INTERNATIONAL STANDARD

ISO 5802

First edition
2001-07-15

Industrial fans — Performance testing *in situ*

Ventilateurs industriels — Essai de fonctionnement in situ

Reference number
ISO 5802:2001(E)

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Printed in Switzerland

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Foreword

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

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 3.

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

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

International Standard ISO 5802 was prepared by Technical Committee ISO/TC 117, *Industrial fans*.

Annexes A to E form a normative part of this International Standard.

Introduction

The need to revise existing methods of site testing has been apparent for some time. Bearing in mind the extent of these revisions, it was felt appropriate to expand the method of site testing into a "stand-alone" document. This would enable the velocity area methods to be fully detailed for all commonly encountered airway cross-sections. It would also allow the addition of descriptive annexes covering the selection of suitable measuring stations and instrument calibration.

In accordance with recent International agreements, it will be noted that fan pressure is now defined as the difference between stagnation pressure at the fan inlet and outlet. Stagnation pressure is the absolute pressure which would be measured at a point in a flowing gas if it were brought to rest isentropically. For Mach numbers less than 0,2 the gauge stagnation pressure is within 0,6 % of the total pressure.

Less emphasis is placed on the use of "fan static pressure" as this is a conventional quantity only. It is to be anticipated that its use will cease with time. All fluid losses are essentially losses in stagnation pressure and this has been reflected in the definitions now specified.

It should be recognized that the performance of a fan measured under site conditions will not necessarily be the same as that determined from tests using standardized airways. The reasons for such differences are not only due to the inherently lower accuracy of a site test, but also due to the so-called "system effect factor" or "installation effect", where the ducting connections at fan inlet and/or outlet modify its performance. The need for good connections cannot be understated. This International Standard specifies the use of "common parts" immediately adjoining the fans for the consistent determination of pressure and also to ensure that air/gas is presented to the fan as a symmetrical velocity profile free from swirl and undue distortion. Only if these conditions are met, will the performance under site conditions equate with those measured in standardized airways.

It should also be noted that this International Standard specifies the positioning of velocity-area measuring points according to log-Tchebycheff or log-linear rules. Arithmetic spacing can lead to considerable error unless a very high number of point readings are taken. (These would then have to be plotted graphically and the area under the curve obtained using planimetry. The true average velocity would be this area divided by the dimensional ordinates).

It is outside the scope of this International Standard to assess the additional uncertainty where the lengths of straight duct either side of the measuring station are less than those specified in annex C. Guidance is, however, given in ISO/TR 5168 and ISO 7194, from which it will be seen that where a significant radial component exists, uncertainties can considerably exceed the normally anticipated 4 % at 95 % confidence levels.

Industrial fans — Performance testing *in situ*

1 Scope

This International Standard specifies tests for determining one or more performance characteristics of fans installed in an operational circuit when handling a monophase fluid.

2 Normative references

The following normative documents contain provisions which, through reference in this text, constitute provisions of this International Standard. For dated references, subsequent amendments to, or revisions of, any of these publications do not apply. However, parties to agreements based on this International Standard are encouraged to investigate the possibility of applying the most recent editions of the normative documents indicated below. For undated references, the latest edition of the normative document referred to applies. Members of ISO and IEC maintain registers of currently valid International Standards.

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

ISO 5801:1997, *Industrial fans — Performance testing using standardized airways.*

IEC 60034-1, *Rotating electrical machine — Part 1: Rating and performance.*

IEC 60051-8, *Direct acting indicating analogue electrical measuring instruments and their accessories — Part 8: Special requirements for accessories.*

3 Terms, definitions and symbols

3.1 Terms and definitions

For the purposes of this International Standard, the following terms and definitions apply.

The quantities referred to are time-averaged mean values. Fluctuations which affect the quantities being measured may be accounted for by repeating measurements at appropriate time intervals. Mean values may then be calculated which are taken as the steady-state value.

3.1.1
air
air or other gas, except when specifically referred to as atmospheric air

3.1.2
standard air
atmospheric air having a density of exactly 1,2 kg·m^{-3}

NOTE Atmospheric air at a temperature of 16 °C, a pressure of 100 000 Pa and a relative humidity of 65 %, has a density of 1,2 kg·m^{-3}, but these conditions are not part of the definition.

3.1.3
fan
rotary machine which maintains a continuous flow of air at a pressure ratio not normally exceeding 1,3

3.1.4
impeller
rotating part of a fan which, by means of its blades, transfers energy to the air

3.1.5
casing
those stationary parts of a fan which direct the flow of air from the fan inlet opening(s) to the fan outlet opening(s)

3.1.6
duct
airway in which the air velocity is comparable with that at the fan inlet or outlet

3.1.7
chamber
airway in which the air velocity is small compared with that at the fan inlet or outlet

3.1.8
transition piece
section
airway along which there is a gradual change of cross-sectional area and/or shape

3.1.9
test enclosure
room, or other space protected from draught, in which the fan and test airways are situated

3.1.10
area of the conduit section
A_x
area of the conduit at section x

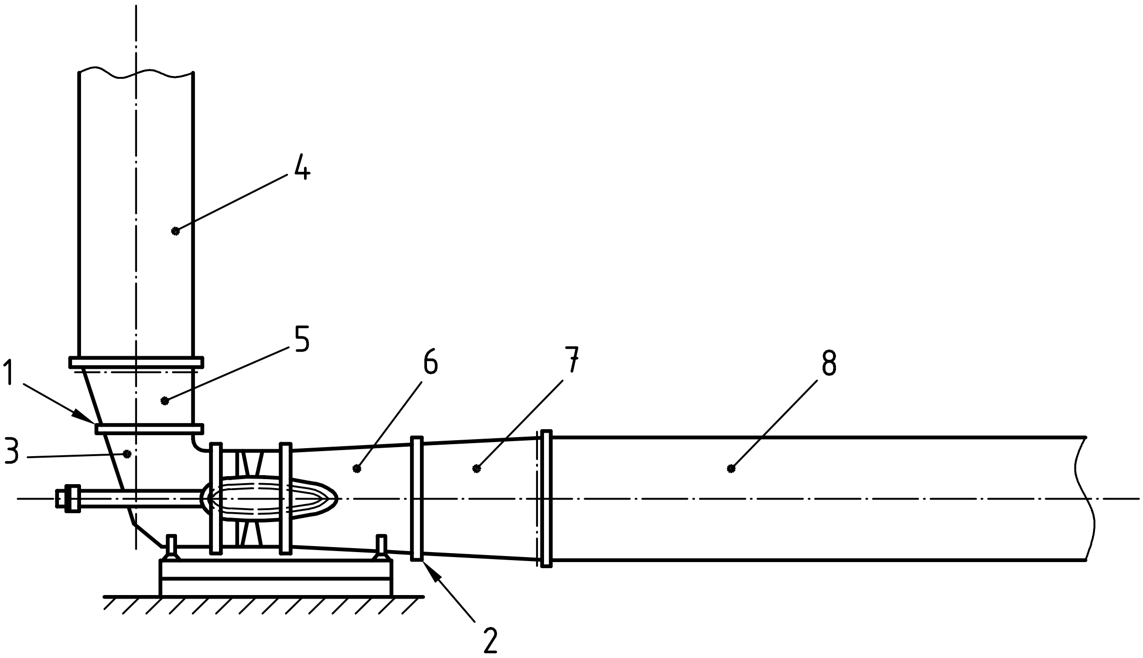
3.1.11
fan inlet area
A_1
by convention, the gross area in the inlet plane inside the casing

NOTE The fan inlet plane should be taken as that surface bounded by the upstream extremity of the air moving device. In this International Standard the fan inlet plane is indicated by plane 1 (see Figure 1).

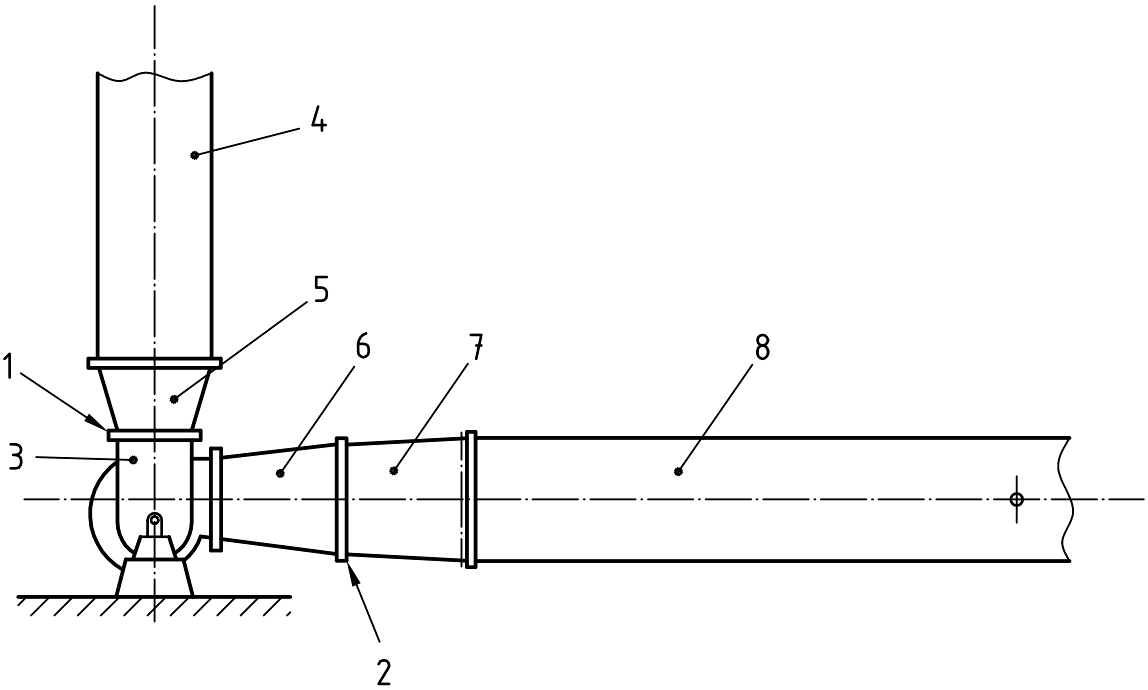
3.1.12
fan outlet area
A_2
by convention, the gross area in the outlet plane inside the casing without deduction for motors, fairings or other obstructions

NOTE The fan outlet plane should be taken as that surface bounded by the downstream extremity of the air moving device. In this International Standard the outlet is indicated by plane 2 (see Figure 1).

3.1.13
temperature
t
air or fluid temperature measured by a temperature sensor



a) Axial fan



b) Centrifugal fan

Key

1 Plane 1
2 Plane 2
3 Inlet box
4 Inlet duct
5 Transition
6 Diffuser
7 Transition
8 Outlet duct

Figure 1 — Location of pressure measurement planes for site testing

3.1.14
absolute temperature
θ
thermodynamic temperature measured above absolute zero

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

3.1.15
stagnation temperature at a point
θ_{sg}
absolute temperature which results if an ideal gas flow is brought to rest isentropically without addition of energy or heat

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

3.1.16
static or fluid temperature
θ
absolute temperature of a thermal sensor moving at the fluid velocity

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

where v is the fluid velocity (m/s)

3.1.17
dry bulb temperature
t_d
air temperature measured by a dry temperature sensor in the test enclosure, near the fan inlet or airway inlet

3.1.18
wet bulb temperature
t_w
air temperature measured by a temperature sensor covered by a water-moistened wick and exposed to air in motion

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

3.1.19
stagnation temperature at a section
θ_{sgx}
mean value in time of the stagnation temperature averaged over the area of the specified airway cross section

3.1.20
static or fluid temperature at a section
θ_x
mean value in time of the static or fluid temperature averaged over the area of the specified airway cross section

3.1.21
specific gas constant
R
for an ideal gas, the equation of state is written

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

3.1.22
inlet stagnation temperature
θ_{sg1}
temperature in the test enclosure near the fan inlet or the inlet duct at a section where the fluid velocity is less than 25 m/s

NOTE In this case the stagnation temperature may be considered equal to the ambient temperature

$$\theta_{sg1} = \theta_a = t_a + 273{,}15$$

3.1.23
isentropic exponent
κ
for an ideal gas and an isentropic process

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

3.1.24
specific heat at constant pressure
c_p
for an ideal gas:

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

3.1.25
specific heat at constant volume
c_V
for an ideal gas

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

3.1.26
compressibility factor
Z

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

and Z is a function of the ratios $\frac{p}{p_c}$ and $\frac{\theta}{\theta_c}$

where

p_c is the critical pressure of the gas

θ_c is the critical temperature of the gas

NOTE For an ideal gas Z = 1.

3.1.27
absolute pressure at a point
p
pressure measured from absolute zero, which is exerted at a point at rest relative to the air around it

3.1.28
atmospheric pressure
p_a
absolute pressure of the free atmosphere at the mean altitude of the fan

3.1.29
gauge pressure
p_e
value of a pressure when the datum pressure is the atmospheric pressure at the point of measurement

NOTE It may be negative or positive:

$$p_e = p - p_a$$

3.1.30
absolute stagnation pressure at a point
p_{sg}
absolute pressure which would be measured at a point in a flowing gas if it were brought to rest via an isentropic process

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

where Ma is the Mach number at this point

3.1.31
dynamic pressure at a point
p_d
pressure calculated from the velocity v and the density ρ of the air at the point

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

3.1.32
total pressure at a point
p_t
absolute stagnation pressure minus the atmospheric pressure

$$p_t = p_{sg} - p_a = p_e + p_d$$

NOTE When the Mach number is less than 0,2, the Mach factor is less than 1,01 and the absolute stagnation pressure p_{sg} is very close to the sum of the gauge pressure, the atmospheric pressure and the dynamic pressure:

$$p_{sg} \cong p_e + p_a + p_d$$

3.1.33
average gauge pressure at a section x
p_{ex}
mean value in time of the gauge pressure averaged over the area of the specified airway cross section

3.1.34
average absolute pressure at a section x
p_x
mean value in time of the absolute pressure averaged over the area of the specified airway cross section

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

3.1.35
conventional dynamic pressure at a section x
p_{dx}
dynamic pressure calculated from the average velocity and the average density at the specified airway cross section

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

3.1.36
fan dynamic pressure
p_{dF}
conventional dynamic pressure at the fan outlet calculated from the mass flow, the average air density at the outlet and the fan outlet area

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

3.1.37
absolute stagnation pressure at a section x
p_{sgx}
sum of the conventional dynamic pressure p_{dx} corrected by the Mach factor coefficient F_{Mx} at the section and the average absolute pressure p_x

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

NOTE The absolute stagnation pressure may be calculated by the expression:

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

3.1.38
average total pressure at a section x
p_{tx}
when the Mach number is less than 0,122, the Mach factor F_M may be neglected so

$$p_{tx} = p_{ex} + p_{dx} = p_{sgx} - p_a$$

3.1.39
fan pressure
p_F
difference between the stagnation pressure at the fan outlet and the stagnation pressure at the fan inlet

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

3.1.40
fan static pressure
p_{sF}
conventional quantity defined as the fan pressure minus the fan dynamic pressure corrected by the Mach factor at the fan outlet area

$$p_{sF} = p_{sg2} - p_{dF} F_{M2} - p_{sg1} = p_2 - p_{sg1}$$

3.1.41
Mach number at a point
Ma
ratio of the fluid velocity at a point and the velocity of sound in the fluid

NOTE For an ideal gas:

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

3.1.42
Mach number at a section x
Ma_{x}
ratio of the fluid average velocity by the velocity of sound at the specified airway cross section

$$Ma_{\mathrm{x}} = \frac{v_{\mathrm{mx}}}{\sqrt{\kappa R_{\mathrm{w}} \theta_{\mathrm{x}}}}$$

3.1.43
Mach factor
F_{M}
correction factor which is applied to the dynamic pressure at a point given by the expression

$$F_{\mathrm{M}} = \frac{p_{\mathrm{sg}} - p}{p_{\mathrm{d}}}$$

NOTE The Mach factor may be calculated by:

$$F_{\mathrm{M}} = 1 + \frac{Ma^2}{4} + \frac{Ma^4}{40} + \frac{Ma^6}{1600} + \ldots \qquad \text{for } \kappa = 1,4$$

3.1.44
stagnation inlet density
ρ_{sg1}
density calculated from the stagnation inlet pressure p_{sg1} and the stagnation inlet temperature θ_{sg1}

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

3.1.45
average density at a section x
ρ_{x}
fluid density calculated from the absolute pressure p_{x} and the static temperature θ_{x}

$$\rho_{\mathrm{x}} = \frac{p_{\mathrm{x}}}{R_{\mathrm{w}}\, \theta_{\mathrm{x}}}$$

3.1.46
mean density
ρ_{m}
arithmetic mean value of inlet and outlet densities

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

3.1.47
mean mass flowrate at a section
q_m
mean value over time of the mass of fluid which passes through the specified airway cross section per unit of time

NOTE The mass flow will be the same at all cross sections within the fan airway system, apart from leakage. When the fan is not gastight, the mass flow is taken as either that at the fan inlet or outlet, as appropriate.

3.1.48
inlet stagnation volume flow
$q_{V\mathrm{sg1}}$
mass flowrate divided by the stagnation inlet density

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

3.1.49
outlet stagnation volume flow
$q_{V\mathrm{sg2}}$
mass flowrate divided by the stagnation outlet density

$$q_{V\mathrm{sg2}} = \frac{q_m}{\rho_{\mathrm{sg2}}}$$

3.1.50
volume flow at a section x
$q_{V\mathrm{x}}$
mass flow at the specified airway cross section divided by the corresponding mean value in time of the average density at that section

$$q_{V\mathrm{x}} = \frac{q_m}{\rho_{\mathrm{x}}}$$

3.1.51
average velocity at a section x
v_{mx}
volume flow at the specified airway cross section divided by the cross-sectional area A

$$v_{\mathrm{mx}} = \frac{q_{V\mathrm{x}}}{A_{\mathrm{x}}}$$

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

3.1.52
fan work per unit mass
y
mechanical energy increment per unit mass of the fluid passing through the fan

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

NOTE y may be calculated as in 3.1.57.

3.1.53
fan static work per unit mass
y_s

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

3.1.54
fan pressure ratio
$r_{\mathrm{F}p}$
ratio of the average absolute stagnation pressure at the outlet section of a fan to that at its inlet section

$$r_{\mathrm{F}p} = \frac{p_\mathrm{sg2}}{p_\mathrm{sg1}}$$

3.1.55
density ratio of inlet density to mean density
k_ρ
fluid density at the fan inlet divided by the mean density in the fan

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

3.1.56
compressibility coefficient
k
ratio of the mechanical work done by the fan on the air to the work that would be done on an incompressible fluid with the same mass flow, inlet density and pressure ratio

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

NOTE 2 k is given by the expression:

$$k = \frac{(\kappa - 1)\, \rho_\mathrm{sg1}\, P_\mathrm{r} \log_{10} r_{\mathrm{F}p}}{\kappa\, q_m\, p_\mathrm{F} \log_{10}\left[1 + \dfrac{(\kappa - 1)\, \rho_\mathrm{sg1}\, P_\mathrm{r}\, (r - 1)}{\kappa\, q_m\, p_\mathrm{F}}\right]}$$

3.1.57
fan air power
P_u
conventional output power which is the product of the mass flow by the fan work per unit mass or the product of the inlet volume flow, the compressibility coefficient k and the fan pressure

$$P_\mathrm{u} = q_m\, y \cong q_{V\mathrm{sg1}}\, p_\mathrm{F}\, k$$

3.1.58
fan static air power
P_us
conventional output power which is the product of the mass flow q_m by the fan static work per unit mass or the product of the inlet volume flow, the compressibility coefficient k and the fan static pressure p_sF

$$P_\mathrm{us} = q_m\, y_\mathrm{s} \cong q_{V\mathrm{sg1}}\, k\, p_\mathrm{sF}$$

3.1.59
impeller power
P_r
mechanical power supplied to the fan impeller

3.1.60
fan shaft power
P_a
mechanical power supplied to fan shaft

3.1.61
motor output power
P_o
shaft power output of the motor or other prime mover

3.1.62
motor input power
P_e
electrical power supplied at the terminals of an electric motor drive

NOTE With other drive forms it is not usual to express the input to the prime mover in terms of power.

3.1.63
rotational speed
N
number of revolutions of the fan impeller per minute

3.1.64
rotational frequency
n
number of revolutions of the fan impeller per second

3.1.65
tip speed
u
peripheral velocity of the impeller blade tips

3.1.66
peripheral Mach number
Ma_u
non-dimensional parameter equal to the ratio of the tip speed to the velocity of sound in the gas at the stagnation conditions of fan inlet

$$Ma_u = \frac{u}{\sqrt{\kappa\ R_w\ \theta_{sg1}}}$$

3.1.67
fan Mach number
Ma_F
conventional quantity used as a scaling parameter

NOTE It is the fan tip speed divided by the speed of sound in standard air:

$$Ma_F = \frac{\pi D_r n}{c}$$

where c = 340 m·s^{-1} for ambient temperature.

3.1.68
fan impeller efficiency
η_r
fan air power P_u divided by the impeller power P_r

3.1.69
fan impeller static efficiency
η_{sr}
fan static power P_{us} divided by the impeller power P_r

3.1.70
fan shaft efficiency
η_a
fan air power P_u divided by the shaft power P_a

NOTE The fan shaft power P_a includes bearing losses whilst the impeller power does not.

3.1.71
fan motor efficiency
η_M
fan air power P_u divided by the motor output power P_o

3.1.72
overall efficiency
η_e
fan air power P_u divided by the input power for the fan and motor combination

3.1.73
kinetic energy factor at a section x of area A_x
α_{Ax}
non-dimensional coefficient equal to the time-averaged flux of kinetic energy per unit mass through the considered area A_x divided by the kinetic energy per unit mass corresponding to the mean air velocity through this area

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

where

v is the local absolute velocity

v_n is the local velocity component normal to the cross section

NOTE It has been agreed for this International Standard that by convention in fan technology α_{Ax} equals one.

3.1.74
kinetic index at a section x
i_{kx}
non-dimensional coefficient equal to the ratio of the kinetic energy per unit mass at the section x and the fan work per unit mass

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

3.1.75
Reynolds number at a section x
Re_x
product of the local velocity, the local density and a relevant scale length (duct diameter, blade chord) divided by the dynamic viscosity

$$Re_x = \frac{\rho_x \, v_{mx} \, D_x}{\mu_x}$$

NOTE It is a non-dimensional parameter which defines the state of development of a flow and is used as a scaling parameter.

3.1.76
fan Reynolds number
Re_F
product of the fan tip speed, the inlet density and the impeller diameter divided by the dynamic viscosity of the fluid at the fan inlet

$$Re_F = \frac{\rho_1 \, \pi \, n \, D_r^2}{\mu_1}$$

NOTE It is a conventional quantity used as a scaling parameter.

