

Industrial fans –

Performance testing in situ

ICS 23.120

National foreword

This British Standard is the UK implementation of EN ISO 5802:2008+A1:2015. It is identical to ISO 5802:2001 incorporating amendment 1:2015. It supersedes BS EN ISO 5802:2008 (dual numbered BS 848-3:2001) which is withdrawn.

The UK participation in its preparation was entrusted to Technical Committee RHE/2, Ventilation for buildings, heating and hot water services

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

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English Version

Industrial fans - Performance testing in situ (ISO 5802:2001)

Ventilateurs industriels - Essai de performance in situ (ISO 5802:2001)

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Foreword

The text of ISO 5802:2001 has been prepared by Technical Committee ISO/TC 117 “Industrial fans” of the International Organization for Standardization (ISO) and has been taken over as EN ISO 5802:2008 by Technical Committee CEN/TC 156 “Ventilation for buildings” the secretariat of which is held by BSI.

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of an identical text or by endorsement, at the latest by April 2009, and conflicting national standards shall be withdrawn at the latest by April 2009.

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Endorsement notice

The text of ISO 5802:2001 has been approved by CEN as a EN ISO 5802:2008 without any modification.

Foreword to amendment A1

This document (EN ISO 5802:2008/A1:2015) has been prepared by Technical Committee ISO/TC 117 “Fans” in collaboration with Technical Committee CEN/TC 156 “Ventilation for buildings” the secretariat of which is held by BSI.

This Amendment to the European Standard EN ISO 5802:2008 shall be given the status of a national standard, either by publication of an identical text or by endorsement, at the latest by December 2015, and conflicting national standards shall be withdrawn at the latest by December 2015.

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

NOTE Atmospheric air at a temperature of 16°C , a pressure of $100\,000 \text{ Pa}$ and a relative humidity of 65% , has a density of $1,2 \text{ kg}\cdot\text{m}^{-3}$, 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

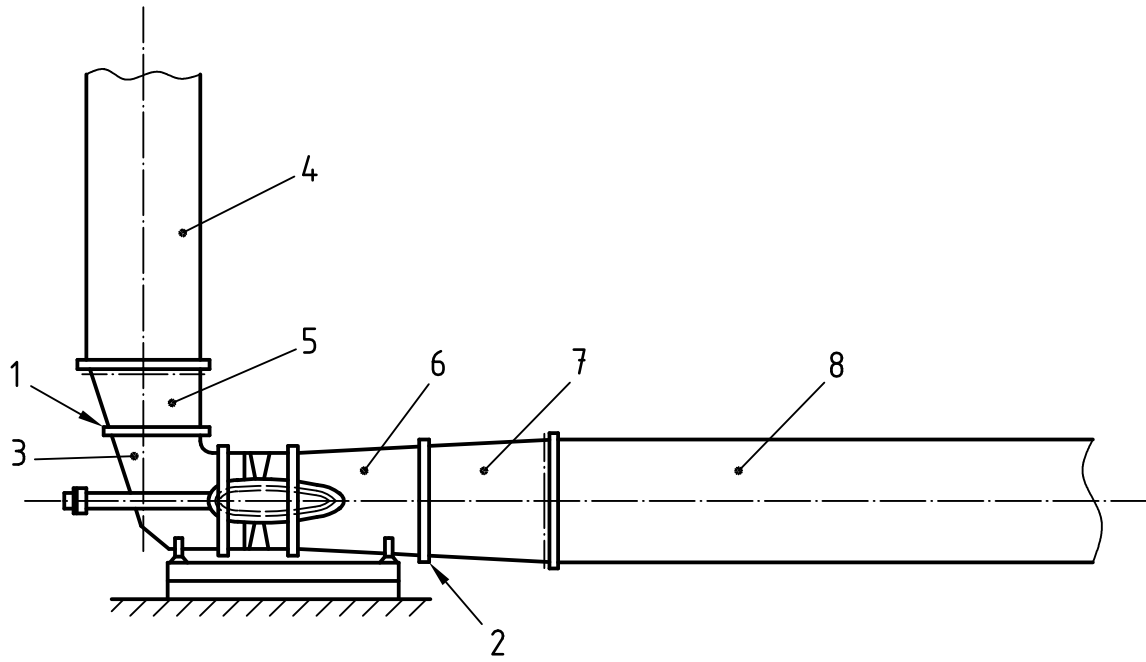
NOTE The fan outlet plane should be taken as that surface bounded by the downstream extremity of the air moving device. In this International Standard the outlet is indicated by plane 2 (see Figure 1).

