EUROVENT / CECOMAF

EUROVENT 2/9 - 1996

EXPERIMENTAL DETERMINATION OF MECHANICAL ENERGY LOSS COEFFICIENTS OF AIR HANDLING COMPONENTS

EUROVENT 2/9 - 1996

EXPERIMENTAL DETERMINATION OF MECHANICAL ENERGY LOSS COEFFICIENTS OF AIR HANDLING COMPONENTS

EUROVENT 2/9 - 1996

This document has been prepared by the EUROVENT Working Group « WG 2 ». It is in accordance with a European Standard in preparation in CEN/TC 156.

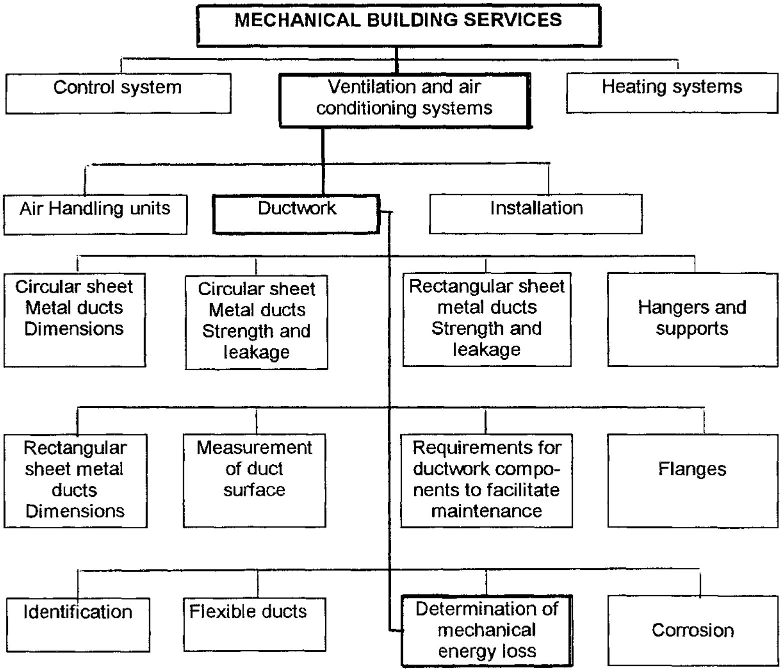
Published by EUROVENT
15 rue Montorgueil
F - 75001 PARIS

Tel 33.1.40.26.00.85
Fax 33.1.40.26.01.26

This EUROVENT Document is a part of series of standards for ductwork used for ventilation and air conditioning of buildings for human occupancy. The position of this standard in the whole field of the standards for the mechanical building services is illustrated in figure 1.

No existing European Standard is superseded.

Figure 1 - Ductwork



CONTENTS

APPENDIX A

PARTICULAR TEST ARRANGEMENTS

APPENDIX B

MEASURING TECHNIQUES

1 - SCOPE

The purpose of this EUROVENT Document is to prescribe unified test procedures and conditions for the experimental determination of mechanical energy loss coefficients for ductwork components such as ducts, elbows, diffusors, converging junctions and diverging junctions.

2 - REFERENCES

ISO 5221 Air Flow measurement in an air handling duct.

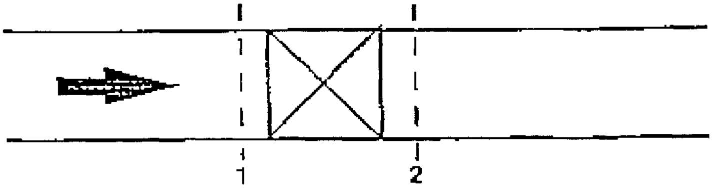
3 - DEFINITIONS

All definitions shall be taken out from the relevant standard prepared by the CEN/TC 156.

4 - GENERAL TEST METHOD

4.1 - Principle

In principle it is possible to give a very rigorous definition of energy loss produced by a component of air distribution systems.



The mechanical energy loss in the flow within the component is equal to the difference between the energy entering the component through Section I and the energy leaving the component through Section 2.

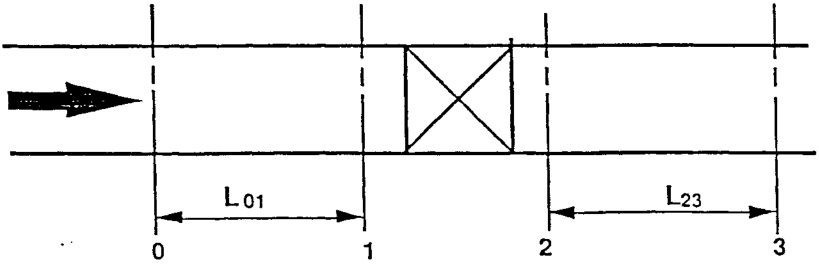
By applying the generalized Bernoulli formula which takes into account the fact that the air is compressible (its density varies through the component) and that it is a real fluid, (the velocity distribution in a section is not uniform), the energy loss by unit mass (J/kg) is expressed as:

$$[\Delta y]_1^2 = \frac{p_1 - p_2}{\rho_{12}} + \alpha_{A1}\frac{Vm^2{}_1}{2} - \alpha_{A12}\frac{Vm^2{}_2}{2} + g(Z_1 - Z_2)$$

p	absolute pressure
Vm	mean flow velocity
Z	altitude
ρ_{12}	fluid density
g	free fall acceleration
α_A	kinetic energy factor

The kinetic energy factor α_A can be found by Pitot-tube exploration in the cross section under consideration. The density ρ_{12} depends on the flow variation through the component.

<u>In practice</u> the mere presence of an air handling component in a duct system modifies the flow structure upstream and downstream of the component. For this reason the practical determination of the mechanical energy losses is generally made on the following test installation:



A straight duct of the length L_{01} is installed upstream of the component and a straight duct of the length L_{23} downstream. The measurement sections (0 upstream and 3 downstream) are consequently distant from the component. From the test values obtained in these sections the characteristics of flow are calculated for the sections I and 2 and then used in the generalized Bernoulli formula to obtain the mechanical energy loss.

