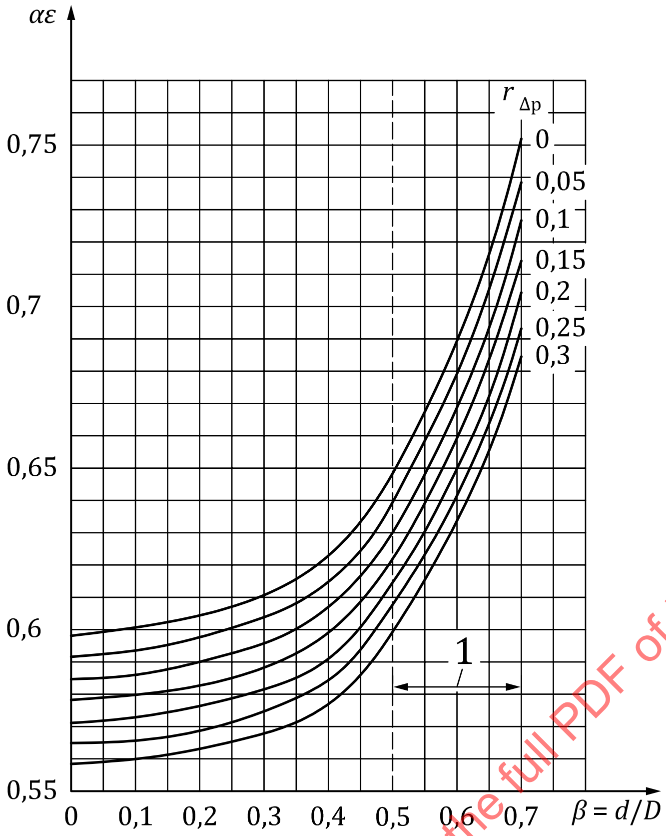
$A = 0{,}596 + (2{,}04)^{-1} \cdot \beta^{3{,}2}$ for $\beta \geq 0{,}5$

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

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

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

The uncertainty with which $\alpha\varepsilon$ is known may be taken as ±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.



Key

1 zone of reduced accuracy

Figure A.10 — Compound flow rate coefficient, $\alpha\varepsilon$, of outlet orifices with wall taps

A.6.4.5 Inlet orifice with wall tappings

Formula (A.10) shall apply:

$$\Delta p = p_a - p_{do} \tag{A.10}$$

$\rho_{up} = \rho_a$

$\beta' = d/D$ 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 A.6.4.

$\alpha = 0{,}598$

$$\varepsilon = 1 - r_{\Delta p}\left(0{,}249 - 0{,}0757 \cdot r_{\Delta p}\right)$$

$$r_{\Delta p} = \frac{\Delta p}{\left(p_a - \Delta p\right)}$$

The uncertainty with which α is known may be taken as ±1,0 % provided that $Re_D \leq 5 \times 10^4$ and $r_{\Delta p} \leq 0{,}3$.

A.6.4.6 Orifice plate with wall tappings in the test chamber

Formula (A.11) shall apply:

$$\Delta p = p_{up} - p_{do} \tag{A.11}$$

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

$$\theta_{up} = \theta_{sgup} = T_{up} + 273{,}15$$

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

$\alpha\varepsilon$ is determined in accordance with A.6.4.4.

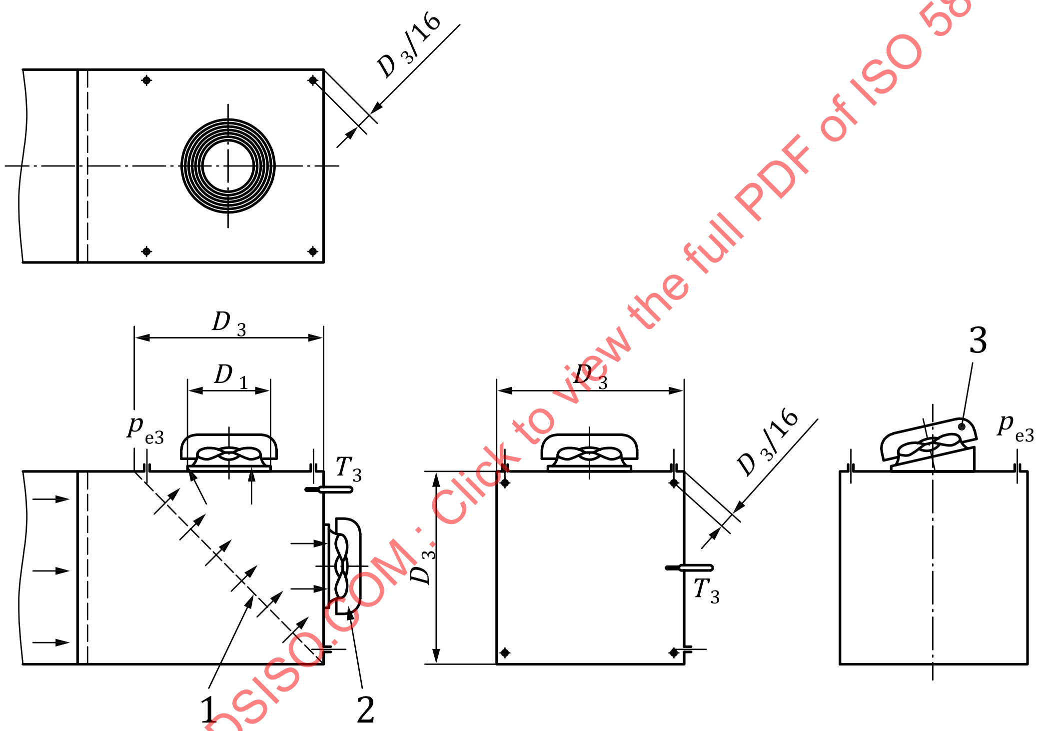
The other remarks of A.6.4.4 shall apply.

