TECHNICAL REPORT

ISO/TR 16219

First edition
2020-09

Fans — System effects and system effect factors

Ventilateurs — Effet système et facteurs d'effet système

Reference number
ISO/TR 16219:2020(E)

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Published in Switzerland

Contents

Page

Foreword

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

Any feedback or questions on this document should be directed to the user's national standards body. A complete listing of these bodies can be found at www.iso.org/members.html.

Introduction

ISO 5801 provides the information for accurately measuring the performance of fans when tested under standardised laboratory conditions. The ducting where specified ensures a fully developed symmetrical velocity profile at the fan inlet. There may also be sufficient straight ducting at the fan outlet to ensure efficient conversion of the distorted velocity profile at the fan outlet to a measurable stable and homogeneous profile at the measuring station.

This document shows how fan performance is affected by both inlet and outlet connections to it. System designers must not only look at the ideal performance curve and calculated system pressure drop but also take into account the losses at the entry and exit points of the fan. These are described in the document.

The concept of the system effect factor (SEF) was introduced to the fan industry by AMCA in 1973. Since its inception it has become widely accepted worldwide. In more recent years it has been realized that the SEF depends not only on the fan type and the fitting geometry but also on the fan design and manufacturing. Some less efficient fans may sometimes be less sensitive to system effect induced by poor inlet flow conditions than more efficient fans of the same type.

Furthermore, the origin of the system effect induced by a fitting at the fan inlet is different from the one due to the same fitting located on the fan outlet. That is why two different definitions of SEF are proposed in this document according to whether the appurtenance is at the fan inlet or fan discharge.

Fans — System effects and system effect factors

1 Scope

This document deals with the likely degradation of air performance of fans tested in standardized airways in accordance with ISO 5801 when compared with the performance of fans tested under actual site conditions. It deals with the performance of a number of generic types of fan and fittings. The results given are intended as guidelines and only provide trends, as the system effect depends on the exact geometry of the fan and disturbing component.

The test data presented in this document are taken from an extensive experimental program conducted 20 years ago by NEL (National Engineering Laboratory, UK), mainly on axial and centrifugal fans. Data are also taken from several research projects financially supported by ASHRAE, some of them being carried out in the AMCA laboratory in Chicago, as well as from results published previously by individual fan manufacturers.

2 Normative references

There are no normative references in this document.

3 Terms, definitions and symbols

No terms and definitions are listed in this document.

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

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

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

The following symbols are used:

Symbol	Description	SI units	I-P units
A_2	Fan outlet area	m^2	ft^2
C	System effect (SE) coefficient (see 5.2)	Dimensionless	Dimensionless
p_C	Conventional pressure loss (see 5.2)	Pa	in. wg
p_f	Fan pressure	Pa	in. wg
p_{fd}	Fan dynamic pressure (see Clause 4)	Pa	in. wg
p_{fs}	Fan static pressure	Pa	in. wg
p_{SE}	System effect (see 5.2)	Pa	in. wg
p_{SEo}	Additional pressure loss due to non-uniform flow (see 5.2)	Pa	in. wg
q_{V1}	Volume flow rate of the fan	m^3/s	cfm
S_{EF}	System effect factor	Dimensionless	Dimensionless
ξ	Loss coefficient (see 5.1)	$(m^3/s)/(Pa^{0,5})$	
ρ	Density of air	kg/m^3	lbm/ft^2
ρ_{std}	Standard air density	kg/m^3	lbm/ft^2
NOTE The term “fan dynamic pressure” or “dynamic pressure” is used throughout this document and is equivalent to the term “velocity pressure” as used in some countries.			

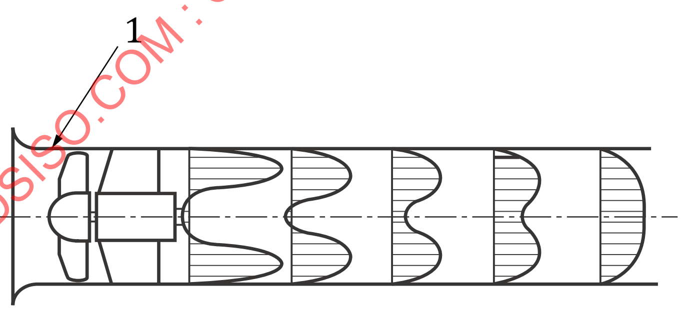
4 Origin of fan system effects

Manufacturers' fan performance ratings are mostly based on tests carried out in a laboratory under ideal conditions. Ideal conditions refer to uniform, swirl-free air velocity profiles at fan inlet and outlet, like those of the test rigs described in ISO 5801 and AMCA 210. In 'real life' fan installations, such ideal conditions may not be present due to improper connection of the fan to the system. Such improper connections include obstacles at fan inlets and outlets that alter the aerodynamic characteristics of the fan and lead to deficient performance in relation to catalogue ratings, even when the system pressure losses have been estimated accurately. The term "system effect" is a measure of this degradation of fan performance.

The origin of system effect is different at fan outlet and at fan inlet. At the fan outlet, for example in the case of an improperly connected outlet fitting such as an elbow, damper or duct branch, the system effect is linked to less-than-optimum non-uniform flow profiles induced by the fan at the entrance to the fitting (Figure 1). This degraded flow will create more pressure loss across the fitting than would be the case when measuring the fitting loss assuming uniform homogeneous flow profiles or when estimating it from standard handbooks such as the ASHRAE Handbook of Fundamentals[14].

When the fitting is at the fan inlet, for example an elbow or a fan inlet duct/box (Figure 2), the velocity profiles at the inlet to the fitting may be uniform and the fitting pressure loss as measured or estimated from standard handbooks may be valid. However, the flow patterns at the fan inlet (or fitting outlet) may be disturbed with the presence of a vortex, spin or vena-contracta. This less than optimum flow condition at fan inlet caused by the fitting will lead to a reorganization of the flow inside the impeller and therefore a deterioration of fan performance in relation to catalogue ratings. Not only the fan curve may be affected by this disturbing obstacle but also sometimes, but not always, the fan power curve. A companion document will be drafted at a later date to show the influence of the inlet obstacles on the fan power curve for the same configurations of fans and fittings as in this document.

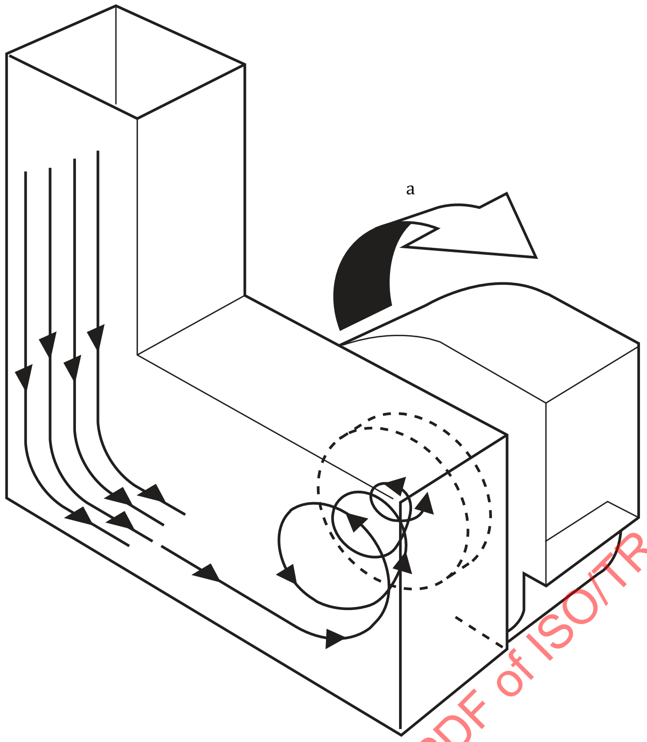
In both cases, the resulting air flow of the fan-system combination deteriorates, but for distinct physical reasons. For this reason, two different definitions and treatment of fan system effect are incorporated, depending on whether the fitting is at the fan inlet or fan outlet. It is also recognized that in some situations, obstacles very close to fan discharge (e.g. side walls at a short distance of a plenum fan impeller as shown in Figure 20) may also deteriorate fan performance in the same manner as components located at fan inlet.



Key

1 axial fan

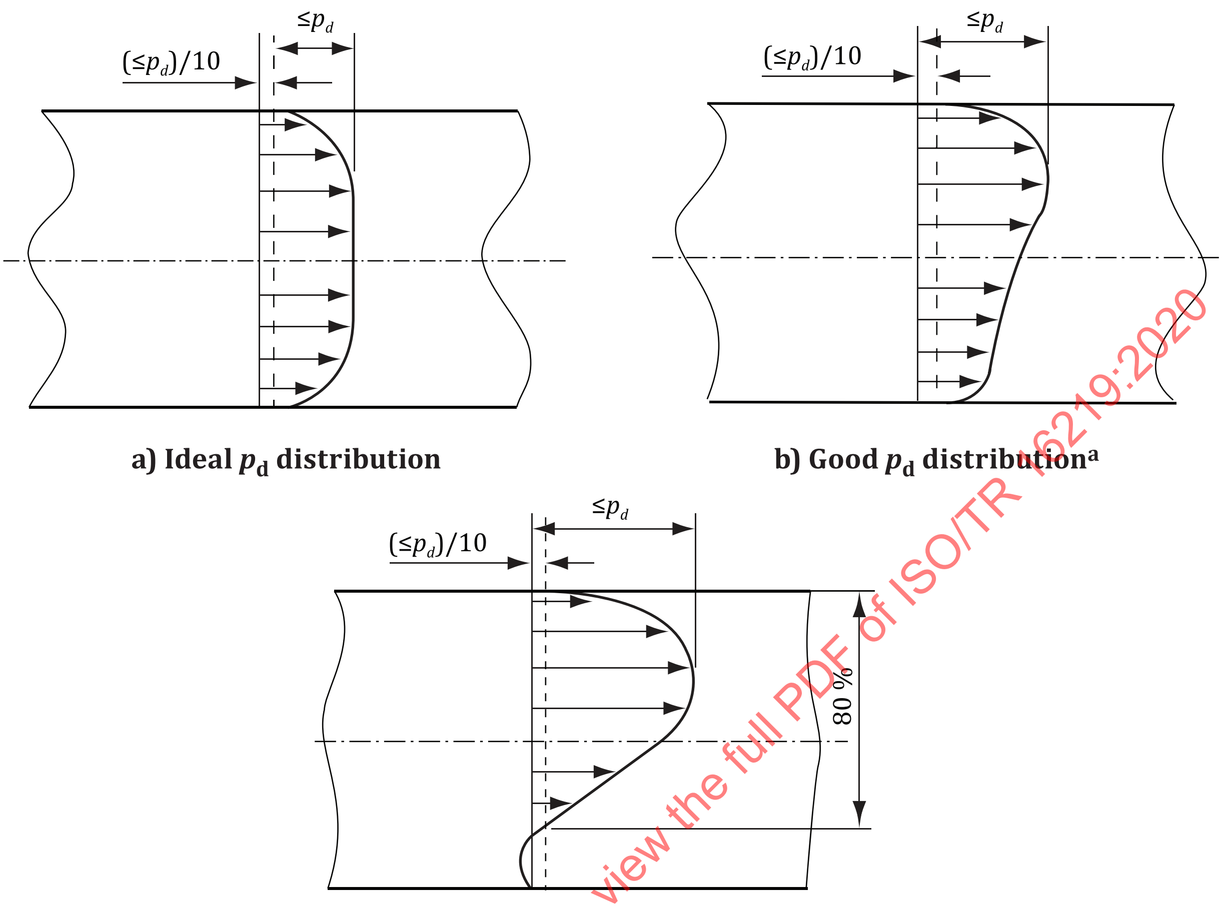
Figure 1 — Non-uniform velocity profiles at fan outlet



a Impeller rotation.

Figure 2 — Vortex at fan inlet

An ideal connection to a fan would be one which results in a velocity distribution across the fan inlet connection plane which is relatively uniformly distributed and without appreciable swirl component, as shown in Figure 3.



a) Ideal p_d distribution

b) Good p_d distribution[a]

c) Satisfactory p_d distribution[b]

Key

p_d mean dynamic pressure of the duct flow

[a] Also satisfactory for flow into fan inlets, but may be unsatisfactory for flow into inlet boxes, may produce swirl in boxes.

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

Figure 3 — Ideal fan connections

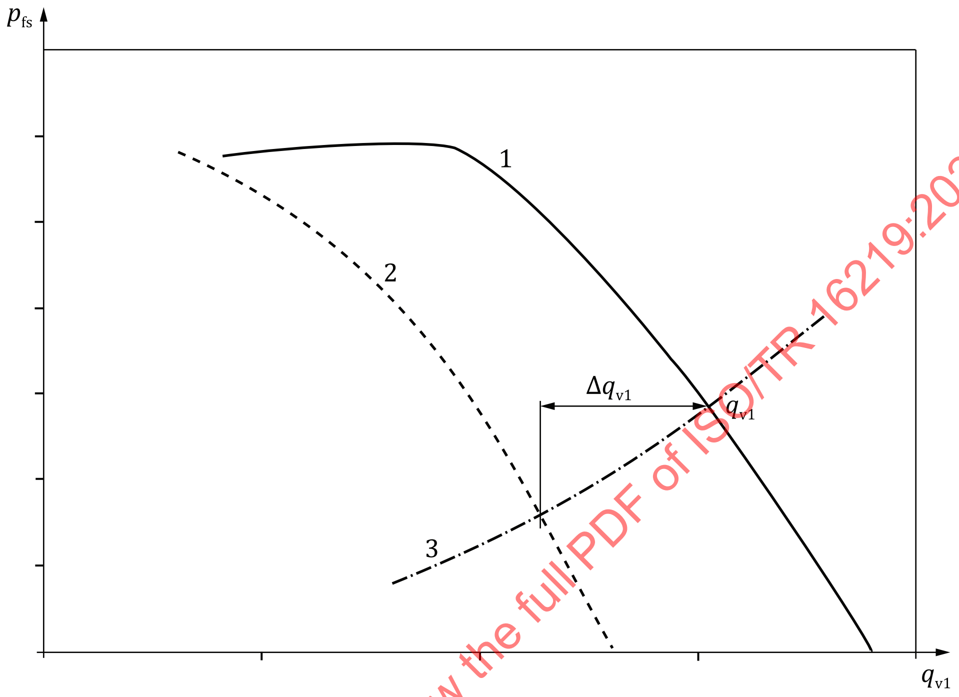
5 Definitions of system effect factor (SEF)

5.1 Inlet SEF

With a component at the fan inlet, the SEF is defined as the relative airflow drop $\Delta q_{v1}/q_{v1}$ along a given system line as shown in Figure 4. In this figure, the solid curve and the dotted line curve are the static pressure curves without and with system effect, respectively. The curve with system effect is obtained by adding the pressure loss of the fitting for each flow rate increment, when it may be measured or estimated from guidebooks (e.g. IDEL'CIK), to the static pressure of the fan + inlet fitting combination. This procedure allows for the assessment of the installation effect related to the degradation of the fan curve itself without accounting for the pressure loss of the fitting.

To quantify the system effect on the whole fan curve, the quantity $\Delta q_{V1}/q_{V1}$ is plotted versus the system resistance coefficient $\xi = q_{V1} / \sqrt{p_{fs}}$ (p_{fs} being the fan static pressure at q_{V1})[1)] in Figure 5.

The SEF for a given fan + inlet fitting configuration is the average of $\Delta q_{V1}/q_{V1}$ over the ξ range, presented as a percentage in the results. $\Delta q_{V1}/q_{V1}$ is positive when the flow with the inlet fitting is lower than that of the free inlet configuration.



Key

q_{V1} volume flow rate of the fan

p_{fs} fan pressure

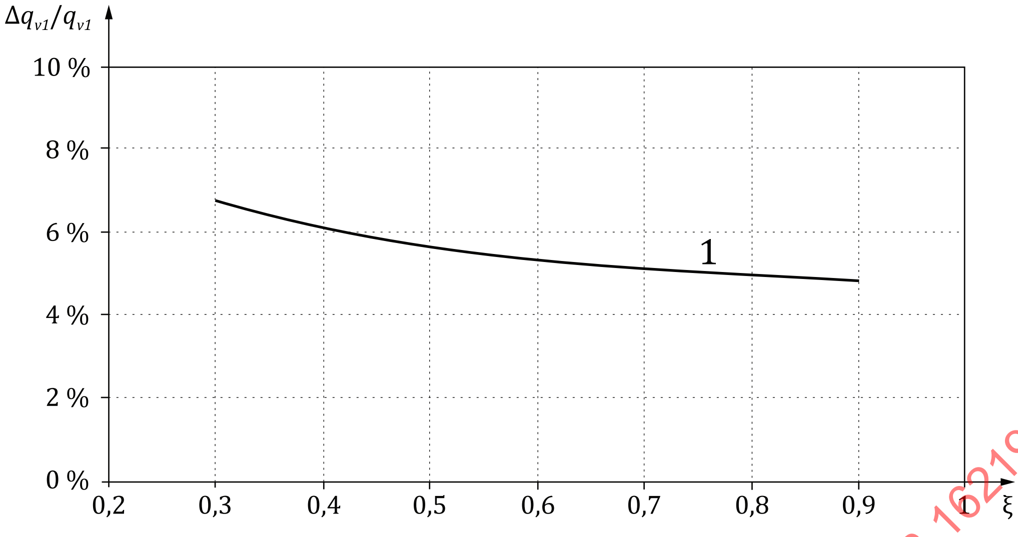
1 fan curve without system effect

2 fan curve with system effect

3 system line

Figure 4 — Definition of q_{V1} and Δq_{V1} on a given system line

1) q_{V1} is either in cfm or m^3/s while p_{fs} is either in in. wg or Pa.



Key

$\Delta q_{V1}/q_{V1}$ relative flow drop in volume flow rate of the fan

ξ system resistance coefficient

1 system effect curve

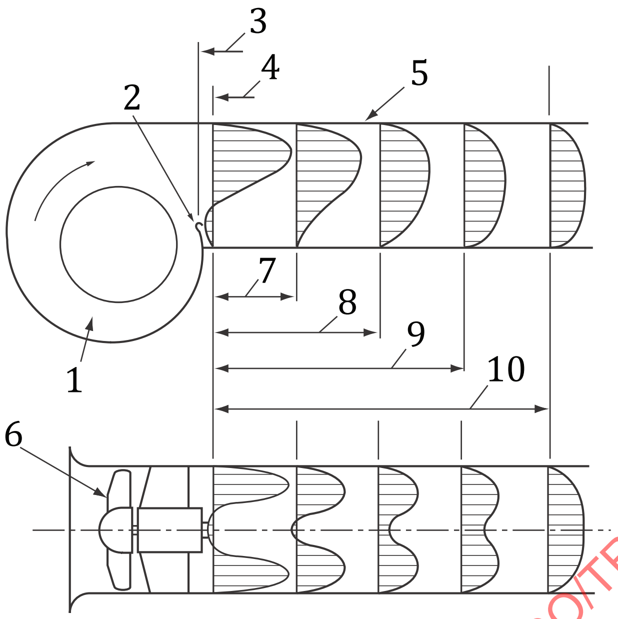
Figure 5 — Example of relative flow drop $\Delta q_{V1}/q_{V1}$ versus system resistance coefficient ξ

Clause 6 describes various situations resulting in inlet system effects.

5.2 Outlet system effect

Outlet system effect is a measure of the pressure losses across fan outlet appurtenances such as an outlet duct, elbow, volume control damper, duct branch or plenum, due to non-uniform outlet flow induced by the fan and improper outlet connections.

Most fans, for applications requiring systems connected at their outlets, are tested and rated for performance with an outlet duct 2 to 3 'equivalent duct diameter' long. The outlet duct helps control the diffusion of the outlet flow and establish a uniform velocity profile (Figure 6). In most cases, it is not practical for the fan manufacturer to supply this duct as part of the fan, but rated performance will not be achieved unless a comparable duct is included in system design.