The choice of lengths L_{01} and L_{23} and the assumptions concerning the flow through these duct sections can cause differences in the final results. Therefore an agreement on the choice of lengths must be established before the start of the experimental work.

There is no intrinsic value of energy loss coefficient for an air handling component. For each upstream flow condition a different value will be found. Consequently the use of a long straight duct upstream of the component is just one of many possible conditions. However the different lengths of this duct and different entry conditions can product variations in the flow pattern.

Therefore, it is important to specify in detail all characteristics of the installation upstream of the component. According to this standard the upstream straight duct has a length equal to 20D and a specified perforated plate at the entrance. The measuring section is located at a distance 5D from the component.

The downstream flow pattern depends on the component under test. Usually a very long straight duct is used and the measuring section is a distance away in order to allow for the correct measurement. The energy loss of the ducting must be taken into account in the calculation of the energy loss coefficient of the component under test. For the same length of straight duct this energy loss may be very different depending on the flow pattern (essentially in the presence or in the absence of swirl).

As the actual loss is not known the conventional energy loss corresponding to the fully established flow without swirl is normally used.

According to this standard instead of a very long duct (it may be as long as 40D) a specified flow straightener (as used for fan performance testing now applied in the ISO Standard and in many countries) is installed immediately downstream of the component under test. The correct measurement of the pressure is therefore possible whereas the loss on the straightener and associated ducting is taken into account conventionally.

An important advantage of this method is the elimination of the necessity to measure the kinetic energy factor $\propto_A$ in the upstream section as well as in the downstream section. It is assumed that $\propto_A$ is equal to one. If a particular component produces a very strong swirling flow with an irregular velocity distribution, the energy loss in the straightener will be far greater than the conventional value used for the calculation. The energy loss coefficient of the component under test will appear higher.

It is considered that this is the correct way to present these characteristics because in practice the rotational energy in fluid flow will be lost anyway and this loss is produced by the component (though not in the component itself). It may be noted that in usual methods (a long straight duct downstream) this assumption is also applied but the measurement is more difficult and the scatter of results obtained in different laboratories may be important.

4.2 - Test installation

The standard test installation is shown on *Figure* I. Following specification shall be used:

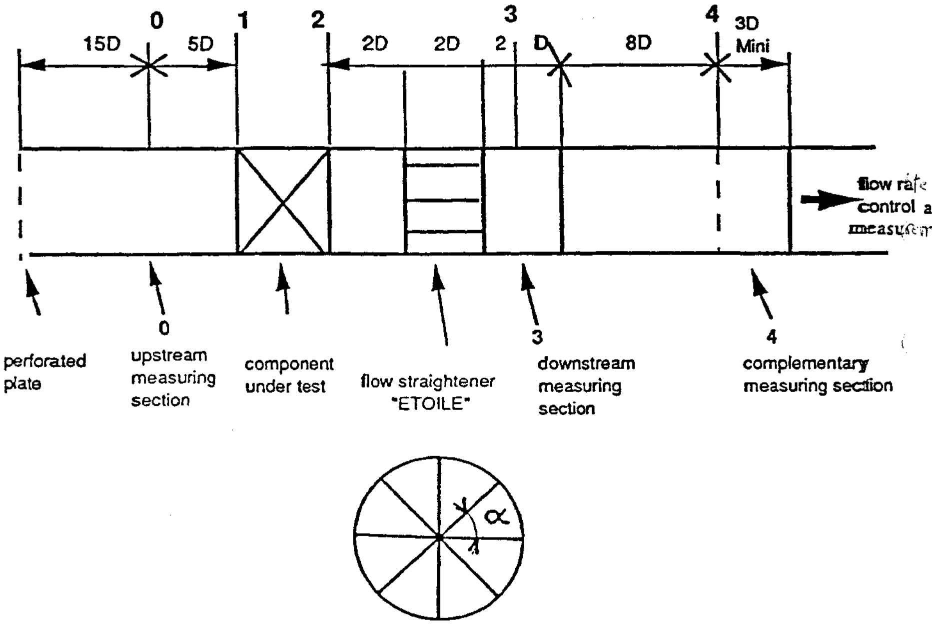


Figure 1

Standard Test Installation

a) Duct diameter : Equal to the diameter of the component under test

b) Perforated plate at the inlet :

   diameter of holes : 5 mm

   distance between axes : 7,5 mm

   free area/total area: 0.40

c) Duct roughness : Smooth metal duct

d) Flow straightener "ETOILE" in accordance with the drawing

   Length : 2D (tolerance I%)

   Thickness : <0.007D

   Tolerance : ∝ = 45°± 2.5°

4.3 - Rectangular and other non-circular ducts and components

For ducts and components with non circular cross sections (essentially rectangular and oval) the notion of the hydraulic diameter shall be introduced. The hydraulic diameter is calculated as four times cross section divided by the perimeter.

For a rectangular cross section with sides a and b this gives :

$$D_h = \frac{4ab}{2(a+b)} = \frac{2ab}{a+b}$$

The standard test installation shall be made with upstream and downstream ducts of the same cross section as the component under test ; using D_h instead on D for circular duct all calculations will be the same.

As an alternative solution a test installation with circular ducts may be used. The component under test shall be connected to the upstream and downstream ducts using a transition with the following specifications :

- the cross section area of the circular duct shall be equal to the cross section area of the component with a tolerance of ± 10 %

- the length of the transition shall be equal to one diameter of circular duct

- for the calculation of the energy loss coefficient of the component under test, the energy loss in the transition shall be considered equal to the loss in a straight duct having the same length.