Annex B
(informative)

Fan-powered roof exhaust ventilators

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. Figure B.1 shows a possible configuration.

Figure B.1 shows one modification which involves the inclination of the final screen of 45 % maximum free area 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 $D_1 \leq 0{,}5D_3$ where D_1 is the diameter of the opening in the roof or the larger side of the rectangular opening.

Figure B.1 — Setup of a fan-powered roof exhaust ventilator on an inlet 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.

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.

Test results shall not be altered to compensate for leakage in the test chamber.

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_0 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 given in Formula (C.1):

$$p \cdot V = m \cdot R \cdot \theta$$

or (C.1)

$$p = \rho \cdot R \cdot \theta$$

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;

θ is the absolute air temperature;

ρ is the air density.

Differentiating with respect to time,

$$V \cdot \frac{\mathrm{d}p}{\mathrm{d}t} = \frac{\mathrm{d}m}{\mathrm{d}t} \cdot R \cdot \theta$$

and

$$Q = \frac{1}{\rho} \cdot \frac{\mathrm{d}m}{\mathrm{d}t} \text{ or } \rho \cdot Q = \frac{\mathrm{d}m}{\mathrm{d}t}$$

Substituting and rearranging gives:

$$\frac{\mathrm{d}p}{\mathrm{d}t} = \frac{\rho \cdot Q \cdot R \cdot \theta}{V} \text{ or } Q = \frac{V}{\rho \cdot R \cdot \theta} \cdot \frac{\mathrm{d}p}{\mathrm{d}t}$$

and Formula (C.2):

$$Q = \frac{V}{p} \cdot \frac{\mathrm{d}p}{\mathrm{d}t} \qquad \text{(C.2)}$$

or

$$Q = \frac{V}{p} \cdot \frac{\Delta p}{\Delta t}$$

where

Q is the leakage air flow rate.

Leakage rate, Q, can be determined from Formula (C.3) once the pressure decay curve [Figure C.1 c)] is known for the chamber:

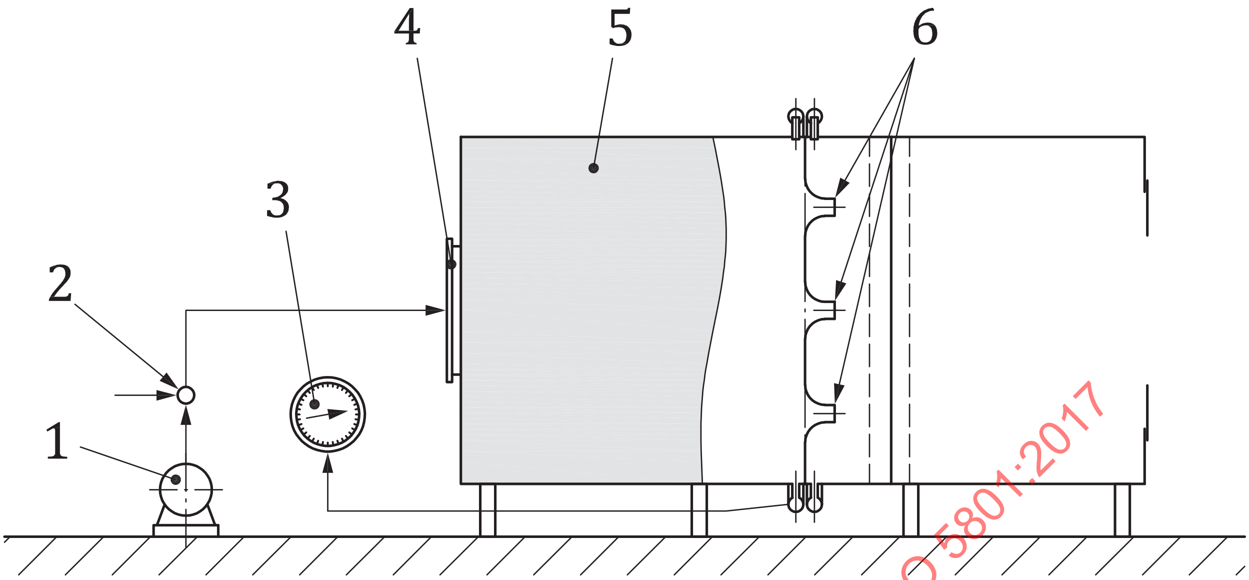
$$Q = \frac{V}{p_{\mathrm{Te}}} \cdot \frac{\Delta p_{\mathrm{Te}}}{\Delta t} \qquad \text{(C.3)}$$