Key

1 centrifugal fan
2 cutoff
3 blast area
4 outlet area
5 discharge duct
6 axial fan
7 25 % effective duct length
8 50 % effective duct length
9 75 % effective duct length
10 100 % effective duct length

Figure 6 — Velocity profiles at fan outlet

The techniques documented to estimate pressure losses of a fitting such as an elbow or the published pressure drop performance from a manufacturer of a fitting such as a damper are based upon uniform approach velocity profiles. The pressure loss so estimated is referred to as the 'conventional pressure loss' across the fitting. Unless uniform approach velocity profile is ensured, there will be additional pressure losses across these fittings. Outlet system effect is used to estimate the actual pressure loss across the fitting in a given installation.

Clause 7 describes various situations resulting in outlet system effects. The total outlet system effect, p_{SE} (Pa), for a given situation (fitting) is defined as:

$$p_{SE} = p_c + p_{SEo}$$

where

p_c is conventional pressure loss (Pa);

p_{SEo} is additional pressure loss due to non-uniform flow (Pa).

p_{SEo} can be expressed as a function of flow by the following formula:

$$p_{SEo} = C \times p_{fd2}$$

where

p_{fd2} is dynamic pressure at fan outlet $0,5*\rho*(q_{V1}/A_2)^2$;

q_{V1} is fan airflow rate, m^3/s;

A_2 is fan outlet area in m^2;

C is system effect coefficient;

ρ is air density in kg/m^3.

The outlet system effect p_{SE} at each flow rate q_{V1} must be added to the design system curve to obtain the actual system curve (Figure 7).

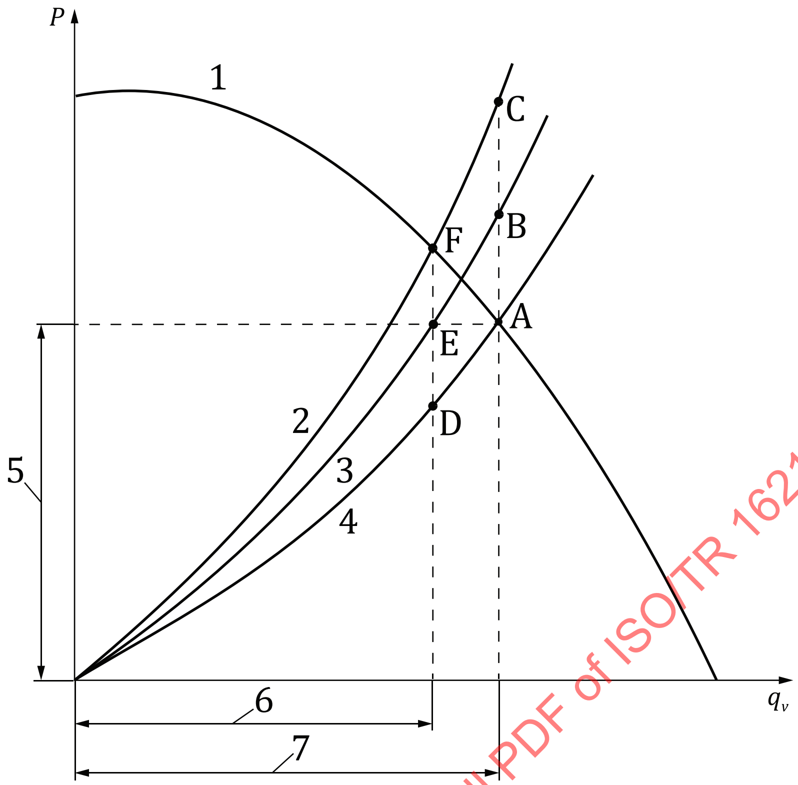
The system effect coefficient C is averaged over the fan curve to obtain what is called the outlet SEF in Clause 7.

$P_B - P_A$ = fitting conventional pressure drop at design flow

$P_C - P_B$ = outlet system effect, p_{SEo}, at design flow

$P_E - P_D$ = fitting conventional pressure drop at actual flow

$P_F - P_E$ = outlet system effect, p_{SEo}, at actual flow



Key

q_V fan volume flow rate
P fan pressure
1 fan catalogue pressure-flow curve
2 actual system curve
3 system curve with fitting conventional pressure drop
4 system curve without conventional pressure drop and no allowance for system effect
5 design pressure
6 actual flow
7 design flow

Figure 7 — Modification of design system curve due to outlet system effect

In some cases the conventional pressure loss p_c cannot be estimated or is not relevant, like for instance with side walls close to the impeller of a plenum fan in the example of 7.2.2.2. In this case the system effect is due to the disturbed flow in the impeller induced by the proximity of the walls.

6 Examples of inlet SEF

6.1 Introduction

Examples of inlet system effect are taken from different dedicated research programs carried out since the 1990s. The National Engineering Laboratory (NEL) in the UK performed an extensive experimental study on nine different types of fans and six ductwork fittings at the fan inlet. A summary of the test configurations and main results obtained is given in References [3] and [4]. Otherwise, several research programs have been financially supported by ASHRAE in which the tests were performed mainly by AMCA to quantify the SEF on:

— a backward inclined/airfoil centrifugal fan – ASHRAE Research Project 1216-RP[5];

— a forward curved centrifugal fan – ASHRAE Research Project 1272-RP[6];

— two airfoil centrifugal plenum fans – ASHRAE Research Project 1420-TRP[7];

— three sizes of propeller fans of the same series – ASHRAE Research Project 1223-RP[8].

Finally, a test was done more recently by AMCA on a forward curved centrifugal fan with an inlet 90° segmented elbow at various orientations.

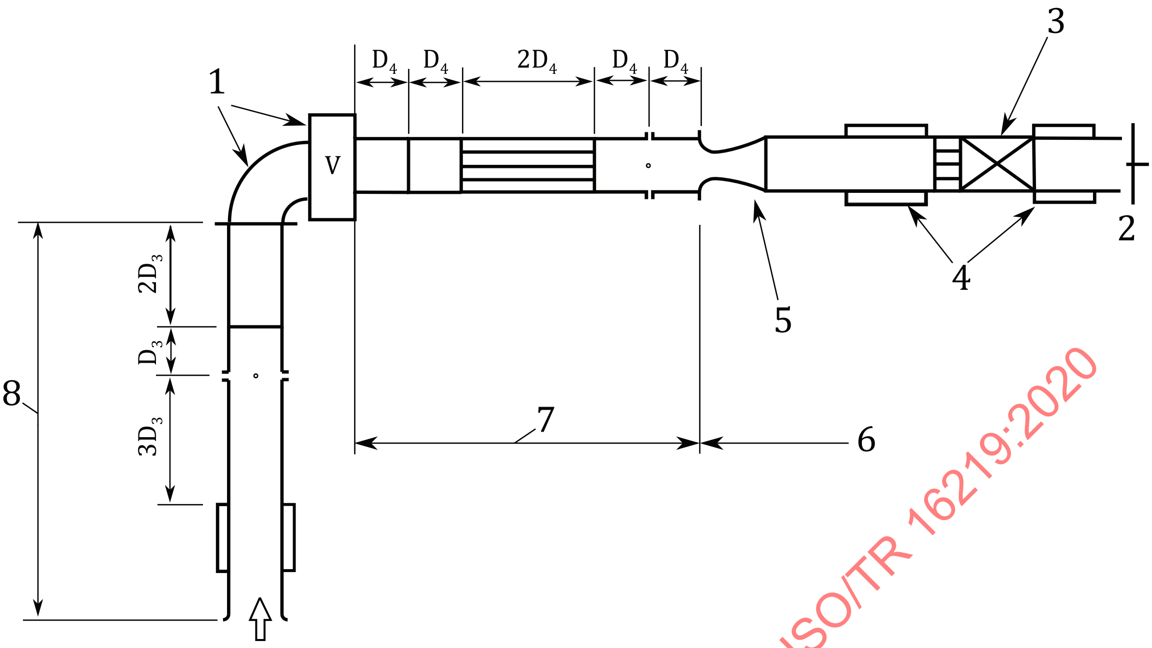
6.2 Axial fans

6.2.1 Experimental setups

6.2.1.1 NEL

All the tests were performed on a ductwork of D = 630 mm, where D is the duct diameter. A layout of the test ductwork with a bend connected to the fan inlet is shown in Figure 8. The distance between the inlet fitting and the fan is varied from 0D, as in Figure 8 to 2D.

Details of the experimental program and measurement procedure are given in Reference [4] and private reports. The test data used in the present analysis are the performance curves of the fan alone and fan + inlet fitting and the measured pressure losses of the fittings. All the fan curves, initially based on total pressure, were transformed into static pressure curves by subtracting the dynamic pressure at the fan outlet according to ISO 5801.



Key

1 test fan and fitting
2 throttle
3 auxiliary boost fan
4 silencers
5 flow measurement nozzle
6 flow measurement and control section
7 outlet duct
8 inlet duct

SOURCE Based on content from National Engineering Laboratory (NEL) Fan Connected Ductwork Study for FETA (FET001) February 1992, reproduced with permission from the Fan Manufacturers Association, FETA UK.

Figure 8 — Test rig for determination of installation effect — Fitting at fan inlet

Table 1 gives the main characteristics of the axial fans tested by NEL while Figure 9 shows views of the fans, including centrifugal fans. Figure 10 presents sketches of the fittings that were connected to the fan inlet (or outlet) via transition elements.

They include:

a) rectangular/circular transition, section 800 × 400 → D = 630 mm, length 950 mm;

b) short square bend 90°, section 630 × 630, curvature radius 100 mm;

c) square mitred bend 90°, section 630 × 630, with guide vanes;

d) circular five-piece segmented bend, D = 630 mm;

e) rectangular to rectangular box fitting, section 800 × 400, length 2 400 mm;

f) rectangular splitter silencer, section 800 × 400, length 1 200 mm;

g) banjo connector, section 1 260 × 630, length 1 890 mm.

Table 1 — Main characteristics of the axial fans tested by NEL

Fan	Fan type	Blade setting °	Hub/tip ratio	Speed rpm
1	tubeaxial	24	0,223	1 440
2	tubeaxial	30	0,223	1 440
3	vaneaxial	24	0,389	1 440
4	vaneaxial	32	0,389	1 440
5	tubeaxial	24	0,389	2 900
6	tubeaxial	32	0,389	2 900

NOTE All the fans have a diameter of 630 mm.

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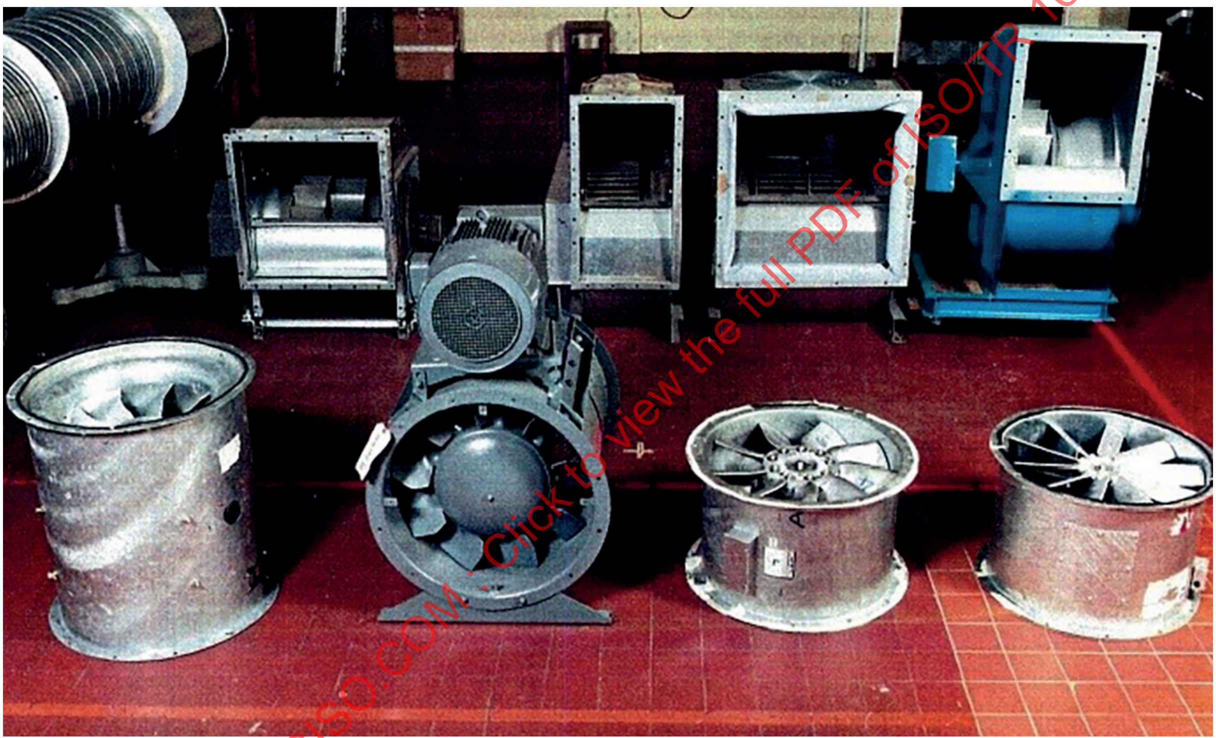

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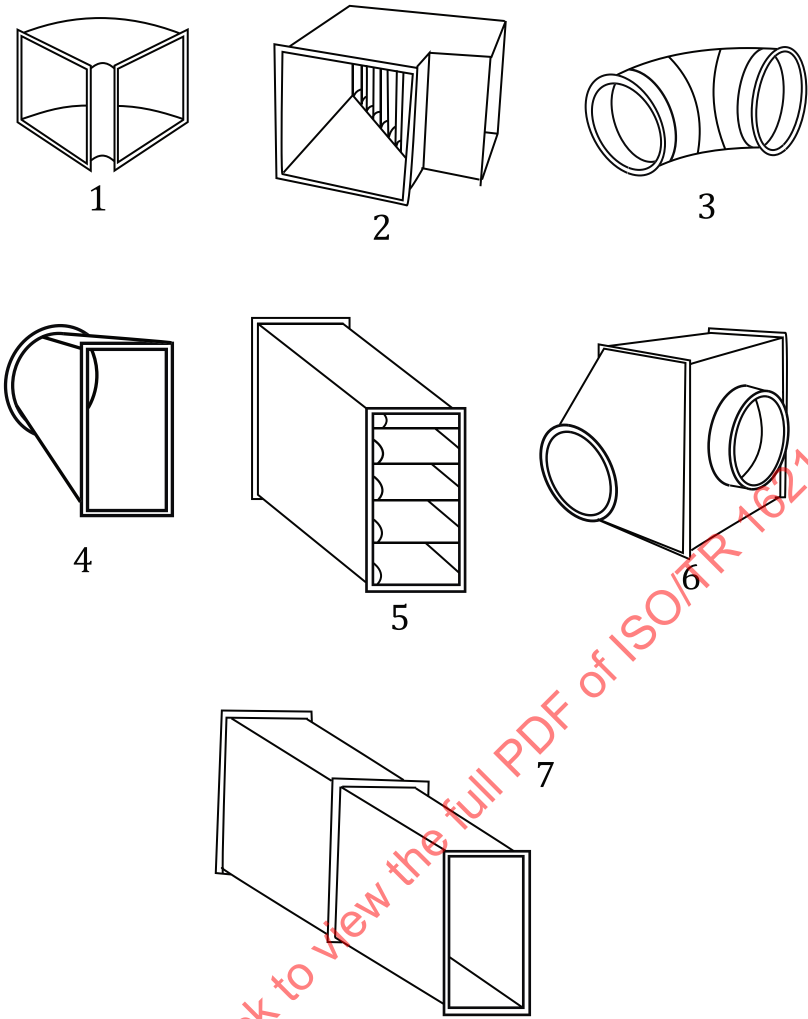
Figure 9 — Views of the NEL test fans



Key

1 short square bend 90°
2 square mitred bend 90°
3 circular five-piece segmented bend
4 rectangular/circular transition
5 rectangular splitter silencer
6 banjo connector
7 rectangular to rectangular box fitting

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Figure 10 — NEL fittings

6.2.1.2 ASHRAE 1223-RP

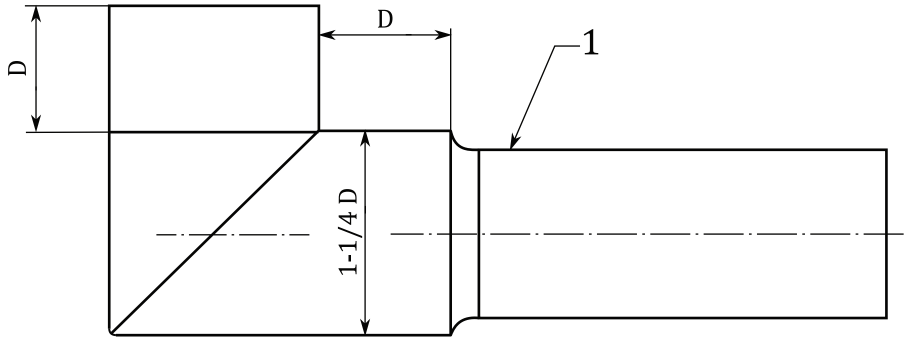
This project dealt with the inlet installation effect on three propeller fans of diameters D = 610 mm, 914 mm and 1 219 mm each running at two speeds. The speeds range from 259 rpm to 908 rpm, according to the diameter. The inlet fittings are:

— a 90° round mitred elbow with five angular positions ranging from 0° to 270° (Figure 11);

— three round inlet ducts with contraction area ratios of 1,0, 1,25 and 1,5 (Figure 12);

— a wall at several distances ranging from 0,25D to 2D in front of the fan inlet (Figure 13).

Details of the experimental setups are presented in Reference [8].



Key

1 basic fan set up

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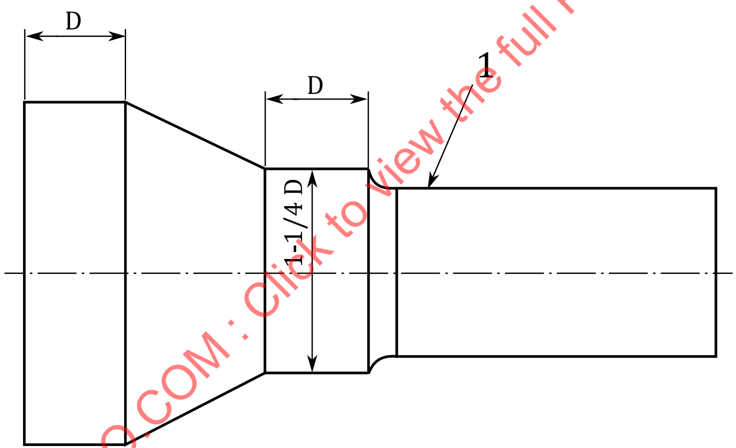
Figure 11 — 90° Round mitred elbow



Key

1 basic fan set up

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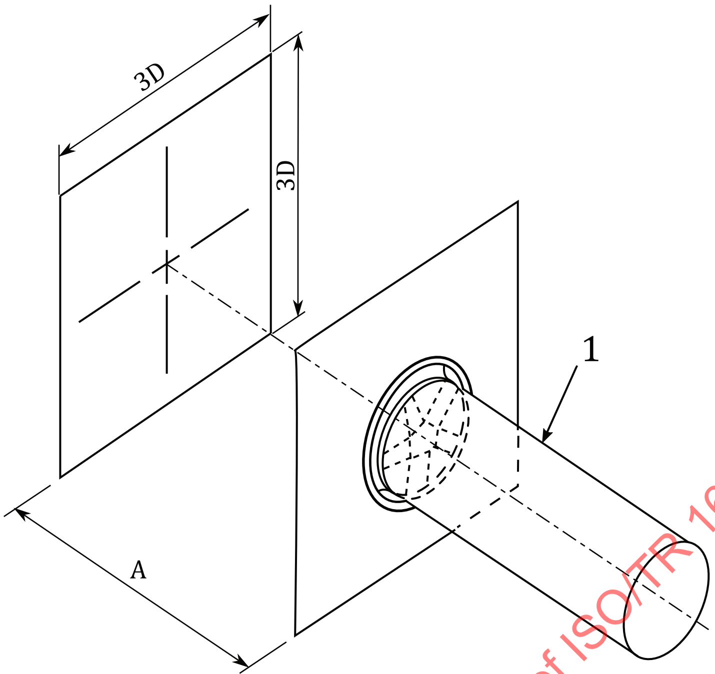
Figure 12 — Inlet duct with contraction



“A”	
1/4	D
1/2	D
3/4	D
1	D
1–1/4	D
1–1/2	D
2	D

Key

1 basic fan set up

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Figure 13 — Wall in front of the fan inlet

6.2.2 Results

6.2.2.1 Introduction

As stated in 5.1 the SEF due to inlet fittings is the relative flow drop $\Delta q_{v1}/q_{v1}$ averaged over the flow range. If the pressure loss of the appurtenance is known, it is added for each flow rate increment to the fan + fitting curve in order to quantify the system effect due only to the degradation of the fan curve by the flow distortion induced by the fitting.