4.4 - Measurements

The following quantities shall be measured:

I) Atmospheric pressure P_{atm} (Pa)

2) Air temperature θ (°C) [$T = 273{,}15 + \theta$ (K)]

3) Air humidity - either dry and wet bulb or dew point temperature

4) Static pressure in the section 0 (mean value of four individual readings)

P_{so} (Pa)

5) Static pressure in the section 3 (mean value of four individual readings)

P_{s3} (Pa)

6) Differential pressure between the section 0 and 3.

ΔP_{so3} (Pa)

7) Static pressure in the section 4 (mean value of four individual readings)

P_{s4} (Pa)

8) Mass flow rate (by an appropriate standardized device as given in Paragraph 6.I.

q_m (kg/s)

4.5 - Calculations

1) Absolute pressures are calculated for the sections 0 and 3.

$P_0 = P_{atm} - P_{s0}$

$P_3 = P_{atm} - P_{s3}$

2) Mean air density (which will be considered constant throughout the test installation) is calculated from

$$\rho = \frac{Pm}{287T} \cdot f$$

where $Pm = \frac{Po + P3}{2}$ and f, the humidity factor, given by :

$f = 1 - 0.378\,\frac{Pv}{Pm}$ where Pv is the partial vapour pressure

3) Reynolds number is calculated by $\mathrm{Re} = \dfrac{4qm}{\pi\mu D}$

where the dynamic viscosity is given by
$\mu = (17.1 + 0.048\,\theta)\,10^{-6}$

4) Mean air velocity is calculated from $V = \dfrac{4q_m}{\rho D^2 \pi}$

5) Pressures in sections l and 2 are calculated from

$$P_1 = P_0 - \zeta_{01}\frac{\rho V^2}{2}$$

$$P_2 = P_3 + \zeta_{23}\frac{\rho V^2}{2}$$

where $\zeta_{01} = 5\,(0.005 + 0.45\,\mathrm{Re}^{-0.30})$

$\zeta_{23} = 0.95\,\mathrm{Re}^{-0.12} + 3\,(0.005 + 0.42\,\mathrm{Re}^{-0.30})$

Values of ζ_{01} and ζ_{23} for some Re values are :

Re	ζ_{01}	ζ_{23}
50 000	0.11	0.32
100 000	0.09	0.29
200 000	0.08	0.27
400 000	0.07	0.24

6) Differential pressure between sections l and 2 shall be then :

$$p_1 - p_2 = \Delta p_{03} - (\zeta_{01} + \zeta_{23})\,\frac{\rho V^2}{2}$$

7) Mechanical energy loss per unit mass is given by:

$$[\Delta y]_1^2 = \frac{p_1 - p_2}{2}$$

8) Energy loss coefficient of the component tested is given by

$$\zeta = \frac{p_1 - p_2}{\rho \frac{V^2}{2}}$$

4.6 - UNCERTAINTIES

To determine the uncertainties of test results we have to start with the formula used for calculating the energy loss coefficient of a component:

$$\zeta = \frac{p_1 - p_2}{\rho \frac{V^2}{2}}$$

In practice p_1-p_2 is not measured directly ; from the measured differential pressure $\Delta\ p_{03}$ the value of $(p_1 - p_2)$ is calculated by:

$$p_1 - p_2 = \Delta p_{03} - (\zeta_{01} + \zeta_{23}) \cdot \rho \frac{V^2}{2}$$

The energy loss coefficient is therefore given by :

$$\zeta = \frac{\Delta p_{03}}{\frac{\rho V^2}{2}} - (\zeta_{01} + \zeta_{23})$$

The term $(\zeta_{01} + \zeta_{23})$ represents a calculated conventional value of the correction to be applied. It varies only slightly with Reynolds number ; for a Reynolds number variation of 100 % (for instance from 10^5 to 2.10^5) this coefficient varies only 8 %. As the Reynolds number can be easily known with an uncertainty of less than 2 %, it is clear that the uncertainty on $(\zeta_{01} + \zeta_{23})$ is very small. Therefore, the uncertainty on ζ will be closely related to the uncertainty on the term

$$\frac{\Delta p_{03}}{\frac{\rho V^2}{2}}$$

In fact the absolute uncertainty on ζ will be very close to the absolute uncertainty on this term.

The term $\frac{\Delta p_{03}}{\frac{\rho V^2}{2}}$ may be developped as follows:

$$\frac{\Delta p_{03}}{\frac{\rho V^2}{2}} = \frac{\Delta p_{03}}{\frac{\rho}{2}\left[\frac{4qm}{\rho D^2 \pi}\right]^2} = \frac{\Delta p_{03}.D^4.\rho\pi^2}{8q_m^2}$$

In order to determine the overall uncertainty, the following expression must be used :

$$\zeta = \frac{\Delta p_{03}.D^4.\rho\pi^2}{8q_m^2} - (\zeta_{01} + \zeta_{23})$$

We can put $\zeta = X - Y$

where $X = \frac{\Delta p_{03}.D^4.\rho\pi^2}{8q_m^2}$

and $Y = (\zeta_{01} + \zeta_{23})$

To find the uncertainty associated with ζ it must be kept in mind that ζ is given as a difference (and not a product) between two terms. Therefore the absolute (and not relative) values of uncertainties for each term have to be used.

If the absolute uncertainty on X is Δ x and on Y ΔY, it may be considered that the absolute uncertainty on ζ will be equal to:

$$\Delta\zeta = \sqrt{\Delta X^2 + \Delta Y^2}$$

As these are absolute uncertainties it is not possible to produce a general statement for all possible situations ; the relative magnitude between X and Y is very important.

As an example let's consider the results obtained for a given component for Re = 100 000 With X = 0.73 and Y = 0.38 the loss coefficient is ζ = X - Y = 0.35.

If X is obtained with a relative uncertainty of 3 % and Y with a relative uncertainty of 0.3 % these values should be transformed into absolute uncertainties.

Therefore $\Delta X = 0.0219$ and $\Delta Y = 0.00114$.