where

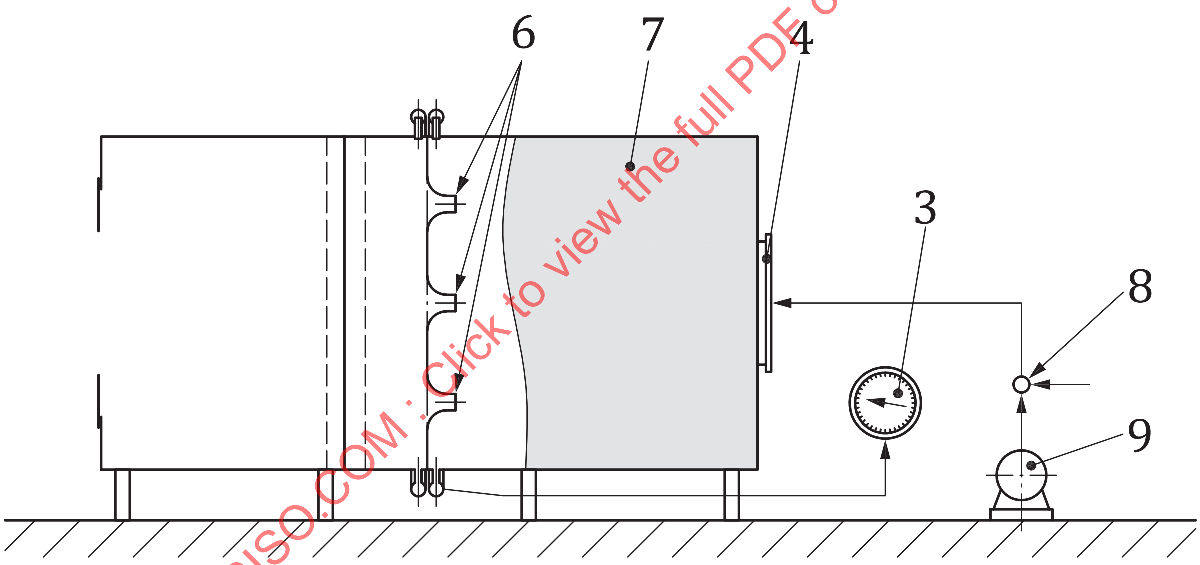
p_{Te} is the test pressure

$\frac{\Delta p_{\mathrm{Te}}}{\Delta t}$ is the slope of curve in Figure C.1 c);

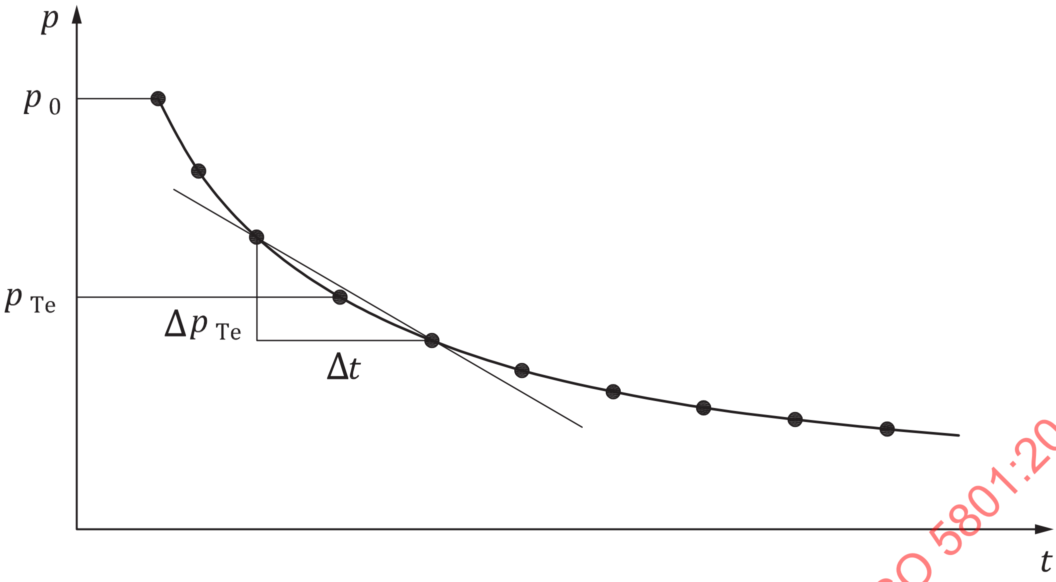
Δt_{min} is the minimum difference.



a) Setup for outlet-side chamber



b) Setup for inlet-side chamber



c) Graph showing the decay in chamber pressure with time

Key

1 fan or air compressor
2 valve
3 pressure gauge
4 test fan location
5 test chamber
6 nozzles plugged
7 chamber
8 valve test
9 vacuum pump
p pressure, in pascals
t time, in seconds

Figure C.1 — Pressure decay method for leakage test

C.2.2 Procedure

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.

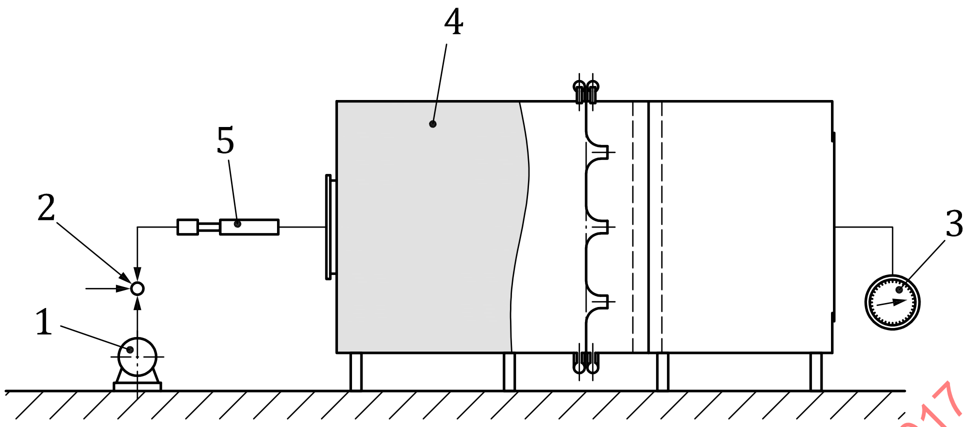
At time $t = 0$, start a stopwatch and record the pressure at periodic time intervals (a minimum of three readings is recommended) to get a decay curve as in Figure C.1 c). Continue to record until the pressure reaches a state in which the pressure does not change significantly.

Quick pressure changes indicate substantial leakage which shall 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 shall 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

C.4.1 General

For test chambers divided in two parts by a partition, like multiple-nozzle chambers, the single step leakage test methods proposed in 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.4.2 First phase

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.

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.

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.

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_{Sa}, is measured together with the differential pressure across the nozzle wall, Δp_a.

C.4.3 Second phase

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.

All the nozzles in the nozzle wall are sealed.

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.

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_{Sb}, together with the differential pressure across the nozzle wall, Δp_b.