3.1.77
friction loss coefficient
$(\zeta_{x\text{-}y})_y$
non-dimensional coefficient for friction losses between sections x and y of a duct, calculated for the velocity and density at section y

NOTE For incompressible flow:

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

3.1.78
fan flow coefficient
ϕ
non-dimensional quantity equal to the mass flowrate divided by the product of the mean density, the peripheral speed of the impeller and the square of the diameter of the impeller

$$\phi = \frac{q_m}{\rho_m \, u D_r^2}$$

3.1.79
fan work per unit mass coefficient
Ψ
non-dimensional quantity equal to the fan work per unit mass divided by the square of the peripheral speed of the impeller

$$\Psi = \frac{y}{u^2}$$

3.1.80
fan power coefficient
λ_P
non-dimensional quantity equal to the impeller power divided by the product of the mean density with the cube of the peripheral speed of the impeller and the square of the diameter of the impeller

$$\lambda_P = \frac{P_\mathrm{r}}{\rho_\mathrm{m} u^3 D_\mathrm{r}^2}$$

3.2 Symbols

A	Area of the conduit section	m^2
A_x	Area of the conduit of section x	m^2
A_w	Correcting coefficient for partial pressure of water vapour at a given temperature	
A_1	Fan inlet area	m^2
A_2	Fan outlet area	m^2
b	Distance from the wall to the nearest measuring point	m
c	Speed of sound in air	$\mathrm{m{\cdot}s^{-1}}$
c_p	Specific heat at constant pressure	$\mathrm{J{\cdot}kg^{-1}{\cdot}K^{-1}}$
c_V	Specific heat at constant volume	$\mathrm{J{\cdot}kg^{-1}{\cdot}K^{-1}}$
d	Diameter of the head of the velocity probe	mm
D	Internal diameter of a circular cross-section duct	m
D_a	Minimum inner diameter of an annular duct	m
D_e	Equivalent diameter of a non-circular cross-section duct	m
D_h	Hydraulic diameter of the duct	m
D_r	Diameter of the impeller	m
e	Thickness of the ring in an annular duct	m
$e_{p\mathrm{F}}$	Fan pressure uncertainty	
e_q	Flowrate uncertainty	
e_Δ	Characteristic uncertainty	
f	Additional uncertainty	
f_i	Weighting coefficient	
F	Proximity coefficient	
F_M	Mach factor	
g	Gravitational acceleration	$\mathrm{m{\cdot}s^{-2}}$
h	Horizontal distance of the probe from the reference wall when the orthogonal coordinates are used	m
h_u	Relative humidity $\left(h_\mathrm{u} = \frac{p_\mathrm{v}}{p_\mathrm{sat}}\right)$	
H	Height of the rectangular section of a duct	m
i_k	Discharge kinetic index	

i_{kx}	Kinetic index at section x	
I	Line current	A
k	Compressibility coefficient	
k_ρ	Density ratio	
l	Length of traverse line	m
l_a	Length of traverse line at distance a from reference wall	m
l_0	Length of traverse line at distance 0 from reference wall	m
l_x	Length of traverse line at distance x from reference wall	m
L	Length of the rectangular section of a duct, or greatest possible length of a section having any one form	m
L_{D}	Length of a duct	m
L_{p}	Inner dimension of the duct in a direction perpendicular to the nearest wall to the probe	m
Ma	Mach number at a point	
Ma_{F}	Fan Mach number	
Ma_{x}	Mach number at section x	
n	Rotational frequency of impeller	$\mathrm{r{\cdot}s^{-1}}$
N	Rotational speed of impeller	$\mathrm{r{\cdot}min^{-1}}$
N_{r}	Number of traverse lines	
p	Mean pressure in space and time of the fluid, i.e. absolute static pressure	Pa
p_{a}	Atmospheric pressure (absolute)	Pa
p_{d}	Dynamic pressure at a point	Pa
p_{dx}	Dynamic pressure at section x	Pa
p_{dF}	Fan dynamic pressure	Pa
p_{e}	Gauge pressure	Pa
p_{esgx}	Gauge stagnation pressure at section x	Pa
p_{eX}	Average gauge pressure at section x	Pa
p_{F}	Fan (stagnation) pressure	Pa
p_{l}	Inverse of the exponent of the characteristic law of the evolution of velocities at the wall (taking into account the measurement results of the surface roughness of the walls and the value of the Reynolds numbers)	
p_{sat}	Saturation vapour pressure	Pa
p_{sF}	Fan static pressure	Pa
p_{sg}	Absolute stagnation pressure at a point	Pa
p_{sgx}	Absolute stagnation pressure at section x	Pa
p_{t}	Total pressure at a point	Pa
p_{tx}	Total pressure at section x	Pa
p_{v}	Partial pressure of water vapour	Pa
p_{x}	Average absolute pressure at section x	Pa
p_1	Absolute static pressure in the inlet section	Pa
p_2	Absolute static pressure in the outlet section	Pa
P_{a}	Mechanical power output to the fan shaft	W

P_e	Motor input power	W
P_f	Friction losses power	W
P_o	Power available at the output shaft of the drive	W
P_r	Mechanical power supplied to the impeller of the fan	W
P_u	Fan air power	W
P_{us}	Fan static power	W
q_m	mass flowrate	$kg \cdot s^{-1}$
q_V	Volume flowrate	$m^3 \cdot s^{-1}$
q_{Vr}	Actual volume flowrate	$m^3 \cdot s^{-1}$
q_{Vs}	Volume flowrate corresponding to standardized conditions of use of the DP device	$m^3 \cdot s^{-1}$
q_{Vsg1}	Volume flowrate at the inlet at stagnation conditions	$m^3 \cdot s^{-1}$
q_{Vsg2}	Volume flowrate at the outlet at stagnation conditions	$m^3 \cdot s^{-1}$
q_{Vx}	Volume flowrate at section x	$m^3 \cdot s^{-1}$
r	Radius of the duct	m
r_{Fp}	Fan pressure ratio	
r_A	Area ratio of an orifice plate	
R	Specific gas constant	$J \cdot kg^{-1} \cdot K^{-1}$
R_D	Extreme value of a duct radius	m
Re_x	Reynolds number at section x	
R_w	Specific gas constant of humid air	$J \cdot kg^{-1} \cdot K^{-1}$
S	Characteristic proportional slope of equivalent orifice	
t	Air or fluid temperature measured by a temperature sensor	°C
t_d	Dry bulb temperature	°C
t_x	Static temperature at section x	°C
t_w	Wet bulb temperature	°C
u	Peripheral speed of the impeller	$m \cdot s^{-1}$
U	Voltage of electrical current	
v	Local absolute velocity	$m \cdot s^{-1}$
v_a	Axial velocity at test section	$m \cdot s^{-1}$
v_m	Mean value of v over time	$m \cdot s^{-1}$
v_{m1}	Mean value of v in the inlet section over time	$m \cdot s^{-1}$
v_{m2}	Mean value of v in the outlet section over time	$m \cdot s^{-1}$
v_{mx}	Mean value of v in section x over time	$m \cdot s^{-1}$
v_n	Local velocity normal to the cross-section	$m \cdot s^{-1}$
$v_x(y)$	Profile of velocities along the segment of the exploration line of the abscissa x	$m \cdot s^{-1}$
V	Volume of fluid	m^3
y	Vertical distance of the probe from the reference wall when orthogonal coordinates are used	m
y_F	Fan work per unit mass	$J \cdot kg^{-1}$
y_{Fs}	Fan static work per unit mass	$J \cdot kg^{-1}$
z	Mean altitude of the fan from reference plane	m

z_1	Mean altitude of fan inlet from reference plane	m
z_2	Mean altitude of fan outlet from reference plane	m
Z	Compressibility factor	
α_{A}	Kinetic energy coefficient of the flow	
α_{A1}	Value of the coefficient α in the inlet section of area A	
α_{A2}	Value of the coefficient α in the outlet section of area A	
δq_V	Absolute uncertainty in the volume flowrate q_V	$\mathrm{m^3{\cdot}s^{-1}}$
Δp	Differential pressure	Pa
Δq_V	Absolute limit error on the determination of the volume flowrate q_V	$\mathrm{m^3{\cdot}s^{-1}}$
Δz	Altitude at barometer minus altitude of fan	m
ε	Expansion factor	
η_{a}	Fan shaft efficiency	
η_{e}	Overall efficiency (or unit efficiency)	
η_{Ms}	Motor shaft efficiency	
η_{M}	Motor efficiency	
η_{r}	Fan impeller efficiency	
η_{sr}	Fan impeller static efficiency	
η_{tr}	Drive efficiency	
κ	Ratio of specific heats (at constant pressure and volume)	
λ	Darcy friction factor	
λ_P	Fan power coefficient	
ζ	Friction loss coefficient ($\zeta = \lambda{\cdot}L{\cdot}D_{\mathrm{h}}^{-1}$)	
μ_{x}	Dynamic viscosity of the fluid at section x	Pa·s
μ_1	Dynamic viscosity of the fluid at the fan inlet	Pa·s
ρ	Density of fluid	$\mathrm{kg{\cdot}m^{-3}}$
ρ_{m}	Mean density	$\mathrm{kg{\cdot}m^{-3}}$
ρ_{12}	Arithmetic mean value over time of inlet and outlet densities	$\mathrm{kg{\cdot}m^{-3}}$
ρ_{x}	Average density at section x	$\mathrm{kg{\cdot}m^{-3}}$
ρ_1	Mean density in the inlet section	$\mathrm{kg{\cdot}m^{-3}}$
ρ_2	Mean density in the outlet section	$\mathrm{kg{\cdot}m^{-3}}$
ρ_{sg1}	Stagnation inlet density	$\mathrm{kg{\cdot}m^{-3}}$
θ	Absolute temperature	K
θ_{sg}	Stagnation temperature at a point	K
θ_{sgx}	Stagnation temperature at section x	K
θ_{x}	Static or fluid temperature at section x	K
ϕ	Fan flow coefficient	
ψ	Azimuth	radians
Ψ	Fan work per unit mass coefficient	

4 Quantities to be measured

The flow of fluid in a fan and in the installation it serves is never completely steady. However, the quantities relating to the state and displacement which characterize this flow do have steady mean values over time, at least in the normal operating zone of the fan, when the system resistance is kept constant and the rotational speed of the fan is maintained to within 0,5 %.

The fluctuations which affect the characteristics investigated may be taken into account by repeating the measurements at appropriate intervals of time so that mean values may be calculated truly representing the desired mean values over time, which then become virtually steady values.

For a permanent flow of fluid of this nature generated by an industrial fan operating in an airtight section of an airway without a branch pipe (inlet section 1; outlet section 2 of Figure 1), the following expression serves as a basis for defining the effect of the fan on the flow under consideration:

$$y = \frac{P_u}{q_m} = \frac{p_2 - p_1}{\rho_m} + \frac{\alpha_{A2}\, v_{m2}^2}{2} - \frac{\alpha_{A1}\, v_{m1}^2}{2} + g\,(z_2 - z_1)$$

By convention, for this International Standard $\alpha_{A2} = \alpha_{A1} = 1$

5 General conditions and procedures concerning *in situ* tests

5.1 General recommendations

Tests on site shall be carried out after an initial check that the fan is functioning properly.

There shall be no significant leakage of gas into or out of the airway between the fan and any flow or pressure measuring plane. There shall be no unintended recirculation of gas between the inlet and outlet of the fan.

The measures necessary for the safety of the test operators and for the prevention of damage to the fan shall not have any appreciable effect on the performance characteristics of the machine under test.

Before beginning the acceptance tests, the supplier shall have the right to check that the fan is in good working order and to make any necessary adjustments.

5.2 Selection of test point when only the system resistance can be varied

If, for a fan without an adjustment device (e.g. variable pitch, adjustable blades or inlet vane control), a single specified operating point is to be checked and only if the system resistance can be varied, measurements shall be taken at at least three operating points selected as follows.

a) For the point of least flow, the value of the flowrate or of the flow coefficient shall be less than that at the specified point, and shall be between 85 % and 90 % of this latter value if possible.

b) For the point of greatest flow, the value of the flowrate or of the flow coefficient shall be greater than that at the specified point, and shall be between 110 % and 115 % of this latter value if possible.

c) For the intermediate point, the value of the flowrate or of the flow coefficient shall be as close as possible to that at the specified point, and shall be between 97 % and 103 % of this latter value if possible.

If, for a fan without an adjustment device, more than one specified operating point is to be checked and if only the system resistance of the airway can be varied, the measuring point shall be selected as follows.

d) A test point shall be selected corresponding to each specified point so as to obtain a value for the flowrate corrected if necessary to take account of a variation in speed in relation to the specified speed, or for the value of the flow coefficient of the fan, as close as possible to that at the specified point and, if possible, within 3 %.

e) The variation in the flowrate or in the flow coefficient between two adjacent test points may not exceed 10 % of the arithmetical mean of the flow coefficients at the specified point;

f) The range of the test points shall extend on both sides beyond the range of the specified points.

The number and the range of operating points may be reduced by mutual agreement between the parties.

5.3 Fans fitted with adjustment devices

When the fan is fitted with a geometric adjustment device, a measuring point shall be obtained by setting both the adjustment device of the fan and the system resistance of the airway such that the values of the flow and pressure coefficients at this test point are as close as possible to those of the corresponding specified point, the deviation being, if possible, less than 4 %.

It is recommended that the proper settings of the adjustment devices be determined by means of a preliminary test.

Supplementary points shall be added to each measuring point thus obtained, keeping the adjustment device in the same position, modifying only the system resistance and adhering to the recommendations laid down for the case of a single specified operating point.

5.4 System throttling devices allowing the system resistance to be altered

To obtain different points on the characteristic curves of the fan, the flowrate shall be reduced by throttling the system or increased by opening a by-pass. These devices shall be located so that they do not disturb the flow either in the measuring section or in the fan.

It is advisable to avoid positioning the two restricting devices in series as this may create pulsating flow.

The system throttling devices shall, as far as possible, be symmetrical and shall cause no swirl. They shall preferably be positioned downstream from the fan. If this is impossible, they shall be positioned as far as possible upstream of the fan inlet. It shall be ensured that these positions are such that the resulting disturbance has no appreciable effect either on the measurements or on the operation of the fan.

In any case, the system throttling devices shall be placed at a minimum of 5 D_h downstream or 10 D_h upstream from the fan[1], D_h indicating the hydraulic diameter of the duct[2].

It should be noted that the proposed distances are not always adequate to reduce the disturbance of flow in the fan to negligible proportions.

In cases of serious doubt, an appropriate test shall be carried out in order to control the flow conditions.

It is also permissible to use any other means (e.g. fans in series or in parallel) which can alter the operating point of the fan without disturbing the flow conditions in the fan and the measuring section.

5.5 Selection of the test point when the system resistance cannot be varied

When the system resistance of the airway cannot be varied, the measurement can only be made for one operating point. In this case an agreement between the parties is necessary to the effect that the test can only be carried out at this single point.

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1) Provided that these lengths are sufficient to avoid inaccuracies in measurement of the flowrate and the pressure of the fluid on both sides of the fan.

2) The hydraulic diameter is equivalent to four times the sectional area divided by the internal perimeter. For a circular section, the hydraulic diameter is equal to the geometrical diameter of the section.

5.6 When correction of the coefficient deduced from the test is not necessary

When the values of the density and viscosity of the fluid and the rotational speed of the fan measured during a test do not differ by more than 10 % in relation to the specified value of the fan Reynolds number, it is not necessary to correct non-dimensional coefficients deduced from the test.

6 Instrumentation

6.1 Instrumentation for measurement of pressure

6.1.1 Barometers

The atmospheric pressure in the test enclosure shall be determined with an uncertainty not exceeding ± 0,3 %.

Barometers of the direct-reading mercury column type should be read to the nearest 100 Pa (1 mbar) or to the nearest 1 mm of mercury. They should be calibrated and corrections applied to the readings as specified in ISO 5801. Correction may be unnecessary if the scale is preset for the regional value of g (to within ± 0,01 m/s^2) and for room temperature (within ± 10 °C).

Barometers of the aneroid or pressure transducer type may be used provided they have a calibrated accuracy of ± 200 Pa and the calibration is checked at the time of test.

The barometer should preferably be located in the test enclosure. If it is placed elsewhere in the locality, a correction, $\rho_a \cdot g \cdot \Delta z$, in pascals, should be applied for any difference in altitude exceeding 10 m.

6.1.2 Manometers

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

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

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

Calibration should be a series of steady pressures, taken both in rising and falling sequence to check for any difference. The reference instrument should be a precision manometer or micromanometer capable of being read to an accuracy of 0,25 % or 0,5 Pa, whichever is the greater.

Manometers should be located and calibrated at the mean altitude of the fan or, alternatively, where the difference exceeds 10 m a correction as given in 6.1.1 should be applied.

6.1.3 Damping of manometers

Rapid fluctuations of manometer readings should be limited by damping so that it is possible to estimate the average reading within ± 1,0 % of the significant pressure. The damping may be in the air connections leading to the manometer or in the liquid circuit of the instrument. It should be linear, and of a type which ensures equal resistance to movement in either direction. The damping should not be so heavy that it prevents the proper indication of slower changes. If these occur, a sufficient number of readings should be taken to determine an average within ± 1,0 % of the significant pressure.

If linear damping is required, this may be achieved by inclusion of short lengths of small bore tube or glass capillary on either side of the manometer.

6.1.4 Checking of manometers

Liquid column manometers should be checked in their test location to confirm their calibration near the significant pressure. Inclined tube instruments should be frequently checked for level and rechecked for calibration if disturbed. The zero reading of all manometers shall be checked before and after each series of readings without disturbing the instrument. Care should be taken to ensure that all tubing and connections to other instruments are free from blockage or leakage.

6.2 Measurement of air velocity

6.2.1 Pitot-static tube

The Pitot-static tube described in ISO 5801 shall be used without preliminary calibration (see also ISO 3966 and ISO 7194).

Using the differential pressure (Δp) measured by this instrument in conjunction with a manometer, the local fluid velocity may be calculated by the formula:

$$v = \left[\frac{2\Delta p}{\rho} \right]^{1/2}$$

The lower limit of the differential pressure measurement depends on the accuracy required for this determination and the accuracy of the micromanometer selected. Under normal industrial conditions, the use of the Pitot-static tube is not recommended in sections where the differential pressure at any measuring point is less than 10 Pa.

In order to keep the error in the flowrate, resulting from a velocity gradient along the measuring section, within negligible limits, the ratio d/D_h of the diameter d of the head of the tube to the hydraulic diameter D_h of the duct shall not exceed 0,02.

Pitot-static tubes shall be used subject to the following conditions.

a) The Pitot-static tube shall be manufactured in conformity with the dimensional specifications stipulated and shall be in good condition.

b) The axis of the head of the Pitot-static tube shall be parallel to the axis of the duct to within $\pm$ 5°; appropriate devices shall be provided for this purpose.

c) The Pitot-static tube shall be kept firmly in place during the measurements.

d) The distance between the axis of the Pitot-static tube and the wall shall be greater than the diameter of the head of the tube.

e) The local Reynolds number, related to the diameter of the head of the tube, shall be greater than 500. This means that for air at atmospheric pressure and temperature the local velocity v in metres per second should be not less than $v = 7{,}5/d$, where d is the diameter of the head of the tube, in millimetres.

f) The angle formed by the direction of flow at each point and the axis of the duct shall, in general, not exceed 10° except for the relatively small number of points for which this value might reach 15°.

This angle may be measured for instance by one of the following methods:

— cylindrical probe with three holes and at least two manometers; this is the more simple method using the same hole as the Pitot-static tube;

— weathercock with indicator;

— winch or anemometer with radial blades and measure of rotational speed.

The marking devices of the measuring points shall be placed downstream of the measuring section and the total blockage area shall not be greater than 2,5 % of the area of the measuring section.

The velocity probes shall be fixed in such a way that they vibrate as little as possible. Branch pipes and electric cables used for the measurements shall be so located that they do not disturb the measurement itself.

The openings for probes, pipes and cables shall be sufficiently airtight as not to influence the measurements made near the wall.

The geometric shape of the measuring section shall be as simple as possible.

When the Mach number exceeds 0,2 (corresponding to approximately 70 m/s in standard air) a correction factor taking account of the compressibility effects shall be included in the formula from which the local fluid velocity can be calculated using measurements taken by means of the Pitot-static tube. In this case:

$$v = \varepsilon \left[\frac{2\Delta p}{\rho} \right]^{1/2}$$

with $\varepsilon = \left[1 + \frac{Ma^2}{4} + \frac{2-\kappa}{24} Ma^4 \right]^{-1/2}$

$$\varepsilon \cong \left[1 - \frac{1}{2\kappa} \cdot \frac{\Delta p}{p} + \frac{\kappa + 1}{6\kappa^2} \left(\frac{\Delta p}{p} \right)^2 \right]^{1/2}$$

The validity of this formula is limited to

$$\frac{\Delta p}{p} \leqslant 0{,}3 \quad \text{for } \kappa = 1{,}4$$

6.2.2 Rotating vane anemometers

The use of rotating vane anemometers is limited to conditions where there is no significant fluctuation in velocity level at any point in the measuring plane. They may be used subject to the following conditions.

a) The appliance shall be in good condition and shall be calibrated by an authority recognized by the parties concerned, before and after the tests (see annex E for recommended procedure).

b) The axis of the anemometer shall be as parallel as possible to the axis of the duct. The deviation in the flow in relation to the axis of the anemometer shall not exceed 5° at any measuring point if the error is to be maintained at less than 1 %.

c) The diameter of the appliance shall be less than 1/10 of the smallest dimension of the measuring section.

d) When it is agreed that an abnormal velocity distribution exists, consideration should be given to the introduction of a smaller diameter anemometer and an increased number of measuring points.

e) The distance between the centre of the appliance and the wall shall be not less than 3/4 of the diameter of the appliance.

f) The appliance shall be mounted on a support which is sufficiently rigid to prevent vibrations but which will disturb the flow as little as possible.

g) As the accuracy of measurement depends very much on the value of the reading and the uniformity of the flow, the lowest reading shall be at least three times the velocity at which the anemometer commences to rotate.

6.2.3 Other appliances

The use of other devices (e.g. Venturi probes, swinging vane anemometers, thermal anemometers, etc.) is recommended if the velocities are so low that the Pitot-static tubes or rotating vane anemometers cannot provide a good accuracy.

The conditions set out in 6.2.2 for rotating vane anemometers also apply to the instruments mentioned above.

However it should be noted that the specified calibrations concern the complete instrument comprising the head, the connections and the indicator.

It is also pointed out that the thermal anemometers are particularly suitable for measurements close to the wall.

6.3 Measurement of temperature

6.3.1 Thermometers

Instruments for the measurement of temperature shall have an accuracy of ± 1,0 °C after the application of any calibration correction. The corrected test reading should be recorded to the nearest 0,5 °C.

6.3.2 Thermometer location

Temperature measurements taken inside an airway shall be made with the sensing element directly in the airstream located on a horizontal diameter, one third of the airway diameter or 100 mm from the wall, whichever is the lesser. Instruments should be withdrawn from the airstream when performance readings are taken, unless it is demonstrated that their presence does not affect the determination of performance.

6.3.3 Humidity

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

The wet bulb thermometer should be located in an air stream of velocity of at least 3 $m \cdot s^{-1}$. The sleeving should be clean, in good contact with the bulb, and kept wetted with pure water. Relative humidity may be measured provided the apparatus used has an accuracy of ± 2 %.

6.3.4 Influence of air velocity

The uncertainty of temperature measurement will increase if the temperature sensing element is placed in an airstream of velocity exceeding 60 $m \cdot s^{-1}$ with atmospheric air (0,15 Mach number). A thermometer located in the ductwork indicates an intermediate temperature between the stagnation temperature and static temperature, but closer to the stagnation temperature.

If the air velocity is equal to 25 $m \cdot s^{-1}$ the difference between stagnation and static temperatures is 0,31 °C; for 35 $m \cdot s^{-1}$, the same difference is 0,61 °C; and for 50 $m \cdot s^{-1}$ the difference is 1,24 °C.

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

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

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

If it is not possible to meet this requirement, sensing elements can be put inside an airway on a horizontal diameter, at least 100 mm from the wall or one third of the airway diameter, whichever is the lesser.

6.4 Determination of density

6.4.1 Density of the air in the test enclosure

The density of the air in the test enclosure (in kg·m^{-3}) is given by the following expression:

$$\rho_a = \frac{3{,}484\,(p_a - 0{,}378\,p_v)}{1\,000\,(273{,}15 + t_a)}$$

However, when testing with standardized or similar airways, the effect of water vapour is usually negligible. At temperatures below 23 °C the following simplified expression may be used with an uncertainty not exceeding ± 0,5 %.

$$\rho_a = \frac{3{,}468\,p_a}{1\,000\,(273{,}15 + t_a)}$$

This latter expression may also be used under site conditions when the moisture content of the air is less than 1,5 % by mass.

6.4.2 Average density of the air in an airway section

The average density of the air in an airway section x, where the average gauge pressure in pascals is p_{ex}, and the average air temperature in degrees Celsius is t_x, may be obtained for high pressure according to ISO 5801.

6.4.3 Determination of vapour pressure

The vapour pressure (in pascals) may be obtained from the following expression:

$$p_v = p_{sat} - p_a A_w (t_a - t_w)$$

where

p_{sat} is the saturation vapour pressure at the wet bulb temperature t_w;

t_a is the dry bulb temperature;

A_w = 6,66 × 10^{-4} °C^{-1} when t_w is between 0 °C and 150 °C;

A_w = 5,94 × 10^{-4} °C^{-1} when t_w is less than 0 °C.

Table 1 lists values of saturated vapour pressure (p_{sat}) in 0,5 °C increments of water or ice in contact with air over the temperature range – 4,0 °C to 49,5 °C. The air relative humidity h_u can also be directly measured in order to obtain:

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

where $(p_{sat})_{ta}$ is the satured vapour pressure at the dry bulb temperature t_a obtained from Table 1 with t_a instead of t_w.

6.5 Measurement of rotational speed

6.5.1 Fan shaft speed

The fan shaft speed shall be measured at regular intervals throughout the period of test for each test point so as to ensure the determination of average speed during each such period with an uncertainty not exceeding ± 0,5 %. No device used should significantly affect the speed of the fan under test or its performance.

6.5.2 Examples of acceptable methods

6.5.2.1 Digital counter measuring the revolutions for a measured time interval

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

6.5.2.2 Revolution counter

This shall be free from slip and timed over a period of not less than 60 s per operation.

6.5.2.3 Direct indicating mechanical or electrical tachometer

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

6.5.2.4 Stroboscopic methods

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

6.5.2.5 Frequency meter

When the fan is direct driven by a synchronous or induction motor, by measurement of the supply frequency and, in the latter case, also counting the slip frequency, the frequency meter shall have an uncertainty of not more than 0,5 %. Alternatively, a digital instrument of lower class index, i.e. smaller uncertainty, is permissible. The device used for indicating slip frequency shall be used in such a manner as to permit direct counting with an uncertainty not exceeding ± 0,25 % of the shaft speed.

7 Determination of fan pressure

7.1 Location of pressure measurement plane

7.1.1 For the purposes of determining the fan pressure, the static pressure shall be measured at planes on the inlet and/or the outlet side of the fan sufficiently close to it to ensure that the pressure losses between the measuring planes and the fan are calculable in accordance with available friction factor data without adding excessively to the uncertainty of fan pressure determination. Friction factors for smooth ducts are given in annex D.