The <u>absolute</u> uncertainty on ζ may then be estimated to be equal to

$$\Delta\zeta = \sqrt{\Delta X^2 + \Delta Y^2}$$

The relative uncertainty on ζ is therefore 6.2 %.

With the same relative uncertainties on X and Y, but with a different relative magnitude of X and Y, the relative uncertainties on ζ will be different.

If X has about twice value of Y for the component A and about three times value of Y for the component B, the uncertainty on ζ will be about twice the uncertainty on X for the component A and about 1.5 times the uncertainty on X for the component B.

Looking at the term X it may be seen that four quantities must be measured

- pressure Δp_{o3}
- duct diameter D
- density ρ
- flow rate q_m

It is easy to see that the flow rate must be measured with a very high accuracy. An error of 2% on qm will give 4% for the term X and 8% ζ for the component A used in the above example.

The second important quantity is Δp_{o3} . Here an error of 2% will give 2% for X and 4% for ζ for the component A.

Highly sensitive micromanometer must be used as there is no generally recognized national or international standard but it is not possible to correctly evaluate the uncertainty associated with pressure measurement. Only a direct comparison between the pressure measuring instruments may allow this evaluation.

4.7 - Number of test points

The mechanical energy loss coefficient of air handling components usually varies with Reynolds number. The variation is strong for low Reynolds numbers and generally the coefficient decreases when Reynolds number increases. For Reynolds numbers higher that 150 000 the variation of the coefficient may be considered negligeable.

Therefore, in order to decide the number of test points, the useful range of Reynolds numbers shall be first defined. A minimum of three tests shall be carried out at :

- minimum required Re
- maximum required Re
- an intermediate Re

If minimum required Reynolds number is above 150 000 a single test for one Reynolds number shall be sufficient.

4.8 - Presentation of data

The test report shall contain :

- clear definition of the component tested
- energy loss coefficient at corresponding Reynolds numbers

APPENDIX A

PARTICULAR TEST ARRANGEMENTS

A.1 - Components with inlet different from outlet (diverging or converging)

If the inlet diameter of a component is different from its outlet diameter, the test installation shall be built according to the general principles using the inlet diameter D_1 for the installation upstream of the component and the outlet diameter D_2 for the installation downstream of the component. The upstream mean velocity V_1 and Reynolds number Re_1 shall be calculated using the measured mass flow rate and corresponding upstream cross section ; the same calculation shall be applied for the downstream part of the test installation in order to calculate the downstream mean velocity V_2 and Reynolds number R_{02}, Differential pressure between Sections I and 2 shall be:

$$p_1 - p_3 = \Delta p_{03} - \zeta_{01} \frac{\rho V_1^2}{2} - \zeta_{23} \frac{\rho V_2^2}{2}$$

where : $\zeta_{01} = 5\ (0.005 + 0.42\ Re_1^{-0.30})$

$\zeta_{23} = (0.95\ Re_2^{-0.12} + 3\ (0.005 + 0.42\ Re_2^{-0.30})$

The energy loss coefficient of the component is given by :

$$\zeta = \frac{p_1 - p_2}{\frac{\rho V_1^2}{2}}$$

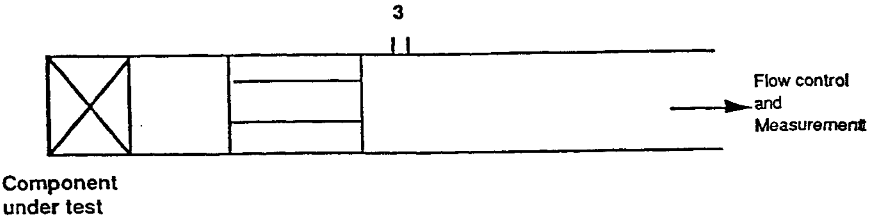
A.2 - Components with free inlets

The components intended to be used at the inlet of air handling installations shall be tested in the condition corresponding to their normal use. Only downstream part of the general test installation (*Figure* 2) shall be used. All calculations shall be the same assuming that the differential pressure used to calculate the energy loss coefficient shall be:

$$p_1 - p_2 = p_3 - \zeta_{23} \frac{\rho V_2^2}{2}$$

Figure 2

Testing Components with free inlet

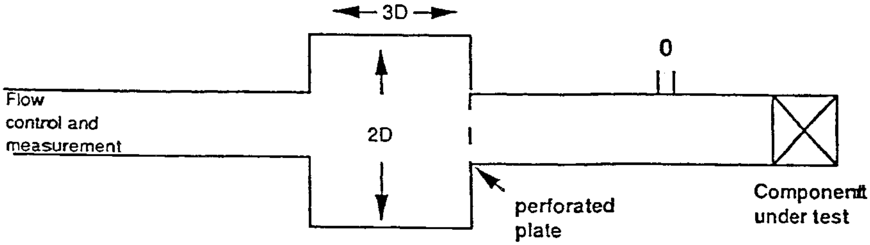


A.3- Components with free outlets

The components intended to be used at the outlet of air handling installations shall be tested in the conditions corresponding to their normal use. Only inlet part of the general test installation shall be used. In order to allow the flow-rate control and measurement without changing test conditions, an inlet chamber shall be used as shown on *Figure* 3. Flow measuring and control system shall be installed upstream of the component under test.

Figure 3

Testing Components with free outlet



A.4 - Components with two inlets (convering junctions)

By convention the inlet branches shall be designated as Branch A and Branch B and the common section with total flow as C.

The upstream test installation of both Branches A and B shall be built in accordance to the general test method (perforated plate at the inlet, 20D long straight duct, pressure taps at 5D upstream of the component under test). The downstream test installation (common Section C) shall be built in accordance with the general test method (straight duct of 17D minimum length, straightener and pressure tops at 5D downstream of the component under test).
The test installation is shown on *Figure* 4.