In the results analysed below the pressure losses of the components have been measured (or estimated from a database) in the NEL tests and in ASHRAE 1223-RP for the mitred elbow and the contractions[2].

2) In the configuration with the wall in front of the fan (Figure 13) there is no pressure drop of the inlet disturber as that is not relevant.

In the following table of results the inlet SEF is shown as a percentage with the following convention depending on whether the system effect is considered as small, medium or high.

x %	SEF < 5 %
y %	5 % ≤ SEF < 10 %
z %	SEF ≥ 10 %

6.2.2.2 NEL

The pressure loss coefficients of the fittings assessed from NEL measurements with axial fans are presented in Table 2. Except for the rect/circ transition, where the loss coefficient is very low, the coefficients of the 90° bends of various shape are similar.

Table 2 — Pressure loss coefficients of the NEL fittings at the axial fan inlet

Fitting	Pressure loss coefficient
Rect/circ transition (a)	0,07
Short square bend (b)	0,36
Square mitred bend (c)	0,40
Segmented bend (d)	0,32

Table 3 shows the SEF calculated for the axial fans of Table 1 and some of the inlet fittings described in 6.2.1.1 for four gaps L/D between the component and the fan. The SEF is small or even insignificant in most of the combinations of fans and fittings, except on fan 2 where the SEF is higher with bends (b) and (c). In this case the larger the gap, the lower the SEF.

Table 3 — SEF of axial fans with inlet fittings

		Average Dq_{v1}/q_{v1}					
		Axial					
	L/D	1	2	3	4	5	6
Rect/circ transition (a)	0	−0,4 %	−0,9 %	0,0 %	−0,1 %	−0,3 %	−1,2 %
	0,5						−0,5 %
	1	0,5 %	0,1 %	0,3 %	0,0 %	−0,4 %	0,3 %
	2	−1,0 %	0,3 %	−0,4 %	−0,4 %	−0,6 %	0,7 %
Short square bend (b)	0	3,7 %	**6,3 %**	0,9 %	2,7 %	0,4 %	−0,2 %
	0,5	2,6 %	**6,2 %**	−0,8 %	2,0 %		
	1	0,3 %	3,8 %	1,0 %	−0,6 %	−0,3 %	0,6 %
	2	0,5 %	0,7 %	−0,6 %	−0,4 %	0,3 %	−1,2 %
Square mitred bend (c)	0	0,2 %	**6,4 %**	2,5 %	0,0 %	2,0 %	0,5 %
	0,5		**5,6 %**	2,2 %	−0,5 %		
	1	0,1 %	**5,4 %**	1,9 %	−0,1 %	0,8 %	0,5 %
	2	−0,2 %	2,5 %	−0,8 %	0,0 %	−0,3 %	−0,2 %
Segmented bend (d)	0	−1,0 %	−1,0 %	−1,9 %	−0,3 %	−1,6 %	−0,3 %
	0,5	1,5 %	1,9 %	−0,2 %	−0,3 %		
	1	3,6 %	0,6 %	0,5 %	0,0 %	−1,4 %	0,9 %
	2	1,8 %	1,3 %	1,0 %	−0,9 %	−1,3 %	−0,7 %
NOTE Blank cells represent no data.							

6.2.2.3 ASHRAE 1223-RP

Table 4 presents the results obtained on the three impellers with the different fittings. For each fan, the SEF has been averaged over the test speeds. The pressure losses of the elbows and contractions have been taken into account to calculate the SEF but the losses at the entrance of these components have not been accounted for, which may be important since there is no inlet bell at the entrance of the ductwork (see Figure 11 and Figure 12). Even if some differences are observed between the results of the three fans for reasons that may be linked to a size effect, the fact that the fans are not absolutely geometrically similar or both, the general trend observed is close.

The influence of the contraction decreases as the area ratio increases from 1 to 1,5. That the SEF is the highest for an area ratio of 1 can be explained by the fact that the pressure loss at the entrance of the fitting is not considered in the calculation. The duct velocity and thus the pressure loss are indeed higher for the area ratio of 1 than for the two other ratios. If this loss is taken into account, the difference between the SEF of the three contraction ratios is very small.

The elbow induces a strong SEF, and its orientation has some influence with a maximum effect obtained at 0° (elbow oriented vertically upwards). The wall in front of the fan inlet has also a strong effect when its distance to the fan is 0,25D. The effect is still not negligible when this distance increases, especially with the propeller of 1 219 mm.

Table 4 — SEF of propeller fans with inlet fittings

	Propeller 610 mm	Propeller 914 mm	Propeller 1 219 mm
Inlet fittings	Average $\Delta q_{v1}/q_{v1}$		
Contraction AR = 1	**5,4 %**	*4,7 %*	**8,9 %**
Contraction AR = 1,25	*2,3 %*	*2,5 %*	*4,6 %*
Contraction AR = 1,5	*0,6 %*	*0,5 %*	*3,1 %*
Elbow oriented 0°	**10,7 %**	**12,7 %**	**13,7 %**
Elbow oriented 45°	**6,7 %**	**11,9 %**	**11,8 %**
Elbow oriented 90°	**5,5 %**	**9,9 %**	**13,3 %**
Wall at 0,25D	**17,1 %**	**12,8 %**	**11,3 %**
Wall at 0,5D	*4,2 %*	*1,3 %*	**5,7 %**
Wall at 0,75D	*3,2 %*	*1,5 %*	**6,1 %**
Wall at 1D	*3,6 %*	*1,2 %*	**6,4 %**
Wall at 1,25D	*2,6 %*		

NOTE Blank cells represent no data.

6.3 Centrifugal and mixed-flow fans

6.3.1 Experimental setups

6.3.1.1 NEL

Table 5 gives the main characteristics of the centrifugal and mixed-flow fans tested by NEL. The fittings tested are described in 6.2.1.1. The pressure loss coefficients of these fittings measured by NEL are shown in Table 6.

Table 5 — Main characteristics of the centrifugal and mixed-flow fans tested by NEL

Fan	Fan type	Impeller diameter	Speed (rpm)
7	Mixed-flow with guide vanes	630	1 470
8	FC centrifugal, single inlet	630	850
9	BC centrifugal, single inlet	610	2 600
10	FC centrifugal, double inlet	630	850
11	BC centrifugal, double inlet	510	1 800
SOURCE: Based on content from National Engineering Laboratory (NEL) Fan Connected Ductwork Study for FETA (FET001) February 1992, reproduced with permission from the Fan Manufacturers Association, FETA UK.			

Table 6 — Pressure loss coefficients of the fittings

Fitting	Pressure loss coefficient
Rect/circ transition	0,07
Short square bend	0,36
Square mitred bend	0,40
Segmented bend	0,32
Rect splitter silencer	2,42
Banjo connection	4,16

6.3.1.2 ASHRAE 1216-RP

In this project, a backward inclined (BI) airfoil centrifugal fan of impeller diameter D = 762 mm is tested according to test configuration B with various inlet fittings, i.e. five bearings with their supports (see an example in Figure 14) and two cabinets of heights 2D and 3D and width L varying from 2D to 0,25D (Figure 15). The tests have been carried out at three rotation speeds: 796, 1 327 and 1 731 rpm. Details of the experimental setups are presented in Reference [5].

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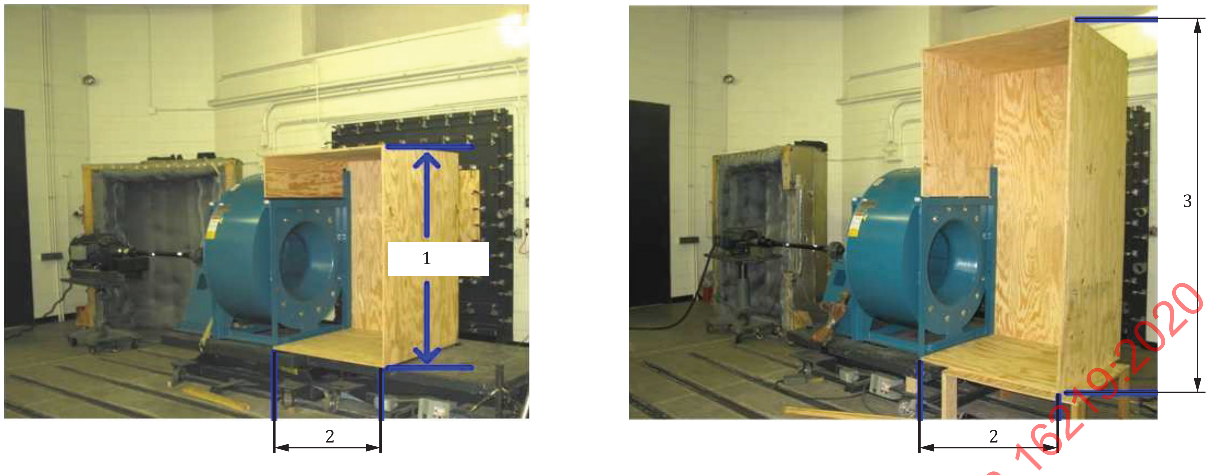
Figure 14 — BI centrifugal fan with inlet bearing obstruction



1 2D height
2 width L
3 3D height

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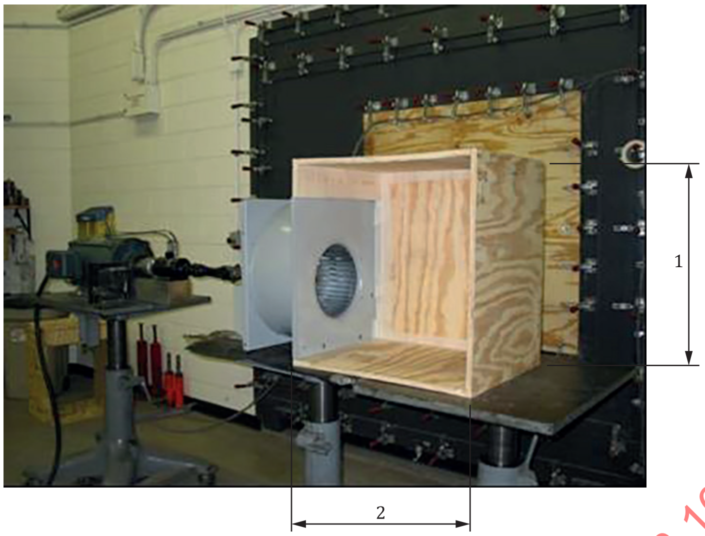
Figure 15 — BI centrifugal fan with inlet cabinets 1 (left) and 2 (right)

6.3.1.3 ASHRAE 1272-RP

The objective and the test procedure of this project are similar to those of ASHRAE 1216-RP. The test fan is a forward-curved centrifugal fan of impeller diameter D = 321 mm, and the inlet fittings are four bearings of various types and two cabinets of heights 2D and 3D and width L varying from 2D to 0,25D. The test speeds are 1 000, 1 500 and 2 000 rpm. Details of the test setup are shown in Reference [6]. Figure 16 shows a view of the fan with inlet cabinet 1 of 2D height.



Key

1 2D height

2 width L

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Figure 16 — FC Centrifugal Fan with inlet cabinet 1

6.3.1.4 ASHRAE 1420-TRP

This project was carried out on two airfoil centrifugal plenum fans of impeller diameters D = 381 mm and 686 mm, according to test configuration A (Figure 17). The rotation speeds are 3 150 rpm for the small impeller and 1 575 rpm for the large impeller. For each fan the system effects due to inlet fittings were determined. Details of the test setup are presented in Reference [7].

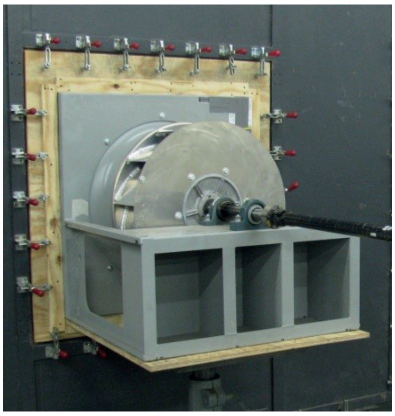

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Figure 17 — Views of the two centrifugal impellers

For each fan the inlet appurtenances are:

— a pillow block bearing with support (Figure 18);

— a coil in an inlet cabinet at four locations of the fan 1D, 0,75D, 0,5D and 0,25D (Figure 19);

— 10 configurations of inlet cabinet of 1D depth with the right, top and left walls at four locations ranging from 1D to 0,25D (Figure 20 and Table 7). One of the cabinets was also tested with an inlet cone to determine if it was necessary for all cabinet tests;

— a return fan inlet cabinet of 1D width with the right side wall at four locations 1D, 0,75D, 0,5D and 0,25D (Figure 21) plus a 45° baffle (Figure 22).

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Figure 18 — Inlet pillow block bearing with support

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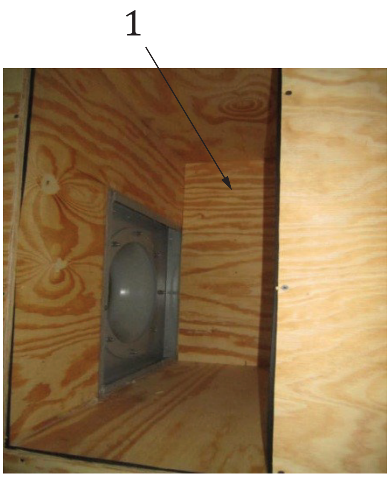
Figure 19 — Coil in an inlet cabinet

Figure 20 — Inlet cabinet with side wall spacings 1D right, 1D top and 1D left

Table 7 — Configurations of inlet cabinet with various side wall spacings

	Right	Top	Left
Box 1	1D	1D	1D
Box 2	0,75D	1D	1D
Box 3	0,75D	0,75D	1D
Box 4	0,75D	0,75D	0,75D
Box 5	0,5D	1D	1D
Box 6	0,5D	0,5D	1D
Box 7	0,5D	0,5D	0,5D
Box 8	0,25D	1D	1D
Box 9	0,25D	0,25D	1D
Box 10	0,25D	0,25D	0,25D



Key

1 right side wall

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Figure 21 — Return fan inlet cabinet with perpendicular right side wall at 0,25D from the impeller

Key

1 45° baffle

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Figure 22 — Return fan inlet cabinet with right side wall at 45°

6.3.1.5 Elbow at the inlet of a forward-curved centrifugal fan

This test was done on a FC centrifugal fan with an impeller diameter D = 305 mm and a 90° segmented elbow of inlet diameter 203 mm according to test configuration B. The fan speed was 1 750 rpm. The system effect was assessed for 12 orientations of the elbow from position 12 o'clock, corresponding to the inlet orientated upwards (Figure 23).

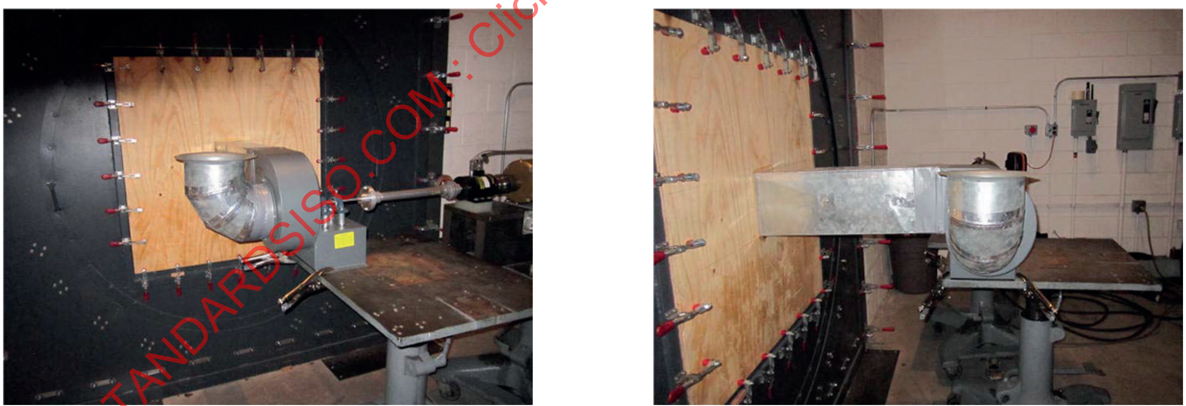


Figure 23 — Views of the FC centrifugal fan with the inlet elbow at 12 o'clock

6.3.2 Results

6.3.2.1 NEL

Table 8 shows the SEF calculated for the mixed-flow fan and two centrifugal fans of Table 5 and some of the inlet fittings described in 6.2.1.1 for four gaps L/D between the component and the fan. The pressure losses of the fittings are taken into account in the SEF assessment.

Table 8 — SEF due to inlet fittings

		A Average $\Delta q_{v1}/q_{v1}$ Mixed flow and centrifugal		
	L/D	**7**	**8**	**9**
Rect/circ transition (a)	0	*0,5 %*	*−0,4 %*	*0,1 %*
	0,5			
	1	*0,9 %*	*−0,3 %*	*0,4 %*
	2	*0,7 %*	*−0,1 %*	*0,7 %*
Short square bend (b)	0	*1,0 %*	*1,3 %*	*2,0 %*
	0,5			
	1	*0,7 %*	*2,6 %*	*1,0 %*
	2	*1,2 %*	*0,0 %*	*1,0 %*
Square mitred bend (c)	0	*1,2 %*	*2,5 %*	*1,0 %*
	0,5			
	1	*1,2 %*	*2,7 %*	*1,0 %*
	2	*1,6 %*	*1,8 %*	*1,0 %*
Segmented bend (d)	0	*1,1 %*	*−0,2 %*	*−0,1 %*
	0,5			*0,3 %*
	1	*1,4 %*	*1,2 %*	*0,4 %*
	2	*1,7 %*	*−0,1 %*	*0,8 %*
Rect splitter silencer (f)	0	*2,1 %*		
	0,5			
	1	*1,8 %*		
	2	*1,5 %*		
Banjo connection (g)	0			**16,8 %**
	0,5			
	1			
	2			**19,6 %**
NOTE Blank cells represent no data.				

Table 8 reveals that the SEF is small or insignificant in most of the combinations of fans and fittings. A notable exception is the banjo connector with fan 9 (BC centrifugal fan). For this configuration, according to the authors of the study the flow was very unstable and the performance dropped considerably. A similar trend was obtained with the banjo connector fitted to fan 8 (FC centrifugal fan), for which no results appear in Table 8 due to inconsistencies. Therefore, a banjo connector at the inlet of a centrifugal fan is not recommended at all.

6.3.2.2 ASHRAE 1216-RP

The tests have been done at three fan speeds, and the results presented here are values averaged over the three speeds.

Table 9 presents the system effect due to the different inlet bearings. The effect is nearly negligible, the maximum SEF being obtained with the 10 % flange with crossing supports (configuration shown in Figure 14).