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

Temperature	Saturation vapour pressure p_{sat} of water, hPa									
°t	0,0	0,1	0,2	0,3	0,4	0,5	0,6	0,7	0,8	0,9
–4	4,55	4,51	4,48	4,44	4,41	4,37	4,35	4,31	4,28	4,24
–3	4,89	4,87	4,83	4,79	4,76	4,72	4,68	4,65	4,61	4,59
–2	5,28	5,24	5,20	5,16	5,12	5,08	5,04	5,01	4,97	4,93
–1	5,68	5,64	5,60	5,56	5,52	5,47	5,44	5,39	5,36	5,32
0	6,11	6,07	6,03	5,97	5,93	5,89	5,84	5,80	5,76	5,72
0	6,11	6,16	6,19	6,24	6,29	6,33	6,37	6,43	6,47	6,52
1	6,56	6,61	6,67	6,71	6,76	6,80	6,85	6,91	6,96	7,00
2	7,05	7,11	7,16	7,21	7,25	7,31	7,36	7,41	7,47	7,52
3	7,57	7,63	7,68	7,73	7,79	7,85	7,91	7,96	8,01	8,08
4	8,13	8,19	8,24	8,31	8,36	8,43	8,48	8,53	8,60	8,65
5	8,72	8,79	8,84	8,91	8,96	9,03	9,09	9,16	9,21	9,28
6	9,35	9,41	9,48	9,53	9,61	9,68	9,75	9,81	9,88	9,95
7	10,01	10,08	10,15	10,23	10,29	10,36	10,43	10,51	10,57	10,65
8	10,72	10,80	10,87	10,95	11,01	11,09	11,17	11,24	11,32	11,40
9	11,48	11,55	11,63	11,71	11,79	11,87	11,95	12,03	12,11	12,19
10	12,27	12,36	12,44	12,52	12,61	12,69	12,77	12,87	12,95	13,04
11	13,12	13,21	13,29	13,39	13,47	13,56	13,65	13,75	13,84	13,93
12	14,01	14,11	14,20	14,29	14,39	14,48	14,59	14,68	14,77	14,87
13	14,97	15,07	15,17	15,27	15,36	15,47	15,57	15,67	15,77	15,88
14	15,97	16,08	16,19	16,29	16,40	16,51	16,61	16,72	16,83	16,93
15	17,04	17,16	17,27	17,37	17,49	17,60	17,72	17,83	17,96	18,05
16	18,17	18,29	18,41	18,52	18,64	18,76	18,88	19,00	19,12	19,25
17	19,37	19,49	19,61	19,73	19,87	19,99	20,12	20,24	20,37	20,51
18	20,63	20,76	20,89	21,03	21,16	21,29	21,43	21,56	21,69	21,83
19	21,96	22,11	22,24	22,39	22,52	22,67	22,80	22,95	23,09	23,23
20	23,37	23,52	23,67	23,81	23,96	24,11	24,25	24,41	24,56	24,71
21	24,87	25,01	25,17	25,32	25,48	25,64	25,80	29,95	26,11	26,27
22	26,43	26,60	26,76	26,92	27,08	27,25	27,41	27,59	27,75	27,92
23	28,09	28,25	28,43	28,60	28,77	29,95	28,12	29,31	29,48	29,65
24	29,84	30,01	30,19	30,37	30,66	30,75	30,92	31,11	31,29	31,48
25	31,68	31,87	32,05	32,24	32,44	32,63	32,83	33,01	33,21	33,41
26	33,61	33,81	34,01	34,21	34,41	34,61	34,83	35,03	35,24	35,44
27	35,65	35,87	36,08	36,28	36,49	36,71	36,93	37,15	37,36	37,57
28	37,80	38,03	38,24	38,47	38,69	38,92	39,15	39,37	39,60	39,83
29	40,05	40,29	40,52	40,76	41,00	41,23	41,47	41,71	41,95	42,19
30	42,43	42,68	42,92	43,17	43,41	43,67	43,92	44,17	44,43	44,68
31	44,93	45,19	45,44	45,71	45,96	46,23	46,49	46,75	47,01	47,28
32	47,56	47,83	48,09	48,37	48,64	48,92	49,19	49,47	49,75	50,03
33	50,31	50,60	50,88	51,16	51,45	51,73	52,03	52,32	52,61	52,91
34	53,20	53,51	53,80	54,11	54,40	54,71	55,01	55,32	55,63	55,93
35	56,24	56,55	56,87	57,17	57,49	57,81	58,13	58,45	58,77	59,11
36	59,43	59,76	60,08	60,41	60,75	61,08	61,41	61,75	62,08	62,43
37	62,77	63,11	63,45	63,80	64,15	64,49	64,85	65,20	65,56	65,91
38	66,27	66,63	66,99	67,35	67,72	68,08	68,45	68,83	69,19	69,56
39	69,95	70,32	70,69	71,07	71,45	71,84	72,23	72,61	73,00	73,39
40	73,79	74,17	74,57	74,97	75,37	75,77	76,17	76,59	76,99	77,40
41	77,81	78,23	78,64	79,05	79,47	79,89	80,32	80,73	81,16	81,69
42	82,03	82,45	82,89	83,32	83,76	84,20	84,64	85,08	85,53	85,97
43	86,43	86,88	87,33	87,79	88,25	88,71	89,17	89,64	90,11	90,57
44	91,04	91,52	91,99	92,47	92,95	93,43	93,91	94,40	94,88	95,37
45	95,87	96,36	96,85	97,35	97,85	98,36	98,85	99,36	99,88	100,39
46	100,89	101,41	101,93	102,45	102,97	103,51	104,04	104,57	105,09	105,63
47	106,17	106,71	107,25	107,79	108,33	108,89	109,44	109,99	110,55	111,11
48	111,67	112,23	112,80	113,37	113,93	114,51	115,08	115,65	116,24	116,83
49	117,41	118,00	118,59	119,17	119,79	120,37	120,99	121,57	122,19	122,80

7.1.2 Before the commencement of observations, the pressure at the measuring section shall be surveyed to determine the uniformity of the readings. Four cases may be identified as follows.

a) Where the difference in pressure between any of four wall tappings constructed in accordance with 7.2.2.2 is less than 5 % of the arithmetic average, then these tappings may be interconnected by a manifold as shown in Figure 2 and the pressure so measured taken to be the average gauge pressure.

b) Where the difference between any of the measurements at these four wall tappings is greater than 5 % but less than 10 % of the arithmetic average, then the tappings shall be replaced by a Pitot-static tube. This shall be inserted into the airway at the points defined and under the conditions stated in 7.2.2.4. Provided the difference between each of these four readings and their arithmetic average is less than 10 %, then this average may be taken or alternatively four separate Pitot-static tubes may be interconnected in the manner described in 7.2.2.2.

c) Where the difference between any of these four Pitot-static readings and the arithmetic average is greater than 10 % but less than 15 %, then a full traverse shall be taken in accordance with the requirements of 6.2.1 and at positions as defined in 8.4. The arithmetic average of all readings shall be taken.

d) Where the difference between any traverse reading and the arithmetic average of the traverse exceeds 15 % of the average, then the pressure measurement plane shall be considered unsatisfactory for site measurements.

The method described under c) above may also be used for situations which comply with a) and b).

7.1.3 If conveniently close to the fan, the "test length" selected for air flow measurement shall also be used for pressure measurement. Other planes used for pressure measurement shall be not closer than 1,5 D_e from the fan inlet and no closer than 5 D_e from the fan outlet (Figure 3). It may be shorter, provided that the flow conditions are checked for stability. The plane of pressure measurement shall be selected at least 5 D_e downstream of any bend, expander or obstruction likely to cause separated flow or otherwise interfere with uniformity of pressure distribution. The fan shall be taken to include all appartenances such as inlet boxes, dampers, diffusers, etc. In all cases the plane chosen for pressure measurements shall be such that the mean air velocity at the plane can also be determined either by calculation from readings taken elsewhere or by direct measurement by the traverse method.

7.1.4 Where a fan is connected directly to a plenum chamber on its inlet or outlet side the pressure measurement plane shall be located as close as possible to the face of the plenum chamber to which the fan is connected so that the points of pressure measurement are, in fact, in "dead zones" where there is no significant air velocity.

7.2 Measurement of fan pressure

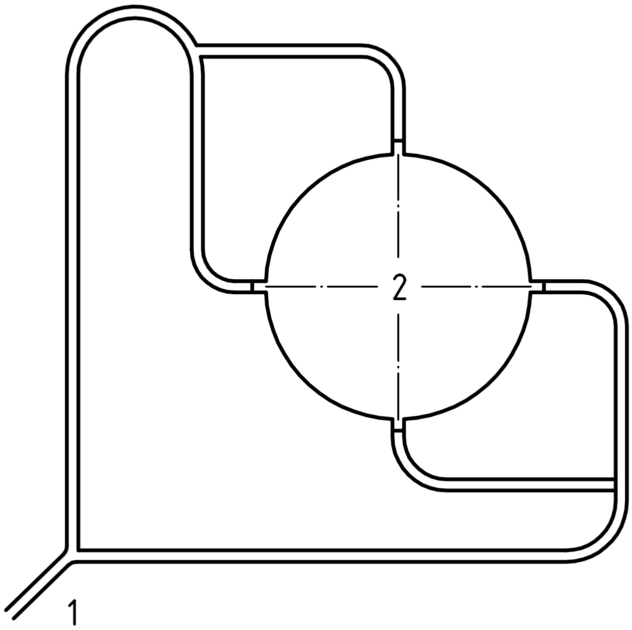
7.2.1 General

Care shall be taken to ensure that the measurements of the static pressure on the inlet and outlet sides of the fan are taken relative to the atmospheric pressure, or to that existing within a common test enclosure. When this is not possible the method given in 7.2.3.4 should be used.

7.2.2 Measurement of static pressure on site

7.2.2.1 This shall be carried out using a manometer as described in 6.1.2 to 6.1.4 in conjunction with wall tappings or with the static pressure connection of a Pitot-static tube as described in 7.1.2.

7.2.2.2 Under conditions of reasonably uniform flow [case a) of 7.1.2], free from swirl and separation, static pressure may be measured by use of four wall tappings (see Figure 2) evenly spaced around the perimeter of the duct (and in the centre of the sides in the case of a rectangular duct) (Figure 4) provided such tappings are finished flush and free from burrs inside and the adjacent duct walls are smooth, clean and free from undulations and discontinuities (Figure 5).

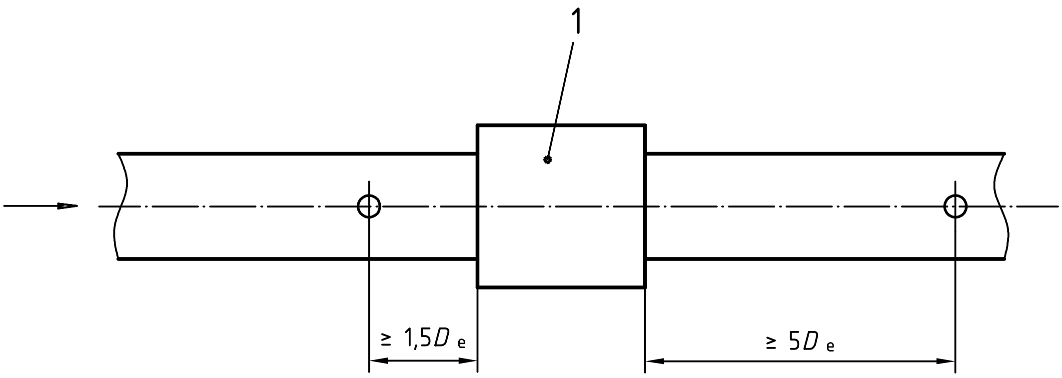


Key

1 To manometer

2 Airway

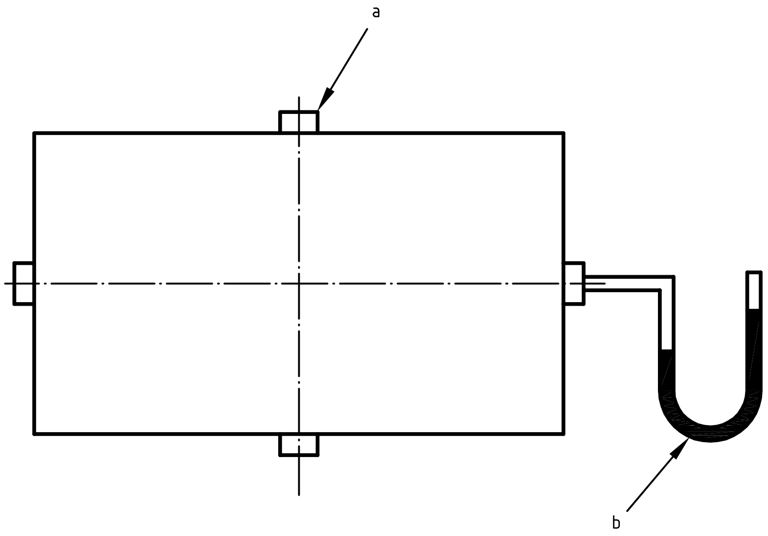
Figure 2 — Tapping connections to obtain average static pressure in circular airway
(shown inter-connected to single manometer)



Key

1 Fan

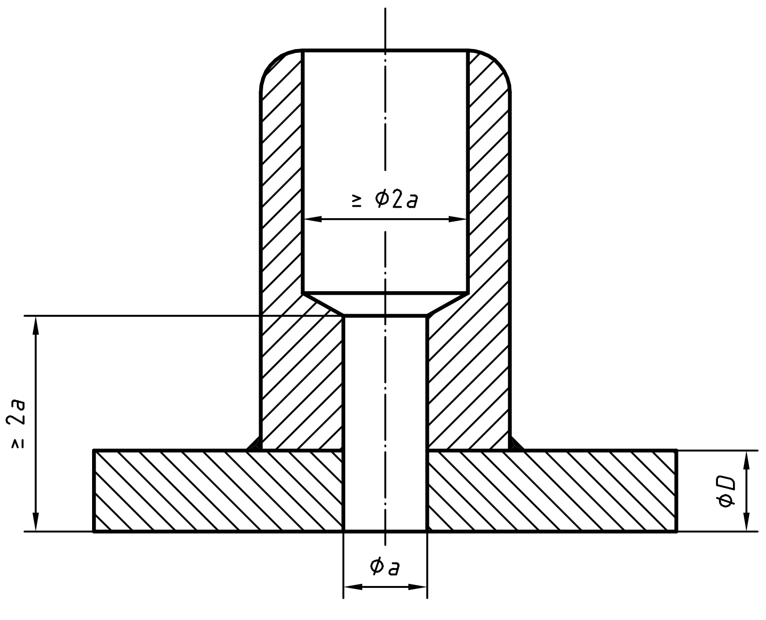
Figure 3 — Location of pressure measurement planes for site testing



a Minimum of four taps, located 90° apart and near the centre of each wall.

b Static pressure measurement required at each tap. Use the average of the measurements as the static pressure for the plane.

Figure 4 — Tapping connections to obtain average static pressure in rectangular airway
(separate manometer connections shown)



Key

$\varnothing D$ Airway diameter

Figure 5 — Construction of wall pressure tappings

7.2.2.3 Care shall be taken to ensure that all tubing and connections are free from blockage and leakage.

Before the commencement of any series of observations, the pressure at the four side tappings should be individually measured at a flowrate towards the maximum of the series. If any one of the four readings lies outside a range equal to 5 % of the rated fan pressure, the tappings and manometer connections should be examined for defects, and if none are found, the flow shall be examined for uniformity.

7.2.2.4 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 one-eighth of the airway diameter from the wall or, in the case of a rectangular airway, one-eighth of the duct width from the centre of each wall. Under steady flow conditions, a static pressure reading should be taken at each point and the average calculated.

7.2.2.5 Where the pressure measurement plane is located adjacent to the fan inlet or outlet in a chamber, the static pressure may be measured by use of either wall tappings or a Pitot-static tube suitably located to transmit the static pressure to a manometer.

7.2.3 Distinction between installation categories

7.2.3.1 General

ISO 5801 recognizes four installation categories, according to which the fan performance may vary:

— Type A: free inlet, free outlet,

— Type B: free inlet, ducted outlet,

— Type C: ducted inlet, free outlet,

— Type D: ducted inlet, ducted outlet.

When testing it is essential to ensure that the inlets of Types A and B fans are unobstructed. Failure to observe this will result in an additional unmeasurable resistance. Guidance on the desirable free space may be found in ISO 5801.

7.2.3.2 Type A installation

This International Standard describes methods of flowrate and pressure measurement for use in ducts, and is not, therefore, applicable to Type A unducted situations.

7.2.3.3 Type B installation

Where a fan is installed as a blowing fan under category B conditions (Figure 6), the average gauge pressure at the test section on the outlet side of the fan shall be measured in accordance with 7.2.2.

The reference (or effective) pressure is in this case the fan pressure p_{F} defined as the gauge stagnation pressure p_{esg2} at the fan outlet minus the gauge stagnation pressure p_{esg1} at the fan inlet (in this case zero).

The gauge stagnation pressure p_{esg2} is given by the equation:

$$p_{\mathrm{esg2}} = p_{\mathrm{e2}} + p_{\mathrm{d2}}\,(F_{\mathrm{M2}})$$

where the Mach factor F_{Mx} at any station x is given by the equation:

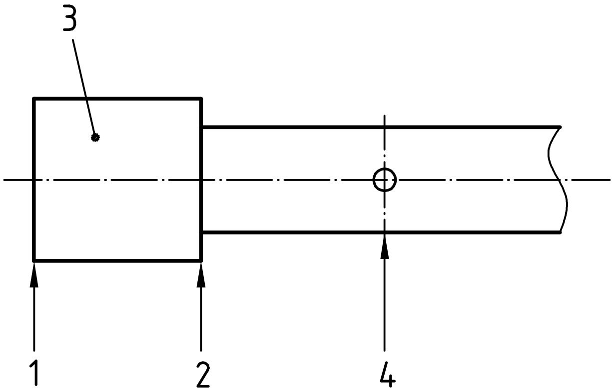
$$F_{\mathrm{M}} = 1 + \frac{Ma^2}{4} + \frac{Ma^4}{40} + \frac{Ma^6}{1600} + \ldots$$

and the value of local Mach number Ma_x at that station is given approximately by the equation:

$$Ma_x = \frac{q_m / \rho_x A_x}{\sqrt{\kappa R_w \, (t_x + 273{,}15)}}$$

When the air may be considered as incompressible ($p_F \leqslant 2000$ Pa, $Ma_2 \leqslant 0{,}15$ or by agreement between the user and manufacturer) then $F_{M2} = F_{M4} = 1$ and the following method is applied.

The gauge pressure p_{e2} at the fan outlet is calculated by adding an allowance for friction $(\zeta_{2\text{-}4})_4 p_{d4}$ (see annex D) to the gauge pressure p_{e4} measured at the test section on the outlet side. A correction for any difference in cross-section area up to 7 % at the two stations shall be allowed.



Key

3 is fan

Figure 6 — Type B installation

The formula for gauge pressure p_{e2} is:

$$p_{e2} = p_{e4} - p_{d2} \left[1 - \left(\frac{A_2}{A_4} \right)^2 \right] + (\zeta_{2-4})_4 \, p_{d4}$$

The conventional dynamic pressure at any station x in the airway is given by the equation:

$$p_{dx} = \rho_x \frac{v_{m\,x}^2}{2} = \frac{1}{2 \, \rho_x} \left[\frac{q_m}{A_x} \right]^2$$

where $\rho_x = \rho_2 = \rho_4 = \rho_a = \dfrac{p_a}{R_w (t_a + 273{,}15)}$

The fan pressure p_F is calculated by the equation:

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

$$p_{esg2} = p_{e4} - p_{d2} \left[1 - \left(\frac{A_2}{A_4} \right)^2 \right] + (\zeta_{2-4})_4 \; p_{d4} + p_{d2} = p_{e4} + p_{d4} \; \left[1 + (\zeta_{2\text{-}4})_4 \right]$$

where $(\zeta_{2-4})_4 = \left[\dfrac{\lambda}{D_{h4}}\right] L_{2\text{-}4}$

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

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

EXAMPLE

p_{esg1} = 0 Pa; p_{e4} = 932 Pa; p_{d4} = 60 Pa; v_{m2} = 10 m·s^{-1}; F_{M2} = 1; ρ_a = 1,2 kg·m^{-3}; p_{d2} = 60 Pa; $(\zeta_{2\text{-}4})_4$ = 0,35; $A_2 = A_4$

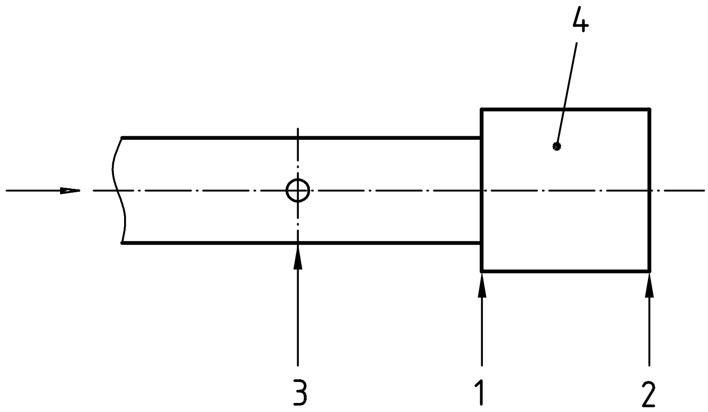
Then p_F = 932 – 0 + 0,35 × 60 + 60 × 1,0 = 1 013 Pa

When $p_F \geqslant$ 2 000 Pa and/or Ma_2 > 0,15 the methods described in ISO 5801:1997, subclauses 14.4 to 14.9.1 and 33, should be applied.

7.2.3.4 Type C Installation

Where a fan is installed as an extract fan under category C conditions (Figure 7), the average gauge pressure at the test section on the inlet side of the fan shall be measured in accordance with 7.2.2.

The reference (or effective) pressure is in this case the fan static pressure $p_{sF,}$ defined as the gauge stagnation pressure p_{e2} at the fan outlet (in this case 0) minus the gauge stagnation pressure p_{esg1} at the fan inlet.



Key

4 Fan

Figure 7 — Type C installation

The gauge stagnation pressure at the fan inlet p_{esg1} is given by the equation:

$$p_{esg1} = p_{e1} + p_{d1}\,(F_{M1})$$

where the Mach factor F_{Mx} at any station x is given by the relationship:

$$F_M = 1 + \frac{Ma^2}{4} + \frac{Ma^4}{40} + \frac{Ma^6}{1600} + \ldots$$

and the value of local Mach number Ma_x at that station is given approximately by the equation:

$$Ma_x = \frac{q_m / \rho_x A_x}{\sqrt{\kappa R_w (t_x + 273,15)}}$$

When the air may be considered as incompressible ($p_F \leqslant 2\,000$ Pa, $Ma_2 \leqslant 0,15$ or by agreement between the user and manufacturer) then $F_{M1} = F_{M3} = 1$ and the following method is applied.

The gauge pressure p_{e1} at the fan inlet is calculated by substracting an allowance for friction $(\zeta_{3\text{-}1})_1 p_{d1}$ (see annex D) from the gauge pressure p_{e3} measured at the test section on the inlet side. A correction for any difference in cross-section area up to 14 % at the two stations shall be allowed.

The formula for gauge pressure p_{e1} is:

$$p_{e1} = p_{e3} - p_{d1}\left[1 - \left(\frac{A_1}{A_3}\right)^2\right] - (\zeta_{3\text{-}1})_3\ p_{d1}$$

where

$$(\zeta_{3\text{-}1})_3 = \left[\frac{\lambda}{D_H}\right] L_{3\text{-}1}$$

and $(\zeta_{3-1})_1 = (\zeta_{3-1})_3 \left(\frac{A_1}{A_3}\right)^2$

The conventional dynamic pressure at any station x in the airway is given by the equation:

$$p_{dx} = \rho_x \frac{v_{m\,x}^2}{2} = \frac{1}{2\ \rho_x} \left[\frac{q_m}{A_x}\right]^2$$

where $\rho_x = \rho_2 = \rho_4 = \rho_a = \dfrac{p_a}{R_w(t_a + 273,15)}$

and the fan static pressure p_{sF} is calculated as:

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

$$= -p_{e3} + p_{d1}\left[1 - \left(\frac{A_1}{A_3}\right)^2\right] + (\zeta_{3\text{-}1})_3\ p_{d1} - p_{d1}$$

$$= -p_{e3} + p_{d1}\left[(\zeta_{3\text{-}1})_1 - \left(\frac{A_1}{A_3}\right)^2\right]$$

$$= -p_{e3} + p_{d3}\left[(\zeta_{3\text{-}1})_3 - 1\right]$$

The fan pressure p_F may be calculated as:

$$p_F = p_{esg2} - p_{esg1} = p_{d2} - p_{esg1} = p_{sF} + p_{d2}$$

NOTE p_{e3} will be negative and numerically greater than the negative terms in the expression; consequently p_{sF} will be positive.

EXAMPLE

p_{e2} = 0 Pa; p_{e3} = –1 000 Pa; v_{m1} = 10 m·s^{-1}; F_{M1} = 1; ρ_a = 1,2 kg·m^{-3}; p_{d1} = 60 Pa; $A_1 = A_3$; $(\zeta_{3-1})_3 = (\zeta_{3-1})_1$ = 0,2

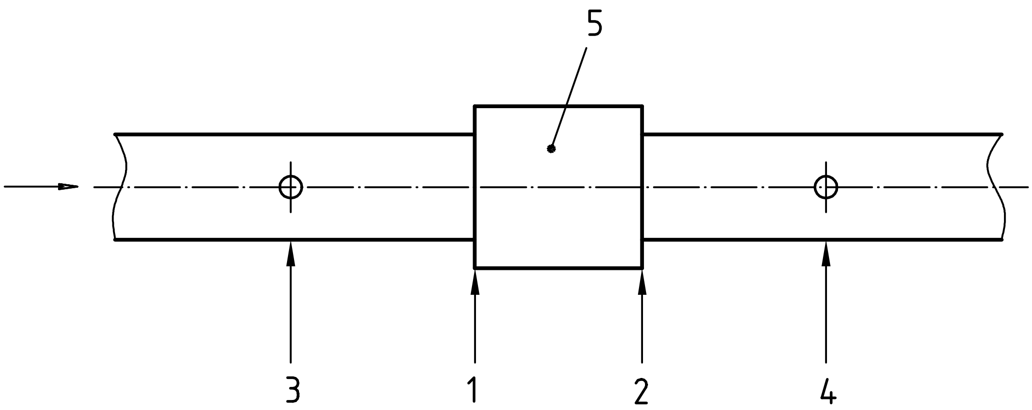
Then p_F = 1 000 + 0 + 0,2 × 60 – 60 × 1,0 = 952 Pa

When $p_F \geqslant$ 2 000 Pa and/or Ma_2 > 0,15 the methods described in ISO 5801:1997, subclauses 14.4 to 14.9.1 and 34 should be applied.

7.2.3.5 Type D Installation

Where a fan is installed as a booster fan under category D conditions (Figure 8), the average gauge pressures at the test section on the outlet side of the fan and at the test section on the inlet side of the fan shall be measured in accordance with 7.2.2.

The reference (or effective) pressure is in this case the fan pressure p_F defined as the gauge stagnation pressure p_{esg2} at the fan outlet minus the gauge stagnation pressure p_{esg1} at the fan inlet.



Key

5 Fan

Figure 8 — Type D installation

The gauge stagnation pressure at the fan outlet p_{esg2} is given by the equation:

$$p_{esg2} = p_{e2} + p_{d2}\,(F_{M2})$$

and the gauge stagnation pressure at the fan inlet p_{esg1} by the equation:

$$p_{esg1} = p_{e1} - p_{d1}\,(F_{M1})$$

where the Mach factor F_{Mx} at any station x is given by the relationship:

$$F_M = 1 + \frac{Ma^2}{4} + \frac{Ma^4}{40} + \frac{Ma^6}{1600} + \ldots$$

and the value of local Mach number Ma_x at that station is given approximately by the equation:

$$Ma_x = \frac{q_m / \rho_x A_x}{\sqrt{\kappa R_w (t_x + 273{,}15)}}$$

When the air may be considered as incompressible ($p_F \leqslant 2\,000$ Pa, $Ma_2 \leqslant 0{,}15$ or by agreement between user and manufacturer) then $F_{M1} = F_{M2} = F_{M3} = F_{M4} = 1$ and the following method is applied:

$$\rho_x = \rho_1 = \rho_2 = \rho_3 = \rho_4 = \rho_a = \frac{p_a}{R_w (t_a + 273{,}15)}$$

The fan pressure p_F is given by the equation (see 7.2.3.2 and 7.2.3.3):

$$\begin{aligned} p_F &= p_{esg2} - p_{esg1} \\ &= p_{e2} + p_{d2} - p_{e1} - p_{d1} \\ &= p_{e4} - p_{d2} \left[1 - \left(\frac{A_2}{A_4} \right)^2 \right] + (\zeta_{(2\text{-}4)})_4 \; p_{d4} + p_{d2} \; - p_{e3} + p_{d1} \left[1 - \left(\frac{A_1}{A_3} \right)^2 \right] + \; (\zeta_{(3\text{-}1)})_1 \, p_{d1} - p_{d1} \\ &= p_{e4} + p_{d4} \; \left[1 + (\zeta_{2\text{-}4})_4 \right] \; - p_{e3} + p_{d3} \left[(\zeta_{3\text{-}1})_3 - 1 \right] \end{aligned}$$

where

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

$$(\zeta_{3\text{-}1})_3 = \left[\frac{\lambda}{D_{h4}} \right] L_{3\text{-}1}$$

and $(\zeta_{3\text{-}1})_1 = (\zeta_{3\text{-}1})_3 \left(\frac{A_1}{A_3} \right)^2$

(see annex D).