For each component two energy loss coefficients shall be defined, one for each branch. The same procedure as in the general test method (Paragraph 4) shall be applied with following specifications:

a) Energy loss coefficient for each branch shall be defined with respect to the dynamic pressure in the common Section C with total flow.

b) When correcting for the energy loss in the straight duct or in the straightener the care shall be taken to apply proper dynamic pressure and Reynolds number.

c) The number of test points shall be defined in accordance with the specification given in Paragraph 4.5. However, for each total flow rate, tests shall be carried out for at least five flow rate ratios between branches (corresponding approximately to the ratio between the flow rate in one branch and the total flow rate equal to : 0, 0.3, 0.5, 0.7 and 1.0).

d) As there are three different flow rates for each case, at least two of them must be measured. It is recommended to use the calibrated inlet perforated plate; the static pressure at section 0 may be used. An exemple of calibration is given on *Figure* 5.

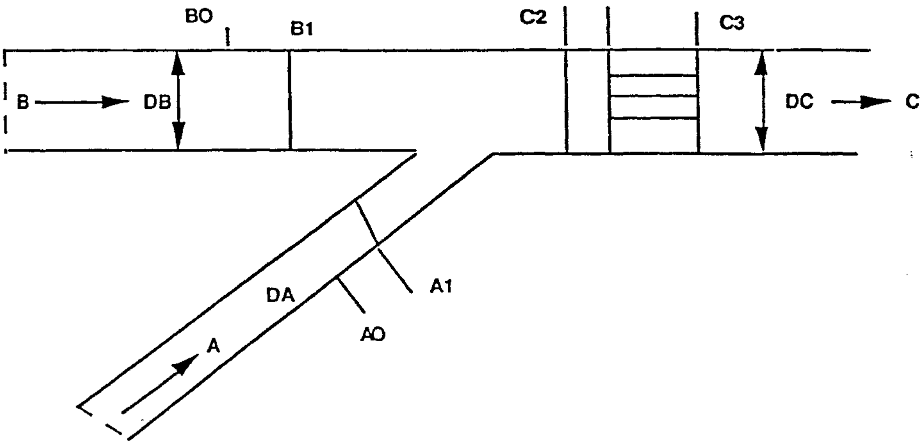
e) In order to change the flow rate ratio between branches it will be necessary to modify the flow resistance of the branches. For this purpose on inlet chamber in accordance with the drawing shown on *Figure* 6 shall be used. This chamber shall have a diameter at least two times larger than the test duct diameter and a length equal to 3D. An orifice plate at the inlet shall be used in order to modify the flow rate. It is considered that the velocity distribution immediately upstream of the component under test will not be modified when such a chamber is used.

   In order to cover the complete range of flow ratios the chamber must be used successively on both inlet branches.

f) An example of calculation is shown on *Figure* 4.

Figure 4

Converging junctions



	BRANCH A (or BRANCH B)	BRANCH C
1. Mass flow-rate	qm_A	qm_C
2. Reynolds Number	$Re_A = \frac{4qm_A}{\pi\mu D_A}$	$Re_C = \frac{4qm_C}{\pi\mu D_C}$
3. Mean Velocity	$V_A = \frac{qm_A}{\rho\pi D_A^2}$	$V_C = \frac{qm_C}{\rho\pi D_C^2}$
4. Measured Pressure Differential	$\Delta P_{AC} = P_{A0} - P_{C3}$	
5. Calculated Pressure Differential	$P_{A1} - P_{C2} = \Delta P_{AC} - K$	

6. Correction Term $K = \zeta A_{01} \frac{\rho V_A^2}{2} + \zeta C_{23} \frac{\rho V_C^2}{2}$

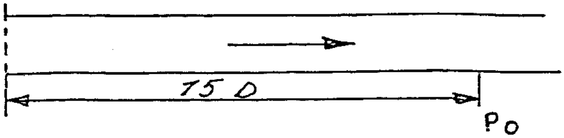
$\zeta A_{01} = 5 (0.005 + 0.42 R_{eA}^{-0.30})$