Table 9 — SEF due to bearings at the inlet of the BC centrifugal fan

Bearings	
Type of bearing	**Average $\Delta q_{v1}/q_{v1}$**
5 % flange	*0,4 %*
10 % flange	*1,6 %*
5 % pillowblock	*0,8 %*
10 % pillowblock	*0,5 %*
10 % flange + cross supp	*2,0 %*

Table 10 shows the effect of the two inlet cabinets of heights 2D (cabinet 1) and 3D (cabinet 2). The SEF is important for the smallest width L = 0,25D, and it is continuously decreasing when L increases. There is a difference between the two cabinets, the SEF being higher whatever L with cabinet 2. This may be due to the asymmetry of the bottom and top walls of the box with respect to the fan axis, while the walls of cabinet 1 are more symmetrical[5].

A width of at least 1D is thus necessary to obtain a relatively small system effect, whatever the cabinet height.

Table 10 — SEF due to the inlet cabinets

Cabinet 1	
L/D	**Average $\Delta q_{v1}/q_{v1}$**
0,25	**15,4 %**
0,5	**5,4 %**
0,75	*1,8 %*
1	*1,1 %*
1,25	*0,2 %*
1,5	*1,0 %*
2	*0,4 %*

Cabinet 2	
L/D	**Average $\Delta q_{v1}/q_{v1}$**
0,25	**16,0 %**
0,5	**7,9 %**
0,75	**6,5 %**
1	*3,3 %*
1,25	*1,7 %*
1,5	*1,3 %*
2	*0,0 %*

6.3.2.3 ASHRAE 1272-RP

The results of system effect are averaged over the three test speeds, as in 6.3.2.2. Table 11 shows the influence of different types of inlet bearings on the SEF. The effect is noticeable with the cartridge bearings, while it is negligible with the pillow block bearings.

Table 11 — SEF due to bearings at the inlet of the FC centrifugal fan

Bearings	
Type of bearing	**Average $\Delta q_{v1}/q_{v1}$**
5 % cartridge	**5,1 %**
10 % cartridge	**5,1 %**
5 % pillowblock	*0,8 %*
10 % pillowblock	*1,9 %*

Table 12 presents the results obtained for the two cabinets of height 2D (cabinet 1) and 3D (cabinet 2). The SEF is very high for L = 0,25D and progressively decreases when L expands. The effect of the asymmetry of the bottom and top walls, observed in Table 10 with the BC centrifugal fan, is still observed here but only for L/D > 0,5. Once again, a minimum width of at least 1D is required to minimize the system effect.

Table 12 — SEF due to the inlet cabinets

Cabinet 1	
L/D	**Average $\Delta q_{v1}/q_{v1}$**
0,25	**30,2 %**
0,5	**14,8 %**
0,75	*4,4 %*
1	*2,0 %*
1,25	*2,0 %*
1,5	*1,1 %*
2	*0,2 %*

Cabinet 2	
L/D	**Average $\Delta q_{v1}/q_{v1}$**
0,25	**25,4 %**
0,5	**13,8 %**
0,75	**7,6 %**
1	*3,2 %*
1,25	*2,7 %*
1,5	*2,3 %*
2	*1,9 %*

6.3.2.4 ASHRAE 1420-TRP

Table 13 to Table 16 present the results obtained with all the inlet fittings and the two plenum fans of 381 mm and 686 mm. In all cases, the SEF are found as insignificant. Nevertheless, it is worthwhile to mention that for the return fan inlet cabinet (Table 16), tests on both impellers were impossible to carry on with the right side wall at 1D and 0,75D from the fan because of the very unstable flow conditions encountered in these fan inlet configurations. Tests of the 686 mm fan were possible only with the right wall at 0,25D from the fan entrance.

Table 13 — SEF due to inlet bearing and inlet box with or without cone

$\Delta q_{v1}/q_{v1}$	Inlet bearing	Inlet box w/o cone	Inlet box with cone
381 mm	*2,0 %*	*1,2 %*	*0,6 %*
686 mm	*1,0 %*	*0,6 %*	*0,6 %*

Table 14 — SEF due to an inlet cabinet with coil at difference distances from the fan

	Inlet cabinet + coil			
$\Delta q_{v1}/q_{v1}$	**Coil 1D**	**Coil 0,75D**	**Coil 0,5D**	**Coil 0,25D**
381 mm	*0,9 %*	*1,2 %*	*1,0 %*	*1,4 %*
686 mm	*0,3 %*	*0,2 %*	*0,6 %*	*2,2 %*

Table 15 — SEF due to an inlet cabinet with different side wall spacings

	Inlet cabinet + different side wall spacings				
$\Delta q_{v1}/q_{v1}$	box 1	box 2	box 3	box 4	box 5
381 mm	*0,7 %*	*1,1 %*	*0,6 %*	*0,6 %*	*1,0 %*
686 mm	*0,4 %*	*0,1 %*	*0,0 %*	*0,1 %*	*0,2 %*
$\Delta q_{v1}/q_{v1}$	box 6	box 7	box 8	box 9	box 10
381 mm	*0,6 %*	*0,9 %*	*1,1 %*	*0,6 %*	*1,2 %*
686 mm	*0,2 %*	*0,4 %*	*0,3 %*	*0,3 %*	*0,6 %*

Table 16 — SEF due to a return fan inlet cabinet with right side wall at two distances from the impeller and 45° baffle

	Return fan inlet cabinet		
$\Delta q_{v1}/q_{v1}$	Right wall 0,25D	Right wall 0,5D	Baffle 45 deg.
381 mm	*0,6 %*	*−0,1 %*	*1,9 %*
686 mm	*0,3 %*	—	*−0,1 %*

6.3.2.5 Elbow at the inlet of a forward-curved centrifugal fan

The pressure loss of the 90° elbow, which is taken into account in the calculation of the SEF, has been measured. The pressure loss coefficient deduced from the curve reproduced in Figure 24 is close to 1 (in SI units), which is high for a segmented bend. A coefficient of 0,32 was indeed obtained for a segmented bend in the NEL tests (Table 2).

Table 17 shows that the SEF due to the elbow is negligible or even beneficial with orientations 6, 7 and 8 ($\Delta q_{v1}/ q_{v1}$ negative).

Therefore, a segmented elbow directly fitted to the inlet of a FC centrifugal fan does not generate a noticeable system effect. This result confirms the trend obtained in the NEL tests on Fan 8 (see Table 8).

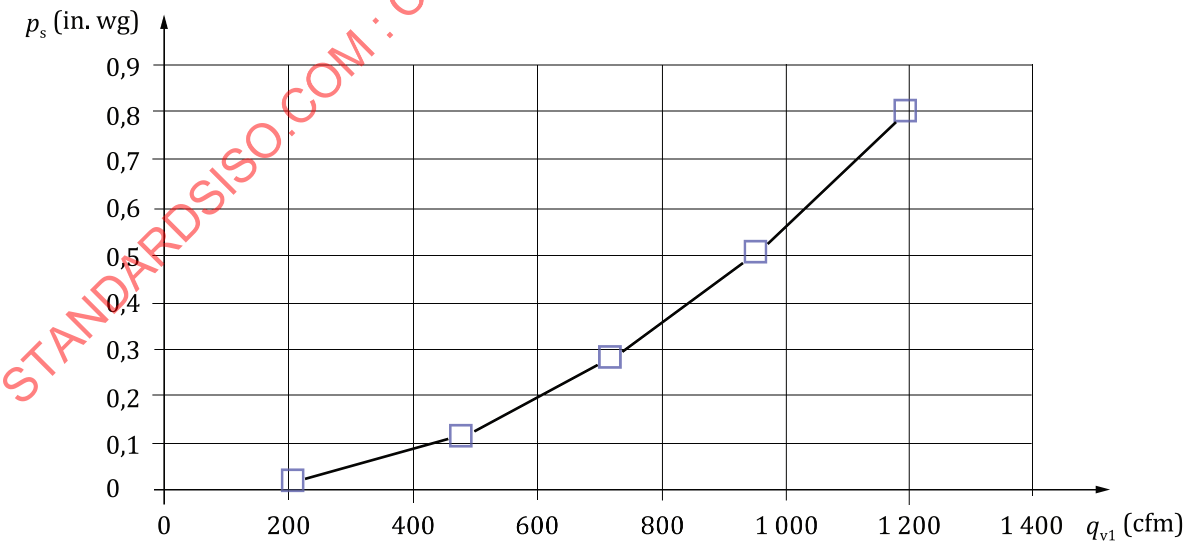


Figure 24 — Pressure loss curve of the 90° elbow

Table 17 — SEF due to the inlet elbow with different orientations

Elbow orientation	1 o'clock	2 o'clock	3 o'clock	4 o'clock	5 o'clock	6 o'clock
$\Delta q_{v1}/q_{v1}$	*−0,6 %*	*0,1 %*	*0,5 %*	*1,0 %*	*0,9 %*	*−2,6 %*

Table 17 *(continued)*

Elbow orientation	7 o'clock	8 o'clock	9 o'clock	10 o'clock	11 o'clock	12 o'clock
$\Delta q_{v1}/q_{v1}$	−3,5 %	−3,7 %	−1,9 %	−0,6 %	−0,4 %	0,1 %

7 Examples of outlet SEF

7.1 Axial fans

7.1.1 General

Examples of system effect of axial fans with outlet fittings are taken from NEL test data only.

7.1.2 Experimental setups

As in the tests with inlet fittings the diameter of the ductwork with outlet fittings is 630 mm. A layout of the NEL ductwork with a 90° bend connected to the fan outlet is shown in Figure 25. The distance between the inlet fitting and the fan is varied from 0D to 2D as previously.

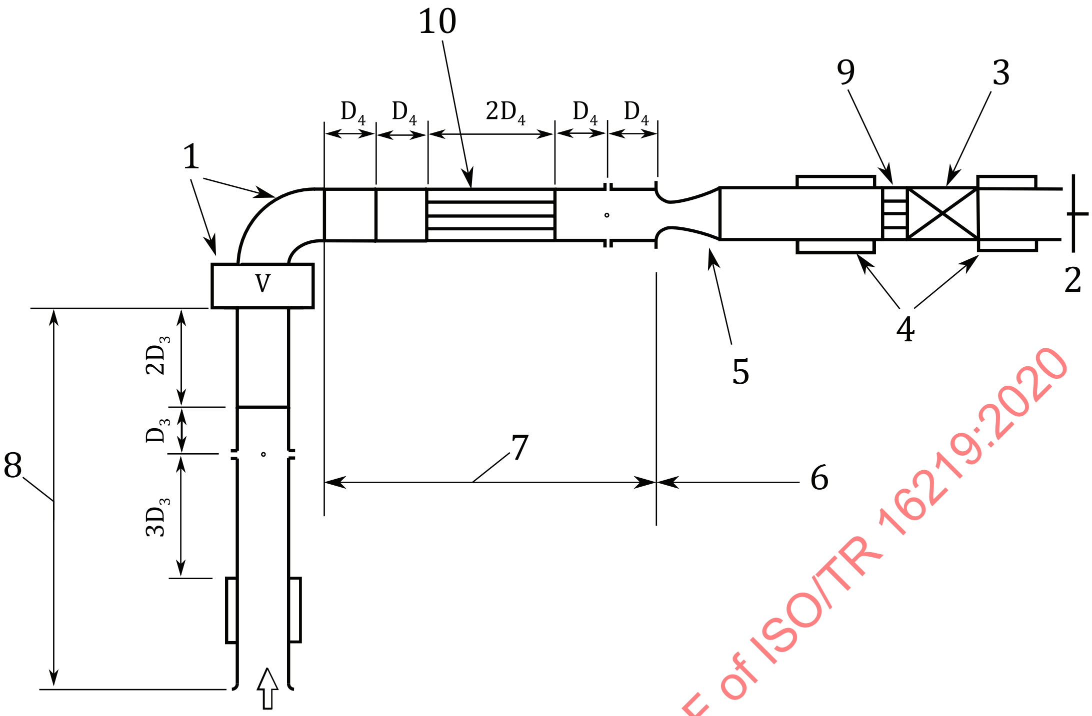
The test data used in the present analysis are the performance curves of the fan alone and fan + outlet fitting and the measured pressure losses of the fittings. All the fan curves are static pressure curves.

The main characteristics of the axial fans tested are shown in Table 1 and a description of the fittings is presented in 6.2.1.1.

7.1.3 Results

The outlet SEF (see 5.2) is presented in Table 18 for six axial fans and five outlet components. To assess the SEF the “conventional” pressure loss coefficients of these fittings have been measured by NEL (see Table 6).

Table 18 shows that for these configurations of fans and fittings the SEF is small or even negative in most cases (favourable effect of the component).



Key

1 test fan and fitting
2 throttle
3 auxiliary boost fan
4 silencers
5 flow measurement nozzle
6 flow measurement and control section
7 outlet duct
8 inlet duct
9 AMCA straightener
10 etoile flow straightener

SOURCE Based on content from National Engineering Laboratory (NEL) Fan Connected Ductwork Study for FETA (FET001) February 1992, reproduced with permission from the Fan Manufacturers Association, FETA UK.

Figure 25 — Test rig for determination of installation effect — Fitting at fan outlet

Table 18 — Outlet SEF

		Axial					
	L/D	**1**	**2**	**3**	**4**	**5**	**6**
Rect/circ transition (a)	**0**	***−0,06***	***−0,01***	***0,05***	***−0,01***	***−0,10***	***−0,39***
	1	***−0,08***	***−0,02***	***−0,05***	***0,03***	***0,07***	***−0,08***
	2	***−0,15***	***0,00***	***−0,12***	***0,08***	***0,20***	***−0,02***
Short square bend (b)	**0**	***−0,17***	***−0,12***	***−0,20***	***−0,03***	***−0,98***	***−1,48***
	1	***−0,12***	***−0,10***	***−0,32***	***−0,13***	***−0,46***	***−0,70***
	2		***−0,01***	***−0,39***	***−0,13***	***−0,32***	***−0,46***
NOTE Blank cells represent no data.							

Table 18 *(continued)*

		Axial					
	L/D	**1**	**2**	**3**	**4**	**5**	**6**
Square mitred bend (c)	**0**	*−0,01*	*0,19*	*0,44*	*0,16*	*−0,18*	*−0,35*
	1	*−0,02*	*0,18*	*−0,01*	*0,05*	*0,26*	*−0,02*
	2	*0,04*	*0,24*	*−0,02*	*−0,05*	*0,29*	*0,14*
Segmented bend (d)	**0**	*−0,32*	*−0,23*	*−0,46*	*−0,25*	*−0,45*	*−0,80*
	1	*−0,32*	*−0,18*	*−0,31*	*−0,28*	*−0,02*	*−0,62*
	2	*−0,23*	*−0,12*	*−0,25*	*−0,22*	*0,23*	*−0,07*
Rect splitter silencer (f)	**0**					*−1,15*	*−1,66*
	1					*−0,92*	*−2,25*
	2					*−0,78*	*−1,21*
NOTE Blank cells represent no data.							

7.2 Centrifugal and mixed-flow fans

7.2.1 Experimental setups

7.2.1.1 NEL

Table 5 gives the main characteristics of the centrifugal and mixed-flow fans tested by NEL. The fittings tested are described in 6.2.1.1 while their pressure loss coefficients are shown in Table 6.

7.2.1.2 ASHRAE 1420-RP

The two centrifugal plenum fans tested are described in 6.3.1.4. The outlet fittings consisted in 10 configurations of outlet cabinet of 2D depth with the right, top and left walls at four locations ranging from 1D to 0,25D (Table 19). Views of one of the configurations are shown in Figure 26.

Table 19 — Configurations of outlet cabinet with various side wall spacings

	Right	**Top**	**Left**
Box 1	1D	1D	1D
Box 2	0,75D	1D	1D
Box 3	0,75D	0,75D	1D
Box 4	0,75D	0,75D	0,75D
Box 5	0,5D	1D	1D
Box 6	0,5D	0,5D	1D
Box 7	0,5D	0,5D	0,5D
Box 8	0,25D	1D	1D
Box 9	0,25D	0,25D	1D
Box 10	0,25D	0,25D	0,25D

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Figure 26 — Outlet cabinet with side wall spacing 1D right, 1D top and 1D left

7.2.2 Results

7.2.2.1 NEL

Table 20 presents the outlet SEF for all the configurations tested with the mixed-flow fan and the four centrifugal fans. The SEF remains weak as on axial fans (see 7.1.3). The strongest effect is observed with fan 7 (mixed-flow fan) and, to a lesser extent, fan 9 (BC centrifugal fan with single inlet).

Table 20 — SEF due to outlet fittings

		Mixed flow and centrifugal				
	L/D	7	8	9	10	11
Rect/circ transition (a)	0	*0,21*		*0,06*	*−0,28*	*−0,10*
	1	*0,53*	*−0,44*	*0,05*	*−0,29*	*−0,08*
	2	*0,14*	*−0,47*	*0,21*	*−0,24*	*−0,02*
Short square bend (b)	0	*0,41*	*−0,30*	*0,10*	*−0,38*	*−0,06*
	1	*0,21*	*−0,40*	*−0,09*	*−0,36*	*−0,25*
	2	*0,12*	*−0,46*	*0,00*	*−0,45*	*−0,21*
Square mitred bend (c)	0	*0,60*	*−0,13*	*0,37*	*−0,24*	*−0,11*
	1	*0,35*	*−0,15*	*0,23*	*−0,21*	*0,02*
	2	*0,41*	*−0,14*	*0,32*	*−0,24*	*−0,01*
Segmented bend (d)	0	*−0,03*	*−0,39*	*−0,01*	*−0,05*	*−0,29*
	1	*0,34*	*−0,39*	*0,09*	*−0,07*	*−0,14*
	2	*0,55*	*−0,46*	*−0,04*	*−0,06*	*−0,15*
Rect/Rect box (e)	0		*−0,44*	*0,21*	*−0,34*	*−0,35*
	1		*−0,39*	*0,15*	*−0,34*	*−0,34*
	2		*−0,50*	*0,28*	*−0,33*	*−0,24*
Rect splitter silencer (f)	0	*0,03*				*−0,14*
	1	*−0,04*				*−0,21*
	2	*0,41*				*−0,20*
NOTE Blank cells represent no data.						

7.2.2.2 ASHRAE 1420-RP

Table 21 shows the outlet SEF of the 10 outlet cabinets with different side wall spacings (Table 19) and two impellers of diameters 381 mm and 686 mm. The system effect coefficient C used to assess the outlet SEF has been calculated without accounting for the conventional pressure loss as this parameter is not relevant in this case. As for the inlet SEF, grey or black cells have been used in the table when the system effect is medium (light grey cell if 1 ≤ SEF ≤ 3) or large (dark grey cell with SEF > 3). The system effect is considered as small when SEF < 1 (white cell). The general trend is a logical increase of SEF when the side walls are closer to the impeller, with a maximum effect obtained with the three side walls at 0,25D from the impeller (box 10). A dissymmetry of the cabinet walls with respect to the fan axis has no noticeable influence on the system effect.

Table 21 — SEF due to outlet cabinet with different side wall spacings

	Outlet cabinet with different side wall spacings									
	Box 1	**Box 2**	**Box 3**	**Box 4**	**Box 5**	**Box 6**	**Box 7**	**Box 8**	**Box 9**	**Box 10**
381 mm	*0,47*	*0,46*	*0,61*	*0,59*	*0,35*	*0,67*	**1,02**	*0,46*	**1,14**	**3,53**
686 mm	*0,40*	*0,47*	*0,61*	*0,63*	*0,42*	*0,72*	**0,92**	*0,59*	**1,14**	**3,32**

8 Reducing system effects

8.1 General

System effects, present in some fan installations, have a detrimental impact on fan performance, resulting in increased noise and vibration, as well as reduced flow and pressure. These system effects, as described previously, are initiated through a number of channels, including improper installation, inadequate system design, after design changes to equipment, as well as in situ fan performance “upgrades”.

Whatever the reason for the system effect, there are a number of practical steps which can be taken to mitigate the impact.