The fan static pressure p_{sF} is given by:

$$\begin{aligned} p_{sF} &= p_{e2} - p_{esg1} \\ &= p_{esg2} - p_{d2} - p_{esg1} \\ &= p_F - p_{d2} \end{aligned}$$

EXAMPLE (all pressures are in pascals)

p_{e4} = 520 Pa; $p_{d4} = p_{d2}$ = 60 Pa; p_{e3} = –390 Pa; ρ_a = 1,2 kg·m^{-3}; $p_{d3} = p_{d1}$ = 50 Pa; $A_1 = A_3$ and $A_2 = A_4$; $(\zeta_{2\text{-}4})_4$ = 0,35; $(\zeta_{3\text{-}1})_1 = (\zeta_{3\text{-}1})_3$ = 0,26;

Then p_F = 520 – 0 + 0,35 × 60 + 60 × 1,0 – (–390) + 0 + 0,26 × 50 –50 × 1,0 = 954 Pa

When $p_F \geqslant 2000$ Pa and/or $Ma_2 > 0{,}15$ the methods described in ISO 5801:1997, subclauses 14.4 to 14.9.1 and 33, 34, 35, should be applied.

7.2.3.6 Booster fans when used in mines

In the case of an underground booster fan, it is not usually possible to measure the gauge pressures in the airways on each side of the fan relative to an ambient pressure; the gauge pressure difference Δp_e between the outlet and inlet sides of such a fan shall be measured by the following method.

The gauge pressure Δp_{e3} at the test section on the inlet side of the fan relative to the static pressure at the centre of the test section on the outlet side of the fan is measured by a gauge pressure tube traverse at the inlet side test section in association with a differential manometer and a stationary static pressure tube in the centre of the outlet side test section (p_{e3} is normally negative). The gauge pressure p_{e4} at the test section on the outlet side of the fan, relative to static pressure at the centre of the test section on the inlet side of the fan, is also measured in a similar manner (p_{e4} is normally positive).

The gauge pressure p_{ec} at the centre point of the test section at the fan outlet is measured relative to the gauge pressure at the centre point of the test section at the fan inlet (p_{ec} is normally positive).

The value of Δp_e is obtained from the following equation:

$$\Delta p_e = p_{e4} - p_{e3} - p_{ec}$$

The fan pressure p_F is obtained from the equation:

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

$$= \Delta p_e + p_{d4} - p_{d3} + \zeta_{(2\text{-}4)4}\, p_{d4} + \zeta_{(3\text{-}1)3}\, p_{d1}$$

NOTE It is anticipated that velocities in mine airways will never be high enough for the Max < 0,1 threshhold to be exceeded.

8 Determination of flowrate

8.1 Choice of measuring method

The flowrate in a section of the actual airway may be determined in two ways: either by determining the velocity at various points in this section and calculating the mean velocity; or by measuring the pressure difference produced by a differential pressure device (orifice plate, Venturi tube, nozzle). The choice of measuring methods will be based on the following considerations.

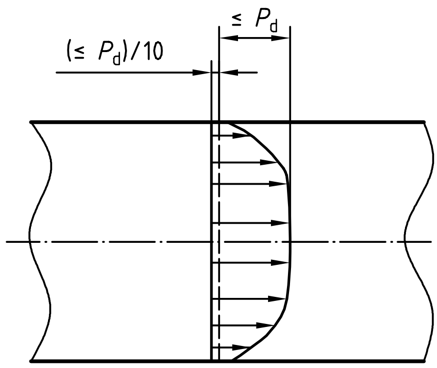
a) Measurements obtained by velocity area methods may be time consuming and require delicate manipulation but they are, in many cases, the only appropriate methods. It is desirable to carry out a preliminary test to determine the conditions (number of readings and duration of observations) in which such measurements are to be taken.

b) Differential pressure devices make it easy to obtain a properly reproducible measurement of the mean value in time of the flowrate even when these measurements are taken by different people at different times. Their use is, in particular, restricted by their own resistance, the required straight length of ducts and the need to use them in ducts of circular cross section.

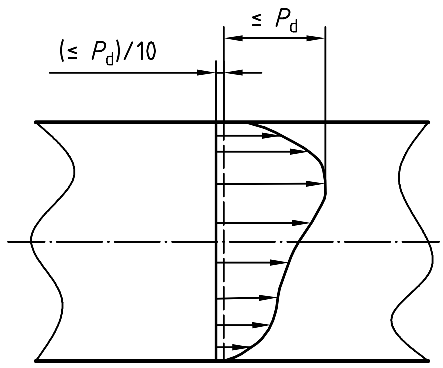
8.2 Choice of measuring section

8.2.1 Freedom from swirl and straightness of flow

The section for measuring the flowrate shall be selected such that there is no appreciable swirl of fluid in it and such that the flow lines are approximately parallel and very close to the direction perpendicular to the plane measurement.

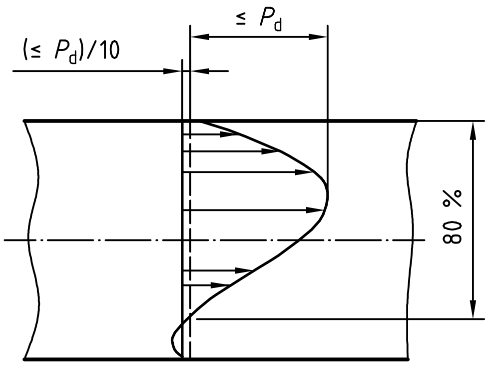


a) Ideal p_d distribution



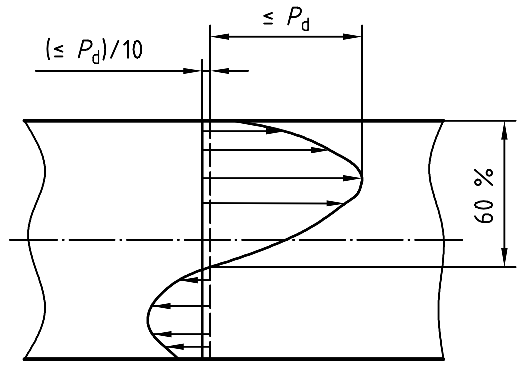
NOTE Also satisfactory for flow into fan inlets, but may be unsatisfactory for flow into inlet boxes; may produce swirl in boxes.

b) Good p_d distribution



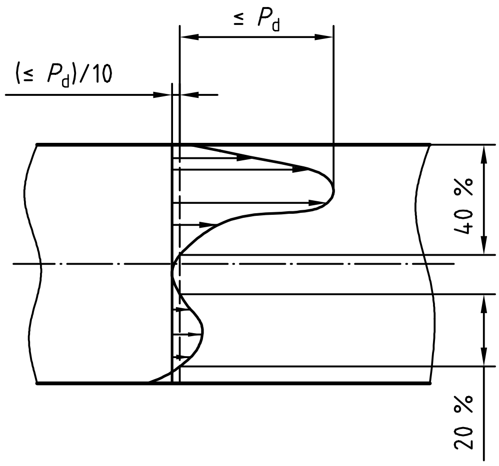
NOTE More than 75 % of p_d readings greater than p_{dmax}/10 (unsatisfactory for flow into fan inlets or inlet boxes).

c) Satisfactory p_d distribution



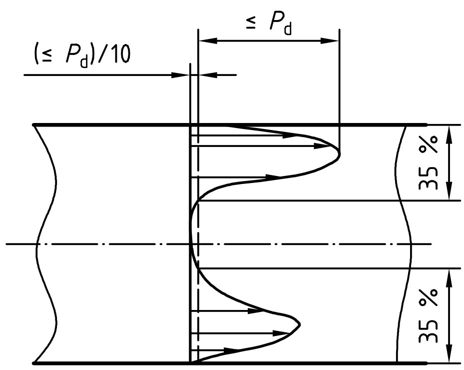
NOTE Less than 75 % of p_d readings greater than p_{dmax}/10 (unsatisfactory for flow into fan inlets or inlet boxes).

d) Unsatisfactory p_d distribution



NOTE Less than 75 % of p_d readings greater than p_{dmax}/10 (unsatisfactory for flow into fan inlets or inlet boxes).

e) Unsatisfactory p_d distribution



NOTE Less than 75 % of p_d readings greater than p_{dmax}/10 (unsatisfactory for flow into fan inlets or inlet boxes).

f) Unsatisfactory p_d distribution

Figure 9 — Typical dynamic pressure distributions encountered in dynamic pressure measurement planes in fan system installations

In cases where it is difficult to satisfy these conditions, it is permissible to add an anti-swirl device upstream of the measuring plane by agreement between the parties. This device shall be positioned in relation to the measuring plane such that the flow is almost axial and free from swirl in this measuring plane. This device shall not affect the flow conditions at the intake and discharge of the fan. It is also permissible to modify the airway over a limited length, by means of an internal lining, for example, in order to improve the shape of the cross section of measurement.

Finally, if it proves impossible to find a plane fulfilling the above conditions, a plane for measuring the flowrate shall be selected by common agreement, but it is stressed that this will affect the accuracy of the measurement.

8.2.2 Acceptability of velocity profile

The velocity distribution should be uniform throughout the traverse plane. This uniformity of distribution is considered acceptable when more than 75 % of the dynamic pressure measurements are greater than a tenth of the maximum measurement (see Figure 9).

8.3 Determination of flowrate using differential pressure devices

Standardized differential pressure devices constructed and used in accordance with ISO 5167-1 may be used without preliminary calibration provided that it is ensured that the flow conditions existing for the extreme flowrates required for the test are acceptable and allow the use of the numerical data given in this International Standard.

Differential pressure devices involving the primary elements defined in ISO 5167-1 but used with the straight lengths described in annex C may be used without preliminary calibration provided that it is ensured that the flow conditions existing for the extreme flowrates required for the test are acceptable and allow the use of the numerical data given in this International Standard taking into account the requirements of annex C.

The use of non-standardized differential pressure devices is permitted provided that the supplier and user agree on the choice of the device and the method of calibration which shall have the same accuracy as that of the standardized device. The non-standardized differential pressure devices which may be used in these conditions are:

a) at the inlet of the duct: an orifice plate, a nozzle, a Venturi tube, a flared inlet or a conical inlet or a Borda inlet,

b) at the outlet of the duct: an orifice plate, a nozzle, a Venturi tube.

8.4 Determination of flowrate by velocity area methods

8.4.1 General recommendations

8.4.1.1 As far as possible, the mean velocity shall be sufficiently high to allow the use of a measuring instrument in the range where there is a high level of accuracy.

8.4.1.2 The flow measurement plane shall be located in any suitable straight length where the airflow conditions are substantially axial, symmetrical and free from swirl or flow reversal. This implies taking due account of the disturbance to the flow caused by bends, sudden expansion or contraction, obstacles or by the fan itself.

8.4.1.3 If possible the flow measurement plane shall be chosen in a straight length of airway of uniform cross section, free from any obstruction which might modify the flow in the measuring plane. This straight length, known as the test length, shall be at least twice the hydraulic diameter D_h of the airway.

The flow measurement plane should, if possible, be at a distance of at least 1,5 D_h from the fan inlet if located on the inlet side of the fan or at least 5 D_h from the fan outlet if located on the discharge side of the fan.

The adoption of these minimum distances does not imply that the requirements of 8.4.1.2 are fulfilled.

If it is not possible to choose a measuring plane which fulfills these conditions, its position shall be chosen by common agreement between the parties. In this case, the validity of the results shall be the subject to mutual agreement.

8.4.1.4 A sufficient number of measuring points in the cross section shall be selected having regard to both the wall effects and possible velocity variations in the central area.

8.4.1.5 When measuring the flowrate by velocity area methods, the flowrate shall be kept as constant as possible throughout the procedure.

For this purpose, the necessary precautions shall be taken to keep the following factors as constant as possible throughout the whole procedure:

a) the equivalent orifice or the resistance of the airway expressed in any other terms;

b) the speed of rotation of the fan;

c) the pressure and temperature of the fluid in the system.

8.4.1.6 When carrying out a dynamic pressure (or velocity) traverse under site conditions and, to some extent under laboratory conditions also, it is not uncommon to notice some fluctuation of the reading at a single point even though the total flowrate and system resistance are kept substantially constant. This is due to the nature of turbulent fluid flow where slight random changes of velocity profile do occur. For this reason a good visual average reading shall be taken at each traverse point over a period of not less than 15 s. The total flowrate shall then be determined from the area of the duct and the average of all the separate velocity readings or the average of the square roots of all the dynamic pressure readings. The complete traverse should then be repeated one or more times until the flow measurement calculated from two successive traverses does not differ by more than 2 %. The mean of those two measurements should then be taken as the correct value.

8.4.2 Siting of measuring points

8.4.2.1 General

The measuring probe should be located in the duct with a tolerance equal to the smaller of the following two values:

a) – 0,05 y (y being the distance of the probe to the nearest duct wall);

b) – 0,005 L_p (L_p being the inner dimension of the duct perpendicular to the nearest wall to the probe).

If one or other of these tolerances is less than 1 mm, the tolerance shall be taken as 1 mm.

8.4.2.2 Circular sections

For circular sections the mean diameter is taken as equal to the arithmetic mean of the measured values on the basis of at least three diameters of the measuring section, with roughly equal angles between them. If the difference between two adjacent diameters is greater than 1 %, the number of diameters measured shall be doubled.

The dimensions of the duct in the plane of the measuring section shall be determined with an uncertainty of less than 0,25 %.

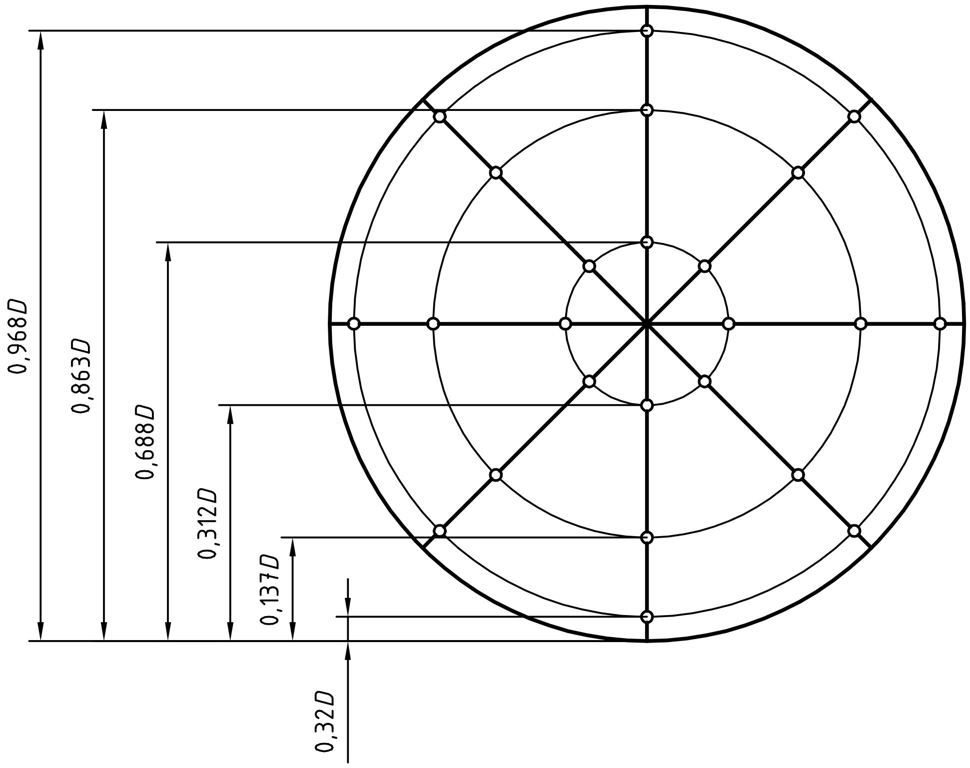
The minimum number of measuring points is 24. The measuring points shall be spread over a minimum of three diameters with at least three points per radius, in accordance with the provisions set out in one of the following two methods: log-Tchebycheff or log-linear.

By way of example, it is possible to take four diameters with three measuring points per radius (see Figure 10) or three diameters with four measuring points per radius (see Figure 11).

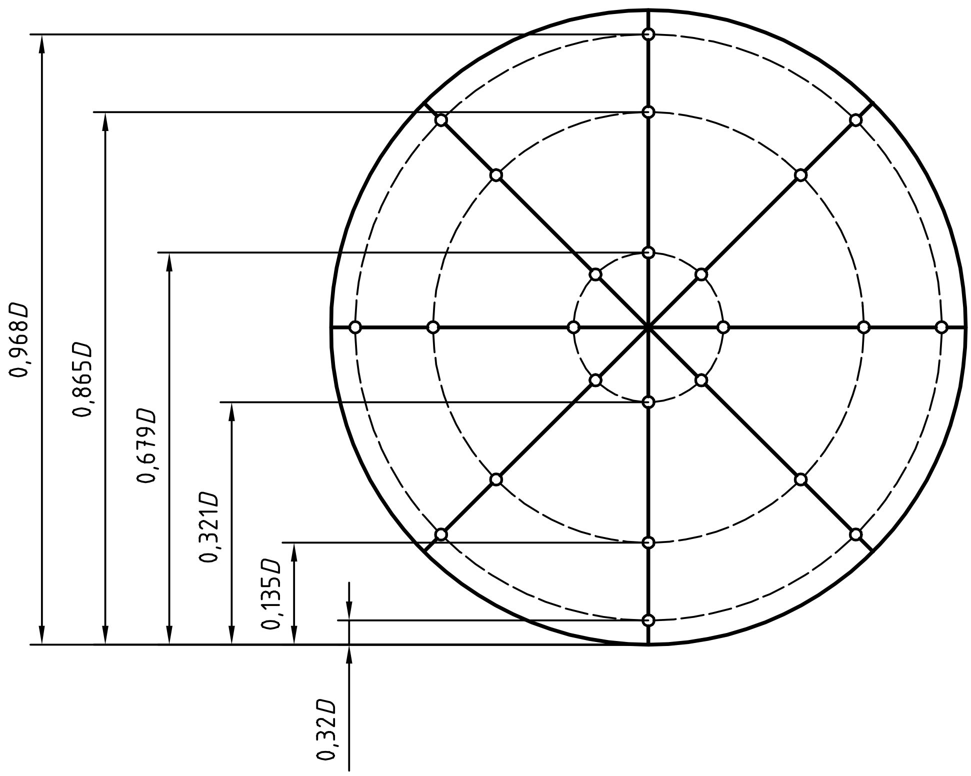
The Tables 2 and 3, respectively, give the siting of the measuring points on the basis of the log-Tchebycheff and log-linear rules, viz:

— for three points per radius (Table 2);

— for four points per radius (Table 3).

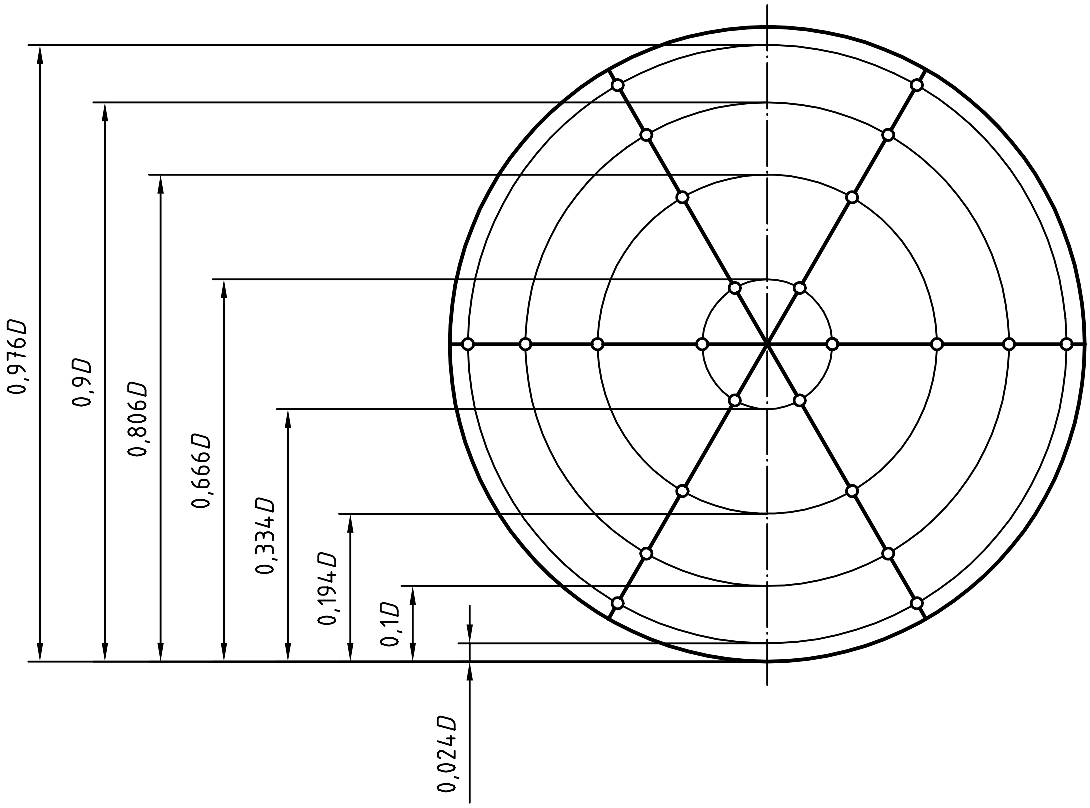


a) Log-Tchebycheff method

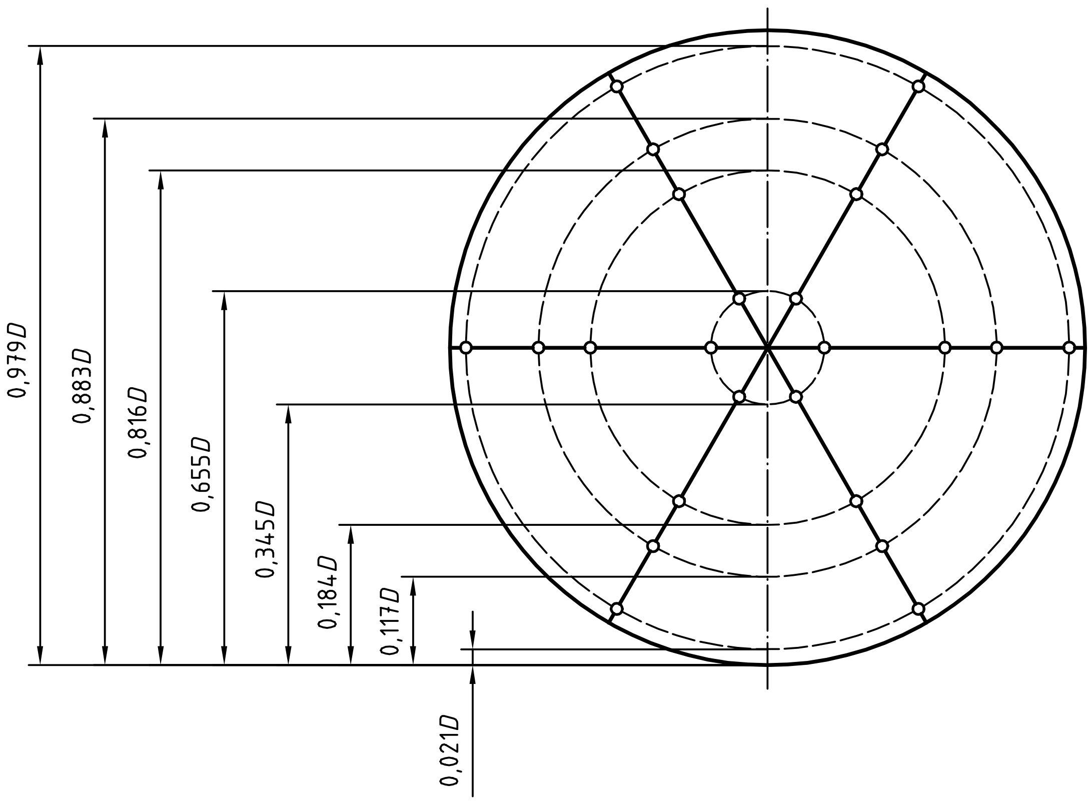


b) Log-liner method

Figure 10 — Siting of measuring points in a circular section with four diameters and three measuring points per radius



a) Log-Tchebycheff method



b) Log-liner method

Figure 11 — Siting of measuring points in a circular section with three diameters and four measuring points per radius

Table 2 — Three points per radius

Point	log-Tchebycheff y/D	log-linear y/D
1	0,032	0,032
2	0,137	0,135
3	0,312	0,321
4	0,688	0,679
5	0,863	0,865
6	0,968	0,968

Table 3 — Four points per radius

Point	log-Tchebycheff y/D	log-linear y/D
1	0,024	0,021
2	0,100	0,117
3	0,194	0,184
4	0,334	0,345
5	0,666	0,655
6	0,806	0,816
7	0,900	0,883
8	0,976	0,979

The mean velocity in the duct is obtained by calculating the arithmetic mean of the velocities at the individual points.

The volume flowrate shall be calculated by multiplying this mean velocity by the area calculated using the mean diameter.

8.4.2.3 Annular sections immediately upstream from an axial flow fan

The velocity area method may be used for measuring the flowrate in annular sections provided that the following conditions are fulfilled.

a) The minimum number of equally spaced radii shall be six.

b) The minimum number of four measuring points per radius shall be spread out along the radii in accordance with the log-linear rule.

   The positioning of the measuring points (Figure 12) depends on the value of the ratio of the diameters D_a/D and is given in Table 4 (for four points per radius). For intermediate values, the position of the measuring points will be located by linear interpolation of the data in this table.