Solving Formula (C.4), 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, A_W, can be estimated, in the same units used for A_{tn}:

$$\begin{aligned} \sqrt{dp_b}\,(A_{tn}+A_W) &= \sqrt{p_{Sa}}\cdot A_C \\ \sqrt{dp_b}\cdot A_W &= \sqrt{p_{Sb}}\,(A_{tn}+A_W) \end{aligned} \qquad \text{(C.4)}$$

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 Formula (C.5), 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.

$$Q_L = Q_C + Q_W = A_C\cdot\sqrt{\frac{2p_{sf}}{\rho}} + A_W\cdot\sqrt{\frac{2\Delta p}{\rho}} \qquad \text{(C.5)}$$

Annex D
(informative)

Fan outlet elbow in the case of a non-horizontal discharge axis

In the case of centrifugal fans, test configuration 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 shall be the smallest possible.

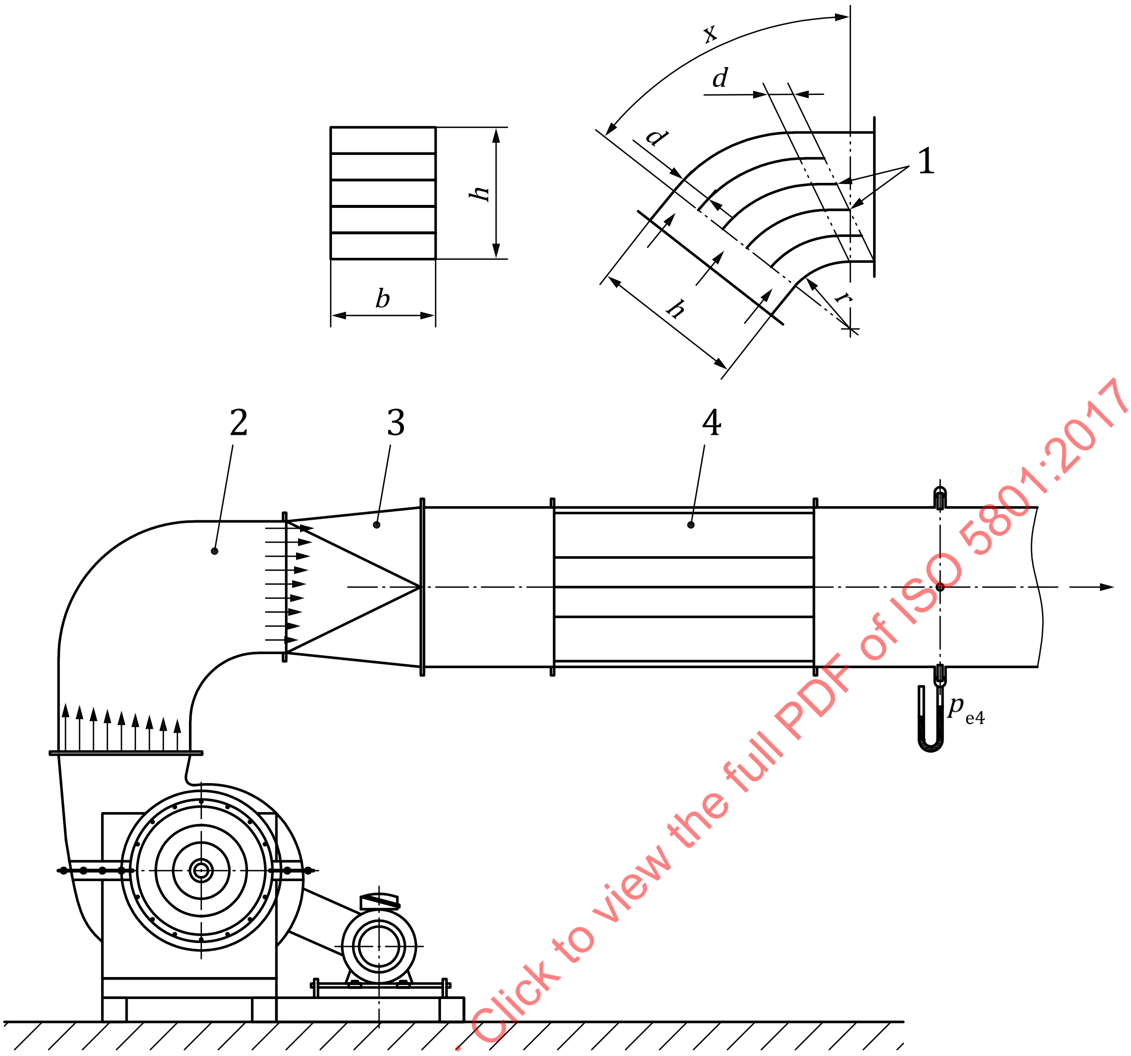
The elbow section shall be located between sections A_2 and A_4 and shall be of a uniform cross-section with splitter vanes.

A conventional friction-loss coefficient is given by Formula (D.1):

$$(\xi_C)_4 = \left[\frac{\chi}{2\pi} \cdot \left(\frac{h}{b}\right)^{\frac{1}{6}}\right] \left(\frac{A_4}{A_C}\right)^2 \tag{D.1}$$

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 c.