8.2 Inlet effects

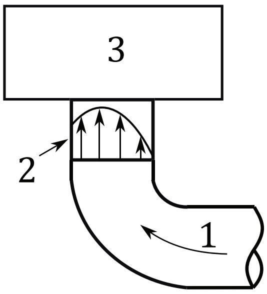
8.2.1 General

Proper flow into the fan is critical to optimizing the installed fan’s efficiency. Ideally, the flow into the fan will be via a straight duct, well matched to the fan’s inlet connection. In practice, that is frequently not the case. Inlet effects can be divided into three (3) general categories:

1) Non-uniform flow

2) Swirl or vorticity

3) Inlet blockage



Key

1 inlet duct

2 non-uniform velocity profile

3 fan

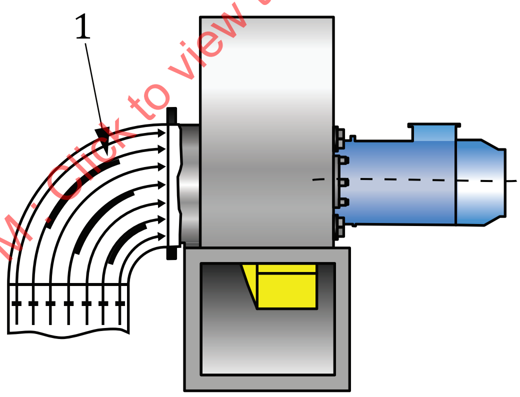
Figure 27 — Upstream elbow

8.2.2 Non-uniform flow

Elbows immediately upstream to the fan inlet need to be avoided if possible (see Figure 27). If this is not possible then the following strategies can minimize the effect:

— Large radius: minimize the separation and inertia effects of the flow. This can be further improved through the addition of turning vanes.

— Include turning vanes: to guide the air and minimize turbulent eddies at the fan inlet. See Figure 28.

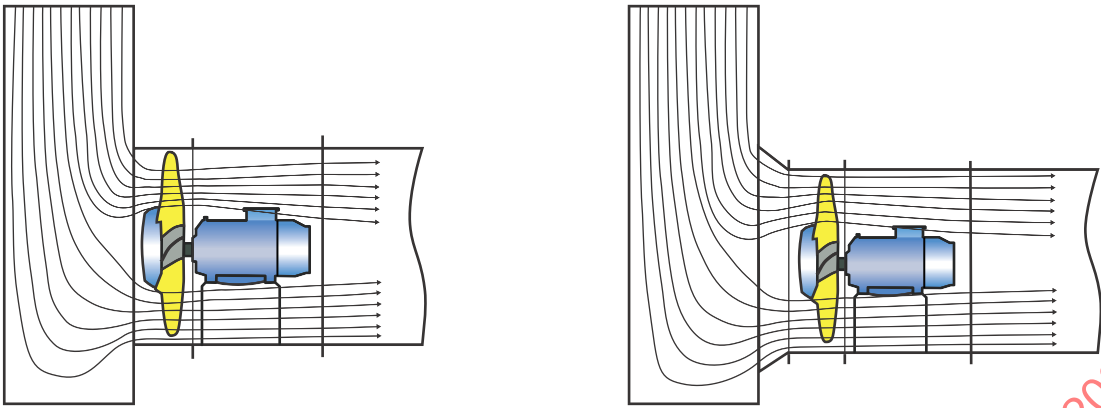


Key

1 cascade turning vanes fitted

Figure 28 — Addition of turning vanes in an upstream elbow

— Sharp entrance to fan inlet: avoid and replace with a generous inlet bell to aid in guiding the flow through the bend and accelerate into the fan inlet. Often, fan manufacturers have designed fan “inlet boxes” with a known loss which can add certainty to the selection process. See Figure 29.

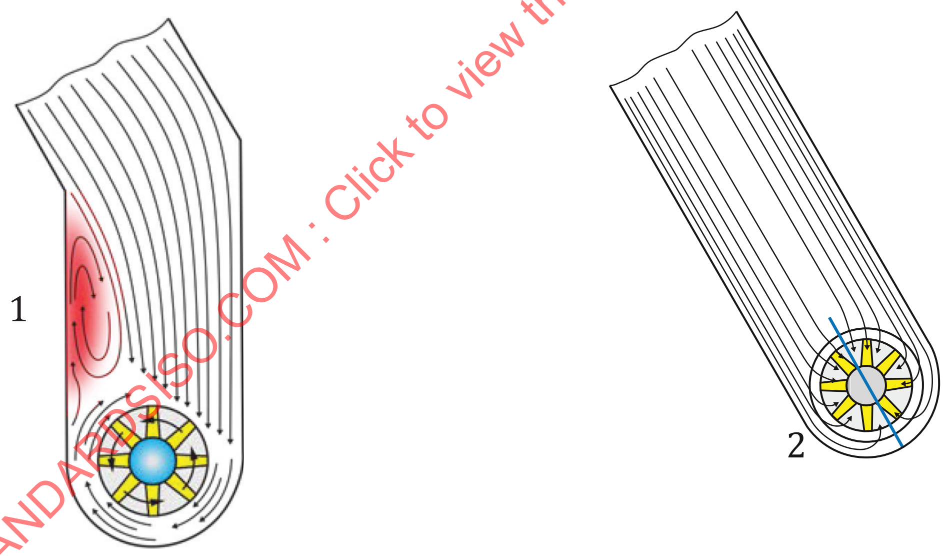


a) Sharp entrance to fan inlet   b) Flare fitted to improve entry

Figure 29 — Examples of fan inlet entrances to a fan inlet

8.2.3 Swirl or vorticity

Bends upstream of the fan inlet box can direct a majority of the flow to one side or the other of the fan inlet box and can create a swirl which is either contrary to or in conjunction with the fan impeller rotation. This can affect the fan performance with unpredictable effects. Either realign the fan inlet box, if at the early design stages, or fit the box with a “splitter plate” which aids in breaking up the swirl in the inlet box. See Figure 30.



a) Effect of upstream bend   b) Fan with splitter plate 1

Key

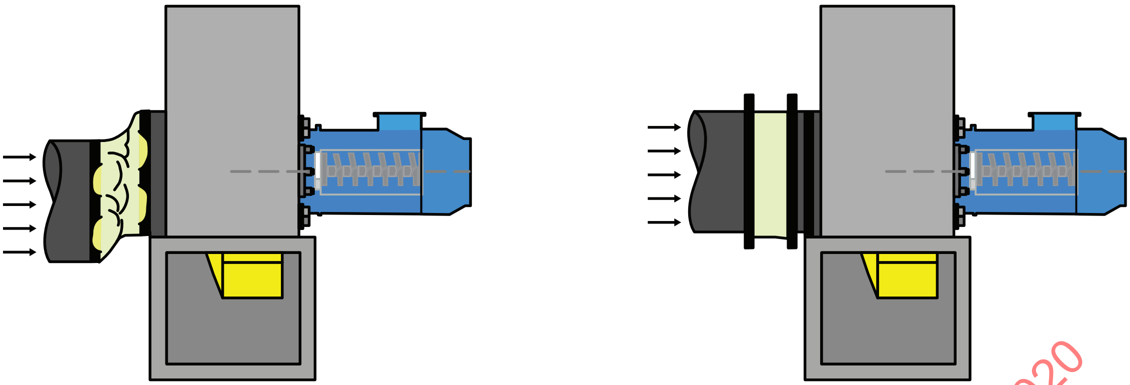
1 swirl can either be with or against the fan impeller rotation

2 central splitter fitted to eliminate swirl

Figure 30 — Upstream bends

8.2.4 Inlet blockage

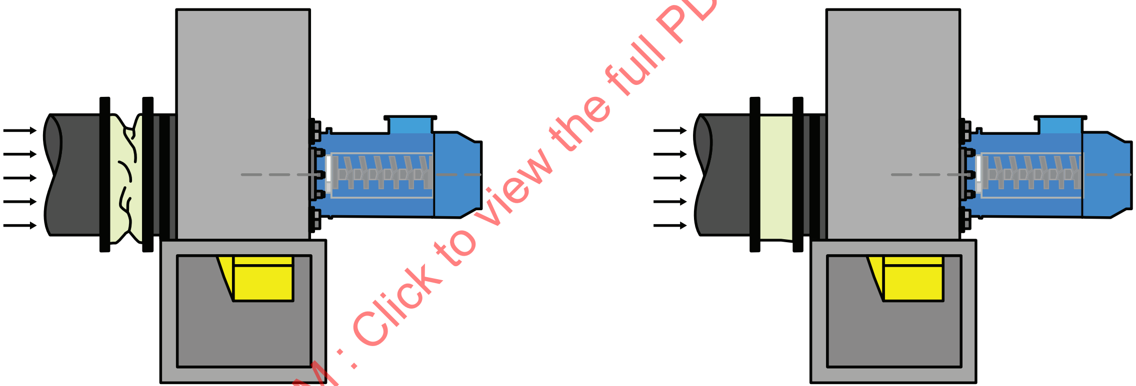
A straight duct, poorly aligned with the inlet, can result in an effective blockage of part of the inlet due to the abrupt change in the duct flow (Figure 31).



a) Bad b) Good

Figure 31 — Inlet blockage example

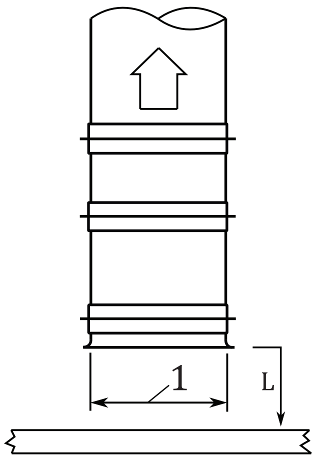
In inlet flexible connection, if not supported with an internal flow guide, can have the internal diameter severely reduced due to the flexibility of the fabric being drawn in due to the high suction pressure of the fan. In addition, for high pressure blowers with relatively small inlets, the flex connection flow liner must be carefully designed so as not to unduly restrict the flow area (Figure 32).



a) Bad b) Good

Figure 32 — Inlet blockage due to flex connection

For open inlet fans, obstructions near the fan inlet can have detrimental effects. It is recommended that the distance between the inlet plane and the obstruction be increased as much as practical (dimension L in Figure 33).

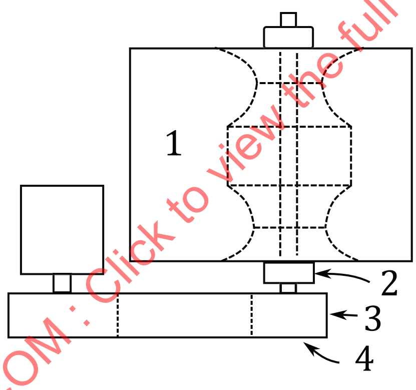


Key

1 diameter of inlet

Figure 33 — Open inlet obstruction avoid

Belt drives and guards in the inlet are not ideal (see Figure 34). Perhaps increased guarding to prevent any access to the fan inlet area located suitably far away would eliminate the need for a belt guard. The guard ought to be as porous as possible (maximize free area in the direction of airflow).



Key

1 fan

2 bearing

3 belts

4 pulley

Figure 34 — Belt obstruction

Bearings in the inlet ought to be mounted on plate pedestals which are open in the axial direction, to allow as much unobstructed flow as possible.

Figure 35 shows an installation with a damper close coupled to an axial fan inlet. In performance terms a static pressure drop for the damper was assumed but no influence of the damper blades on the stability the axial impeller was accounted for. When installed the performance was measured at 80 % of design and an impeller failure was reported. Remedial action was taken to resolve all performance issues.

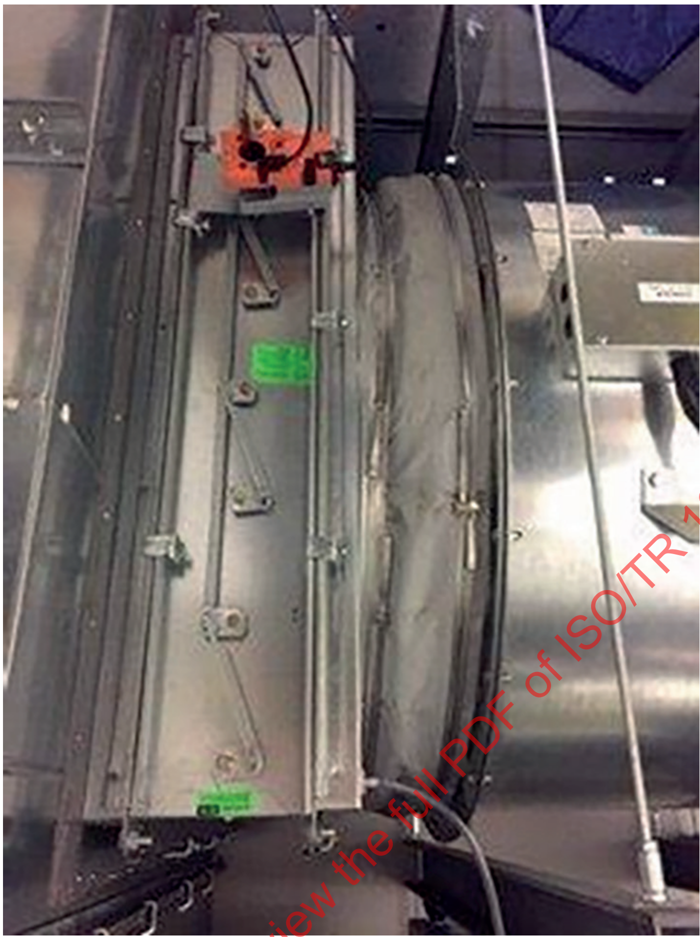


Figure 35 — Close coupled damper 1

8.3 Outlet effects

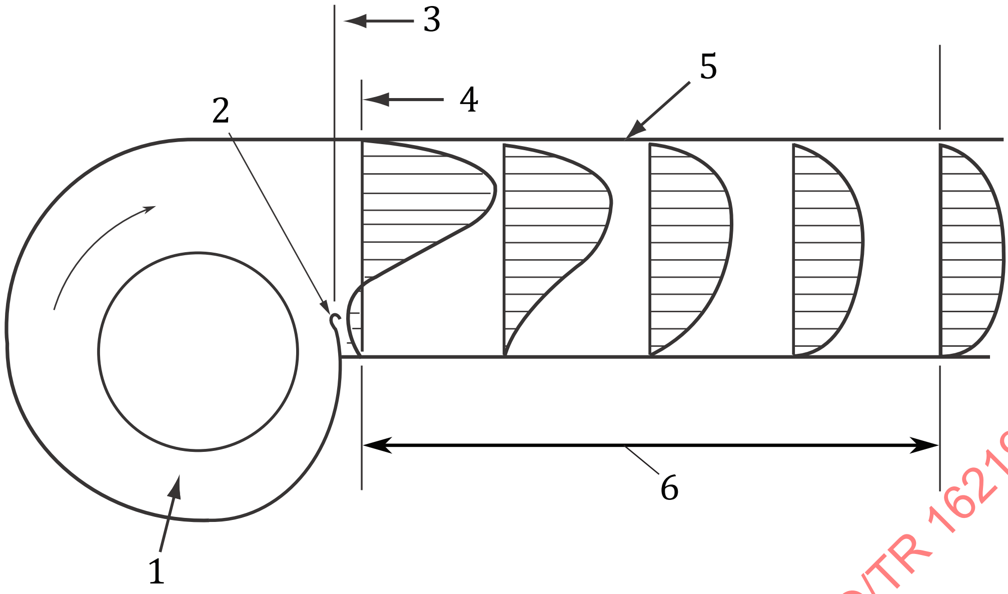
8.3.1 General

Outlet system effects can be broadly categorized as:

1) Insufficient duct length
2) Outlet obstruction
3) Non-uniform flow

8.3.2 Insufficient duct length

Many fans discharge to atmosphere and it is important that the flow be allowed to develop uniformity before discharge to minimize system effects, as shown in Figure 36.



Key

1 centrifugal fan
2 cutoff
3 blast area
4 outlet area
5 discharge duct
6 100 % effective duct length

Figure 36 — Fan outlet velocity profile

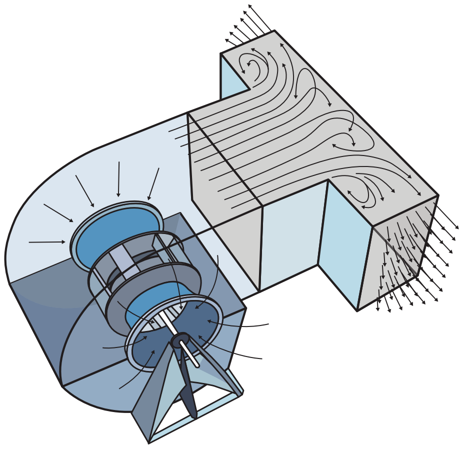
Exhaust fans often have some sort of “weatherhood” to prevent rain from entering the fan. These need to be carefully designed to not cause an obstruction to the flow.

8.3.3 Outlet obstruction

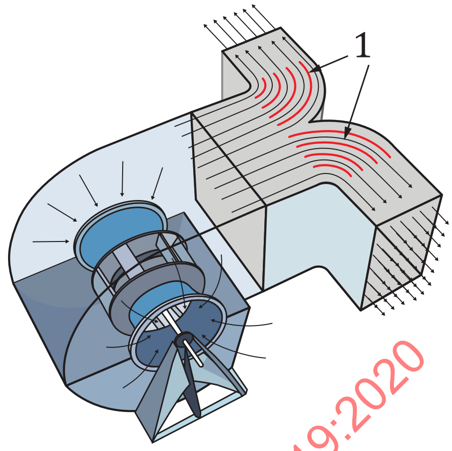
Remove any obvious obstructions or re-orient the fan outlet to avoid those obstructions.

8.3.4 Non-uniform flow

Elbows at the outlet of a fan can have a significant impact, by creating non-uniform flow (see Figures 37, 38 and 39). Adding turning vanes can help in centrifugal fans when the elbow turns the flow in a direction perpendicular to the fan shaft. Adding turning vanes to elbows which direct the flow parallel to the fan shaft can have a decidedly negative effect if the elbow is too close to the fan outlet and the flow has not fully developed.



a) Outlet elbows

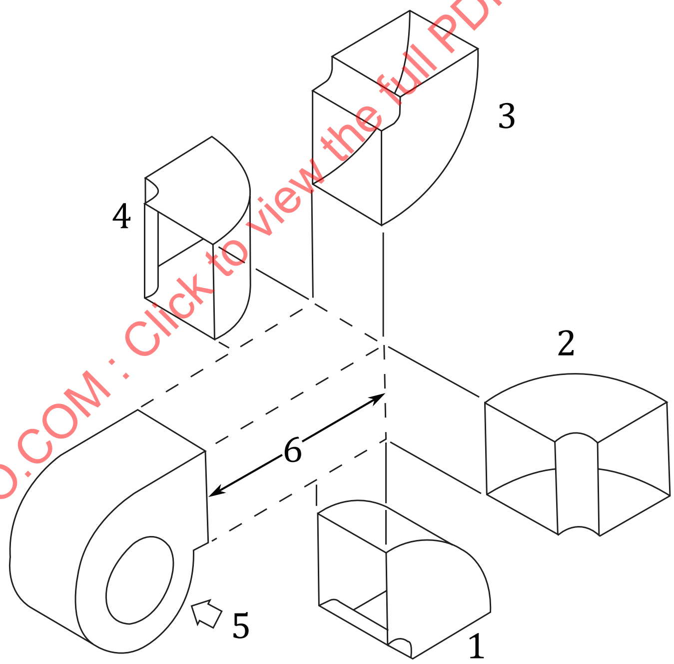


b) Outlet elbow with splitter

Key

1 splitters added to bends

Figure 37 — Outlet elbows



Key

1 position A

2 position B

3 position C

4 position D

5 inlet

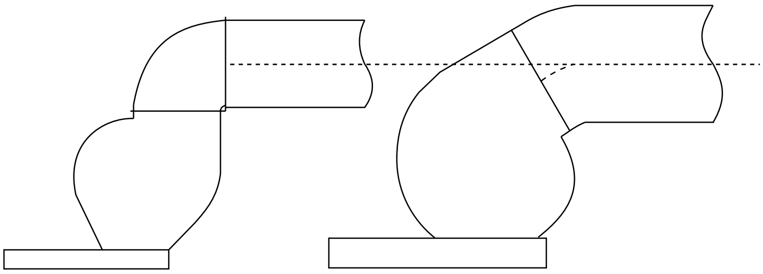
6 % effective duct length

Figure 38 — Outlet elbows

Figure 39 — Poor outlet elbow installation

If none of the above strategies offer sufficient improvement, then it may be possible to improve system performance through a more detailed computational fluid dynamics (CFD) study of the fan and associated ductwork. Perhaps a number of idealized ductwork changes can significantly reduce the demands on the fan to result in improved performance.