   The mean velocity shall be obtained by calculating the arithmetic mean of all the velocities recorded in the section.

c) The flowrate shall be determined by multiplying the area of the cross section by the mean velocity.

d) To determine the area of the section, the internal diameter and the thickness e of the ring shall be measured with a common tolerance of 0,25 %.

In order to reduce any eccentricity error, the thickness shall be taken as the mean of the measurements carried out on the basis of a minimum of four radii spaces at equal angles. If two radial dimensions differ by more than 1 %, the number of dimensions measured shall be doubled. The internal diameter shall be calculated from the measurement of the corresponding perimeter. The area of the annular section is given by the expression:

$$A = \pi\,(D_a + e)e$$

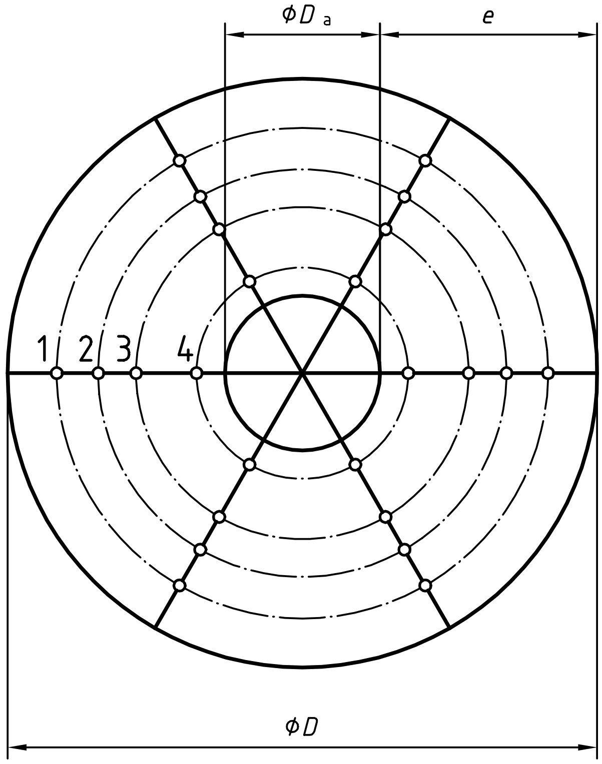


Figure 12 — Siting of the measuring points for an annular section with three diameters and four measuring points per radius

Table 4 — Point distribution in an annular duct

D_a/D	Values of y/D			
	Point 1	**Point 2**	**Point 3**	**Point 4**
0,05	0,023 7	0,097 3	0,202 4	0,349 8
0,1	0,023 5	0,096 5	0,200 4	0,345 2
0,15	0,023 2	0,095 1	0,197 0	0,336 2
0,20	0,022 8	0,093 2	0,192 4	0,324 0
0,25	0,022 2	0,090 8	0,186 5	0,309 7
0,30	0,021 6	0,087 9	0,179 4	0,293 6
0,35	0,020 8	0,084 4	0,171 4	0,276 1
0,40	0,019 9	0,080 4	0,162 2	0,257 5
0,45	0,018 8	0,076 1	0,152 2	0,238 2
0,50	0,017 7	0,071 2	0,141 3	0,218 2
0,55	0,016 4	0,065 9	0,129 6	0,197 6
0,60	0,015 0	0,060 4	0,118 0	0,176 7
0,65	0,013 6	0,053 8	0,104 3	0,155 4
0,70	0,011 9	0,047 2	0,090 7	0,133 7
0,75	0,010 2	0,040 2	0,076 6	0,111 9
0,80	0,008 4	0,032 9	0,062 0	0,089 8
0,85	0,006 3	0,025 1	0,047 1	0,067 6
0,90	0,004 4	0,017 1	0,030 6	0,045 2
0,95	0,002 2	0,008 7	0,016 0	0,022 6

8.4.2.4 Rectangular sections

In the case of ducts with straight rectangular cross sections, the height and length of the section shall be measured along the lines given in Figure 13. If the difference between two adjacent heights or lengths is greater than 1 %, the number of measuring points in this direction shall be doubled. The mean height of the section will be taken as the arithmetic mean of all the heights measured and the mean length of the section as the arithmetic mean of all the lengths measured.

The area of the section shall be conventionally regarded as equal to the mean length multiplied by the mean height.

The dimensions of the duct required for the calculation of the area of the measuring section shall be determined with an uncertainty less than 0,25 %.

The number of cross-lines (parallel to the small side) and the number of measuring points per cross-line shall be a minimum of 5. It is recommended that the number of cross-lines be increased beyond 5 if the aspect ratio of the rectangle (ratio of its length to its height) is very different from 1.

The measuring points are arranged on the basis of the log-Tchebycheff method and Table 5 shows the siting of these measuring points.

The volume flowrate is equal to the area of the section multiplied by the arithmetic mean of the local velocities measured at the various measuring points.

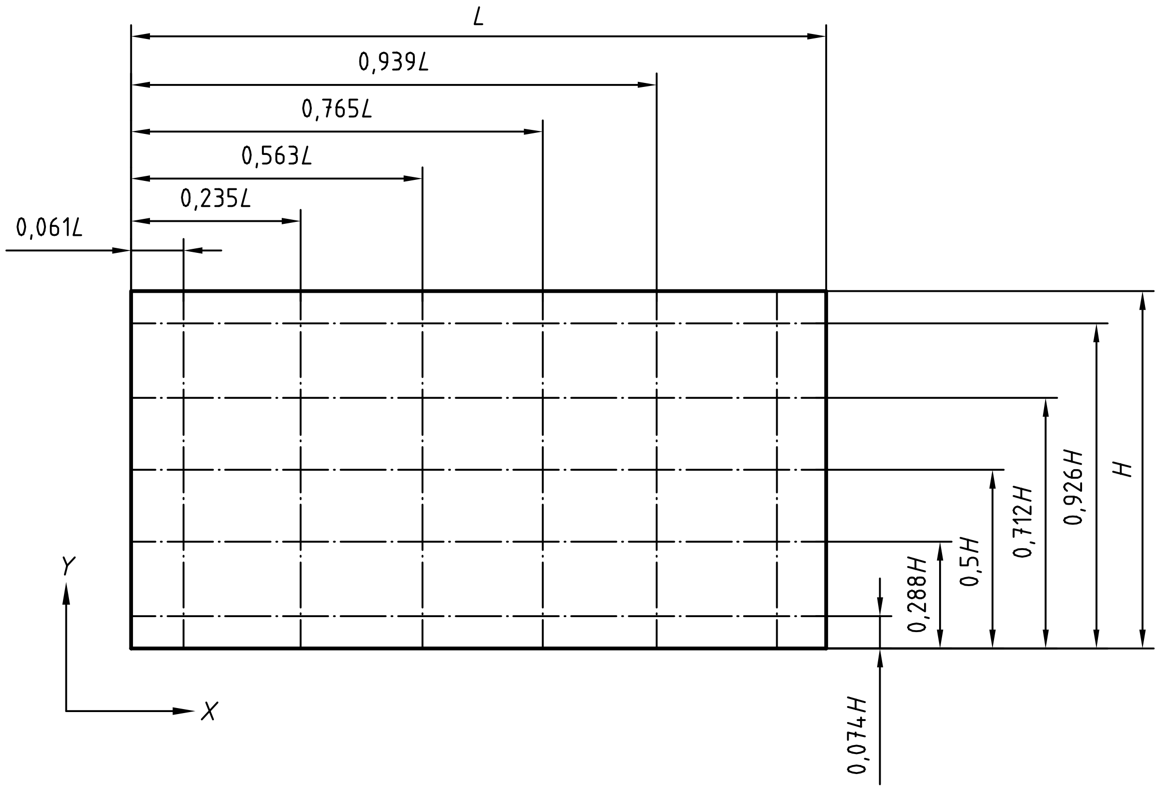


Figure 13 — Rectangular section with six cross-lines and five measuring points per cross-line

Table 5 — Point and line distribution according to log-Tchebycheff in rectangular duct

Number of cross-lines or number of measuring points per cross-line	Point	Values of $\frac{x_i}{L}$ or $\frac{y_i}{H}$
5	1	0,074
	2	0,288
	3	0,500
	4	0,712
	5	0,926
6	1	0,061
	2	0,235
	3	0,437
	4	0,563
	5	0,765
	6	0,939
7	1	0,053
	2	0,203
	3	0,366
	4	0,500
	5	0,634
	6	0,797
	7	0,947

8.4.2.5 Sections of any other shape

Temporary modifications (for instance the insertion of a low resistance lining) may be used to provide a suitable test length of rectangular or circular cross section. However, where this is not possible, the volumetric flowrate of fluid flowing through a straight length of duct or regular, non-re-entrant cross section can, most conveniently, be determined by a modification of the log-Tchebycheff traverse pattern normally applied to a rectangular cross section.

8.4.2.6 Modified log-Tchebycheff traverse pattern

A cross section may be considered to be regular, with the agreement of the parties concerned, if a base line can be drawn parallel to a long straight side or to the major axis of the area with traverse lines set out at right angles to it such that the perimeter line of the area crosses the ends of the traverse lines in a relatively smooth curve or straight line. It is also desirable that the angle between the perimeter line and any traverse line should be not too far removed from a right angle. However, in the procedures outlined below, provision is made for some deviation from this requirement in the case of traverse lines in the marginal zones next to the duct wall and in all the cases set out in Figures 15 to 27.

In order to be able to apply this modified log-Tchebycheff traverse method to the cross section of a non-rectangular duct, typified by Figure 14, it is necessary to weight the average velocity for each traverse line in proportion to its length. In addition, for the highest accuracy, some adjustment shall be made to the position of the traverse lines (Nos. 1 and N_r in the diagram) in the two marginal zones next to the duct wall. A detailed computer analysis has been carried out to determine the correct position for the two marginal traverse lines in thirteen typical duct shapes as shown in Figures 15 to 27.

These take account of the effects of both duct friction and wall shape.

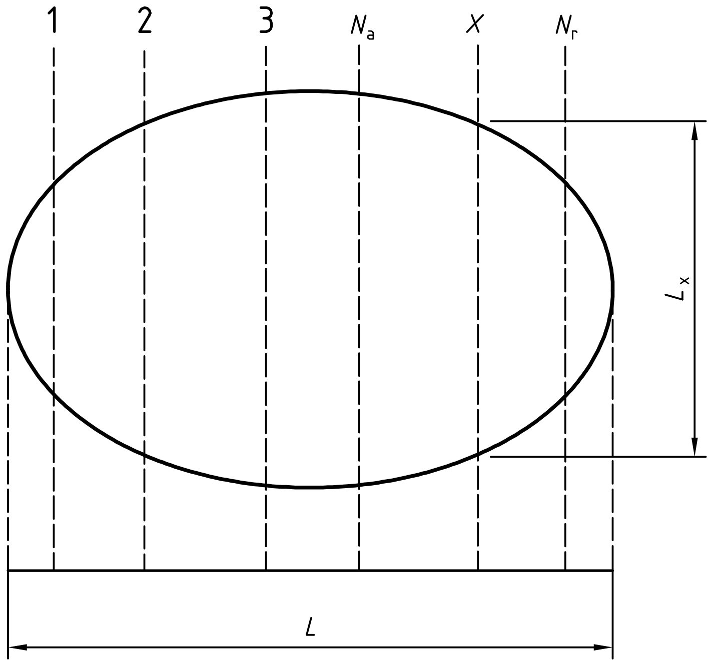
They permit a very simple determination of volume flowrate by multiplying the average velocity for each traverse line by the line length, summing this product for all the traverse lines and multiplying the sum by the length (between duct walls) of the chosen base line and dividing by the number of traverse lines.

The method of determining the precise position of traverse lines in the marginal zones (Nos. 1 and N_r in Figure 14 diagram) is set out in 8.4.2.7 and annex B, and involves the use of the equation:

$$y = kx^{-1/p_l}$$

The value of p_l is dependent on the wall roughness and the Reynolds Number, and is set out in general terms in Table 6.

However, in the great majority of installations, it is not easy to make a precise determination of the parameters involved and, as the variation of the position of the marginal traverse line for p_l values between 5 and 10 is relatively small, the marginal traverse line positions set out in Figures 15 to 27 are all based on the mean p_l value of 7.



Key

L is the baseline length.

L_x is the traverse length.

N_a is the traverse line.

Figure 14 — Regular but non-rectangular or circular cross-section duct showing sample distribution of traverse lines and measuring points

Table 6 — Value of p_l as a function of the surface roughness of the walls and of the Reynolds number

Rough wall with low Reynolds number	$p_l = 5$
Rough wall with high Reynolds number or smooth wall with low Reynolds number	$p_l = 7$
Smooth wall with high Reynolds number	$p_l = 10$

Instructions for carrying out this procedure for any duct shape corresponding to one of those in Figures 15 to 27 are as follows.

a) A "baseline" shall be chosen parallel to the major axis of the duct cross section.

b) Velocity measurements shall be taken at prescribed points along at least six parallel traverse lines at right angles to the baseline and at right angles to the axis of flow.

c) Traverse lines numbers 2 to (N_r – 1) shall be distributed along the baseline according to the log-Tchebycheff rule (see Table 5).

d) Traverse lines 1 and N_r shall be placed in accordance with the appropriate table adjacent to Figures 15 to 27. The value of p_l in these tables shall be selected from Table 6 and if no specific determination of wall roughness can be made, then the value p_l = 7 should be used.

e) At least six measuring points shall be located along each traverse line in accordance with the log-Tchebycheff rule (see Table 5). Where any traverse line is very short, the number of measuring points may be reduced to 5 but the total number of measuring points for the whole area shall not be less than 35.

f) Velocity measurements shall be taken at the prescribed points and the arithmetic mean velocity for each traverse line shall be determined.

g) The volume flowrate for the whole airway is found by

1) multiplying the arithmetic mean velocity for each traverse line by the line length,

2) summing the values so obtained, and

3) multiplying this sum by the baseline length between duct walls and dividing by N_r.

$$q_V = \left[(v_1\, l_1) + (v_2\, l_2)\ \ldots\ldots (v_N\, l_N)\right] \frac{L}{N_r}$$

where

q_V is the total volume flowrate;

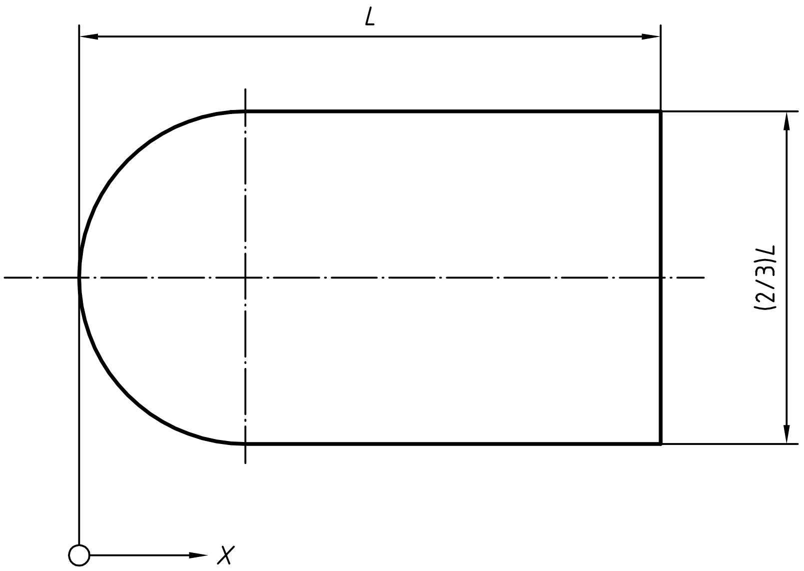
v_N is the arithmetic mean velocity for line x;

l_N is the length of traverse line x;

N_r is the number of traverse lines;

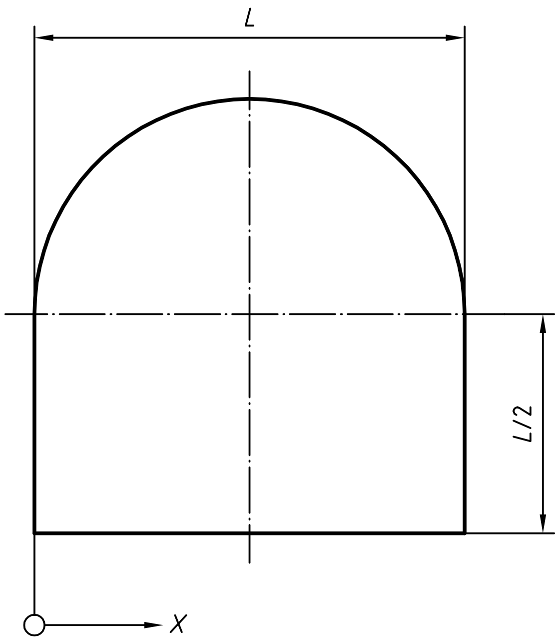
L is the base line length between duct walls.

Several configurations are now examined:



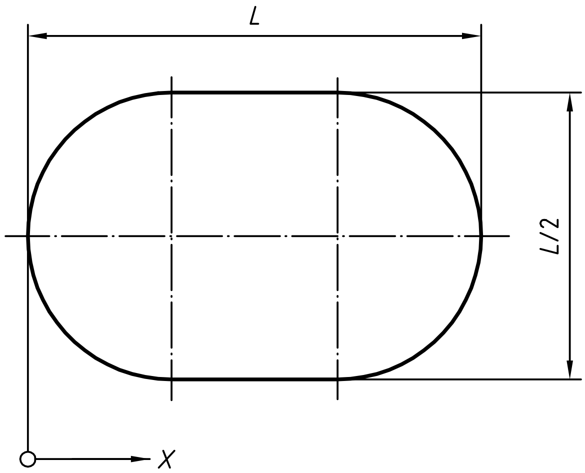
N_r	$\frac{X_1}{L}$	$\frac{X_N}{L}$
6	0,074	0,939
7	0,064	0,947

Figure 15



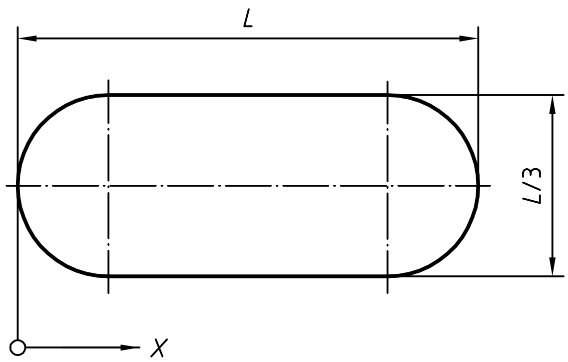
N_r	$\frac{X_1}{L}$	$\frac{X_N}{L}$
6	0,075	0,925
7	0,065	0,935

Figure 16



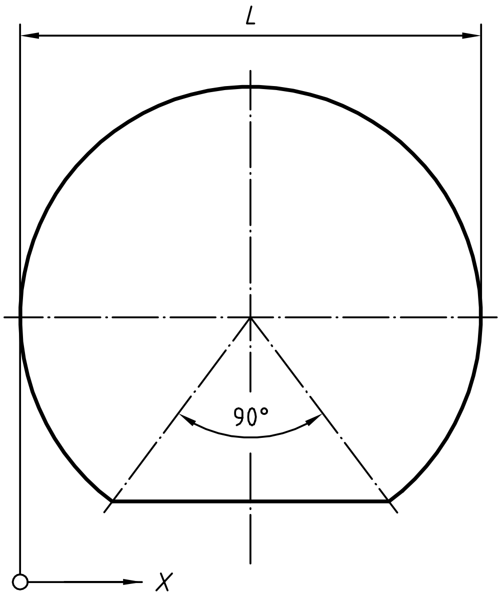
N_r	$\frac{X_1}{L}$	$\frac{X_N}{L}$
6	0,070	0,930
7	0,059	0,941

Figure 17



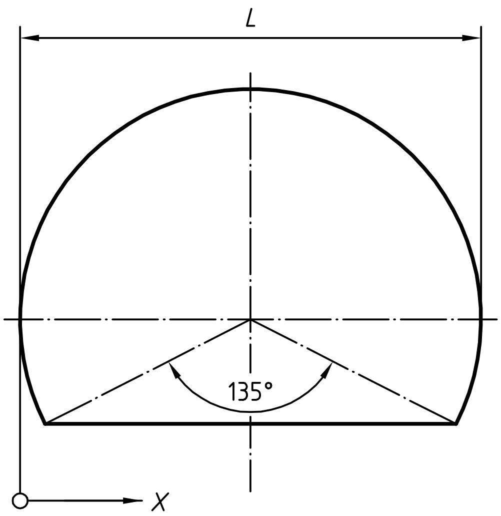
N_r	$\frac{X_1}{L}$	$\frac{X_N}{L}$
6	0,060	0,940
7	0,049	0,951

Figure 18



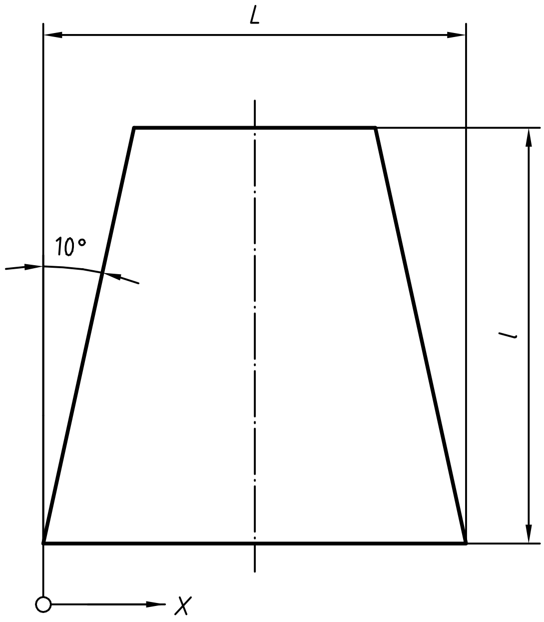
N_r	$\frac{X_1}{L}$	$\frac{X_N}{L}$
6	0,075	0,925
7	0,065	0,935

Figure 19



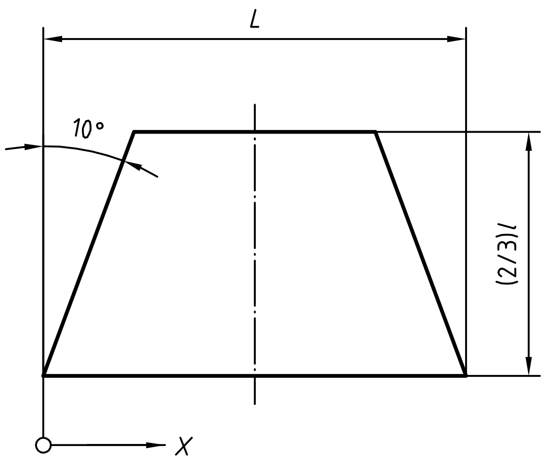
N_r	$\frac{X_1}{L}$	$\frac{X_N}{L}$
6	0,070	0,930
7	0,059	0,941

Figure 20



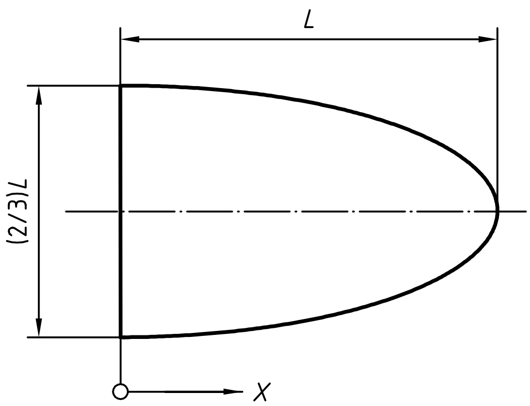
N_r	$\frac{X_1}{L}$	$\frac{X_N}{L}$
6	0,086	0,914
7	0,073	0,927

Figure 21



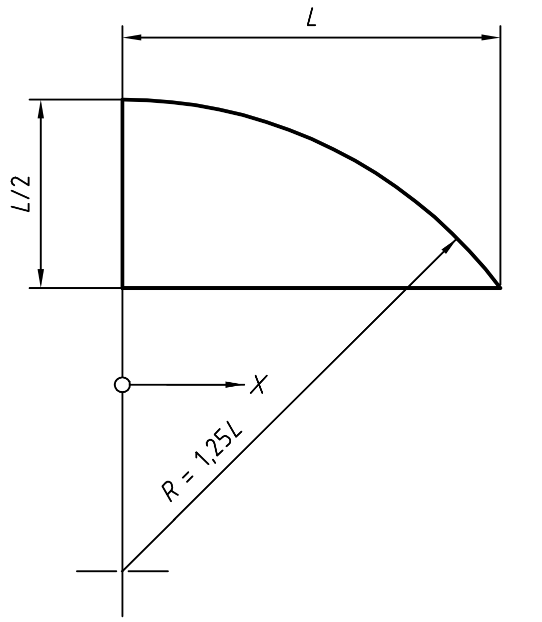
N_r	$\frac{X_1}{L}$	$\frac{X_N}{L}$
6	0,079	0,921
7	0,071	0,929

Figure 22



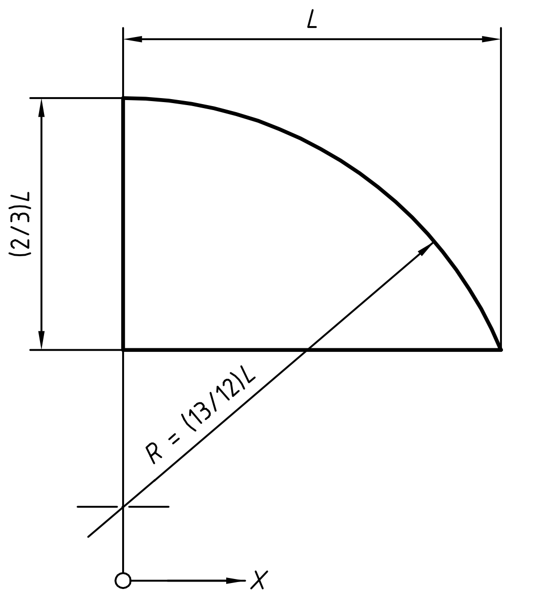
N_r	$\frac{X_1}{L}$	$\frac{X_N}{L}$
6	0,064	0,924
7	0,055	0,935