$\zeta C_{01} = 0.95 R_{eC}^{-0.12} + 3 (0.005 + 0.42 R_{eC}^{-0.3})$

7. Loss Coefficient $\zeta_{AC} = \frac{P_{A1} - P_{C2}}{\frac{\rho V_C^2}{2}}$

Figure 5

Calibrated Perforated Plate



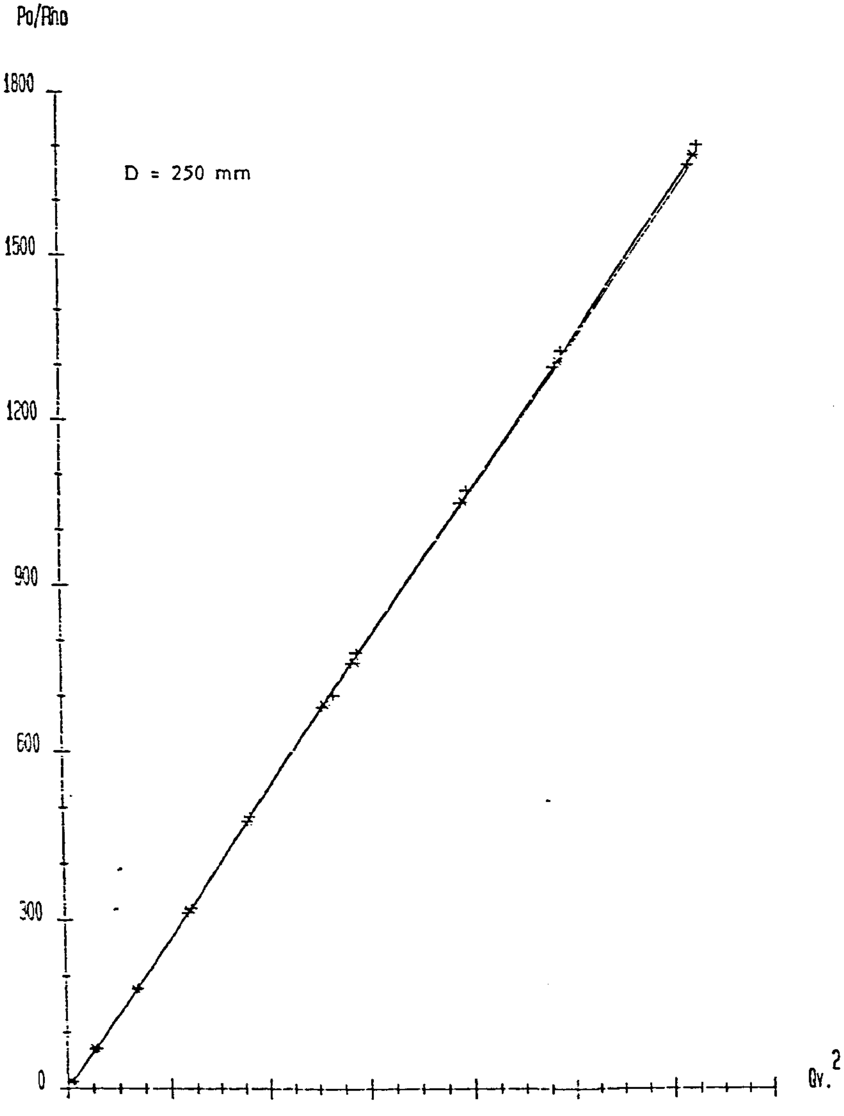
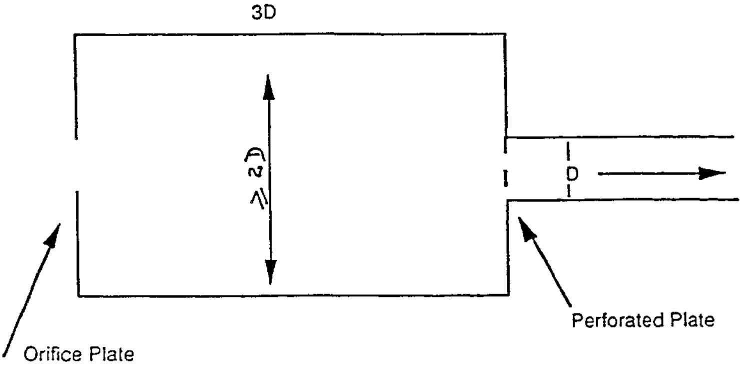


Figure 6

Inlet Chamber for Flow Regulation



A.5 - Components with two outlets (diverging junctions)

By convention the outlet branches shall be designated as Branch A and Branch B and the common inlet section with total flow as C.

The upstream test installation of the common Section C shall be built in accordance with the general test method (perforated plate at the inlet, 20D long straight duct, pressure taps at 5D upstream of the component under test). The downstream installation of both branches A and B shall be built in accordance with the general test method (straight duct of 17D minimum length, straightener and pressure taps at 5D downstream of the component under test). Each Branch A and B shall have its fan, flow rate measuring and controlling system.

The test installation is shown on *Figure* 7 together with an example of calculation.

For each component two energy loss coefficients shall be defined, one for each branch. The same procedure as in the general test method (Paragraph 4) shall be applied with following specifications:

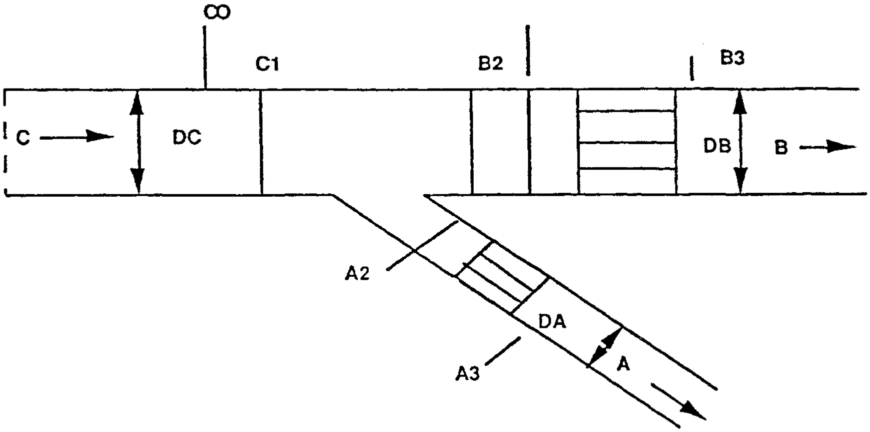
a) Energy loss coefficient for each branch shall be defined with respect to the dynamic pressure in the common Section C with total flow.

b) When correcting for the energy loss in the straight duct or in the straightener the care shall be taken to apply proper dynamic pressure and Reynolds number.

c) The number of test points shall be defined in accordance with the specification given in Paragraph 4.5. However, for each total flow rate, tests shall be carried out for at least five flow rate ratios between branches (corresponding approximately to the ratio between the flow rate in one branch and the total flow rate equal to : 0, 0.3, 0.5, 0.7 and 1.0).