3.1.13

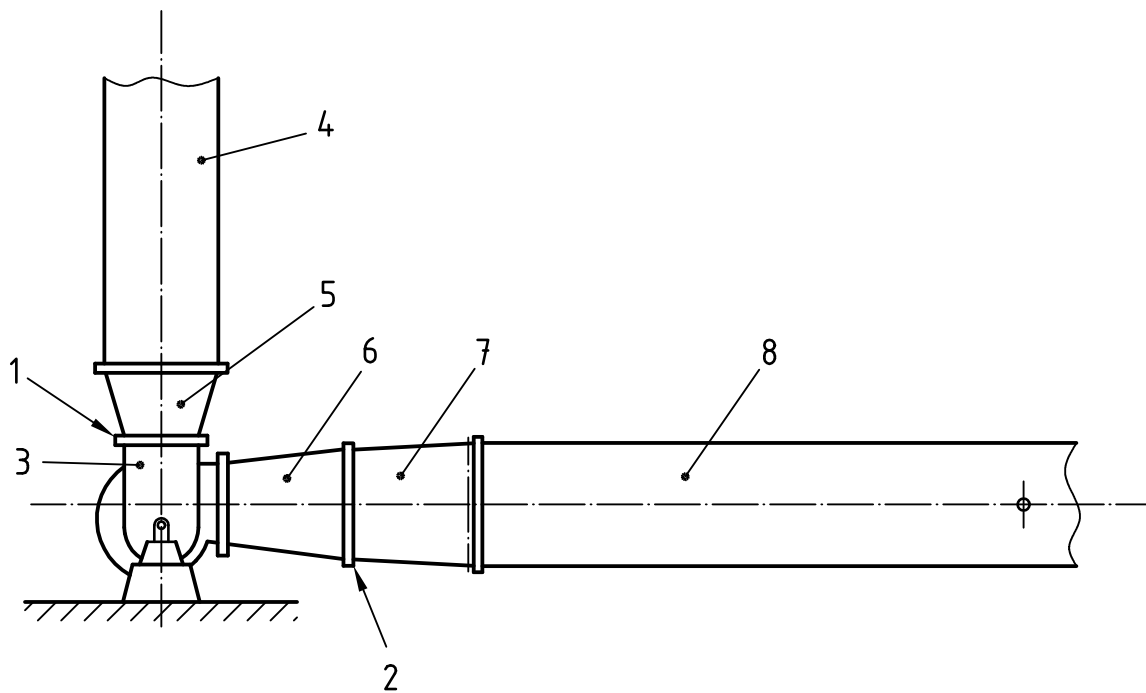
temperature

t

air or fluid temperature measured by a temperature sensor



a) Axial fan



b) Centrifugal fan

Key

- | | |
|--------------|---------------|
| 1 Plane 1 | 5 Transition |
| 2 Plane 2 | 6 Diffuser |
| 3 Inlet box | 7 Transition |
| 4 Inlet duct | 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

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

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:

$$Ma = \frac{v}{\sqrt{\kappa R_w \theta}}$$

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

$$Ma_x = \frac{v_{mx}}{\sqrt{\kappa R_w \theta_x}}$$

3.1.43

Mach factor

F_M

correction factor which is applied to the dynamic pressure at a point given by the expression

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

NOTE The Mach factor may be calculated by:

$$F_M = 1 + \frac{Ma^2}{4} + \frac{Ma^4}{40} + \frac{Ma^6}{1600} + \dots \quad \text{for } \kappa = 1,4$$

3.1.44

stagnation inlet density

ρ_{sg1}

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

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

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

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

3.1.46

mean density

ρ_m

arithmetic mean value of inlet and outlet densities

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

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_{Vsg1}

mass flowrate divided by the stagnation inlet density

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

3.1.49

outlet stagnation volume flow

q_{Vsg2}

mass flowrate divided by the stagnation outlet density

$$q_{Vsg2} = \frac{q_m}{\rho_{sg2}}$$

3.1.50

volume flow at a section x

q_{Vx}

mass flow at the specified airway cross section divided by the corresponding mean value in time of the average density at that section

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

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

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

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

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

NOTE y may be calculated as in 3.1.57.