Figure 23



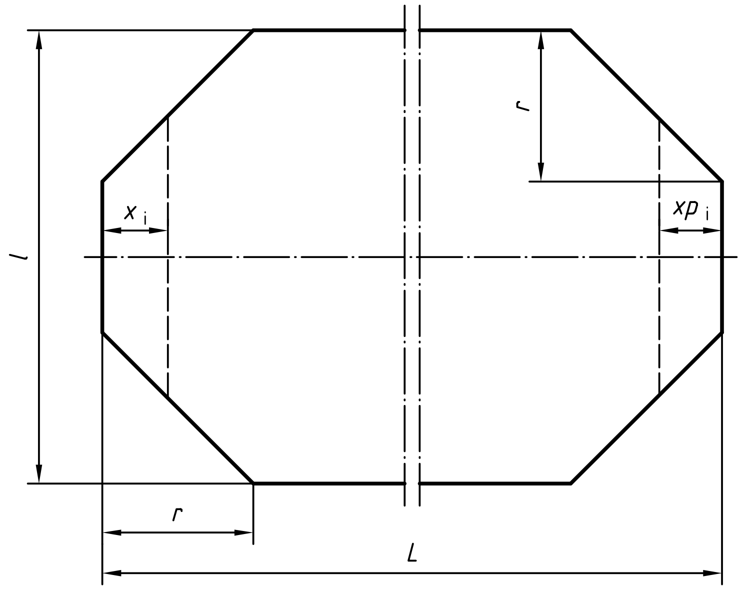
N_r	$\frac{X_1}{L}$	$\frac{X_N}{L}$
6	0,063	0,917
7	0,055	0,929

Figure 24



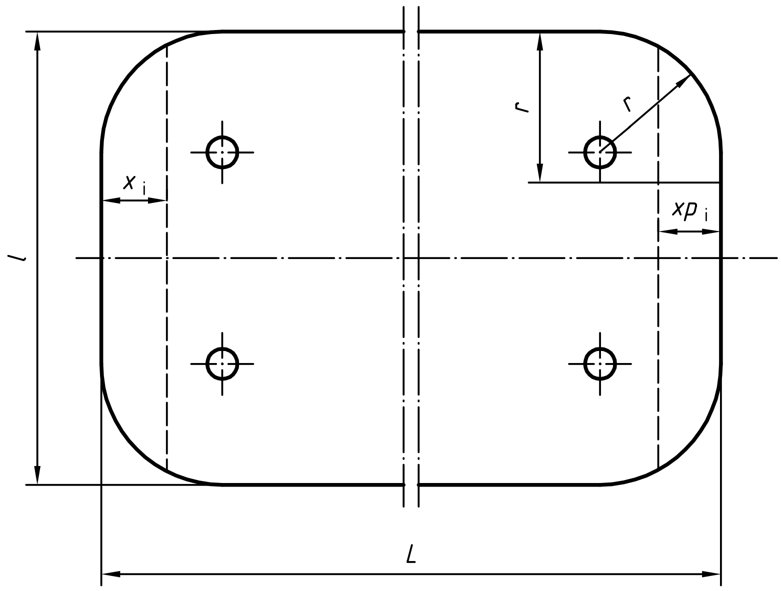
N_r	$\frac{X_1}{L}$	$\frac{X_N}{L}$
6	0,063	0,919
7	0,055	0,930

Figure 25



N_r	$\frac{L}{l}$	$\frac{r}{l}$	$\frac{x_i}{L}$
6	1	1/3	0,081 2
7			0,069 1
6		1/4	0,079 6
7			0,067 6
6	2	1/3	0,083 1
7			0,070 9
6		1/4	0,077 9
7			0,069 1

Figure 26



N_r	$\frac{L}{l}$	$\frac{r}{l}$	$\frac{x_i}{L}$
6	1	1/3	0,072 5
7			0,062 4
6		1/4	0,070 5
7			0,060 9
6	2	1/3	0,068 1
7			0,059 7
6		1/4	0,063 9
7			0,056 7

Figure 27

8.4.2.7 Cases where the duct cross section does not correspond sufficiently closely to one of the shapes in Figures 15 to 27

Provided there are no abrupt changes of wall contour, the one or two marginal segments of width L/N_r at one or either side of the duct cross section are dealt with in the following manner.

Figure 28 shows a family of curves of the equation:

$$y = kx^{-1/p_l}$$

where p_l is equal to 7 (see 8.4.2.5) and k varies from 0 to 1.

The width of the base up to the vertical line of abscissa 1 represents the dimensionless width $\frac{a}{a}$ of the segment being considered. On this should be plotted the vertical dimensionless heights $\frac{l_1}{a}$, $\frac{l_2}{a}$ to $\frac{l_5}{a}$ of the segment at the following abscissae:

0,054 0,242 0,509 0,774 0,954

At the end of the segment, abscissa = 1 as in Figure 28, mark off a vertical height or ordinate $\frac{I}{a}$ corresponding to the sum of $\frac{l_1}{a}$ to $\frac{l_5}{a}$ with the following weightings:

$$\frac{I}{a} = 0{,}083\,\frac{l_1}{a} + 0{,}196\,\frac{l_2}{a} + 0{,}255\,\frac{l_3}{a} + 0{,}226\,\frac{l_4}{a} + 0{,}115\,\frac{l_5}{a}$$

From the top of I (abscissa 1, ordinate $\frac{I}{a}$) trace a line parallel to the family of curves to the point where it intersects the curve representing the upper wall of the segment (ordinates $\frac{l_1}{a}$ to $\frac{l_5}{a}$).

The abscissa b of this point is the correct dimensionless abscissa of the traverse line for this segment. The distance of the traverse line from the segment wall is equal to $b \times a$.

The above instructions apply for the general case where the value of the coefficient p_l in Table 6 is taken as 7. A fuller explanation of the procedure including other values for p_l is given in annex A. A more general treatment of the mathematical process used in determining the traverse line in the marginal zone is given in annex B.

9 Determination of power

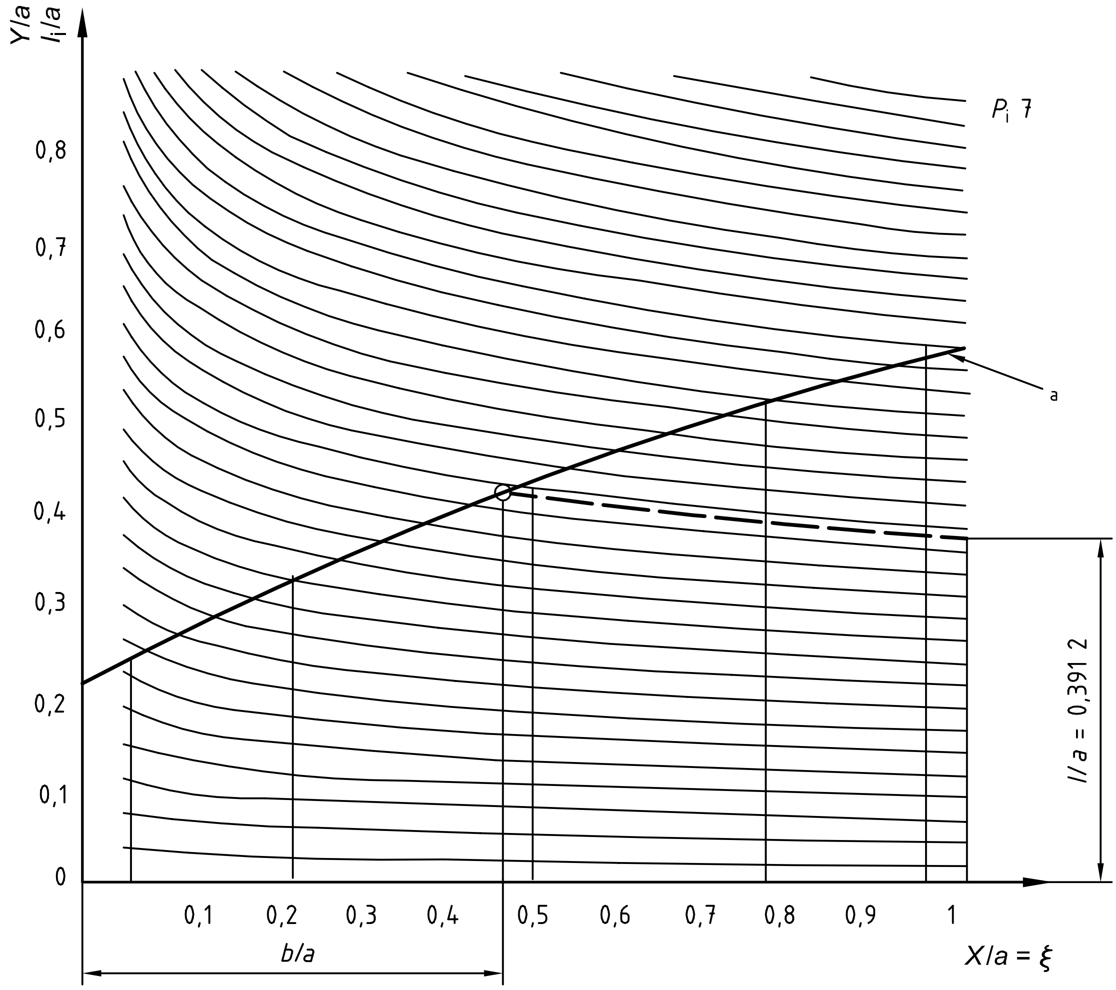
9.1 Definition of performance characteristics relating to the power of a fan

9.1.1 The principal performance characteristics used are defined as follows:

— the fan air power P_u is the product of the mass flowrate q_m and the fan work per unit mass y;

— the impeller power P_r is the power outlet to the fan impeller.

9.1.2 It is seldom possible to determine the impeller power directly, due to difficulty in evaluating the losses in the bearings supporting the fan shaft. However, it is useful to know this power in order to establish the basis characteristic of the fan.



[a] Dimensionless duct height

Figure 28

For this reason, one of the following values defined in a), b) and c) should preferably be measured.

a) The fan shaft power P_a is the mechanical power output to the fan shaft by the drive. It includes losses in bearings, in sealing glands, in bearing cooling devices, etc.

b) Motor shaft power P_o is the power available to the shaft of the drive. It is only equal to power P_a when the fan is directly driven.

In other cases:

$$P_a = \eta t_r P_o$$

where ηt_r is the drive efficiency.

Reasonably precise determination of the drive efficiency is difficult for small power ratings.

c) Motor input power P_e is the power output to the drive input. It is also the total power absorbed by the unit. In the case of drive by electric motor, it is the electric power input to the motor terminals. In other cases, the input power shall be determined from the consumption of fuel, steam, compressed air, etc. in the manner agreed between the parties concerned.

9.1.3 From these powers, the following efficiencies can be defined: the fan efficiency η_r, the fan shaft efficiency η_a, the fan motor efficiency η_m and the overall efficiency η_e, which are obtained by dividing the fan air power P_u by the impeller power, fan shaft power, motor output power and the motor input power, respectively.

Only the performance efficiency reflects the utilization capacity of the power output to the impeller.

The overall efficiency (or unit efficiency), used for motorized fans involves losses in the transmission system and in the drive, in addition to mechanical fan losses.

9.1.4 Agreements between the supplier and purchaser shall always clearly state which power and which efficiency the test should allow to be checked.

9.2 Losses during transmission of power from the motor to the impeller

If the results of tests on site are to be compared with fan performances resulting from bench testing, the power input to the fan should be defined in the same way in both cases.

When it is necessary to determine the fan shaft power, unless the impeller is directly mounted on the motor shaft, an appropriate allowance for losses in the transmission system shall be deducted from the motor shaft power. In order to determine these losses, the method to be used shall be agreed with the supplier of the fan.

If it is necessary to determine the impeller power, bearing and other transmission losses shall be deducted from the motor shaft power. The value of this term shall be determined by agreement between the parties.

9.3 Methods for determination of power

9.3.1 General

In order to obtain the power supplied to the fan, with the needed accuracy level, the mean of a sufficient number of measuring results shall be taken.

The drive shaft power may be determined by applying several methods. It may either be measured directly by means of a torquemeter or, in the case of a drive by electric motor, deduced from the electrical power input to the terminals of the motor. In this latter case the power output of the motor is deduced from its electrical power input by the summation of losses method. For this purpose, measurements of voltage, current, speed and, in the case of a.c. motors, electrical power input and slip of induction motors shall be made for each test point and no-load losses of the motor, when uncoupled from the fan, shall be measured. Alternatively, data concerning the performance of motors identical to that used or performance of a motor which has been calibrated beforehand may be used.

For electrical instruments, the electrical input to the motor during the fan test shall be measured by one of the following methods:

a) for a.c. motors, by the two wattmeter method or by an integrating watt-hour meter and a timing device;

b) for d.c. motors, by measurement of the input voltage and current.

9.3.2 Determination of fan shaft power by means of a torquemeter

The torque which has developed in the fan shaft may be measured by a torquemeter installed between the fan and the transmission system or motor which drives it. The power is calculated by multiplying the torque thus obtained by the rotational speed which has been carefully measured.

The appliance shall have an accuracy greater than 1,5 % of the torque value to be measured so that the power may be determined with an accuracy of 2 % if the rotational speed is measured with an uncertainty of 0,5 %.

Subject to agreement between the parties, the data relating to the static calibration of the appliance may be used, which shall be considered independent of the speed. In this case, the tests should be carried out in the same conditions as for the calibration.

In some cases, the appliance should be calibrated in the conditions encountered during the *in situ* tests, by an authority recognized by the parties, immediately before and after the tests and without any modification having been made to the state of the appliance. The torque shall be measured while the load increases and care shall be taken that, during the readings, the load at no time decreases. When the load decreases, similar precautions shall be taken. If the difference in torque on loading and unloading is greater than 1,5 %, the torquemeter shall be considered unsatisfactory. In the opposite case, the mean of the measurements on loading and unloading shall be retained. The mean of the two calibration curves obtained before and after the test shall be used as the calibration curve for the calculations; provided that the two values measured do not differ by more than 1,5 %.

9.3.3 Determination of the mechanical power output by the electric motor by means of the separate losses method

9.3.3.1 General

The shaft power of the electric drive motor may be calculated by multiplying the electrical input power P_e measured at the terminals by the motor efficiency estimated by the separate losses method.

The estimation of losses is carried out differently depending on whether a three-phase alternating current motor or direct current motor is being concerned.

The losses of an electric motor may be measured by a calorimetric method as described in IEC 60034-2A or by one of the methods described in IEC 60034-2.

9.3.3.2 Case of fan driven by a three-phase induction motor

The following losses shall be taken into consideration.

a) Constant losses:

- losses in active iron, and additional no-load losses in other metal parts;
- losses due to friction in bearings and brushes when they are not measured by running;
- total windage loss in the machine.

b) Load losses:

- losses due to electrical resistance in primary windings;
- losses due to electrical resistance in secondary windings;
- electrical losses in brushes (if any).

c) Additional load losses:

- losses introduced by load in active iron and other metal parts other than the conductors;
- eddy current losses.

The efficiency may be calculated from the total losses, which are assumed to be the summation of the losses determined in the following manner.

a) Constant losses

The sum of the constant losses is determined by running the machine as a motor on no-load. The machine is fed at its rated voltage and frequency. The power absorbed, decreased by the losses due to electrical resistance in primary windings, gives the total of the constant losses. The losses due to electrical resistance in the secondary winding may be neglected.

b) Load losses

Losses due to electrical resistance in the primary windings are calculated from the resistance of the primary windings measured using direct current and corrected to the reference temperature, and from the current corresponding to the load at which the losses are being calculated.

The losses due to electrical resistance in the secondary windings are taken to be equal to the product of the slip and the total power transmitted to the secondary winding.

The additional load losses are assumed to vary as the square of the primary current. The total value at full load is equal to 0,5 % of the rated input of the motor.

9.3.3.3 Case of a fan driven by a direct current motor or by a single-phase induction motor

For the determination of the motor efficiency, IEC 60034-2 should be referred to.

9.3.4 Reference to the performances of motors identical to the one used

When it is impossible to use the separation losses method, it is permitted to use the data given by the motor manufacturer concerning the performances which can be expected from motors identical to the one used for the *in situ* test, providing that an agreement between the parties has been concluded on this subject. The data given by the manufacturer may be considered as sufficient to determine the power output by the motor from the results of measuring the electrical input values.

The power developed by the motor during the tests should be compatible with that developed when the performances were established. The voltage should be stable and its mean should not deviate by more than 2 % from the voltage used when the performances were established.

The useful power supplied by the motor corresponds to the motor input power, and the former can be deduced directly from tables giving one as a function of the other, or indirectly from tables giving either the efficiency or the efficiency and the power factor.

In the second case, the power output by the motor is determined using the equation:

$$P_{\mathrm{o}} = P_{\mathrm{e}}\, \eta_{\mathrm{mot}}$$

where P_{e} is measured by means of a watt-hour meter for single-phase motors or by applying the two watt-hour meter method for three-phase motors.

In the third case, we have:

$$P_{\mathrm{o}} = \cos\varphi\, UI\, \eta_{\mathrm{mot}} \qquad \text{for single-phase motors,}$$

$$P_{\mathrm{o}} = \cos\varphi\, \sqrt{3}\, UI\, \eta_{\mathrm{mot}} \qquad \text{for three-phase motors,}$$

where U and I denote the values of voltages and line currents measured during *in situ* tests. For three-phase motors, they are the means of the values measured on each phase.

9.3.5 Use of a calibrated motor

The use of the calibrated motor allows the power developed by the motor to be deduced from the power output to its terminals by using the curves giving its efficiency or useful mechanical power variation as a function of the electrical input power. These curves which have been drawn up beforehand at the test bench shall be the subject of an agreement between the supplier and the purchaser. These data shall be used in the same way as in the previous case (9.3.4).

The data should be established for voltages ranging from a value 10 % lower than the rated voltage to a value 10 % greater than the rated voltage.

For these measurements the motor shall be allowed to run under load for at least 90 min to ensure that it has reached a temperature as close as possible to that reached during operation.

In order to use the calibration data, it is advisable to ensure that the power output to the motor is compatible with that used during calibration, that the phase voltage is stable, and that its mean does not differ from the calibration voltage by more than 2 %.

In case of any doubt about the stability and values of the frequency of the power supply, it is recommended that a measurement of the frequency should be made. For this purpose reference should be made to the appropriate IEC recommendation.

9.4 Measuring instruments

The powers, voltages and currents shall be measured in all cases with instruments either of class 0,5 (according to IEC 60051-8), for which a correction shall be made in accordance with the calibration curve, or of class 0,2, for which no correction is required. The choice of measuring instrument shall be such that the value read is greater than half the full scale reading in both case. The current and voltage transformers of the measuring apparatus shall be chosen to operate as close as possible to their rated load in order to minimize errors. These appliances are always connected as close as possible to the motor terminals to prevent voltage drops in the cables affecting the measurements.

9.5 Precautions to be taken during *in situ* tests

Care shall be taken that the power measured corresponds to the values of mass flowrate and the fan work per unit mass being measured.

With this aim, it shall be ensured that during the measurement of these two values, there is no sudden or progressive power variation by carrying out a sufficient number of measurements for the input power.

10 Uncertainty associated with the determination of fan performance

10.1 General

These test results will provide an actual value for the flow resistance of the airway system which can be compared with the value specified and will also provide an experimental fan performance point to compare with the fan characteristic determined by test with standardized airways. Discrepancies in either case can be due to any of the following:

— leakage, recirculation or other faults in the system;

— inaccurate estimation of flow resistance of the airway system;

— erroneous application of the standardized test data;

— excessive loss in a system component located too close to the fan outlet or elsewhere;

— disturbance of the fan performance caused by a bend or other system component located too close to the fan inlet;

— errors inherent in site measurement.

In many cases site conditions can be such that the accuracy of performance determination will be substantially inferior to the accuracy expected for fan tests with standardized airways, in which case either a full-scale or model test with a fan operating in a standardized airway configuration may be required in addition to the site test.

A site test may form the basis of a fan acceptance test if so agreed between the manufacturer and the purchaser. If any site test is to form any part of the guarantee between a manufacturer and a purchaser, the manufacturer should be given the opportunity of inspecting the airway installation or layout drawings before the fan is installed on site and to agree the best locations for measurements. Before the installation of the fan on site, the manufacturer should be given the opportunity of indicating if any modifications are likely to be necessary to give a test in accordance with the requirements of this standard.

Where it is impossible to comply strictly with the recommendations of this International Standard, some modifications may be agreed between manufacturer and purchaser, but in such cases it should be understood that the accuracy of the results is likely to be affected.

10.2 Performance errors

Any test for fan performance is subject to error and the range within which these testing errors may be expected to lie is defined, numerically, as the uncertainty of measurement. In addition they differ from that of another nominally identical fan owing to inevitable variations in manufacture. The expected range of this manufacturing variation shall be added to the uncertainty of measurement to determine the minimum tolerance required for a performance specification. Such tolerances are not specified in this International Standard but some recommendations concerning their use are given in 9.3 and 9.4.

10.3 Uncertainty of measurement

In this International Standard uncertainty of measurement is expressed as a percentage of a quantity which is directly measured (e.g. pressure) or one which is determined from other measurements (e.g. volume flow). It is quoted to 95 % confidence limits which implies that, out of a large number of measurements having a normal statistical distribution, 95 % may be expected to be within the limits specified with 2,5 % above the top limit and 2,5 % below the bottom limit.

In clause 17 of ISO 5801:1997, the limits of accuracy required for each individual measurement are specified and may also be taken as the maximum limits of uncertainty. The uncertainty is expressed as a percentage of the quantity relevant to the subsequent calculations; for example the temperature t is to be measured to $\pm$ 1,0 °C but the corresponding uncertainty is taken as $\pm$ 0,5 % of the absolute temperature ($273 + t$) for tests at room temperature.

This example also illustrates the "rounding up" approximation which is appropriate to uncertainty estimation. Taking the range of atmospheric temperature as 0 °C to 40 °C, the corresponding exact calculations of uncertainty would be $\pm$ 0,36 % and $\pm$ 0,32 % respectively. This apparent precision would be quite unjustified, however, and a realistic assessment is $\pm$ 0,5 %.

10.4 Specified uncertainties

The maximum uncertainties of measurement given in Table 7 apply to the quantities determined in accordance with clause 17 of ISO 5801:1997, provided the temperature and pressure at the fan inlet or outlet are within the normal atmospheric range.

10.5 Analysis of uncertainty

10.5.1 General

Basically this analysis deals with the overall uncertainty which will inevitably arise when a series of measured parameters are combined and referred to specific conditions.

The quantities which are determined from the proposed series of measurements are volume flowrate, fan pressure rise and fan absorbed power. The probable uncertainty in the determined quantities depends strongly upon the errors and uncertainties allocated to the individual measurements.

Table 7 — Maximum uncertainties

Quantity	Maximum uncertainties
Atmospheric pressure that is measured by barometer locally calibrated but uncorrected for temperature	± 0,3 % of p_a
Temperature that is measured to ± 1,0 °C and expressed in terms of absolute temperature	± 0,5 % of (273 + t)
Humidity being the uncertainty in air density due to an uncertainty of ± 2 °C in determining wet-bulb temperature t_w at 30 °C dry bulb temperature	± 0,2 % of ρ_a
Static pressure (greater than 150 Pa) combining 1 % manometer and 1 % reading fluctuation uncertainties	± 1,4 % of p_s or Δp
Area of duct or fan outlet	± 0,5 % of A
Area of orifice or nozzle throat	± 0,2 % of $\pi d^2/4$
Rotational speed of impeller	± 0,5 % of n
Power input to impeller, excluding determinations with significant proportion of transmission losses	± 2,0 % of P_r
Density of atmospheric air, including humidity correction in the determination	± 0,4 % of ρ_a
Density of air in an airway with humidity correction (any fan with pressure ratio $r \leqslant 1{,}3$)	± 0,6 % of ρ_a
Density of air in an airway without humidity correction provided in the fan total pressure $p_F \leqslant 10\ 000$ Pa and moisture content $\leqslant 0{,}015$ kg/kg	± 0,6 % of ρ_a

10.5.2 Inlet volume flow (or mass flow)

Where the flow conditions meet the requirements specified in 8.2.1 and when flowrate is determined in accordance with 8.3, the maximum uncertainties are as follows:

± 2,0 % of q_V (or ± 2,0 % of q_m)

When swirl is present or the flow lines are not approximately parallel to each other, the uncertainty will increase.

When the flowrate is determined in accordance with 8.4, the uncertainties of measurement are as follows:

— ± 3,0 % of q_V (or q_m) for measurements taken with a Pitot-static tube in a regularly shaped airway;

— ± 3,3 % of q_V (or q_m) for measurements taken with a Pitot-static tube in an irregularly shaped airway;

— ± 3,5 % of q_V (or q_m) for measurements taken with an anemometer in a regularly shaped airway;

— ± 4,0 % of q_V (or q_m) for measurements taken with an anemometer in an irregularly shaped airway.

10.5.3 Fan dynamic pressure

The maximum uncertainty is approximately ± 4 % of p_{d2}.

10.5.4 Fan pressure

This depends on the installation category (B, C or D) and the ratio of fan dynamic pressure to fan pressure. The dependence on these factors of the maximum uncertainty in p_F is given in Figure 29 for the case where the inlet and outlet duct areas are equal to the fan outlet area. Variations from +7 % to –5 % in these areas are not significant.

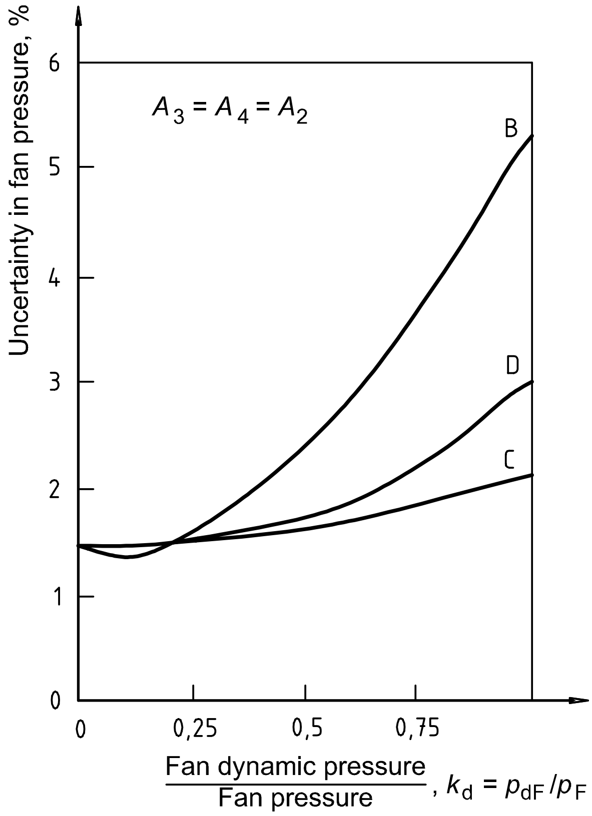
10.5.5 Characteristic uncertainty (see Figure 30)

The maximum uncertainty in volume flowrate when tested at a fixed system resistance depends on the slope of the fan characteristics (see Figure 31). The hatched areas cover proportional slopes from 0 % to –2 % change in p_F for +1 % change in q_V and will usually include the best efficiency point on the characteristic.