Occasionally, an old fan is repurposed in a facility. The fan may be capable of the required performance but is not configured correctly. Due to various pressures, this "fit" issue is ignored and the fan placed and interconnect ducting fitted. Providing a completely new fan, designed to better fit into the system and optimized for the present day operating conditions, would, in most cases, be a better alternative. In the example shown in Figure 40, a smaller fan of CCW rotation and upblast discharge is replaced with a larger fan with CW rotation and top angular up discharge to provide a better fit to the outlet duct configuration with reduced system effects.



Repurposed fan

(poorly configured)

BI 245

Original performance

10,000 cfm, 10 in. wg

η = 67 %

BI 245

Repurposed

20 000 cfm, 10 in. wg.

η = 56 %

Replaced fan

(reduced system effects)

BI 300

Current performance

20 000 cfm, 10 in. wg

η = 80 %

Figure 40 — Fan replacement of a poorly configured fan

8.4 Examples of the effects of poor inlet and outlet connections

Figure 41 shows an example of a fan inlet and outlet where, due to duct sizes, there is a contraction on the fan inlet and expansion on the fan outlet. No account was taken of this when specifying the fan which led to a shortfall in air performance at the commissioning stage. Some modifications were made to the duct layout so that the fan performance was within the tolerance allowed by the contract.

Figure 41 — Example of poor inlet/outlet conditions

9 Conclusions

The objective of this document is to present the main causes and quantification of the system effect (SE) induced by a fitting, appurtenance or obstacle close to the inlet or outlet of a fan. The physical mechanism differs according to the location of the flow disturber (fitting) with respect to the fan: when it is at the inlet the fan curve is modified whereas when it is at the outlet the fan curve is usually not affected but the system pressure drop may increase due to the presence of the appurtenance. That is the reason why two different definitions of the SEF have been proposed, one for the inlet side and the other for the outlet side. In both cases the pressure drop of the obstacle has to be taken into account if relevant in the evaluation of the SEF.

The report presents the results of several dedicated series of tests to assess the SEF of axial and centrifugal fans, which show that in most cases the SEF is not much important except in some special situations such as a wall in front and very close to the inlet of an axial fan, a cabinet with a lateral opening of small cross-section at the inlet of a BC or FC centrifugal fan, or an outlet plenum with side walls close to the impeller of a plenum fan.

A summary of recommendations to reduce the fan SE are then provided with a few examples of very bad fan installations that need be absolutely avoided.

Annexes A and B present basic principles of fan performance representation and fan system calculations, respectively, to help readers who are not familiar with these notions to fully understand the content of this document.

This document only deals with the system effect on the fan air performance curve. As a next step it would be worthwhile to draft a document dealing with the SE on fan power and noise level.

Annex A
(informative)

Basic principles on fan performance representation

A.1 Fan performance curves

Each type and size of fan has different performance, which is normally determined by the proper use of fan tests under laboratory conditions.

These tests are conducted in accordance with appropriate international standards (e.g. ISO 5801 or ISO 5802).

These standards define the equipment, the installations and the procedures required to provide truly reliable and consistent measurements of fan air performance and sound power levels, under standardised laboratory conditions.

An essential part of these standardised conditions is the adoption of one of the five standardised test installations, A, B, C, D or (when applicable) E, as defined in ISO 13349.

It is worth noting that the test set-ups, as specified by these standards, provide nearly ideal flow conditions, and standardised shapes of the ducts or spaces, both at the fan inlet and outlet connections. These conditions are seldom completely met in practice but are essential to provide consistent and repeatable results.

Any further deviation in performance, due to deviations in shape and design of an actual air system from the standardised geometries, or to the effect of any distorted airflow, on the fan air performance and noise, is normally accounted as “system effect”.

Whatever the standard and the installation selected for a specific set of tests, the results are summarized into a set of fan performance curves, or fan characteristic curves. These may vary in form, according to the fan type and the preferences of the fan manufacturer or of the intended market.

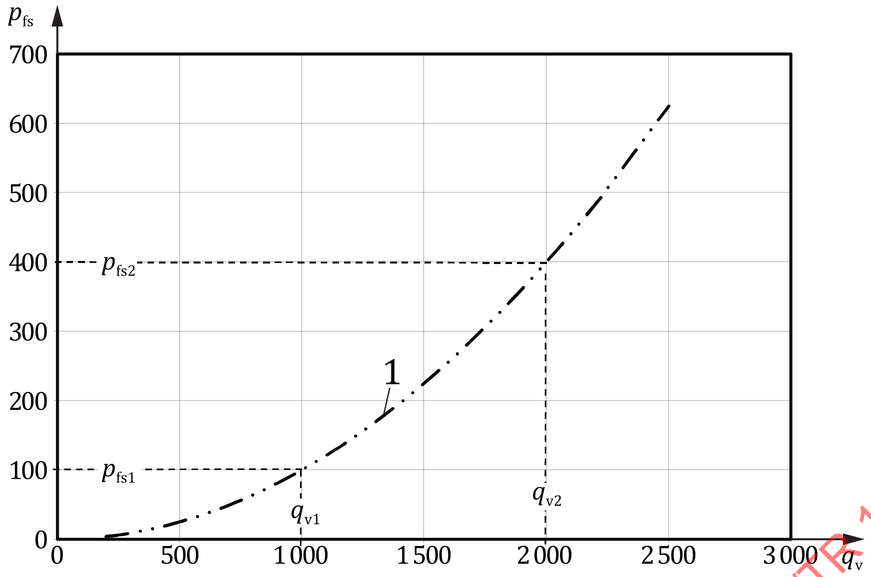
Guidance on the layout and presentation of fan performance can be found in ISO 13348:2007, Clause 8.

A.2 System resistance curves

System resistance is the total sum of all the pressure losses through filters, coils, dampers and ductwork attached to the fan. As a general rule, it is never a constant, but rather an increasing function of the volume flow, and thus airspeed, along the ductwork.

The system resistance curve (Figure A.1) is simply a plot of the pressure that is required to move the air through the system.

For fixed systems, that is with no changes in damper settings, system resistance can often be assumed to be reasonably proportional to the square of the volume flow (q_V). The resistance curve for such a “constant orifice system” is represented by a single parabolic curve.



Key

q_{v1} volume flow rate q_v of the first duty point

p_{fs1} fan static pressure of the first duty point

q_{v2} volume flow rate q_v of the second duty point, along the system resistance curve, with twice the flow rate of the first

p_{fs2} fan static pressure of the second duty point, along the system resistance curve, having four times the static pressure at the first duty point

1 parabolic system-resistance curve

$$\frac{p_{fs2}}{p_{fs1}} = \left(\frac{q_{v2}}{q_{v1}}\right)^2 = \left(\frac{2\,000}{1\,000}\right)^2 = \frac{4}{1}$$

Figure A.1 — System resistance curve

This assumption is strictly correct only in the case when the pressure loss is produced by the dissipation of air-jet kinetic energy, or by surface friction in shear layers under turbulent conditions, without uncontrolled flow separations.

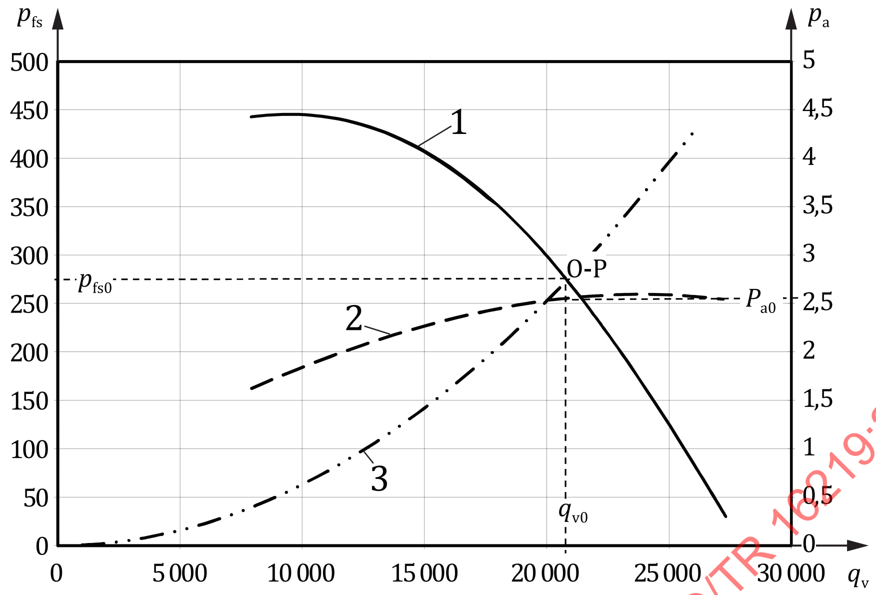
Laminar flow on solid surfaces or through the narrow passages of filters and heat-exchangers, hydrostatic pressure, and flow separations can introduce a component of pressure loss which is not proportional to the square of the average flow speed, but rather linear, constant or unregularly stepping. For limited variations of the fan flow rate, anyway, the use of a square-law pressure loss function is often acceptable.

As an example, consider a system handling 1 000 m^3/h with a total resistance of 100 Pa (static pressure, p_{fs}). If the volume flow q_V is doubled to 2 000 m^3/h, the p_{fs} resistance will increase to 400 Pa, as shown by the squared value of the ratio given in Figure A.1.

The system resistance curve changes, however, as filters are clogged with dirt, coils start condensing moisture or when the position of dampers is changed.

A.3 Operating point

The actual operating point (Figure A.2) of each fan and system combination is determined by the intersection of the system resistance curve with the fan performance curve.



Key

O-P operating point or duty point

q_{V0} fan volume flow rate at the operating point

p_{fs0} fan static pressure at the operating point

P_{a0} fan impeller power at the operating point

1 fan characteristic curve, or static pressure curve, as a function of volume flow rate q_V

2 fan impeller power (or alternatively, *Pa*, fan shaft power), as a function of volume flow rate q_V

3 parabolic system resistance curve or "system line"

Figure A.2 — Operating points

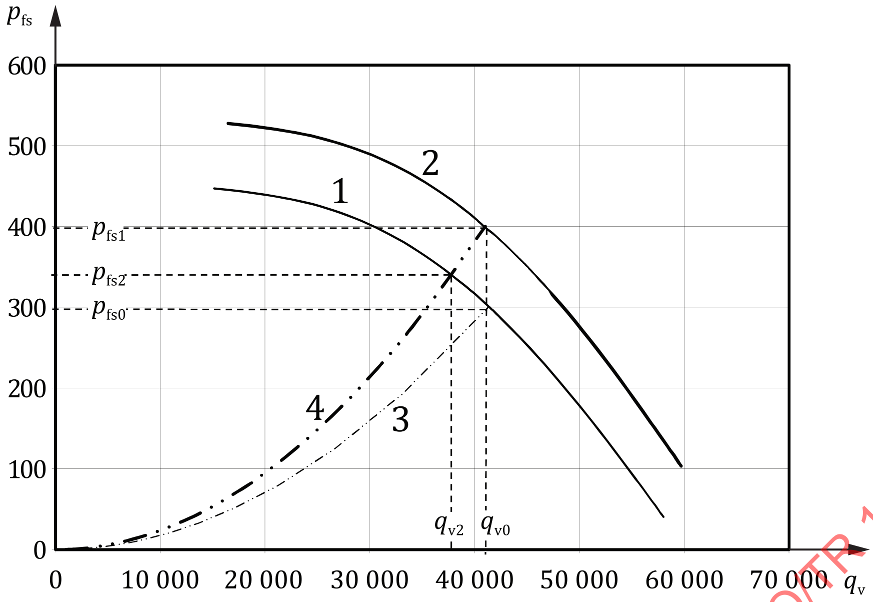
The resulting operating point is the point where the pressure required by the system equals that delivered by the fan at the same volume flow rate.

If the system resistance curve of the system, as actually built, is different from the calculated curve, then the operating point will change and the static pressure and volume flow delivered by the fan will be different from those predicted at system design stage.

In Figure A.3, the actual system has a higher pressure loss than predicted. As a consequence, the air volume is reduced, while, in this case, the static pressure is increased. The shape of the typical power curve of an FC fan would also result in a reduced power requirement.

To compensate for the system performance change, the fan speed would frequently be increased and a higher power would be needed to achieve the desired q_V. Such a move would also increase the noise produced by the fan over the level originally expected, while care needs to be used not to exceed the power and speed safety limits, as provided by the fan manufacturer.

Quite frequently, when there is a difference between actual and calculated fan output, this is due to an unexpected change in system resistance, rather than to any shortcomings of the fan or motor.



Key

p_{fs0} fan static pressure at the originally specified duty point
p_{fs1} fan static pressure at the specified volume after increase of the pressure loss
p_{fs2} fan static pressure at the new duty point, along the fan static-pressure curve at original speed
q_{V0} fan volume flow rate, originally specified
q_{V2} fan volume flow rate at specified speed, after increase of pressure loss
1 fan characteristic curve, at the fan operating speed chosen according to the design operating point
2 fan static pressure curve at the increased speed, necessary to reinstate the specified volume flow rate
3 parabolic system resistance curve through the originally specified duty point
4 parabolic system resistance curve through the new duty points, due to system pressure-loss increase

Figure A.3 — Variations from design system curve

Often, the mistake is made of taking a static pressure reading, across the fan, and concluding that, if the fan pressure is at or above the design requirements, then the volume flow q_V is also at or above design requirements. Figure A.3 demonstrates why such an assumption is wrong, whenever the system pressure loss is different from the expected level.

A.4 Fan performance measurement vs. calculation

Not each fan size, operating condition or duty point must necessarily be measured by testing.

Apart from the obvious interpolation between duty points, measured at different flow rates on the same fan and speed, under certain restrictions the performance of a fan may be calculated from a test carried out on the same fan, but under different conditions. These may include running the test at a different speed, or when running at a different air pressure or temperature, or even when handling atmospheric air (which is normally used in fans during laboratory measurements) while the fan to be rated is specified for use with a gas having different physical properties.

Under further restrictions, the performance of a fan may even be calculated from a test carried out on a different fan sample, being geometrically similar to the original fan, but having a different size.

These calculations are carried out using a set of formulae derived from the classic theory of the turbo-machinery with inviscid flow, and which are generally known as “fan laws”.

A.5 Fan performance calculations: the fan laws vs. interpolation procedures

The following clauses provide a simple description of the incompressible version of the fan laws and of some of their most common applications.

These formulae relate the performance of two fans which are identical or at least geometrically similar, being operated in respective operating points being cinematically similar, and under conditions on the respective Reynolds numbers and Mach numbers.

Changes of fan size, operating speed and air density can thus be accounted for.

The case of the performance of the same fan at different speed and gas properties is a special case of the general one, also entailing size changes.

An extended version of the fan laws, including a correction for compressibility effects, may be used with less restrictive limits to the chance of Reynolds number and compressibility coefficients, and can be found in ISO 5801:2017, 15.2.1.

Another method of calculating the performance of a fan (converted to non-dimensional parameters), by interpolating the performance measured on a limited number of samples, belonging to a series either being geometrically similar, or in any kind of regular geometrical relationship, is provided in ISO 13348:2007, 7.1.6.

This method requires considerably more calculations but has the advantage of being better able to describe the effect on the fan performance of the changing influence of the flow viscosity, of the Mach number and of non-similar but regular changes of dimensions in a fan range. On the other side, it may not allow reliable extrapolations outside the tested range of sizes, speeds and Reynolds or Mach numbers.

When interpolation is not used, the performance of a fan having a larger impeller can then be calculated from the measurement of a similar fan having a smaller diameter, but the performance of smaller fans must not be calculated from measurements achieved on larger-sized fans, as this may easily lead to an overestimate of performance and efficiency.

Both the fan laws and the interpolation procedures provided in ISO 13348 describe the change in fan performance referred to the power delivered to the impeller but cannot predict the effect of a change of motor type, size or speed in the drive system, and the consequent change of efficiency of the drive-system itself.

The effect of any change of the drive system (or of its operating condition), between the original fan model and its operating condition, and the fan model actually tested and its operating condition, cannot be determined with these methods alone, and must be determined by comparison of measured data, specific to the performance and behaviour of the parts of the drive system itself (e.g. motor, transmission, VSD).

Having said this, the computation instrument which is most frequently used to properly predict fan performance is the set of formulae commonly identified as fan laws, in their incompressible form.

Fan manufacturers frequently use the fan laws in the calculation and prediction of the performance of any type of fan, and particularly in the development of the catalogue ratings for a range of geometrically similar fans.

It is impractical to test the performance of every single size of fan, in a range, at every possible speed and duty point. Nor is it possible to simulate every possible inlet density which may be encountered.

On the other side, by use of the fan laws, it is possible to predict, with good accuracy, the performance of a fan at speeds and densities other than those of the original rating test.

Fan users are more frequently interested in the use of the fan laws for the selection of the most appropriate fan size and speed, to meet a specified performance requirement. These laws also allow the prediction of the effects of some changes of fan operating conditions, specifically of the fan speed or of the density of the gas being handled.

A.6 The fan laws

A.6.1 Definitions

The following definitions apply when using the fan laws.

a) Geometrically similar fans

Two fans are geometrically similar when all the corresponding lengths in the two fans are in the same ratio, and all the corresponding angles are equal.

NOTE Most frequently, fans belonging to a homogeneous series of sizes are designed in such a way to be mutually similar.

b) Operating point

An operating point, for a fan, is an operating condition which, for a given standardised installation type, is unambiguously identified by a combination of gas density at inlet, fan inlet pressure, fan pressure and volume flow rate.

c) Cinematically similar operating points

Two operating points, either for the same fan or for two different but geometrically similar fans, are cinematically similar when the angles formed, in corresponding physical points, by the local flow velocities with the stationary and rotating parts of the fans, are the same.

A.6.2 The formulae of the incompressible fan laws

From the analysis of the physical principles which govern the performance of a fan, it is possible to derive a number of formulae, commonly known as "fan laws".

These formulae relate together the performance in two different, cinematically similar operating points of the same fan, or even of two different fans, provided that the two fans are geometrically similar.

These formulae apply to cases in which the pressure rise is less than 2,0 kPa.

The four fan laws in their simpler general form are as follows:

$$q_{V2} = q_{V1} \times \left(\frac{N_2}{N_1}\right) \times \left(\frac{D_2}{D_1}\right)^3 \tag{1)}$$

$$p_{f2} = p_{f1} \times \left(\frac{N_2}{N_1}\right)^2 \times \left(\frac{D_2}{D_1}\right)^2 \times \left(\frac{d_2}{d_1}\right) \tag{2)}$$

$$P_{a2} = P_{a1} \times \left(\frac{N_2}{N_1}\right)^3 \times \left(\frac{D_2}{D_1}\right)^5 \times \left(\frac{d_2}{d_1}\right) \tag{3)}$$

$$\eta_2 = \eta_1 \tag{4)}$$

where

q_V is flow rate (m^3/s);

p_f is pressure (total, static or dynamic) (Pa);

d is gas density (sometimes the Greek letter ρ, "ro", is used instead) (kg/m^3);

N is fan speed (rpm);

D is impeller diameter (mm);

P_a is impeller power (kW);

η is impeller efficiency (fraction of unity).

The most commonly used SI units have been listed beside each dimension. Anyway, the fan laws contain only ratios between the values of these dimensions in the two related operating points, so the units actually used are not essential, provided that the same units (e.g. SI units) are used for both operating points.