Key

1 turning vanes

2 elbow

3 rectangular to round transition section

4 star type flow straightener

d = $h/5$

r = 2,5d

Figure D.1 — Dimensions of outlet elbow for testing large centrifugal fans

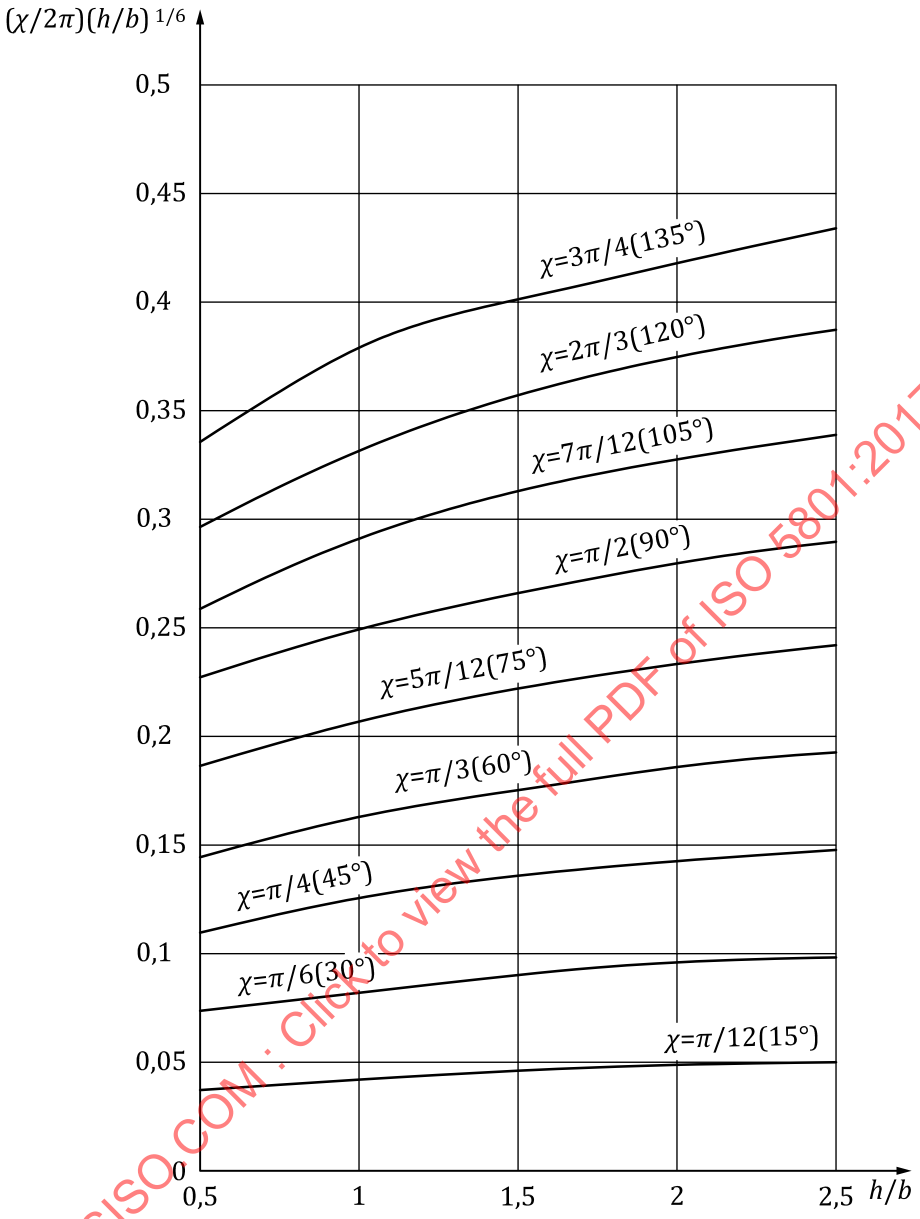


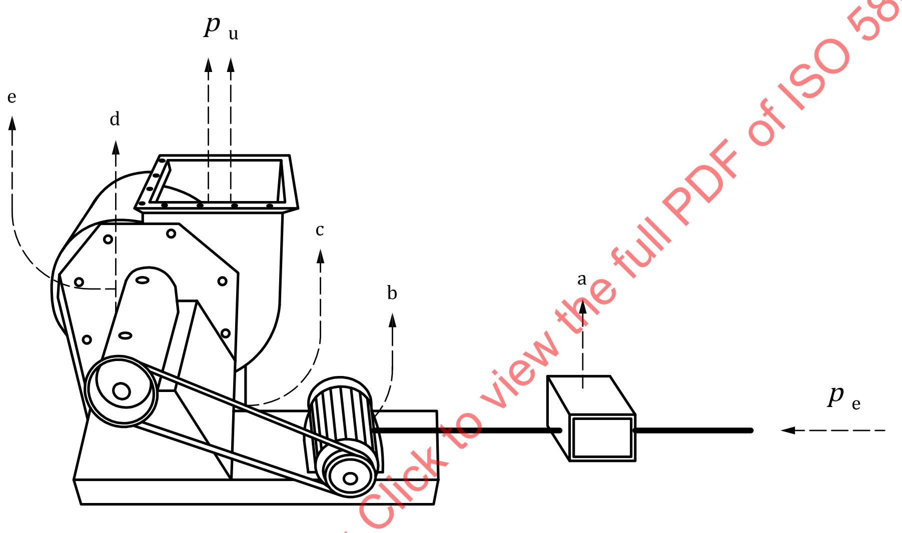
Figure D.2 — Plot of $(\chi/2\pi)(h/b)1/6$ against h/b for calculating the pressure loss in an outlet elbow

Annex E
(informative)

Input power calculation for driven fans at design point

E.1 General

Economic 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. There is, therefore, a need for an agreed approach to the calculation of the electrical input power, P_e. Figure E.1 shows a typical V-belt driven fan and the various losses that occur.



Key

P_e electrical input power

P_u volume flow and pressure (air power)

a Variable speed device loss (heat).

b Motor losses (heat).

c Belt losses (heat).

d Bearing losses (heat).

e Impeller and casing aerodynamic losses (heat).

Figure E.1 — Typical belt driven fan showing power losses

E.2 Power consumption calculation

E.2.1 General

The electrical input power consumed by a fan installation is made up of a number of elements. These may be summarized as follows in E.2.2 to E.2.4.

E.2.2 Impeller power

Impeller power is the mechanical power supplied to the fan impeller in a cased fan. This is denoted as P_r and is expressed in Watts or kilowatts. P_u is the fan air power and fan impeller efficiency is given in Formula (E.1), expressed as a decimal:

$$\eta_r = \frac{P_u}{P_r} \tag{E.1}$$

This is directly applicable to fan arrangements 4, 5, 15 and 16 (see ISO 13349).