NOTE S is the proportional slope of the characteristic, i.e. $S = \Delta p_F / \Delta q_V$, the ratio of the percentage change in fan pressure to the percentage change in volume flowrate. S is negative when the pressure falls as the flow rises.

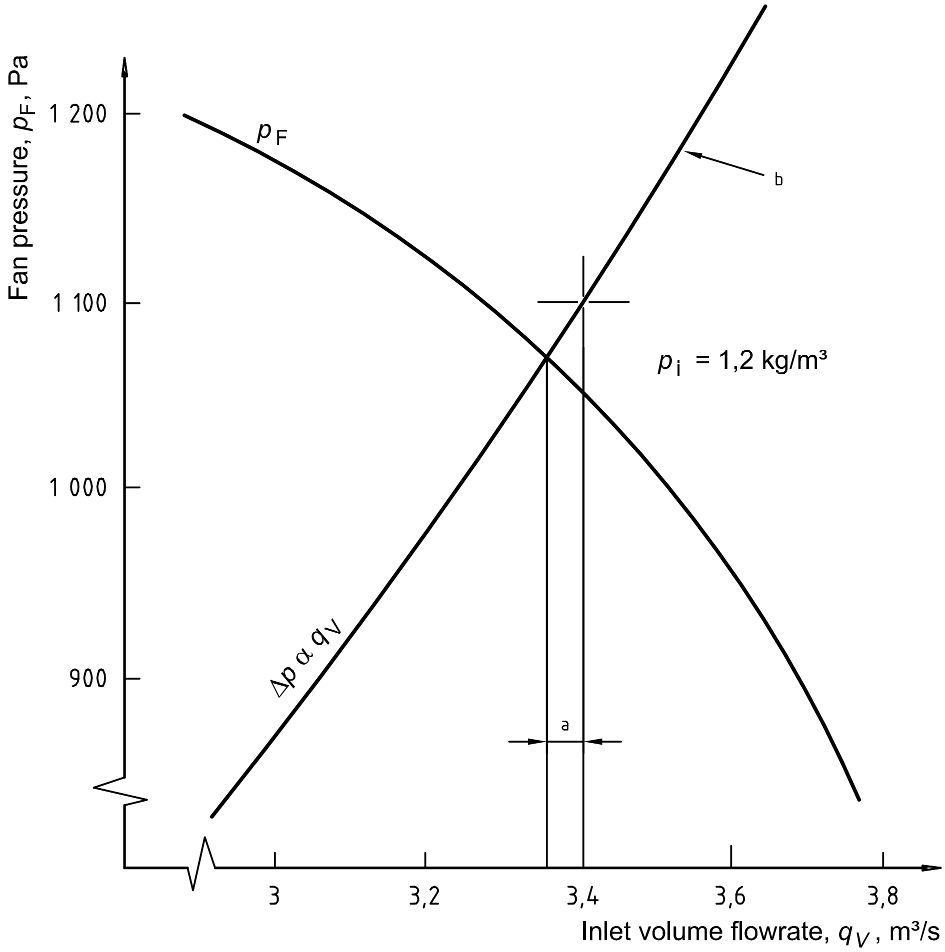
10.5.6 Fan efficiency

This is shown on a separate scale in Figure 31.



NOTE B, C , D are the installation type.

Figure 29 — Uncertainty in fan pressure



[a] Characteristic error

[b] Specified duty 3,4 m³/s at 1 100 Pa

Figure 30 — Example of test for a specified duty

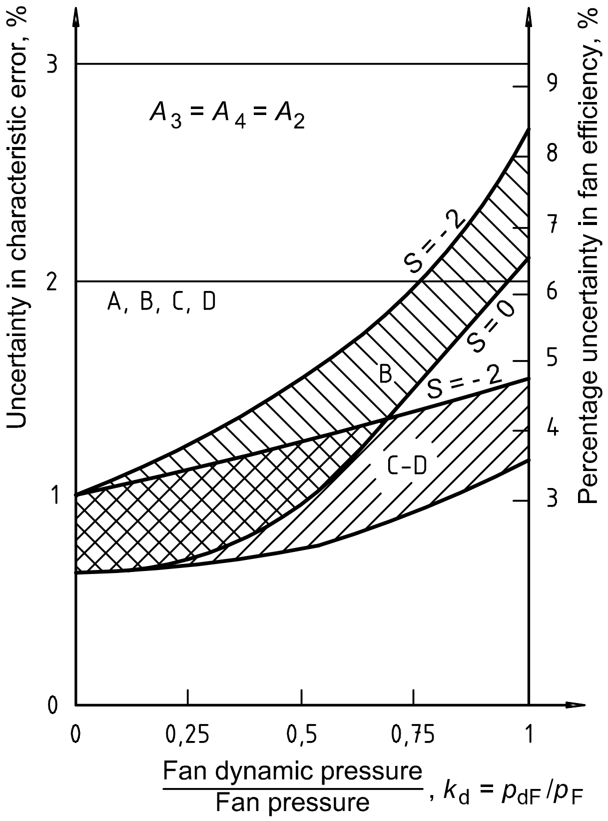


Figure 31 — Characteristic uncertainty and fan efficiency uncertainty

10.5.7 Rotational speed

The effect of conversion from the test speed to the specified speed is to increase the maximum uncertainties expressed in percent as follows:

— in inlet volume flowrate uncertainty: from $\pm e_q$ to $\pm (e_q^2 + 0,25)^{0,5}$

— in fan pressure uncertainty: from $\pm e_p$ to $\pm (e_p^2 + 1,0)^{0,5}$

— in characteristic uncertainty: from $\pm e_\Delta$ to $\pm (e_\Delta^2 + 0,25)^{0,5}$

There is no change of uncertainty in fan efficiency.

10.5.8 Uncertainty in power determination (Electrical methods)

In a three-phase system it has been assumed that the integrating watt-hour meter will make two measurements corresponding to W_1 and W_2.

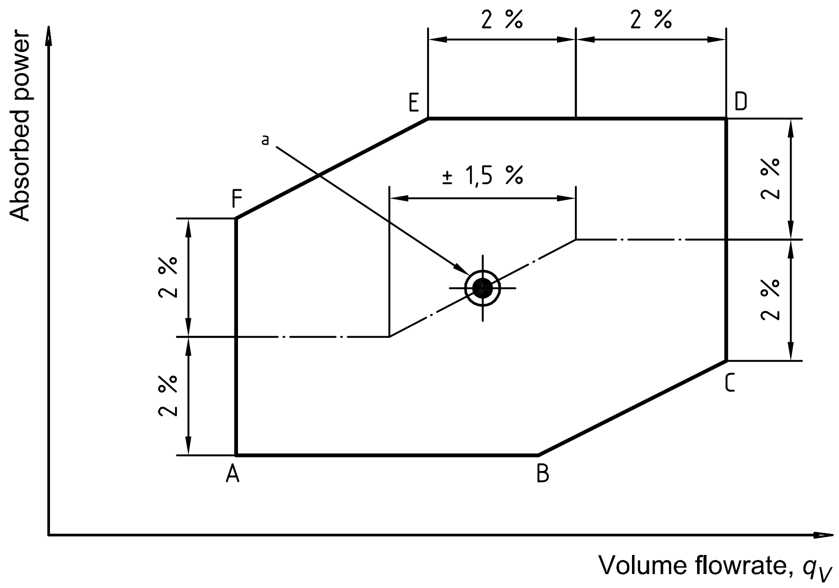
The watt-hour meter should be a precision instrument to class index 0.2, i.e. instrument uncertainty in accordance with IEC 60051-8 is ± 0,2 %. The assessment of input power to the meter will be ± 0,5 % with instruments operating at 50 % or greater of their full-scale value. If all the conditions of IEC 60034-1 are met, then the meter efficiency uncertainty will be ± 0,6 % and the overall uncertainty for absorbed power is ± 2,0 %.

10.5.9 Power absorbed *versus* volume flowrate characteristics

Construction tolerances should not lead to variation in absorbed power of more than ± 1,5 % with a corresponding volume flowrate variation.

Reference to Figure 32 shows that the basic power of the fan shall lie on line a-b, the exact position being dependent on the flowrate of the fan. At any point on this line a-b, the instrument and measurement tolerances are applicable.

At the extreme limits of a and b there will be a ± 2 % flowrate uncertainty together with a ± 2 % power uncertainty and this generates a tolerance envelope within which the measured power and flowrate will be expected to lie. Provided the measured power associated with the flowrate q_V lies within the envelope enclosed by ABCDEF, the fan will be deemed to have met the specified performance in terms of flowrate and absorbed power.



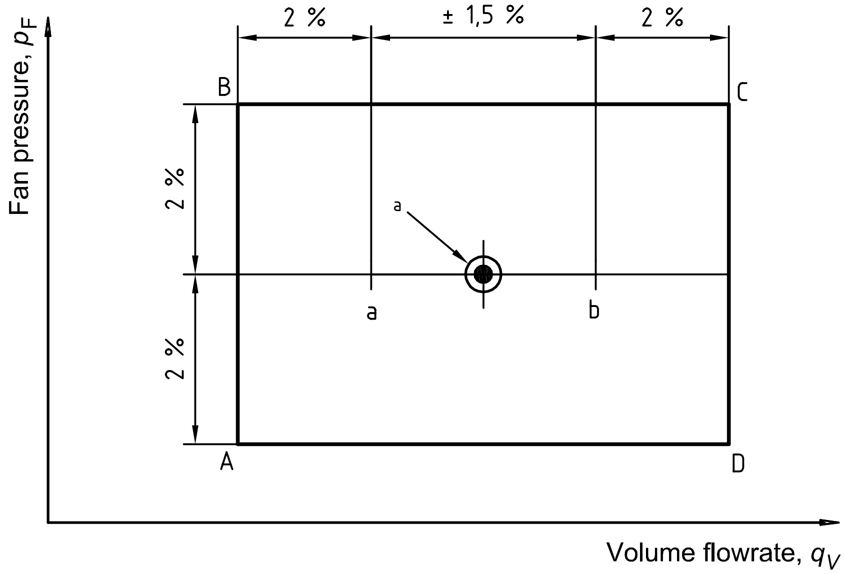
[a] Target duty 36,08 m³/s at 628 kW

Figure 32 — Typical example of power *versus* volume flowrate characteristic

10.5.10 Pressure rise *versus* volume characteristics

Manufacturers' tolerances should not lead to a variation in volumetric flowrate of more than ± 1,5 %. Reference to Figure 33 shows that the fan pressure rise shall lie on line a-b, the exact position being dependent on the flowrate of the fan. At the extreme limit of a and b there will be a ± 2 % flowrate uncertainty together with a ± 2 % pressure rise uncertainty which combine to give a tolerance envelope within which the measured pressure rise and flowrate will be expected.

Provided the measured pressure rise associated with flowrate q_V lies within the envelope enclosed by ABCD the fan will be deemed to have met the specified performance in terms of flowrate and fan pressure.



a Target duty 36,08 m³/s at 628 kW

Figure 33 — Typical example of fan pressure *versus* volume flowrate characteristic

Annex A
(normative)

Position of exploration lines for a marginal wall profile compatible with a general power law

A.1 General

If in a marginal zone of a section, the wall profile is such that the length of the segments intercepted by the walls on perpendiculars to the base varies according to the law (see Figure A.1):

$$l_x = l_0 + (l_\mathrm{a} - l_0)\left(\frac{x}{a}\right)^{1/p_\mathrm{l}'}$$

where

l_x is the length at the running abscissa x;

l_0 is the length at abscissa $x = 0$;

l_a is the length at the abscissa $x = a$ ($a = \frac{L}{m_\mathrm{l}}$ with m_l = number of cross-lines).

The relative position $z = \frac{b}{a}$ of the marginal exploration line may be calculated from the transcendental equation:

$$z^{1/p_\mathrm{l}}\left[l_0 + (l_\mathrm{a} - l_0)\, z^{1/p_\mathrm{l}'}\right] = l_0\,\frac{p_\mathrm{l}}{p_\mathrm{l}+1} + \left(l_\mathrm{a} - l_0\right)\frac{p_\mathrm{l}''}{p_\mathrm{l}''+1}$$

with

$$\frac{1}{p_\mathrm{l}''} = \frac{1}{p_\mathrm{l}} + \frac{1}{p_\mathrm{l}'}$$

$\frac{1}{p_\mathrm{l}}$ being the exponent of the characteristic law of the evolution of velocities at the wall:

$$v_x(y) = v_x(d)\left(\frac{y}{d}\right)^{1/p_\mathrm{l}} \qquad (0 < y < d \text{ for all values of } x)$$

where

$v_\mathrm{x}(d)$ is the velocity at distance y from the wall on the exploration line of abscissa x;

$v_\mathrm{x}(d)$ is the velocity at the measuring point nearest the wall ($y = d$) on the same exploration line;

$\dfrac{1}{p_l{}'}$ is the exponent of the general power law describing the marginal profile of the wall.

This transcendental equation is easily resolved by iteration.

The value of $p_l{}'$ may be determined graphically by plotting the tangent in $x = a$ at the curve giving the variation of l_x as a function of x. Thus (Figure A.1):

$$p_l{}' = \frac{AC}{AB}$$

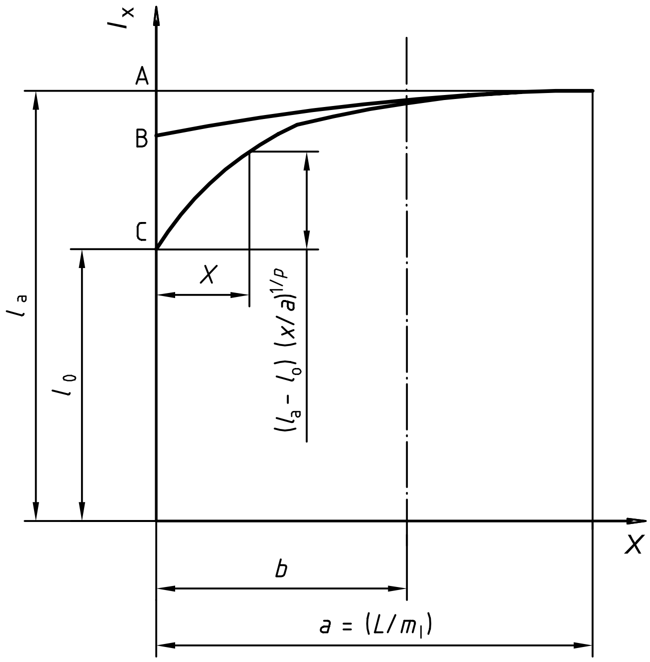


Figure A.1

A.2 Special cases

A.2.1 If $l_0 = 0$, the profile follows a power law (see Figure A.2).

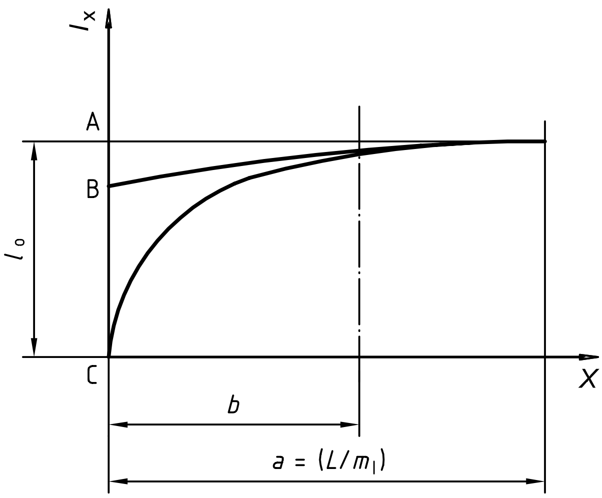


Figure A.2

The relative position $z = \frac{b}{a}$ of the marginal exploration line is given by the equation:

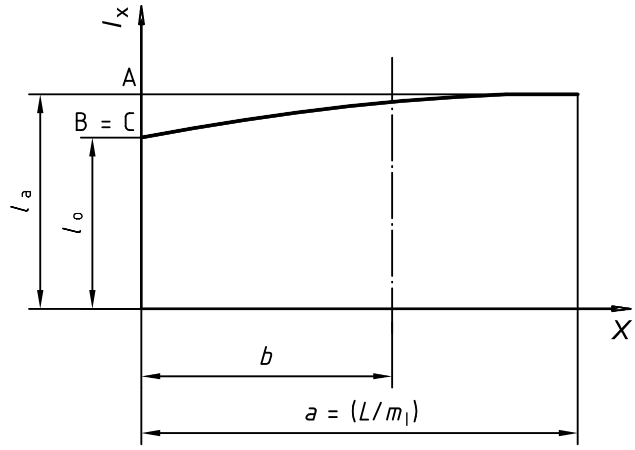
$$\frac{b}{a} = \left(\frac{p_l''}{1 + p_l''} \right)^{p_l''}$$

with $\frac{1}{p_l''} = \frac{1}{p_l} + \frac{1}{p_l'}$ and $p_l' = \frac{AC}{AB}$ (Figure A.2)

A.2.2 If $p_l' = 1$, the marginal section is in the shape of a rectangular trapezium (see Figure A.3).

The relative position $z = \frac{a}{b}$ of the marginal exploration line may be calculated by iteration from the transcendental equation:

$$z^{1/p_l} \left[l_0 + (l_a - l_0) z \right] = l_0 \frac{p_l}{p_l + 1} + (l_a - l_0) \frac{p_l}{2p_l + 1}$$



$\frac{l_a - l_0}{l_0}$	p_l		
	5	7	10
0	0,401 9	0,392 7	0,385 5
0,5	0,468 5	0,469 5	0,472 8
1	0,487 1	0,486 9	0,488 2
2	0,500 5	0,498 7	0,497 9

Figure A.3

A.2.3 If $l_0 = 0$; $p_l' = 1$, the marginal profile is at corner (see Figure A.4).

The relative position $z = \frac{b}{a}$ of the marginal exploration line is given by the equation:

$$\frac{b}{a} = \left(\frac{p_l}{2p_l + 1} \right)^{\frac{p_l}{p_l + 1}}$$

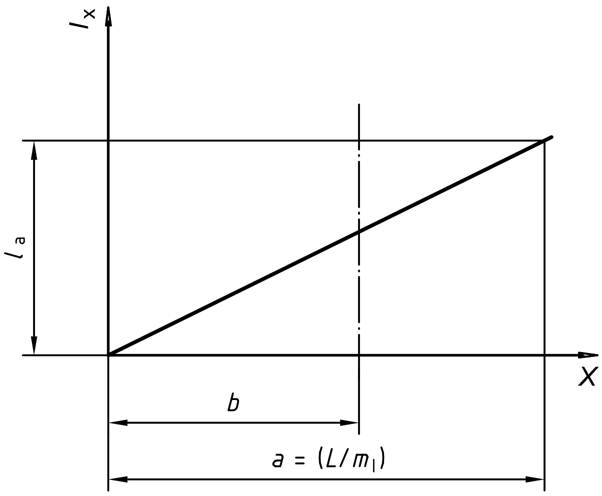


Figure A.4

Annex B
(normative)

Determination of the position of the marginal exploration lines in cases not covered by annex A

In cases not yet studied in which the wall profile in the marginal zone may be reasonably approximated by a polynominal law (degree 9 maximum), the dimensionless or relative position $\frac{b}{a}$ of the marginal cross-line may be determined by the method described below.

a) On a plan covered with the curves of equation

$y = kx^{-1/p_l}$ (see Figure B.1 for $p_l = 7$)

where

x is varying from 0 to 1,

y is varying from 0 to 0,9,

the variation law $l_{x/a}$ is plotted of the dimensionless length of the segments intercepted by the wall on the perpendiculars to the base up to abscissa $1 = \frac{L}{m_l\, a}$ (dimensionless duct height).

b) From this graph are measured five dimensionless heights $l_{i/a}$ of the section, perpendicular to the base, at abscissa points $x_i = \xi_i$, the values of ξ_i being given in Table B.1 for the values of p_l considered.

c) From the weighting coefficient f_i (also given in Table B.1), the integral $I_{/a}$ is calculated by the following equation:

$$I_{/a} = \sum_{i=1}^{5} \xi_i\, l_{i/a}$$

and the length $I_{/a}$ is plotted perpendicularly to the base (at right angles) to abscissa 1.

d) The intersection of the curve of the plan passing through the point $(1, I_{/a})$ with the wall profile $l_{x/a}(x)$ provides the dimensionless abscissa b of the marginal cross-line.

Table B.1 — Calculation of the integral I

$p_l = 5$	ξ_i	0,057	0,246	0,513	0,776	0,953
	f_i	0,072	0,182	0,245	0,221	0,114
$p_l = 7$	ξ_i	0,054	0,242	0,509	0,774	0,954
	f_i	0,083	0,196	0,255	0,226	0,115
$p_l = 10$	ξ_i	0,052	0,238	0,506	0,773	0,954
	f_i	0,092	0,206	0,263	0,230	0,116

EXAMPLE

Take the law of variation of $l_{x/a}$ plotted on Figure B.1 (it is assumed that close to the walls the velocity profile follows the power law of exponent 1/7: $p_l = 7$).

Thus on the dimensionless scale of the drawing, we have the values given in Table B.2.

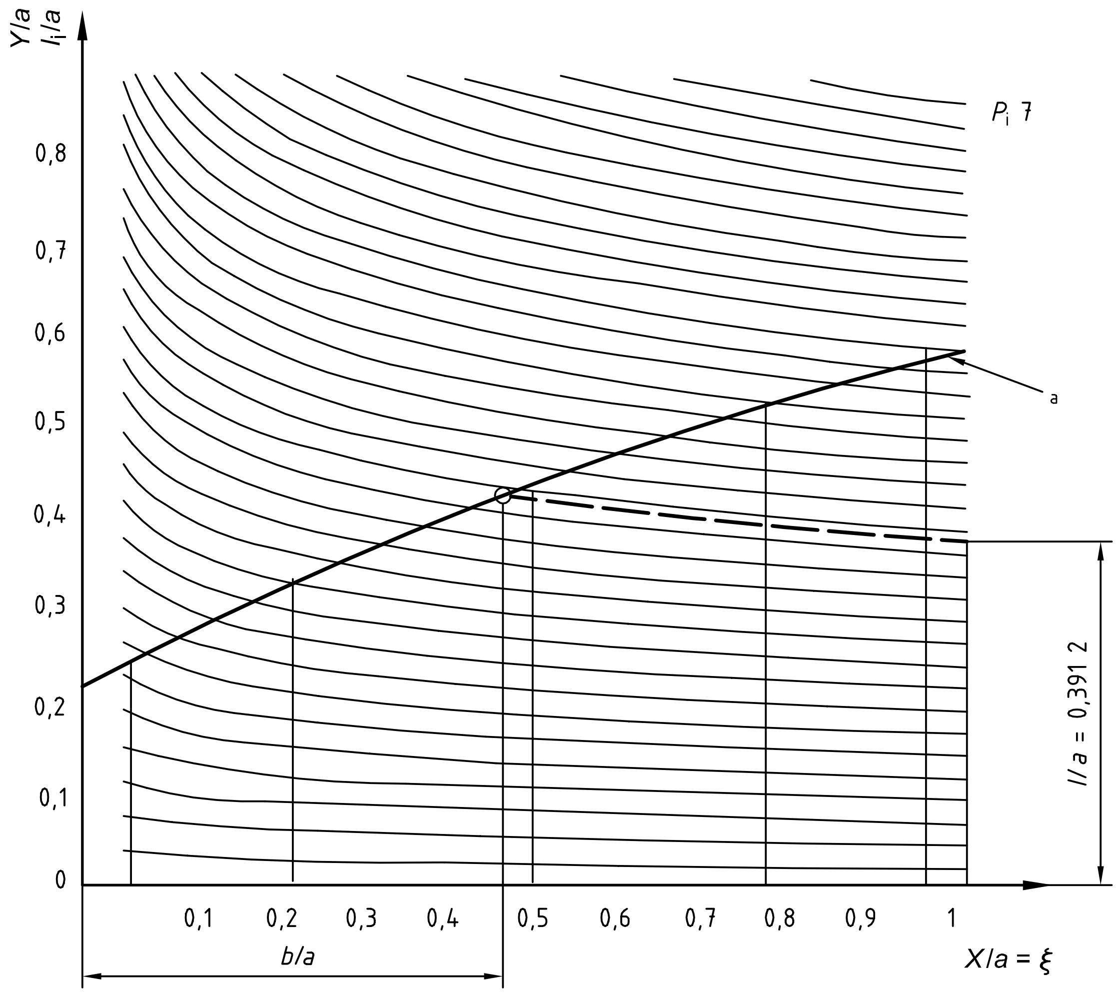
Table B.2 — Example of calculation

ξ_i	x_i mm	$x_{i/a}$	$l_{i/a}$	f_i	$f_i\ l_{i/a}$
0,054	10,8	0,054	0,247 5	0,083	0,020 54
0,242	48,4	0,242	0,340	0,196	0,066 64
0,509	101,8	0,509	0,447 5	0,255	0,114 11
0,774	154,8	0,774	0,540	0,226	0,122 04
0,954	190,8	0,954	0,590	0,115	0,067 85
				$I_{/a} = \Sigma f_i\ l_{i/a}$	0,391 18 ≅ 0,391 2

From these, the dimensionless position of the marginal exploration line is obtained (Figure B.1):

a being equal to 200 mm;

b = 200 × 0,475 = 95 mm.



Key

1 Dimensionless duct height

$y = ks^{-1/p_l}$

Figure B.1

Annex C
(normative)

Minimum straight lengths required upstream and downstream of the differential pressure devices (DP device) used for flow measurement

C.1 General

C.1.1 The presence of disturbances or obstacles upstream or downstream near a differential pressure device intended for flow measurement alters the flow at this point, which affects the use of the flow coefficients given in ISO 5167-1.