It is also worth noting that these laws relate together the fan performance only between cinematically similar operating points of the same fan or of geometrically similar fans. They cannot be used to relate together different points on the characteristic curve of the same fan.

A.6.3 Limitations

Like many physical laws, the fan laws are a mathematical model, developed under some rather arbitrary assumptions, which are required to simplify the resulting formulae, but imply some limitations to their accuracy.

A well-known factor limiting the applicability of the fan laws is gas compressibility: the specific volume of any gas changes with pressure, which has an effect on the performance of any gas-compressing equipment, including fans. For example, if an unchanged mass flow of gas is passing through a fan, the exact volume flow entering the fan must be in some measure larger than that leaving the fan at a higher pressure. One of the main differences between fans and higher-pressure machines, like compressors, is that the pressure increase which they produce is small enough for compressibility effects to be either disregarded completely or treated with simplified formulae.

It is widely accepted that when the fan pressure is below 2 500 Pa any compressibility effect can be ignored without affecting the calculated results.

A set of modified fan laws, including a "compressibility factor", can be adopted for use at higher pressures (see ISO 5801:2017, 15.2.1).

Both practice and calculations confirm, anyway, that the actual threshold, above which compressibility begins to produce effects of any practical significance, is considerably higher, and the simpler form of the fan laws, as described in this document, is commonly used up to 5 000 Pa without problems.

In practice, nearly all the fans used for HVAC&R applications run at pressures low enough to make absolutely safe the use the simpler (incompressible) form of the fan laws.

Two other physical factors which may affect the reliability of the performance predictions achieved using the fan laws are flow viscosity and geometrical deviations.

The ratio of viscous forces and mass forces in corresponding points of two different gas flows may be considerably different, even under conditions of geometrical similarity. This ratio between different kinds of forces acting on the fluid is related with a numerical parameter called the "Reynolds number" (Re), which is given by

$$\mathrm{Re} = \frac{V \times D}{\nu}$$

or, in a simplified form, valid only for fans operating with standard air:

$$\mathrm{Re}_{ND} = N \times D^2$$

where

V is fan peripheral speed (m/s);

N is impeller rotational speed (rpm);

D is impeller diameter (m);

ν is gas cinematic viscosity (m^2/s).

When the values of the Reynolds number are the same in two different operating points, these points are said to be "dynamically similar" to each other.

In practice, dynamic similarity is seldom achieved, and gas viscosity can alter in some way the performance of the fan from the values calculated assuming complete similarity.

The other factor affecting fan law reliability is related with any small deviations from true geometrical similarity: the fan laws require that equally sized fans are identical, and that differently sized fans are geometrically similar.

In practice different fan sizes of the same series frequently have some dimensions which are not truly similar: typically steel plate thickness, shaft diameters and, last but not least, surface roughness.

These deviations from true similarity also introduce a degree of deviation of the real performance of a fan from the predicted values.

The effects of both viscosity and geometrical deviations combine together into what is normally called "scale effect", or a widely demonstrated tendency of larger and faster fans to perform better and more efficiently than fans being similar but smaller, slower or both.

In practice these deviations are small and really significant only when the fan laws are used, by fan manufacturers, to develop very accurate catalogues. Here, appropriately extensive testing is required, to produce truly accurate ratings.

A last factor affecting the reliability of the fan laws is the effect of bearings on the total power requirement of the fan.

The power applied to the fan shaft is partly delivered to the impeller, and partly dissipated by friction in the bearings which support the fan shaft. The power which is governed by the fan laws is, strictly speaking, only the power used by the impeller or impeller power, as properly stated in A.6.2, and, between similar operating points, is proportional to the third power of the fan speed. The power lost because of friction inside the bearings (and heating up those bearings) has a different relationship with speed. In many cases, the power lost in the bearings is proportional to the first power of the fan speed. This is particularly true when most of the friction is coming from any bearing contact-seals, preventing contamination of the bearings and of their lubricating grease.

As an example, when the speed of a fan is reduced by 50 %, the impeller power is reduced by a factor of eight, but the bearing power is reduced only by a half, and its relative weight against impeller power is increased by a factor of four.

In most practical cases, the effect of not calculating separately and correctly the power loss in the bearings may be entirely negligible.

Care should be taken on the smaller, high speed fans (typically very small, backward curved fans running at a few thousand rpm), where the relative importance of the bearings in the overall power requirement may be significant.

Another case which must be treated with care is that of the larger fans, when their speed is reduced considerably, to operate at unusually low pressure, with a power requirement lower than, typically, a few hundred watts.

Except for this final warning, which applies only to a minority of cases, the fan laws can be safely and effectively used to select a fan, or to predict performance changes of an installed fan. The small errors introduced by ignoring these minor effects and using the fan laws in their simpler form are considerably less than the typical measurement uncertainty and, consequently, entirely negligible.

A.6.4 Use of the fan laws to generate catalogue ratings

Sometimes the performance of a range of mutually similar fans is drawn in the form of a non-dimensional curve. Here the horizontal and vertical axes of a diagram show either percentages of wide open flow (maximum volume flow rate at zero fan static pressure) and fully closed pressure (fan static pressure at zero volume flow rate), or otherwise non-dimensional flow and pressure coefficients φ and ψ (see definitions in ISO 13348).

In such a diagram all the possible operating curves of the fans, at different speeds and densities, collapse into a single curve, and each family of cinematically similar operating points collapse into a single point of the curve.

In most cases, the performance diagrams for a specific fan size are drawn as a family of curves in a diagram having volume flow rate on the horizontal axis, and pressure on the vertical one (see Figure A.4). These curves show the pressure produced by that fan as a function of the volume flow rate, at arbitrary steps of fan speed, and at a specific value of the air density which, in principle, is generally the standard density of 1,2 kg/m^3.

In such a "multi-speed" diagram, each family of mutually similar points lays on a curve which represents one of the many possible "square law" parabolas, having the following formula:

$$P = C \times Q^2$$

where each different value of the constant C identifies a different family of mutually similar points.

This formula can be derived by removing the ratio of operating speeds N_2/N_1 from fan law 1 and fan law 2, under the assumption that $D_2/D_1 = 1$ or that the fan diameter is kept unchanged.

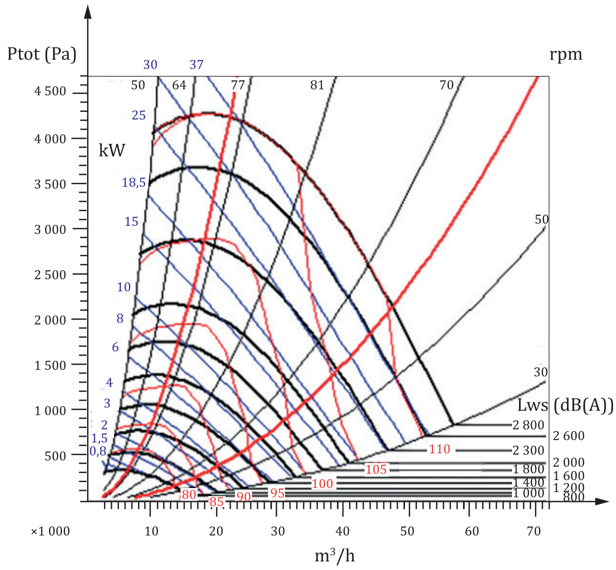


Figure A.4 — Family of constant-speed curves on linear scales

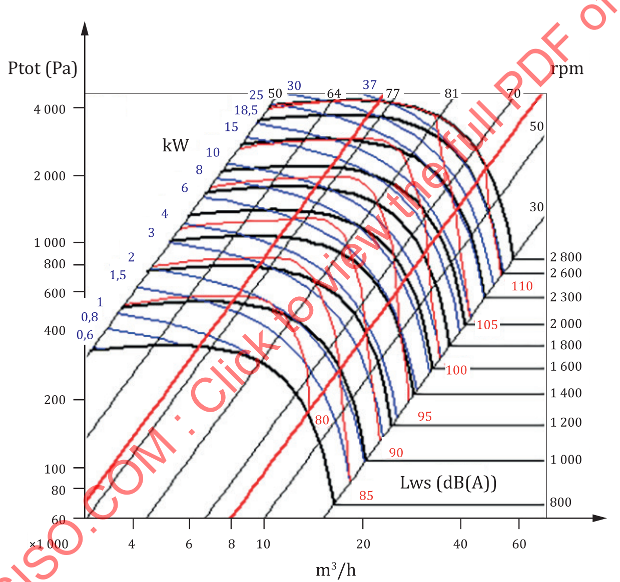


Figure A.5 — Family of constant-speed curves on logarithmic scales

When the “multi-speed” diagram is drawn using logarithmic scales, instead of linear ones, the curves in the diagram are subject to special distortions. The “square law” parabolas, particularly, are converted into straight, upward-sloping lines, parallel to each other as shown in Figure A.5.

If we remember that along each of these lines fan law 4 applies, we can recognize the “constant efficiency” straight lines of the typical logarithmic performance diagram for a belt-driven fan.

A.7 Applications

A.7.1 General

The following subclauses list a number of applications of the fan laws, to solve some typical problems of fan installation and use.

A.7.2 Case 1: Change in fan speed

The first application covers use of the fan laws to calculate the effects of changing only the fan operating speed, supposing that the fan operates at a given air density. Another essential condition is that the fan is connected to an ideal system (constant orifice system), i.e. a system where the pressure loss is exactly proportional to the volume flow squared (curve 3 in Figure A.6).

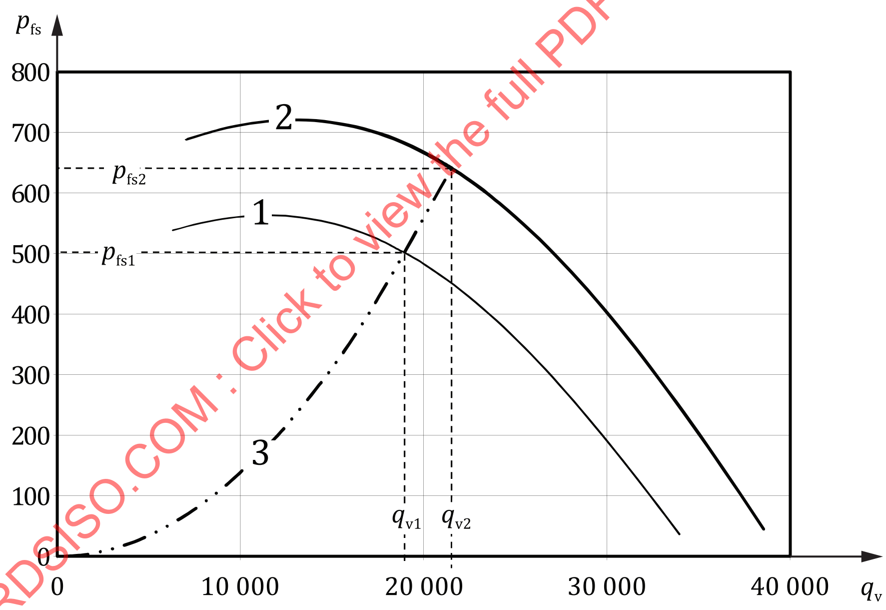
As both the fan diameter and air density remain unchanged, the fan laws from fan law 1 to fan law 4 are simplified as follows:

$$q_{V2} = q_{V1} \times \left(\frac{N_2}{N_1}\right)$$

$$p_{fs2} = p_{fs1} \times \left(\frac{N_2}{N_1}\right)^2$$

$$P_{a2} = P_{a1} \times \left(\frac{N_2}{N_1}\right)^3$$

$$\eta_2 = \eta_1$$



Key

q_{V1} duty point (pressure and volume) along the fan performance curve, before changing speed

q_{V2} duty point (pressure and volume) along the fan performance curve, after changing speed

p_{fs1} fan impeller power at the duty point, before changing speed

p_{fs2} fan impeller power at the duty point, after changing speed

1 fan static pressure curve, as a function of volume flow rate q_v, at speed N_1, before change

2 fan static pressure curve, as a function of volume flow rate q_v, after changing speed to N_2

3 parabolic system line

Figure A.6 — Change in fan speed

The volume flow increases proportionally to the first power of the speed ratio, the pressure to the square of the same ratio, the impeller power to the cube of the speed ratio, while the impeller efficiency will not change. Assuming that the bearing friction is negligible, we can also say that the fan efficiency will also be the same. Under these conditions, if the fan was originally selected to operate at optimum efficiency, changing its speed to alter its volume flow will not reduce its efficiency. At the same time, if the fan has been improperly selected, no speed change can provide a true efficiency improvement.

As most speed changes are applied to increase the volume flow, it is worth remembering that any such change must be kept within the speed and power safety limits established by the fan manufacturer.

A.7.3 Case 2: Change in fan size

Here only the fan diameter is changed, while the two fans are geometrically similar, and rotational speed and air density remain identical (Figure A.7).

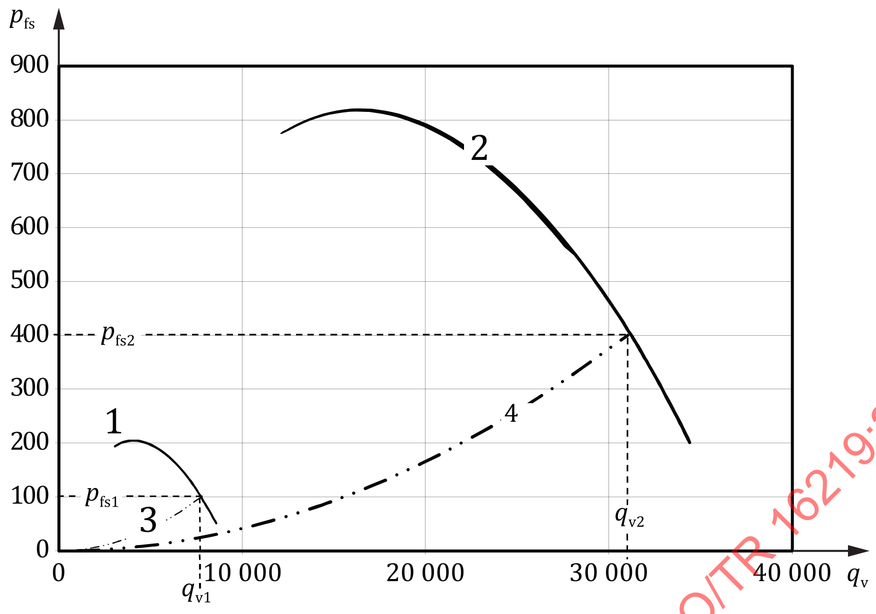
The operating parameters of the cinematically similar operating point for the second fan can be calculated with the following simplified form of the four fan laws:

$$q_{V2} = q_{V1} \times \left(\frac{D_2}{D_1} \right)^3$$

$$p_{f2} = p_{f1} \times \left(\frac{D_2}{D_1} \right)^2$$

$$P_{a2} = P_{a1} \times \left(\frac{D_2}{D_1} \right)^5$$

$$\eta_2 = \eta_1$$



Key

q_{V1} volume flow rate q_v of the duty point of the original fan

q_{V2} volume flow rate q_v of the similar duty point of the fan with new diameter

p_{fs1} static fan pressure p_{sf} of the duty point of the original fan

p_{fs2} static fan pressure p_{sf} of the similar duty point of the fan with new diameter D_2

1 fan static pressure curve, as a function of volume flow rate q_v, of the fan with original diameter D_1

2 fan static pressure curve, as a function of volume flow rate q_v, of the fan with new diameter D_2

3 parabolic system line through the duty point of the original fan D_1

4 parabolic system line through the similar duty point of the fan with new diameter D_2

Figure A.7 — Change in impeller diameter (equal fan speed)

A fifth formula can be written for fan tip speed (V_T):

$$V_{T2} = V_{T1} \times \left(\frac{D_2}{D_1} \right)$$

This calculation procedure is seldom used, alone, to solve practical application problems. It is more frequently used by fan manufacturers to generate performance data for geometrically proportioned "families" of fans.

A.7.4 Case 3: Change in fan size and speed combined

Case 3 shows calculation of the changes in performance produced by inversely proportional changes in fan size and speed, providing constant tip speed, again under the assumption that the density remains unchanged and that the two differently sized fans are geometrically similar (Figure A.8). Under the assumption that

$$N_2 = N_1 \times \left(\frac{D_1}{D_2}\right)$$

which implies also

$$V_{T2} = V_{T1}$$

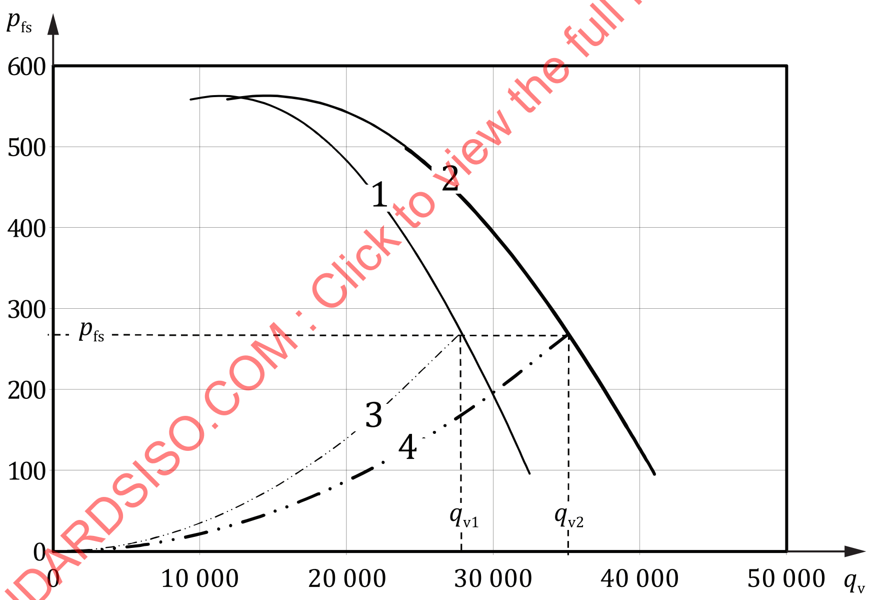
we can calculate the new forms of the fan laws:

$$p_{f2} = p_{f1}$$

$$q_{V2} = q_{V1} \times \left(\frac{D_2}{D_1}\right)^2 = q_{V1} \times \left(\frac{P_{a2}}{P_{a1}}\right)$$

$$P_{a2} = P_{a1} \times \left(\frac{D_2}{D_1}\right)^2$$

$$\eta_2 = \eta_1$$



Key

q_{V1} volume flow rate qv of the duty point of the original fan

q_{V2} volume flow rate qv of the similar duty point of the fan after changing diameter and speed

p_{fs} static fan pressure of the duty point of both fans

1 fan static pressure curve, as a function of volume flow rate q_v, of the first fan with diameter D_1 and speed N_1

2 fan static pressure curve, as a function of volume flow rate q_v, of the second fan with diameter D_2 and speed N_2

3 parabolic system line through the duty point of the original fan

4 parabolic system line through the similar duty point of the second fan with new diameter D_2 and speed N_2

Figure A.8 — Change in fan diameter and speed (constant pressure)

In this case we can see that the similar operating point for the larger fan provides the same pressure, with the same efficiency, but at a volume flow increased by the square of the diameter increase.

Most fan ranges are designed so that they can provide approximately the same tip speed across the size range. These provides the capability to deliver approximately the same pressure in similar operating points, like the best efficiency operating point.

Most metric-sized fan ranges also have sizes taken from the R20 series of normal numbers. An R20 series has an increase of approximately 12,5 % between following, rounded-off sizes. This size-stepping means that, at a given pressure, each fan size can provide optimum efficiency operation at a flow rate some 26 % larger than the smaller size. This also means that when the design flow, which led to the original fan size selection, is altered by more than 20 %, without changing design pressure, a revision of the selected fan size is probably justified, to regain optimum operation.