E.2.3 Fan shaft power

Fan shaft power is the mechanical power supplied to the fan shaft. This is denoted as P_a and is expressed in Watts or kilowatts. P_u is the fan air power and fan shaft efficiency is given in Formula (E.2), expressed as a decimal:

$$\eta_a = \frac{P_u}{P_a} \tag{E.2}$$

This is directly applicable to all other fan arrangements, i.e. 1 to 3, 6 to 14 and 17 to 19 (see ISO 13349).

It differs from the impeller power by the addition of power losses in the fan bearings as a result of friction.

E.2.4 Rolling element bearing friction power

It is preferable to obtain these losses by tests on the difference between arrangements 1 and 4 (see ISO 13349). If this cannot be achieved, then the losses can be estimated from the formulae below.

These losses can be obtained from Formula (E.3):

$$P_b = \frac{\pi}{30 \cdot M \cdot N} \tag{E.3}$$

where

P_b is the power loss in bearings, in Watts;

M is the total frictional moment of the bearings, in Newton metres;

N is the impeller/shaft rotational speed, in r/min.

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 Formula (E.4):

$$M = 0{,}5 \cdot \mu \cdot d_s \cdot C_d \tag{E.4}$$

where

M is the total frictional moment of the bearing, in Newton metres;

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_s is the bearing(s) bore diameter(s), in metres.

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 — full complement	 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, see the 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.

If 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 formula:

$$M_{\text{seal}} = k_1 \cdot d_{\text{s}}^{\text{a}} + k_2 \qquad \text{(E.5)}$$

where

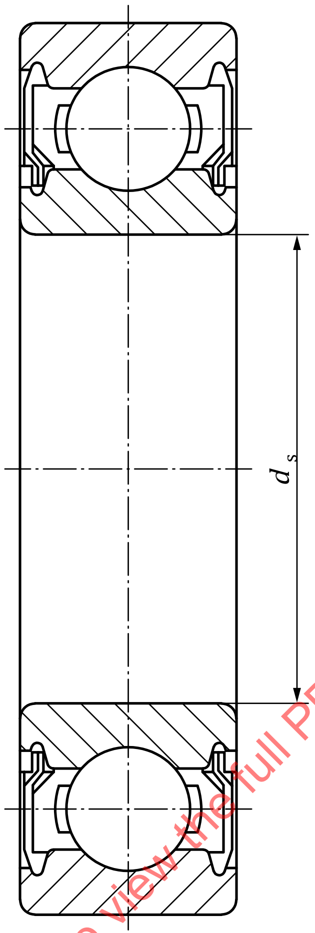
M_{seal} is the frictional moment of seals, in Newton metres;

k_1 is a constant dependent on bearing type;

k_2 is a constant, in Newton metres, dependent on bearing type and seal type;

d_{s} is the bore diameter of bearing, in metres (see Figure E.2);

a is an exponent dependent on bearing and seal type.



Key

d_s bore diameter of bearing

Figure E.2 — Section through a sealed rolling element bearing

a can vary between 0 and 2,3; k_1 can vary between 0 and 0,06 and k_2 can vary between 0 and 50. For confirmation of these values, see the information supplied by the bearing manufacturer, if necessary, noting that different symbols can be used.

As given in Formula (E.6):

$$P_b = P_a - P_r \tag{E.6}$$

the efficiency may be defined as fan bearing efficiency, given in Formula (E.7):

$$\eta_b = \frac{P_r}{P_a} = 1 - \frac{P_b}{P_a} \tag{E.7}$$

and

$$\eta_r \cdot \eta_b = \eta_a$$

NOTE The total moment of the fan bearings is the numerical sum of the individual moments ignoring the sign (the direction of the moments is immaterial).

E.2.5 Transmission power

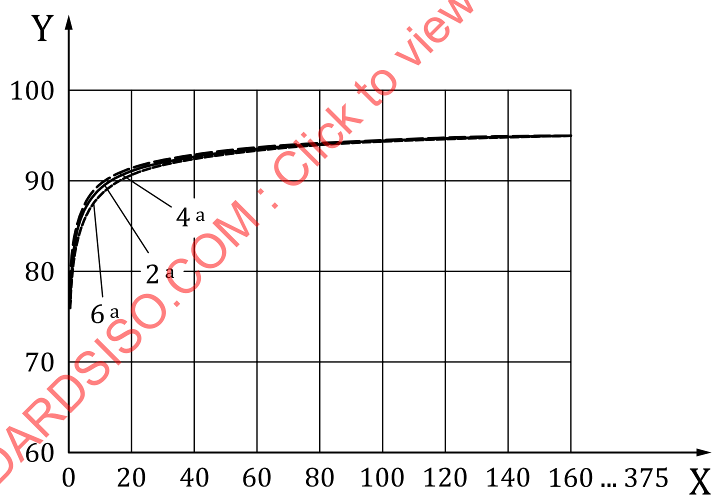
Many fans, especially in the heating, ventilation, air conditioning and refrigeration (HVACR) sector, are driven through pulleys and V-belts. This gives flexibility to fan manufacturers, who can cover a wide duty range with a limited number of models. The system designer can take comfort in the thought that if his or her system resistance calculations prove to be wrong, 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 suffers. Whereas a well-designed drive can exceed 95 % in its efficiency, the provision of additional belts for a direct-online start can often reduce this considerably. A “soft” start can be part of a better solution.

If fans are driven through flexible couplings (see arrangements 7, 8, 9 and 17 in ISO 13349), these are normally assumed to have an efficiency of 97 % unless figures are available from the coupling supplier.