C.1.2 In order to determine the effect of a disturbance on a DP device intended for flow measurement, the following have to be known:

- the type of DP device;
- its area ratio;
- the type and the layout of its pressure tappings;
- the orientation of these tappings in relation to the nearest upstream disturbance to the DP device;
- the type and dimensions of the disturbance;
- the distance between the upstream disturbance and the nearest DP device;
- the relative disposition of any other sources of disturbance when there is interaction between the various disturbances.

C.1.3 For standardized DP devices (orifice plates, nozzles, venturis) and different layouts of pipes upstream of them, this annex gives the values of the minimum straight lengths required upstream of the DP devices which would guarantee the validity of the measurements. It also indicates in each case the correction to be made to the flow coefficient given in the standards for the same ratio, Reynolds number, etc. and also gives an additional degree of uncertainty.

C.2 Proximity coefficient and associated degree of uncertainty

The effect of the disturbance on the result of the measurement carried out by means of the standardized DP device placed in non-standardized conditions shall be taken into account by multiplying the calculated flowrate by a correction factor F called "proximity coefficient", the value of which is a function of the source of the disturbance, the DP device, and the characteristics of the flow (see Tables C.1 and C.2).

The use of this proximity coefficient, however, involves increasing the degree of uncertainty of the flowrate measurement. For each proximity coefficient, and consequently in each particular case, an additional uncertainty f (see Tables C.1 and C.2) is involved, which is added to the uncertainty calculated according to ISO 5167-1 for the type of DP device.

The flowrate in the pipe can then be calculated by using the following equation:

$$q_{\mathrm{vr}} = F q_{V\mathrm{s}} \pm (\delta q_{V\mathrm{s}} / q_{V\mathrm{s}} + f) q_{V\mathrm{s}}$$

where

q_{Vr} is the volume flowrate in the pipe;

q_{Vs} is the volume flowrate estimated using the flow coefficient corresponding to the standardized conditions for use of the DP device involved;

δq_{Vs} is the absolute uncertainty in the flow, q_{Vs} being estimated in accordance with the conditions laid down in ISO 5167-1:1991, subclause 1.1;

F is the proximity coefficient given in Tables C.1 and C.2;

f is the additional uncertainty arising from the use of the DP device in non-standardized conditions.

The following comments should be noted.

a) The value of the proximity coefficients and of the corresponding uncertainties are given in Tables C.1 and C.2 for a non axisymmetric, steady, swirl-free flow, upstream of the source of disturbance preceding the DP device.

When there are two successive sources of disturbances upstream of the DP device, the straight length required upstream of the second source will be determined by the type of disturbance preceding it. The distance between the two sources of disturbances will be at least equal to the length given in ISO 5167-1. The case of a swirling flow is treated in C.6.

b) The differential pressure in a section will be determined by four pressure tappings arranged symmetrically at 45° in relation to the plane of symmetry of the nearest upstream disturbance. Tables C.1 and C.2 give for each case the type of pressure tapping to be adopted.

c) It is desirable to read the pressure individually at each tapping and to take the average of the values measured as the pressure value in that section.

If the solution is not practicable the four pressure tappings can be connected together in a "triple-T" as shown in Figure C.1.

d) Corner pressure tappings are less sensitive to fluctuations of pressure than flanges or D - D/2 tappings. They do, however, require being made with greater precision.

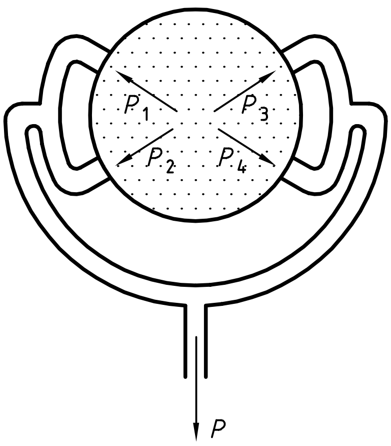


Figure C.1

Table C.1 — Minimum recommended upstress straight length for orifice plates installed downstream of a disturbance

Type of disturbance	Orifice plate area ratio r_A	Type of tappings	Distance between disturbance and orifice plate L	Proximity coefficient F	Uncertainty on F f
1. Single 90° bend					
1.1 Mitre	$r_A \leqslant 0{,}64$	any between the three proposed types	$L \geqslant 8\ D$	0,990	± 0,010
1.2.1 R/D = 1,0	$r_A < 0{,}5$ $0{,}5 \leqslant r_A \leqslant 0{,}64$		$L \geqslant 8\ D$ $L \geqslant 15\ D$	0,990 0,990	± 0,010 ± 0,010
1.2.2 R/D = 1,5	$r_A < 0{,}5$ $0{,}5 \leqslant r_A \leqslant 0{,}64$		$L \geqslant 8\ D$ $L \geqslant 12\ D$	0,990 0,990	± 0,010 ± 0,010
1.2.3 R/D = 4,75	$r_A < 0{,}5$ $0{,}5 \leqslant r_A \leqslant 0{,}64$		$8\ D \leqslant L \leqslant 20\ D$ $12\ D \leqslant L \leqslant 20\ D$	0,990 0,990	± 0,010 ± 0,010
2. Single bend with angle different from 90°			No recommendation possible		
3. Two identical 90° "S" bends					
3.1 Without spacer between bends					
3.1.1 Mitre	$r_A \leqslant 0{,}64$	D and $D/2$ flange or corner	$L \geqslant 30\ D$	1,000	± 0,010
3.1.2 R/D = 1,5	$r_A = 0{,}5$		$L \geqslant 8\ D$	0,990	± 0,010
3.2 With spacer between bends			No results available		
3.2.1 Mitre					
3.2.2 R/D = 1,5					
Spacer = 4D					
4. Two identical 90° "U" bends					
4.1 Without spacer between bends					
4.1.1 Mitre			No results available		
4.1.2 R/D = 1,5	$0{,}25 \leqslant r_A \leqslant 0{,}5$ $0{,}5 < r_A \leqslant 0{,}64$	any	$6\ D \leqslant L \leqslant 30\ D$ $L \geqslant 10\ D$	0,995 0,990	± 0,010 ± 0,010
4.2 With spacer between bends					
4.2.1 Mitre			No results available		
4.2.2 R/D = 1,5	$0{,}25 \leqslant r_A \leqslant 0{,}64$	any	$L \geqslant 8\ D$	0,990	± 0,010
Spacer $\geqslant$ 5 D					
5. Two identical 90° bends in perpendicular planes					
5.1 Without spacer between bends					
5.1.1 Mitre					
5.1.2 R/D = 1,5	$r_A \leqslant 0{,}64$	any	$L \geqslant 40\ D$	1,000	± 0,010
R/D = 4,75	$0{,}25 \leqslant r_A \leqslant 0{,}64$	any	$L \geqslant 15\ D$	1,000	± 0,010
5.2 With spacer between bends	$0{,}25 \leqslant r_A \leqslant 0{,}64$	any	$8\ D \leqslant L \leqslant 24\ D$	0,995	± 0,010
5.2.1 Mitre			No results available		
5.2.2 R/D = 1,5	$r_A \leqslant 0{,}5$	any	$L \geqslant 8\ D$	0,990	± 0,010
Spacer $\geqslant$ 5 D	$0{,}25 \leqslant r_A \leqslant 0{,}64$	any	$L \geqslant 12\ D$	0,990	± 0,010

Table C.1 *(continued)*

Type of disturbance	Orifice plate area ratio r_A	Type of tappings	Distance between disturbance and orifice plate L	Proximity coefficient F	Uncertainty on F f
6. Axisymmetric disturbances					
6.1 Central disc, a = 0,5	r_A = 0,5	corner, flange	$L \geqslant 8\ D$	0,990	± 0,010
6.2 Orifice plate ($r_A \geqslant$ 0,2)	0,1 $\leqslant r_A \leqslant$ 0,64	any	$L \geqslant 25\ D$	0,995	± 0,005
6.3 Plain inlet	0,1 $\leqslant r_A \leqslant$ 0,64	any	$L \geqslant 5\ D$	0,990	± 0,010
7. Valves					
7.1.1 ate valve at least 50 % open	$r_A \leqslant$ 0,5	any	$L \geqslant 12\ D$	0,993	± 0,013
7.1.2 ate at least 65 % open	$r_A \leqslant$ 0,64	any	$L \geqslant 8\ D$	0,993	± 0,013
7.1.3 utterfly valve at least 45° open	$r_A \leqslant$ 0,64	any	$L \geqslant 12\ D$	0,993	± 0,010

Table C.2 — Minimum recommended upstream straight length for nozzles and venturis installed downstream of a disturbance

Disturbance	Type of device	Area ratio r_A	Type of tappings	Distance between disturbance and device L	Proximity coefficient F	Uncertainty on F f
1. Single 90° bend						
1.1 Mitre bend					No results available	
1.2 R/D = 1,5	Nozzle and Venturi nozzle	r_A < 0,35 0,35 < r_A < 0,6	any any	$L \geqslant 5\ D$ $L \geqslant 12\ D$	0,990 0,990	± 0,010 ± 0,010
1.3 Other values R/D		See point 1.2				
2. Single bend with angle different from 90°						
3. Two identical 90° bends perpendicular planes						
3.1 Mitre bends with or without spacer					No results available	
3.2 Swept bends R/D = 1,5 without spacer	Nozzle and Venturi nozzle	r_A < 0,5 0,35 $\leqslant r_A \leqslant$ 0,6	any any	$L \geqslant 5\ D$ $L \geqslant 12\ D$	0,990 0,990	± 0,010 ± 0,010
3.3 Swept bends R/D = 1,5 without spacer		See point 3.2				
4. Other combinations of two identical 90° bends		See point 3.2				

C.3 Minimum straight lengths required upstream of the DP device

C.3.1 Orifice plate

All necessary factors relating to the use of orifice plates in non-standardized conditions can be found in Table C.1.

C.3.2 Nozzles and venturis

All the necessary factors relating to the use of nozzles and venturis in non-standardized conditions can be found in Table C.2.

C.4 Minimum straight lengths required downstream of the DP device

A minimum straight length of $4D$ is required between the standardized DP device intended for flow measurement and any source disturbance situated downstream of it.

C.5 Application of the recommendations

C.5.1 Example for choosing the position of the measuring device in a circuit

The diagrams of the circuits are presented in Figure C.2.

These three diagrams include, in the direction of flow, the following sources of disturbance in series:

— a 90° bend with a radius of curvature defined by the relation $R/D = 1{,}0$ (see Table C.1, disturbance 1.2.1);

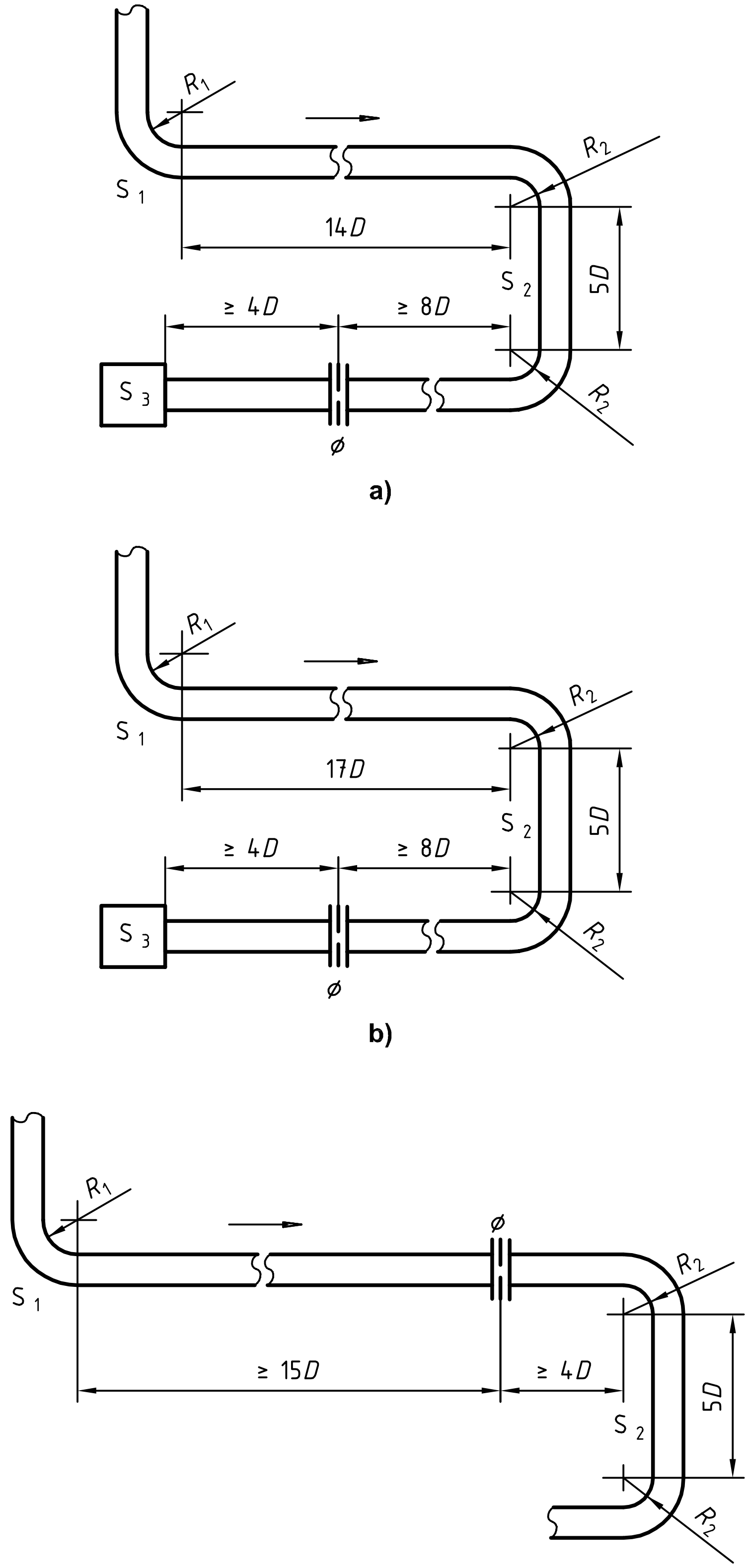
— two identical 90° bends forming a U shape, with a radius of curvature defined by the relation $R/D = 1{,}5$ and separated by a straight length of $5D$ (see Table C.1, disturbance 4.2.2).

The only difference between the three diagrams lies in the distance between the two sources of disturbance; this is $14D$ in the case of Figure C.2 a), $17D$ in Figure C.2 b), and $19D$ in Figure C.2 c).

Installing an orifice plate with an area ratio of $r_A = 0{,}64$ is not permitted between the two sources of disturbance in the first two diagrams. In fact, the minimum distance required between the first source of disturbance and the measuring orifice plate is at least $15D$ (see Table C.1). This condition cannot be met in the first diagram since there is only $14D$ between the two sources of disturbance. In contrast, the second diagram should enable this requirement to be met providing there is no other factor to be taken into account, but it is not possible to have a distance of $4D$ downstream of the orifice plate at the same time.

In these two diagrams, if it is possible to maintain the minimum length of $8D$ between this second disturbance and the measuring orifice plate (see Table C.1) as well as a minimum distance of $4D$ downstream of the measuring orifice plate, the position of the DP device downstream of the second disturbance should be adopted.

The third diagram shows the positioning of the measuring orifice plate between the two disturbances, in compliance with the specifications of this International Standard.



Key

S_1 is 90° bend; $R/D = 1$

S_2 is double bend of 90°; $R/D = 1/5$ in U-spacing $5D$

S_3 is any source of disturbance

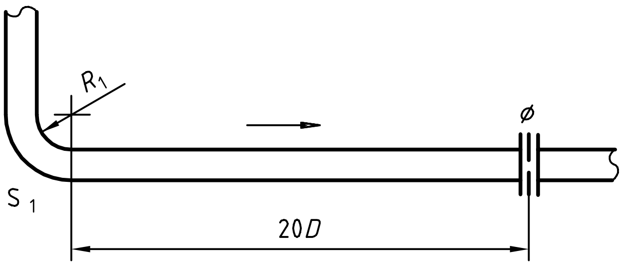
ϕ is orifice plate with $r_A = 0{,}64$

$\rightarrow$ is the direction of flow

Figure C.2

C.5.2 Example of use of the proximity coefficient

To measure the flow using an orifice plate with an area ratio $m = 0,64$ located $20D$ downstream from a source of disturbance of the 90° bend type, $R/D = 1$ (Figure C.3).



Key

S_1 is bend of 90°; $R/D = 1$

ϕ is orifice plate with $r_A = 0,64$

Figure C.3

If the disturbance is disregarded, applying ISO 5167-1:1991, subclause 1.1, would produce the volume flowrate value as:

$$q_{V\mathrm{s}} = 2,015 \pm 0,026$$

with a degree of uncertainty associated with the measurement corresponding to a confidence safety limit of 95 %.

Basing the calculation on Table C.1, in order to take the disturbance into account, a proximity coefficient of 0,99 with an additional uncertainty of ± 1 % will be taken.

The volume flowrate will then be calculated from the following equation:

$$q_{V\mathrm{r}} = 0,99 \times 2,015 \pm (0,026 + 0,01 \times 2,015)$$

$$q_{V\mathrm{r}} = 1,995 \pm 0,046$$

Under no circumstances can the flowrate value be disassociated from its degree of uncertainty.

C.6 Using anti-swirl devices

C.6.1 Where the disturbance is aggravated by swirl, the minimum straight length required between the source of disturbance and the measuring orifice plate can be reduced by placing an anti-swirl device of the etoile type $2D$ long and at least $3D$ downstream of the source of the disturbance (Figure C.4).

C.6.2 For the values $r_A \leqslant 0,64$ and any intensity of swirl a minimum straight length of $6D$ shall be maintained between the downstream end of the anti-swirl device and the measuring orifice plate.

C.6.3 A minimum straight length of $4D$ shall be maintained between the measurement orifice plate and the first source of disturbance downstream of it.

C.6.4 Subject to these conditions, a proximity coefficient $F = 0,99$ associated with an additional uncertainty of ± 1 % will be used.

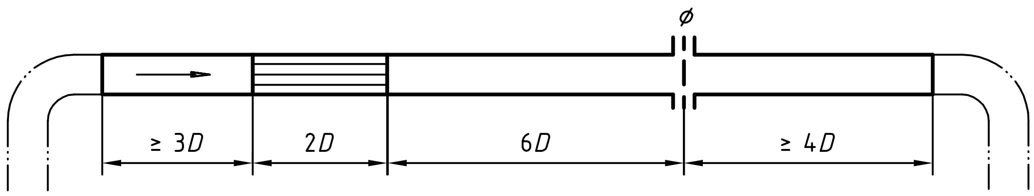


Figure C.4

Annex D
(normative)

Loss allowance for straight, smooth ducts and standardized airways

The pressure drop due to fully developed flow in a uniform, straight, smooth-walled duct of length L_D and hydraulic diameter D_h depends on the Reynolds number Re as shown in Figure D.1. This is based on the following equations:

$$\Delta p = \lambda \frac{L_D}{D_h} \frac{1}{2} \rho_x v_x^2$$

$$\lambda = 0{,}05 + 0{,}42\,(Re)^{-0{,}3}$$

where

$D_h = D$ for a circular cross section

$D_h = \dfrac{2\,b\,h}{b+h}$ for a rectangular cross section of dimensions b and h;

$D_h = 4\left[\dfrac{\text{area of cross section}}{\text{perimeter of cross section}}\right]$ for an airway of any shape;

L_D is the length of the duct;

Re is the Reynolds number for the flow in the pipe.

Spray-painted or zinc-coated rolled steel ducts over 100 mm in diameter and galvanized steel ducts over 500 mm in diameter with widely spaced, well-fitted joints may be treated as being smooth to determine a friction allowance.

The friction factor for other duct materials shall be agreed between the supplier and purchaser.

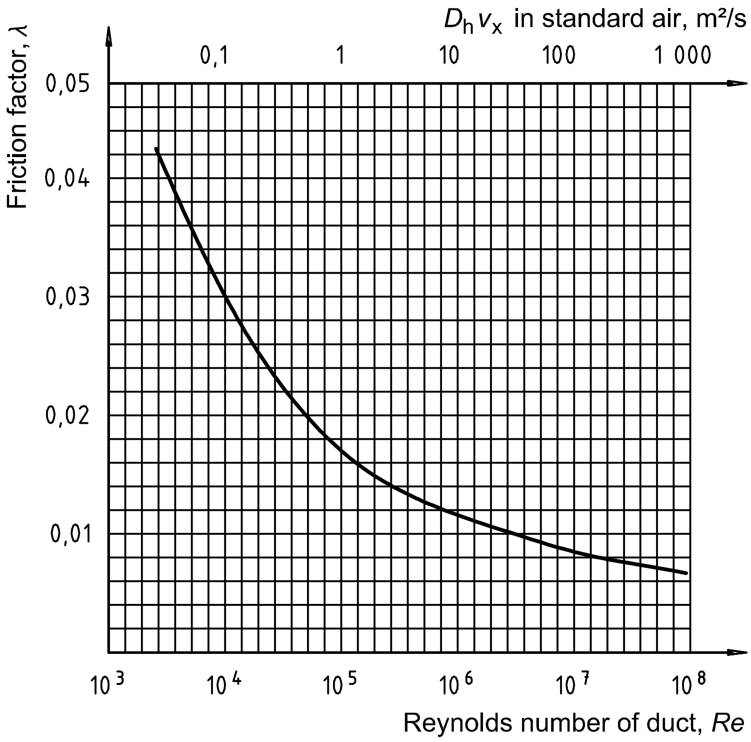


Figure D.1 — Friction factor versus Reynolds number

Annex E
(normative)

Rotating vane anemometer calibration

E.1 General

This annex refers to the calibration of rotating vane anomometers to be used for the measurement of air velocity in the testing of fans to this International Standard and is referred to in 5.2.2.

The following procedure and presentation of results is recommended.

a) The anemometer shall be inspected for damage and any faults found shall be rectified before proceeding with the calibration.

b) The anemometer shall be calibrated in conditions resembling as closely as possible the practical conditions of use with regard to the scale of measurement, the density of the fluid and the orientation of the axis relative to the vertical.

c) The recommended wind tunnel design for use in calibration are the "enclosed working section" type and the "open jet" type. The wind tunnel shall have certified and traceable documentation for the accuracy of velocity measurement.

d) The enclosed working section wind tunnel shall have a cylindrical or octagonal cross-sectional area of at least 25 times the face area of the anemometer. The anemometer shall be placed centrally in the airway.

If this is not possible, then a blockage allowance for the anemometer in the airway shall be made based on the theory for "bluff bodies":

$$\left[\frac{v_o}{v_i}\right]^2 = 1 - 3{,}15\ \frac{S}{C}$$

where

v_o is the unobstructed tunnel velocity;

v_i is the tunnel velocity with the instrument in place;

C is the cross-sectional area of the tunnel;

S is the face area of solid (non-rotating) part of the anemometer.

e) The "open jet wind tunnel" shall have a jet diameter d_g of at least 1,5 times the diameter d_a of the anemometer. The head shall be placed 1 jet diameter from the face of the nozzle (Figure E.1). Where the anemometer head is greater than 0,5 jet diameter, the jet shall be specially calibrated to compensate for the spread of the jet.

f) The anemometer shall be tested at least 10 velocities over the working range of the instrument. Measurements shall be spaced more closely at the lower speeds (similar to R10 series).

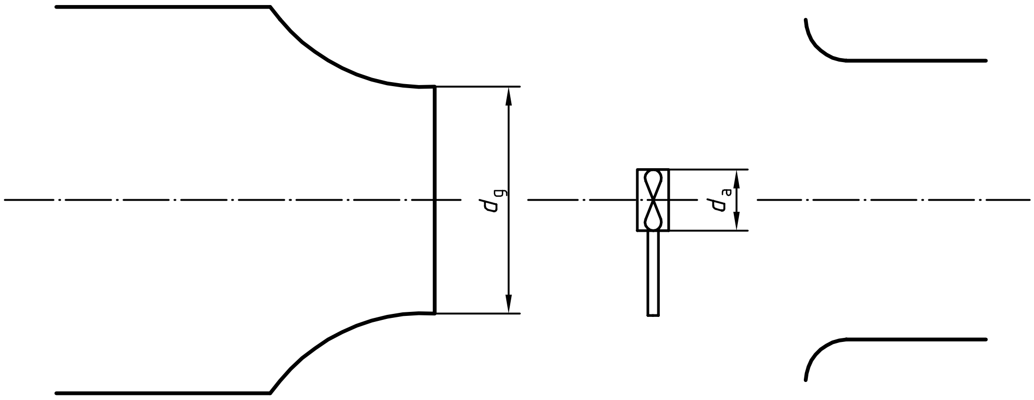


Figure E.1

E.2 Methods of accumulating calibration data

E.2.1 The test wind speed is chosen to coincide with cardinal points on the instrument display (analog). In this method the wind tunnel speed is adjusted until the preferred instrument reading is attained (usually a whole number) and the wind tunnel velocity is subsequently determined.

E.2.2 The test wind speed is chosen to give a series of true values. The wind tunnel is set to generate the required velocity and the instrument reading is noted.

E.2.3 Frequently the adjustments required to establish the above methods are time consuming and require fine control of the wind tunnel speed to obtain accurate results. Where calculation is needed to obtain values of wind speed from the data observed from the wind tunnel and/or instrument, it may be considered preferable to allow the wind tunnel to settle to a series of settings approximating to the required wind speed and the data for the instrument and wind tunnel read at each point.

E.3 Presentation of results

It is possible to present the information in several ways to highlight various aspects of the anemometer performance. There are five systems normally employed.

a) The indicated reading is compared with the true speed in either graphical or tabular form.

b) The indicated reading is compared with the correction required to obtain the true speed.

c) The true speed is compared with the instrument error (value by which the instrument reading departs from the true value).

d) The information obtained by methods b) and c) may also be expressed as a percentage correction or error.

e) In situations where the anemometer is likely to be used in air densities departing from the calibrating conditions by more than 10 %, a calibration curve which is sufficiently independent of density is obtained by plotting:

$$V_{\mathrm{ind}} \sqrt{\rho_{\mathrm{m}}} \text{ against } V_{\mathrm{true}} \sqrt{\rho_{\mathrm{m}}} \text{ or } (V_{\mathrm{true}} - V_{\mathrm{ind}}) \sqrt{\rho_{\mathrm{m}}}$$

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