A.7.5 Case 4: Change in air density only

Case 4 describes the effect of a change of air density, on the same fan, operating at a fixed speed and connected to constant orifice system (i.e. a system where the pressure loss is exactly proportional to the volume flow squared). Another assumption is that the air density, along the system of ducts, is equal or, at least, proportional to the density of the air entering the fan (Figure A.9).

In most practical cases, the pressure loss along a system is proportional to the density of the air along the system. If the air density along the system changes proportionally to the air density in the fan, then both the pressure loss in the system and the pressure generated by the fan will change according to the same ratio of the densities and, consequently:

$$p_{f2} = p_{f1} \times \left(\frac{d_2}{d_1} \right)$$

$$q_{V2} = q_{V1}$$

$$P_{a2} = P_{a1} \times \left(\frac{d_2}{d_1} \right)$$

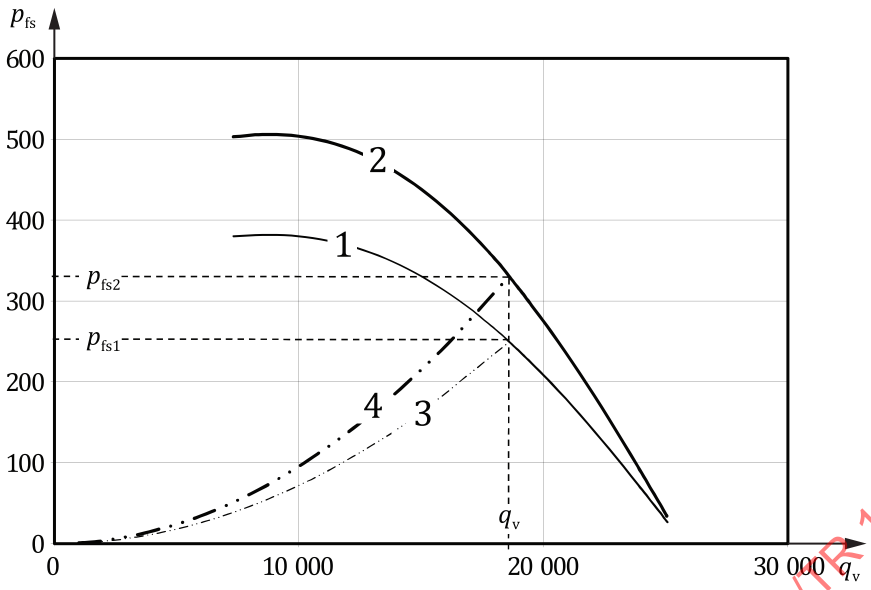
$$\eta_2 = \eta_1$$

In this case, neither the volume flow nor the fan efficiency is altered by the change in density.

The most frequent case of a system operating at varying density is a system where the air temperature is changed significantly, for example a system which must be started at ambient temperature and then run at a temperature significantly higher.

Here the fan is selected to provide the specified volume flow and the pressure required by the system at ambient temperature and density. When the temperature increases, the density is reduced and so are the fan pressure and the power requirement. It is worth noting that the mass flow will be lower at operating temperature than at start-up, as the density will be lower and the volume flow unchanged.

The fan could have been selected using design volume and system pressure at operating temperature and density. The resulting speed would have been the same, but the motor would have been sized for operating power only. Both the motor and the belt drive need to be carefully checked to verify that they can withstand the temporary overload while the system achieves operating temperature.



Key

q_V volume flow rate q_v of the duty points

p_{fs1} volume flow rate q_v of the fan at high temperature and low density

p_{fs2} volume flow rate q_v of the similar duty point of the same fan at standard temperature and density

1 fan static pressure curve, as a function of volume flow rate *qv*, of the fan operating at high temperature and low density

2 fan static pressure curve, as a function of volume flow rate *qv*, of the same fan operating at standard temperature and density

3 parabolic system line through the duty point of the fan at high temperature and low density

4 parabolic system line through the similar duty point of the same fan operating at standard temperature and density

Figure A.9 — Effect of density change (constant flow rate)

A.7.6 Case 5: Change in air density and speed — constant pressure

Case 5 describes an application where a fan of given diameter, connected to a fixed system, is run adjusting its speed to compensate a density change and keep the fan pressure unchanged (Figure A.10).

To produce an unchanged fan pressure,

$$p_{f2} = p_{f1}$$

$$P_2 = P_1$$

we need a speed change given by

$$N_2 = N_1 \times \sqrt{\frac{d_1}{d_2}}$$

$$N_2 = N_1 \times \sqrt{\frac{d_1}{d_2}}$$

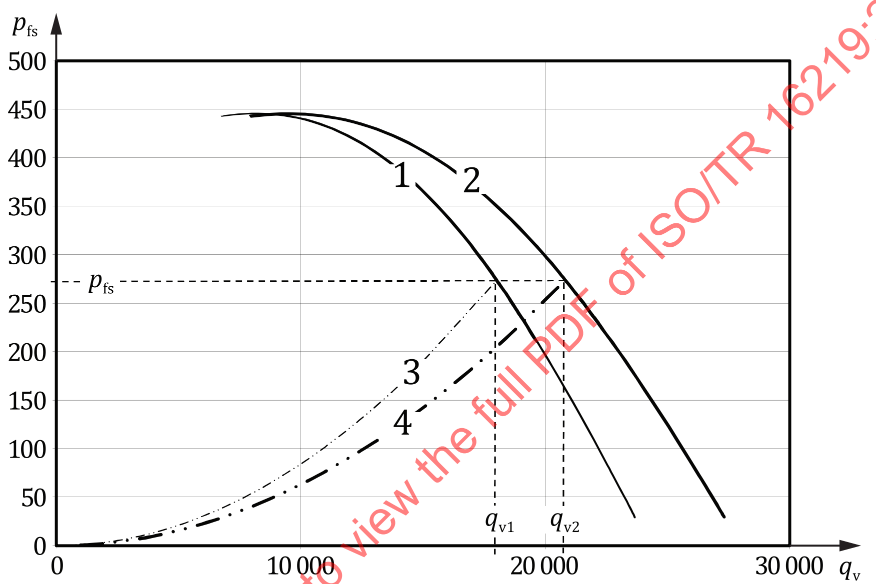
And then the volume flow and power would change to

$$q_{V2} = q_{V1} \times \left(\frac{N_2}{N_1} \right) = q_{V1} \times \sqrt{\frac{d_1}{d_2}}$$

$$P_{a2} = P_{a1} \times \sqrt{\frac{d_1}{d_2}}$$

again, with efficiency unchanged

$$\eta_2 = \eta_1$$



Key

q_{V1} volume flow rate q_v of the of the fan operating at standard temperature and density

q_{V2} volume flow rate q_v of the of the fan operating at lower density and higher speed

p_{fs} fan static pressure at both duty points

1 fan static pressure curve, as a function of volume flow rate qv, of the fan operating at standard temperature and density

2 fan static pressure curve, as a function of volume flow rate qv, of the same fan operating at lower density and higher speed

3 parabolic system line through the duty point of the fan at standard temperature and density

4 parabolic system line through the similar duty point of the same fan operating at lower density and higher speed

Figure A.10 — Density change (constant static pressure)

A.7.7 Case 6: Change in air density and speed — constant mass flow rate

Case 6 describes an application where the fan speed is adjusted to achieve an unchanged mass flow rate, from a fan of given size, connected with a fixed system, when the air density is changed (Figure A.11).

To achieve constant mass flow

$$q_{V2} \times d_2 = q_{V1} \times d_1$$

we need

$$N_2 = N_1 \times \left(\frac{d_1}{d_2}\right)$$

and then

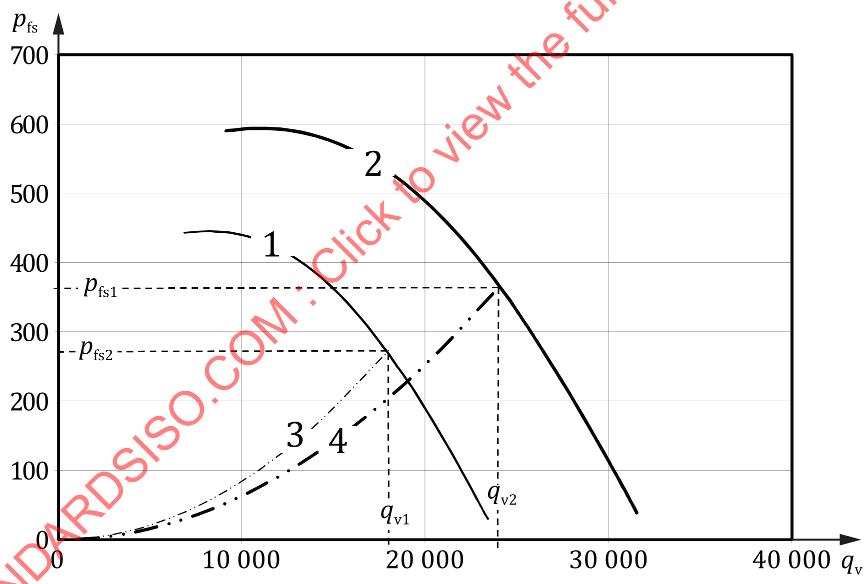
$$p_{f2} = p_{f1} \times \left(\frac{d_1}{d_2}\right)$$

$$P_{a2} = P_{a1} \times \left(\frac{d_1}{d_2}\right)^2$$

and again

$$\eta_2 = \eta_1$$

Application numbers 4, 5 and 6 of the fan laws are the basis for selecting fans for operating densities other than standard, using the catalogue fan tables or diagrams, which are normally based on standard air.



Key

q_{V1} volume flow rate qv of the of the fan operating at standard temperature and density

q_{V2} volume flow rate qv of the of the fan operating at lower density and higher speed

p_{fs1} fan static pressure of the of the fan operating at standard temperature and density

p_{fs2} fan static pressure of the of the fan operating at lower density and higher speed

1 fan static pressure curve, as a function of volume flow rate qv, of the fan operating at standard temperature and density

2 fan static pressure curve, as a function of volume flow rate qv, of the same fan operating at lower density and higher speed

3 parabolic system line through the duty point of the fan at standard temperature and density

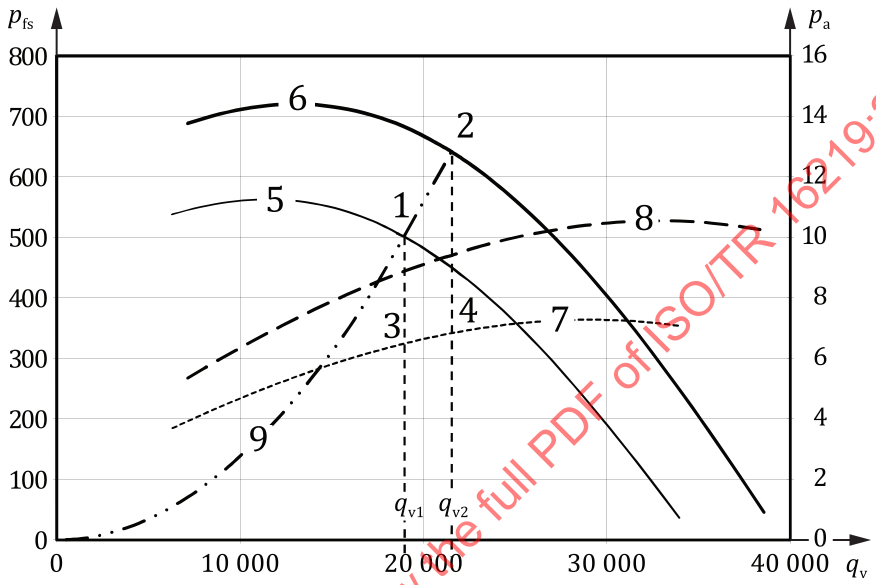
4 parabolic system line through the similar duty point of the same fan operating at lower density and higher speed

Figure A.11 — Density change (constant mass flow)

A.8 Application examples

A.8.1 Example no. 1

An air-conditioning supply fan is operating at a speed of 600 rpm against a static pressure of 500 Pa and requires a mechanical power of 6,5 kW. It is delivering 19 000 m³/h at standard conditions. In order to handle an air-conditioning load heavier than originally planned, more air is required. To increase the volume flow rate to 21 500 m³/h, which new fan speed, static pressure and power are needed? (See Figure A.12.)



Key

q_{V1} fan volume flow rate before changing speed (19 000 m³/h)
q_{V2} fan volume flow rate after changing speed (21 500 m³/h)
1 duty point (at the static pressure of 500 Pa) along the fan performance curve, before changing speed
2 duty point (at the static pressure of 640 Pa) along the fan performance curve, after changing speed
3 fan impeller power (6,5 kW) at the duty point, before changing speed
4 fan impeller power (9,42 kW) at the duty point, after changing speed
5 fan static pressure curve, as a function of volume flow rate qv, at a speed N1 of 600 rpm
6 fan static pressure curve, as a function of volume flow rate qv, after changing speed to 679 rpm
7 fan impeller power, as a function of volume flow rate qv, before changing speed
8 fan impeller power, as a function of volume flow rate qv, after changing speed
9 parabolic system line

Figure A.12 — Example of speed change

Using the fan laws according to case 1:

$$q_{V2} = q_{V1} \times \left(\frac{N_2}{N_1} \right)$$

or

$$N_2 = N_1 \times \left(\frac{q_{V2}}{q_{V1}} \right)$$

$$= 600 \times (21\,500 / 19\,000) = 679 \text{ rpm}$$

$$p_{fs2} = p_{fs1} \times \left(\frac{N_2}{N_1}\right)^2$$

$$= 500 \times (679/600)^2 = 640\,\text{Pa}$$

$$P_{a2} = P_{a1} \times \left(\frac{N_2}{N_1}\right)^3$$

$$= 6{,}5 \times (679/600)^3 = 9{,}42\,\text{kW}$$

Whenever a speed change is applied to increase the volume flow, the new speed and power values need always to be checked to be sure that the speed and power safety limits, established by the fan manufacturer, are not exceeded.

A.8.2 Example no. 2

A fan is operating at a speed of 2 714 rpm with air at 20 °C, against a static pressure of 300 Pa. It is delivering 3 560 m^3/h and requires a mechanical power of 2,84 kW. A 5 kW motor is powering the fan. The system is short capacity but the owner doesn't want to spend any money to change the motor. What is the maximum volume flow from this system with the existing 5 kW motor? What is the maximum speed increase allowed with this motor? What will the flow rate and static pressure be under the new conditions?

Using the fan laws according to case 1 again:

$$N_2 = N_1 \times \sqrt[3]{\frac{P_{a2}}{P_{a1}}}$$

$$= 2\,714 \times (5{,}0/2{,}84)^{1/3} = 3\,280\,\text{rpm}$$

$$q_{V2} = q_{V1} \times \left(\frac{N_2}{N_1}\right)$$

$$= 3\,560 \times (3\,280/2\,714) = 4\,300\,\text{m}^3/\text{h}$$

$$p_{fs2} = p_{fs1} \times \left(\frac{N_2}{N_1}\right)^2$$

$$= 300 \times (3\,280/2\,714)^2 = 440\,\text{Pa}$$

These values, again, need to be always compared with the maximum speed and power recommended by the fan manufacturer, to be sure that no safety limit is exceeded.

A.8.3 Example no. 3

A fan manufacturer wishes to project data, obtained for a fan with a 400 mm diameter, to an 800 mm-diameter fan. At a specific operating point, the 400 mm fan, running at 694 rpm (V_T = 29,07 m/s), delivers 7 755 m^3/h of air at 20 °C, against 100 Pa static pressure. This requires 1,77 kW. What will be the projected flow rate, static pressure, power and tip speed for an 800 mm fan at the same speed? See Figure A.13.

Using the fan laws according to case 3:

$$q_{V2} = q_{V1} \times \left(\frac{D_2}{D_1}\right)^3$$

$$= 7\,755 \times (800/400)^3 = 62\,000 \frac{\text{m}^3}{\text{h}}$$

$$p_{fs2} = p_{fs1} \times \left(\frac{D_2}{D_1}\right)^2$$

$$= 100 \times (800/400)^2 = 400\ \text{Pa}$$

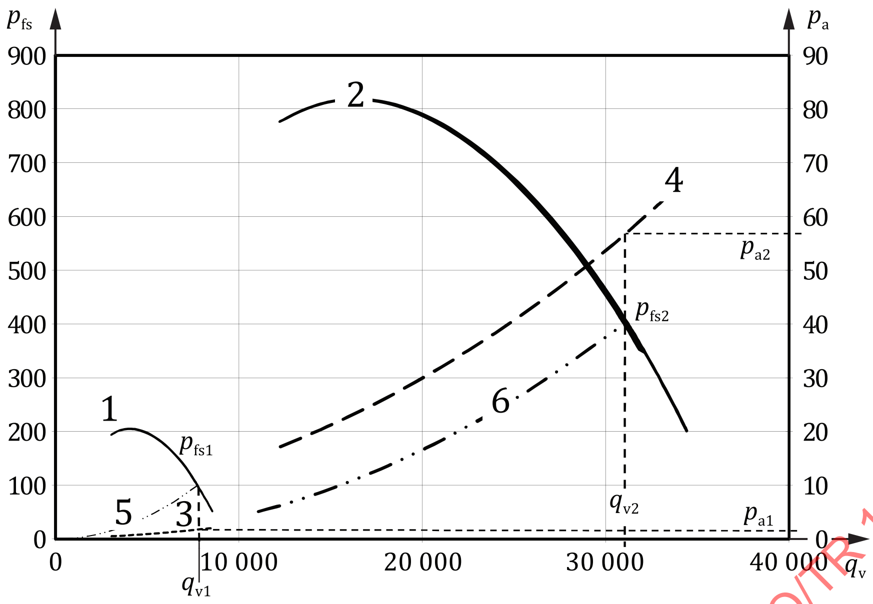
$$P_{a2} = P_{a1} \times \left(\frac{D_2}{D_1}\right)^5$$

$$= 1{,}77 \times (800/400)^5 = 56{,}6\ \text{kW}$$

$$V_{T2} = V_{T1} \times \left(\frac{D_2}{D_1}\right)$$

$$= 29{,}07 \times (800/400) = 58{,}14\ \text{m/s}$$

This example, together with example no. 1, shows how the fan laws can be used to project catalogue data to a number of diameters and speeds from a test on a single fan at one speed.



Key

q_{V1} fan volume flow rate before changing diameter (7 755 m^3/h)

q_{V2} fan volume flow rate after changing speed (31 020 m^3/h)

p_{fs1} duty point (at the static pressure of 100 pa) along the fan performance curve, with original diameter

P_{fs2} similar duty point (at the static pressure of 400 pa) along the fan performance curve, with new diameter

P_{a1} fan impeller power (1,77 kw) at the duty point, with original diameter D1 = 400 mm

P_{a2} fan impeller power (56,6 kw) at the similar duty point, with new diameter D2 = 800 mm

1 fan static pressure curve, as a function of volume flow rate qv, of the fan having diameter 400 mm

2 fan static pressure curve, as a function of volume flow rate qv, after changing diameter to 800 mm

3 fan impeller power, as a function of volume flow rate qv, before changing diameter

4 fan impeller power, as a function of volume flow rate qv, of the fan with new diameter

5 parabolic system line through the original duty point of the fan having diameter D1 = 400 mm

6 parabolic system line (different) through the similar duty point of the fan having diameter D2 = 800 mm

Figure A.13 — Example of change of fan diameter

A.8.4 Example no. 4

A fan drawing air from an oven is delivering 18 620 m^3/h of air at 116 °C against 250 Pa static pressure. It is operating at 796 rpm and requires 9,90 kW. When the oven cools down, the air reaches 21 °C. What happens to the static pressure and to the impeller power required?

using fan laws according to case 4 (Figure A.14):

— density of air at 21 °C = 1,2 kg/m^3

— density of air at 116 °C = 0,9 kg/m^3

$$q_{V2} = q_{V1} = 18\,620\,\mathrm{m}^3/\mathrm{h}$$

$$p_{fs2} = p_{fs1} \times \left(\frac{d_2}{d_1}\right)$$