Figure 7

Diverging junctions



	BRANCH C (or BRANCH B)	BRANCH A
1. Mass flow-rate	qm_C	qm_A
2. Reynolds Number	$Re_C = \frac{4qm_C}{\pi\mu D_C}$	$Re_A = \frac{4qm_A}{\pi\mu D_A}$
3. Mean Velocity	$V_C = \frac{qm_C}{\rho\pi D_C^2}$	$VA = \frac{qm_A}{\rho\pi D_A^2}$
4. Measured Pressure Differential	$\Delta PC_A = P_{C0} - P_{A3}$	
5. Calculated Pressure Differential	$P_{C1} - P_{A2} = \Delta PC_A - K$	
6. Correction Term	$K = \zeta C_{01} \frac{\rho V_C^2}{2} + \zeta A_{23} \frac{\rho V_A^2}{2}$	

$$\zeta C_{01} = 5\,(0.005 + 0.42\,R_{eC}^{-0.30})$$

$$\zeta A_{23} = 0.95\,R_{eA}^{-0.12} + 3\,(0.005 + 0.42\,R_{eA}^{-0.3})$$

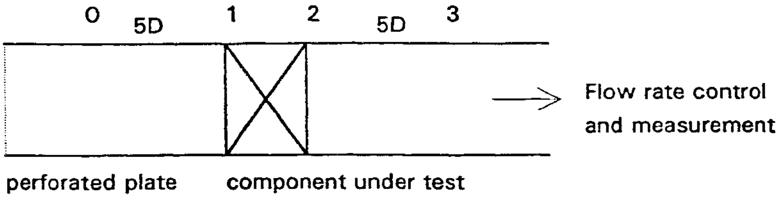
7. Loss Coefficient $\zeta_{CA} = \dfrac{P_{C1} - P_{A2}}{\dfrac{\rho V_C^2}{2}}$

A.6 - Components without swirl

When testing components with which it may be reasonable expected that the swirl cannot be generated (such as straight ducts) the upstream part of the general test installation shall be used as it is necessary to assure the standardized inlet flow conditions. However, as there is no possibility to generate a swirl in the flow the downstream installation shall be built in accordance with the *Figure* 8.

Figure 8

Testing of components without swirl



The differential pressure used to calculate the energy loss coefficient shall be:

$$p_1 - p_2 = p_0 - p_3 - (\zeta_{01} + \zeta_{23})\frac{\rho V^2}{2} = \Delta p_{03} - (\zeta_{01} + \zeta_{23})\frac{\rho V^2}{2}$$

$$\text{where } \zeta_{01} + \zeta_{23} = 10\,(0.005 + 0.42\,\mathrm{Re}^{-0.30})\,\frac{\rho V^2}{2}$$

APPENDIX B

MEASURING TECHNIQUES

B.1 - Flow rate measurement

For the purpose of this standard the flow rate measurement shall be carried out using one of fourteen methods described in the International Standard ISO 5221:
"Air distribution and air diffusion - Rules to methods of measuring air flow rate in an air handling duct".

This EUROVENT Document gives different methods of measuring air flow rate in an air handling duct which, without the need of calibration, meet various specific requirements in the field of air distribution and air diffusion.

The following considerations should be kept in mind:

a) The fluid is air, its temperature and pressure being almost those at ambient conditions.

b) Since the flow rates are sometimes relatively small, the Reynolds numbers to be considered may sometimes correspond to relatively small values (for instance some thousands).

c) The widest possible freedom of choice is provided in order to have methods which can be applied either to laboratory testing or to site testing.

d) The methods of measuring air flow rates in a duct have reached a higher degree in the matter of accuracy than is sometimes necessary for the requirements of air distribution and air diffusion.

The values indicated for the uncertainty of the coefficients given must be increased for the uncertainty of the air flow rate itself when inappropriate manometers are used.

Finally, it should not be forgotten that the values which are mentioned throughout this International Standard would be seriously in error if the flow approaching the measuring devices herein described do not offer any guarantee on this point without the addition of a suitable accessory.

In cases where low Reynolds numbers occur and where reduced requirements concerning accuracy are acceptable, special information has been given in an Annex to ISO 5221.

The use of one of the following devices is proposed:

I) Orifice plate with corner taps
2) Orifice plate with flange taps
3) Orifice plate with D and D/2 tappings
4) ISA 1932 nozzle
5) "Long-radius" nozzle
6) Classical Venturi tube
7) Venturi nozzle
8) Orifice plate with conical entrance
9) "Quarter circle" orifice plate
10) Orifice plate located at the inlet end of the system
11) "Quarter circle" nozzle located at the inlet end of the system
12) Inlet cone
13) Venturi nozzle with sonic throat (critical flow nozzle)
14) Pitot-static tube

B.2 - Pressure measurement

B.2.1 - Atmospheric pressure

The atmospheric pressure in the test enclosure shall be determined at the mean altitude between the inlet and outlet sections with an uncertainty not exceeding ± 0.2%. Barometers of the direct reading mercury column type should be read to the nearest 100 Pa (1 millibar) or to be nearest 1 mm of mercury. They should be calibrated and corrections applied to the readings for any difference in mercury density from standard, any change in length of the graduated scale due to temperature and for the local value of g.

Correction may be unnecessary if the scale is preset for the regional value of g (within ± 0.01 m s^{-2}) and for room temperature (within ± 5°C).

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

B.2.2 - Pressure difference

Manometers for the measurement of pressure difference shall have an uncertainty under conditions of steady pressure, and after applying any calibration corrections (including that for any temperature difference from calibration temperature and for g value) not exceeding ± 1 % of the significant pressure or 1.5 Pa whichever is the greater.

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

Calibration should be at a series of steady pressures, taken both in rising and falling sequence to check for any difference.

The reference instrument should be a precision manometer or micro-manometer capable of being read to an accuracy of ± 0.25 % or 0.5 Pa whichever is greater.

Liquid column manometers should be checked in their test location to confirm their calibration near the significant pressure. Inclined tube instruments should be frequently checked for level and rechecked for calibration if disturbed. The zero reading of all manometers shall be checked before and after each series of readings without disturbing the instrument.

B.2.3 - Use of wall tappings

At each of the sections for pressure measurement in the standardised airways specified in Section 4.2 the average static pressure shall be taken to be the average of the static pressure at four wall tappings.

Each tapping takes the form of a hole through the wall of the airway conforming to the dimensional limits shown in figure 9. It is essential that the hole be carefully produced so that the bore is normal to and flush with the inside surface of the airway, and that all internal protrusions are removed. Rounding of the edge of the hole up to a maximum of 0.1 a is permissible.