E.2.6 Motor power

Perhaps the most common type of motor used in fan installations (certainly above an output of 1 kW) is the squirrel cage a.c. induction design. It is robust, reliable, requires minimum maintenance and is relatively inexpensive. 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. Three efficiency levels are standardized in IEC 60034-30. These are identified as IE1, IE2 and IE3. The minimum efficiencies at the nominal rating are shown in Figures E.3, E.4 and E.5. The efficiency for actual motors at partial loads (around 75 % of nameplate rating) can sometimes be greater than that at full load. This is contrary to earlier designs. It is important to use the efficiency at the actual absorbed power, which may be calculated using any of the methods described in IEC 60034-2-1.



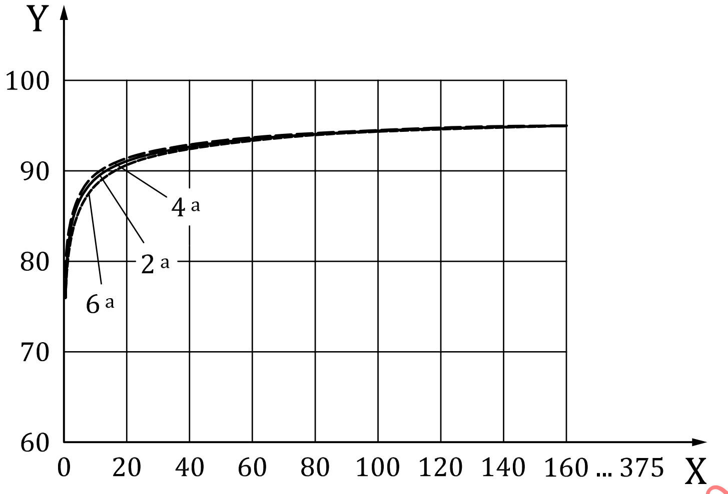
Key

X power, in kW

Y efficiency, in %

a Number of poles.

Figure E.3 — Nominal limits (%) for Standard Efficiency (IE1) 50 Hz electric motors



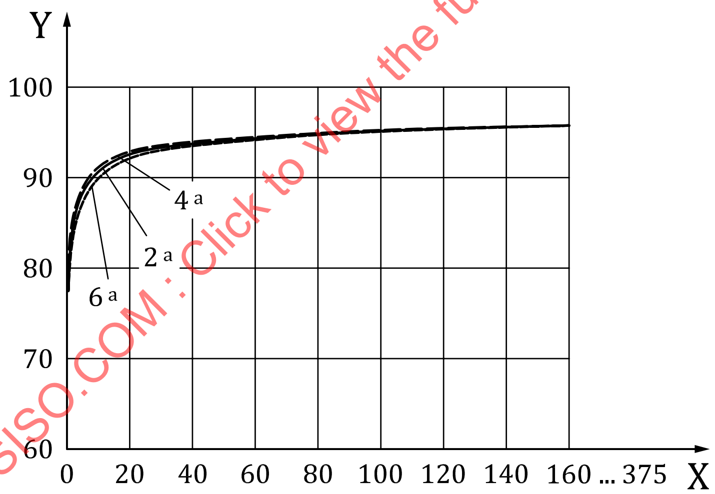
Key

X power, in kW

Y efficiency, in %

a Number of poles.

Figure E.4 — Nominal limits (%) for High Efficiency (IE2) 50 Hz electric motors



Key

X power, in kW

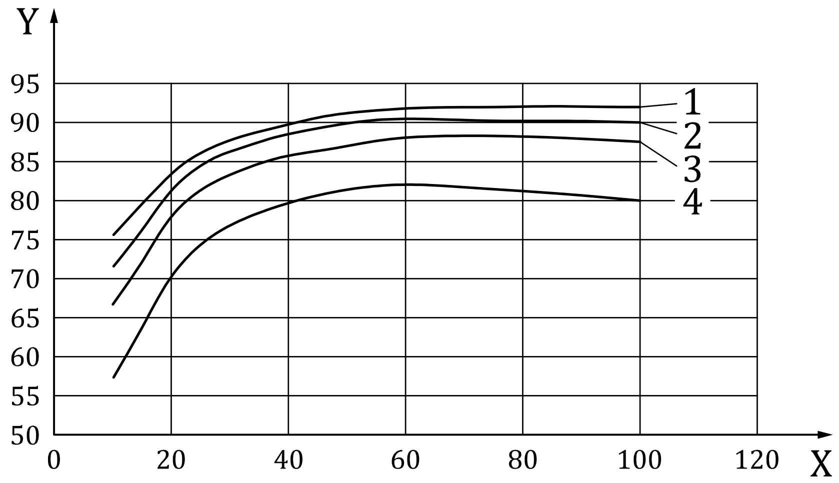
Y efficiency, in %

a Number of poles.

Figure E.5 — Nominal limits (%) for Premium Efficiency (IE3) 50 Hz electric motors

E.2.7 Controls/power loss

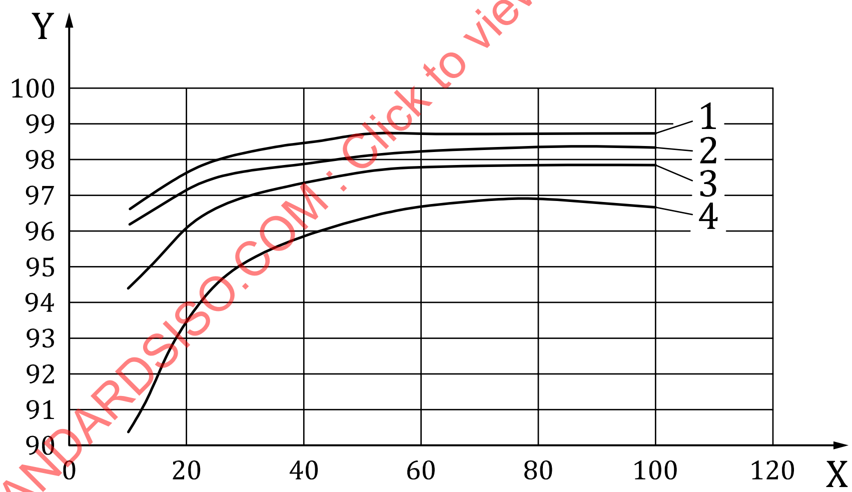
This is often ignored, especially with inverters. The efficiency of controls at high turn-down ratios can be much less than 100 %, although, of course, powers absorbed by the fan are also small. Figures E.6, E.7 and E.8 are typical examples of a 30 kW motor.