3.1.53 fan static work per unit mass

y_s

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

3.1.54 fan pressure ratio

r_{Fp}

ratio of the average absolute stagnation pressure at the outlet section of a fan to that at its inlet section

$$r_{Fp} = \frac{p_{sg2}}{p_{sg1}}$$

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:

$$k = \frac{(\kappa - 1) \rho_{sg1} P_r \log_{10} r_{Fp}}{\kappa q_m p_F \log_{10} \left[1 + \frac{(\kappa - 1) \rho_{sg1} P_r (r - 1)}{\kappa q_m p_F} \right]}$$

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

$$P_u = q_m y \cong q_{Vsg1} p_F k$$

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}

$$P_{us} = q_m y_s \cong q_{Vsg1} k p_{sF}$$

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 $c = 340 \text{ m}\cdot\text{s}^{-1}$ 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

$(\zeta_{x-y})_y$

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

$$\lambda_P = \frac{P_r}{\rho_m u^3 D_r^2}$$

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	m·s ⁻¹
c_p	Specific heat at constant pressure	J·kg ⁻¹ ·K ⁻¹
c_V	Specific heat at constant volume	J·kg ⁻¹ ·K ⁻¹
d	Diameter of 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_{pF}	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	m·s ⁻²
h	Horizontal distance of the probe from the reference wall when the orthogonal coordinates are used	m
h_u	Relative humidity $\left(h_u = \frac{p_v}{p_{\text{sat}}} \right)$	
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	r·s ⁻¹
N	Rotational speed of impeller	r·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·s ⁻¹
q_V	Volume flowrate	m ³ ·s ⁻¹
q_{Vr}	Actual volume flowrate	m ³ ·s ⁻¹
q_{Vs}	Volume flowrate corresponding to standardized conditions of use of the DP device	m ³ ·s ⁻¹
q_{Vsg1}	Volume flowrate at the inlet at stagnation conditions	m ³ ·s ⁻¹
q_{Vsg2}	Volume flowrate at the outlet at stagnation conditions	m ³ ·s ⁻¹
q_{Vx}	Volume flowrate at section x	m ³ ·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·kg ⁻¹ ·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·kg ⁻¹ ·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·s ⁻¹
U	Voltage of electrical current	
v	Local absolute velocity	m·s ⁻¹
v_a	Axial velocity at test section	m·s ⁻¹
v_m	Mean value of v over time	m·s ⁻¹
v_{m1}	Mean value of v in the inlet section over time	m·s ⁻¹
v_{m2}	Mean value of v in the outlet section over time	m·s ⁻¹
v_{mx}	Mean value of v in section x over time	m·s ⁻¹
v_n	Local velocity normal to the cross-section	m·s ⁻¹
$v_x(y)$	Profile of velocities along the segment of the exploration line of the abscissa x	m·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·kg ⁻¹
y_{Fs}	Fan static work per unit mass	J·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	m ³ .s ⁻¹
Δp	Differential pressure	Pa
Δq_V	Absolute limit error on the determination of the volume flowrate q_V	m ³ .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_f	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 ($\zeta = \lambda \cdot L \cdot D_h^{-1}$)	
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	kg.m ⁻³
ρ_m	Mean density	kg.m ⁻³
ρ_{12}	Arithmetic mean value over time of inlet and outlet densities	kg.m ⁻³
ρ_x	Average density at section x	kg.m ⁻³
ρ_1	Mean density in the inlet section	kg.m ⁻³
ρ_2	Mean density in the outlet section	kg.m ⁻³
ρ_{sg1}	Stagnation inlet density	kg.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.

- 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.
- 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.
- 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.

- 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 fan¹⁾, D_h indicating the hydraulic diameter of the duct²⁾.

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.

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 $\pm 0,3 \%$.

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 $\pm 0,01 \text{ m/s}^2$) and for room temperature (within $\pm 10 \text{ }^\circ\text{C}$).

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

The barometer should 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 $\pm 1 \%$ 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 $\pm 1,0 \%$ 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 $\pm 1,0 \%$ 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 $\pm 5^\circ$; 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.