In the case of a cylindrical airway the four tappings should be equally spaced around the circumference. In the case of a rectangular airway they should be at the centres of the four sides. Four similar tappings may be connected to a single manometer but, to avoid averaging error, it is recommended that they should be connected in pairs by two equal lengths of tubing as shown in figure 8. The mid-points of these should be connected by a third tube, and the manometer connected at the mid-point of this.

Note : The bore diameter a to be not less than 1.5 mm not greater than 5 mm and not greater than 0.1 D.

Care shall be taken to ensure that all tubing and connections are free from blockage and leakage and empty of liquid. Before the commencement of any series of observations the pressure at the four side tappings should be individually measured at a flow rate towards the maximum of the series. If any one of the four readings lies outside a range equal to 2 % of the significant pressure, the tappings and manometer connections should be examined for defects, and if none are found the flow should be examined for uniformity.

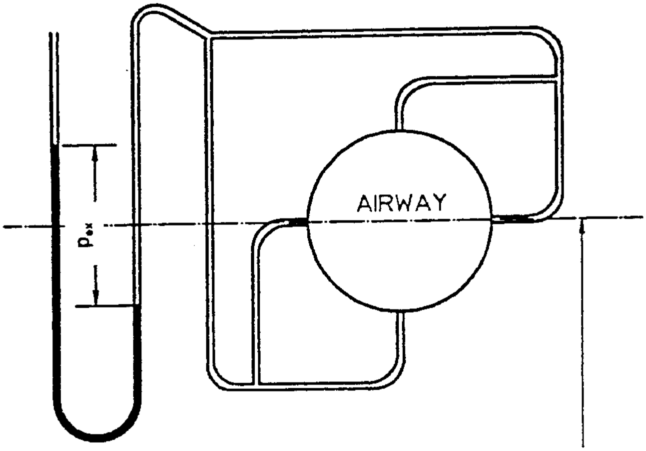


Figure 8 Tapping connections to obtain average static pressure and altitude of manometer

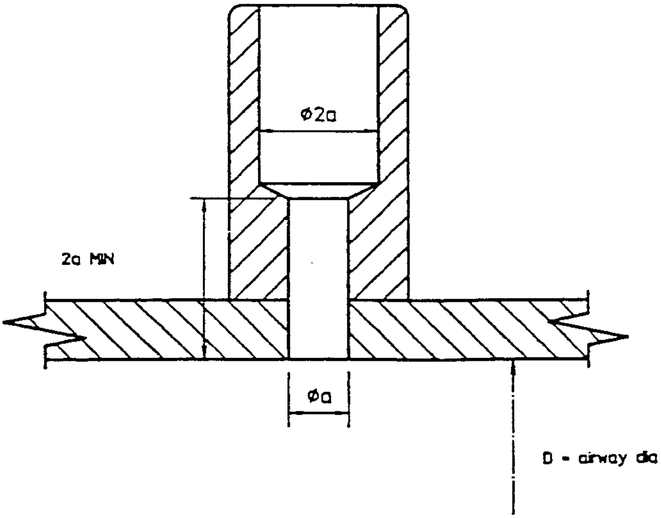


Figure 9 Construction of wall pressure tappings

B.3 - Temperature measurement

Instruments for the measurement of temperature shall have an accuracy of ± 0.5°C after the application of any calibration correction.

B.4 - Humidity measurement

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

The wet-bulb thermometer should be located in an air stream of velocity of at least 3 $m\ s^{-1}$. The sleeving should be clean, in good contact with the bulb, and kept wetted with pure water.

Relative humidity may be measured provided the apparatus used has an accuracy of ± 2 %.

LIST OF THE MEMBER ASSOCIATIONS

BELGIUM FABRIMETAL 21 rue des Drapiers - B-1050 BRUXELLES Tel 32/2/5102518 - Fax 32/2/5102562	***ITALY*** ANIMA - CO.AER Via Battistotti Sassi, 11 - I-20133 MILANO Tel 39/2/73971 - Fax 39/2/7397316
GERMANY **FG ALT im VDMA** Postfach 710864 - D-6000 FRANKFURT/MAIN 71 Tel 49/69/66031227 - Fax 49/69/66031218	***NORWAY*** **NVEF** P.O.Box 850 Sentrum - N-0104 OSLO Tel 47/2/413445 - Fax 47/2/2202875
SPAIN **AFEC** Francisco Silvela, 69-1°C - E-28028 MADRID Tel 34/1/4027383 - Fax 34/1/4027638	***SWEDEN*** **KTG** P.O. Box 55 10 - S-11485 STOCKHOLM Tel 46/8/20800 - Fax 46/8/6603378
FRANCE **UNICLIMA (Syndicat du Matériel Frigorifique, Syndicat de l'Aéraulique)** Cedex 72 - F-92038 PARIS LA DEFENSE Tel 33/1/47176292 - Fax 33/1/47176427	***SWEDEN*** **SWEDVENT** Box 17537 - S-11891 STOCKHOLM Tel 46/8/6160400 - Fax 46/8/6681180
UNITED KINGDOM **FETA (HEVAC and BRA)** Sterling House - 6 Furlong Road - Bourne End GB-BUCKS SL 8 5DG Tel 44/1628/531186 - Fax 44/1628/810423	***FINLAND*** **FREA** PL 37 FIN-00801 HELSINKI Tel 358/9/759 11 66 - Fax 358/9/755 72 46
NETHERLANDS **VLA** Postbus 190 - NL-2700 AD ZOETERMEER Tel 31/79/531258 - Fax 31/79/531365	***FINLAND*** **AFMAHE** Etalaranta 10 - FIN-00130 HELSINKI Tel 358/9/19231 - Fax 358/9/624462
NETHERLANDS **NKI** Postbus 190 - NL-2700 AD ZOETERMEER Tel 31/79/3531258 - Fax 31/79/3531365	***TURKEY*** **ISKID** ARCELIK S.A. Klima Isletmesi - 81719 TUZLA ISTANBUL Tel 90/216 3954515 - Fax 90/216 4232359