Key

X nominal torque (%)
Y efficiency (%)
1 100 % speed
2 75 % speed
3 50 % speed
4 25 % speed

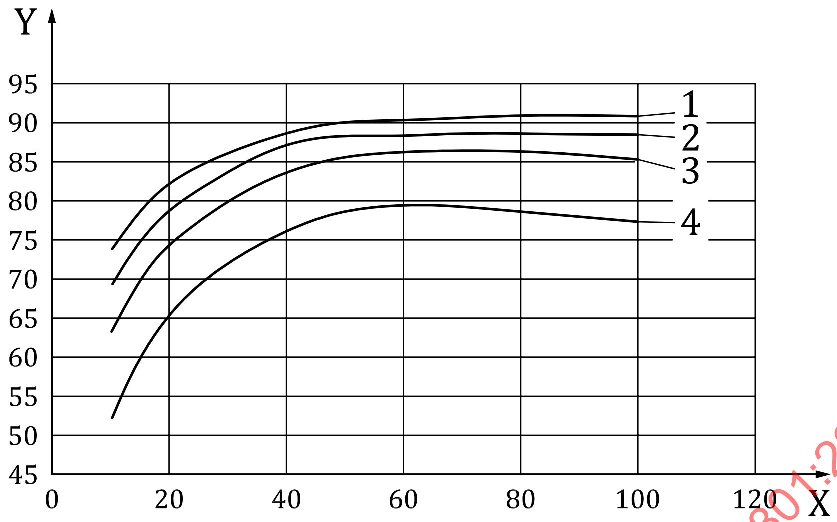
Figure E.6 — Indicative efficiency of a typical motor at various loads



Key

X nominal torque (%)
Y efficiency (%)
1 100 % speed
2 75 % speed
3 50 % speed
4 25 % speed

Figure E.7 — Indicative efficiency of a typical variable frequency drive



Key

X nominal torque (%)

Y efficiency (%)

1 100 % speed

2 75 % speed

3 50 % speed

4 25 % speed

Figure E.8 — Efficiency of a typical motor and variable frequency drive

E.3 Mains power required

The electrical power input from the mains may be calculated using Formula (E.8):

$$P_e = \frac{q_{Vsg1} \cdot p_f \cdot k_P}{\eta_r \cdot \eta_b \cdot \eta_T \cdot \eta_{mot}}$$

and (E.8)

$$P_{ed} = \frac{P_e}{\eta_c}$$

where

k_P is the compressibility coefficient;

P_e is the electrical input power, in kilowatts, alternatively in Watts;

q_{Vsg1} is the flow rate, in cubic metres per second or litres per second (m^3/s or l/s);

p_f is the fan pressure, in kilopascals or Pascal's;

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_{mot} 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, P_e is in Watts; if fan pressure is expressed in kilopascals, P_e is in kilowatts.

NOTE 2 $\eta_r \cdot \eta_b = \eta_a$, where η_a is the fan shaft efficiency.

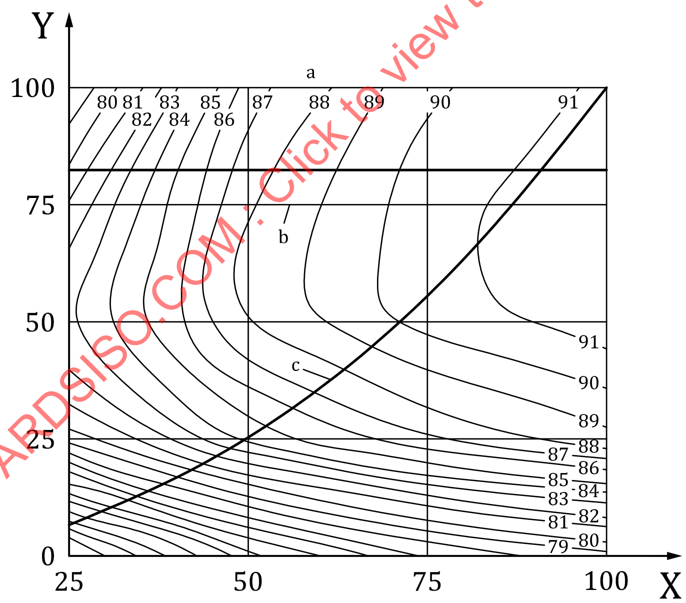
NOTE 3 Fan pressure can also be defined on a static basis provided η_r is also calculated on the same basis. Fan static efficiency is not theoretically correct as it can never be 100 % or 1.

NOTE 4 These calculations are usually conducted at the enquiry stage before an audit can be carried out.

All duties and values should be for the appropriate installation category.

E.4 Presenting results of a typical motor and VFD while driving a fan

The combined efficiency of an induction motor and VFD while driving a fan depends on how the fan pressure varies with flow rate. For many systems, $p_f \propto {q_{Vsg}}^2$ (see Figure E.9). By plotting the torque, t_m required from the motor against speed, it is possible to deduce that torque $t_m \propto$ speed n^2 or N^2. There are, however, other possibilities, e.g. torque required may not necessarily follow the square law, while viscosity effects can reduce the speed index to less than 2. It is also possible that there are fixed resistance elements.



Key

X speed (% nominal)

Y torque (% nominal)

a System efficiency.

b Constant torque load application.

c Variable torque load from an idealised fan system $t_m \propto n^2$.

Figure E.9 — Typical efficiency of motor and VFD if applied to a fan (adapted from P. Angers)

