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Introduction to design and performance data for diffusers

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(Continued on inside back cover)

INTRODUCTION TO DESIGN AND PERFORMANCE DATA FOR DIFFUSERS

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INTRODUCTION TO DESIGN AND PERFORMANCE DATA FOR DIFFUSERS

1. INTRODUCTION

A diffuser is a section of closed duct along which the mean static pressure of a flowing fluid increases as a result of decreasing kinetic energy of the flow without energy input from an external* source. For subsonic flow this generally requires a duct with a cross-sectional area increasing in the direction of flow, although an increase in mean static pressure can occur in ducts of other geometries, such as parallel pipes, when a distorted velocity profile settles to a more uniform condition.

Diffusers are used to match components in internal flow systems by producing an increase in static pressure without further energy input, by providing an efficient reduction in flow velocity or simply by joining a small area duct to a larger one. For example, diffusers are used to improve the performance of compressors by converting some kinetic energy to pressure energy (often also called “flow work”) and to achieve the necessary velocity reduction between compressor and combustion chamber in gas turbines. In air-conditioning systems and elsewhere, diffusers may be used to reduce the flow velocity and hence losses and noise. For convenience in installation and to reduce cost, small pipes may be used for transmission of fluids and are then connected to larger working components by a diverging section.

In all these cases the adoption of a “nearer-to-optimum” design, as opposed to one that simply satisfies the primary requirements, can lead to improvements in overall system performance. Inadequate design leads not only to poor pressure recovery, non-uniformity of exit flow and high energy losses but also to noise and vibration due to fluctuating separation. Furthermore, adverse effects on the performance of duct components downstream of the diffuser may be caused by separated, asymmetric or fluctuating flows.

This Data Item provides a comprehensive survey of diffuser performance which gives background information for non-specialists and enables specialists and non-specialists to assess the effects of geometrical and flow variables on diffuser performance. More detailed performance data for a range of particular diffuser geometries are given in the following Data Items.

ESDU 73024, Performance of conical diffusers in incompressible flow.

ESDU 74015, Performance in incompressible flow of plane-walled diffusers with single-plane expansion.

ESDU 75026, Performance of circular annular diffusers in incompressible flow.

In the present Item, Section 2 introduces some parameters used to define diffuser performance and Section 3 briefly discusses some analytical performance-prediction methods. Section 4 describes the principal features of performance trends with overall diffuser geometry and includes comments on the effects of cross-sectional shape and surface finish while Section 5 covers the effect of inlet flow conditions on performance. Section 6 summarises briefly some methods that have been suggested for improving diffuser performance, especially of wide-angle diffusers, and guides the user to sources of more detailed information. Some of these methods have particular application as remedial measures for diffusers whose performance has been found to be unsatisfactory in service.

* External energy input or output may be used to improve diffuser performance, for example by removal of potentially separating boundary layers.

2. NOTATION AND DEFINITIONS

2.1 Notation

		<i>SI units</i>
A	cross-sectional area of duct	m^2
A_E	effective flow area, see Section 2.3.3.1	m^2
B	blockage ratio, see Section 2.3.3.1	–
b	width of rectangular diffuser (between parallel walls for single-plane expansion), see Sketches 2.2 and 2.3	m
C_{pr}	static-pressure recovery coefficient, see Section 2.3.1	–
C_{pr}^*	optimum value of C_{pr} for given length ratio (L/R_1 , L/w_1 and L/h_1 for conical, plane-walled with single-plane expansion and annular diffusers respectively)	–
C_{pr}^{**}	optimum value of C_{pr} for given area ratio A_2/A_1	–
C_{prI}	value of C_{pr} for an ideal diffusion process, see Section 2.3.1	–
D	diameter of circular section	m
D_h	equivalent or hydraulic diameter, $4A/\text{cross-sectional perimeter}$	m
h	annulus width, $R_o - R_i$	m
K_t	total-pressure loss coefficient, see Section 2.3.2	–
K_t^*	minimum value of K_t for given length ratio (L/R_1 , L/w_1 and L/h_1 for conical, plane-walled with single-plane expansion and annular diffusers respectively)	–
K_t^{**}	minimum value of K_t for given area ratio A_2/A_1	–
L	axial length of diffuser	m
L_d	length of straight, parallel duct downstream of diffuser (tailpipe)	m
L_u	length of straight, parallel duct upstream of diffuser	m
m	total mass flow rate through section	kg/s
p	static pressure	$^*\text{Pa}$
p_t	total pressure	$^*\text{Pa}$
R	radius of circular duct	m

For footnotes see end of Notation Section.

Re	Reynolds number, $\bar{V}_1 D_1 / \nu_1$, $\bar{V}_1 w_1 / \nu_1$, and $\bar{V}_1 h_1 / \nu_1$ for conical, plane-walled with single-plane expansion and annular diffusers respectively	–
r	ratio of inner to outer wall radii (diameters) for annular cross section, R_i / R_o	–
V	local absolute velocity, $\sqrt{v_x^2 + (v_y^2 + v_z^2)}$	m/s
$\bar{V}$	mean velocity, $m / \rho A$, (i.e. area-averaged mean axial velocity)	m/s
v	component of local velocity	m/s
w	width of rectangular duct (between diverging walls for single-plane expansion), see Sketches 2.2 and 2.3	m
x	axial distance measured from start of area change, positive downstream, see Sketch 2.5	m
x_ε	distance from diffuser entry at which roughness commences (see Section 4.6.3)	m
α	kinetic energy coefficient, see Section 2.3.2	–
ε	equivalent roughness height, see ESDU 66027 (Reference 1) for typical values	m
η	diffuser effectiveness, see Section 2.3.1 (also called static-pressure recovery efficiency)	–
μ	dynamic viscosity of fluid	N s/m ²
ν	kinematic viscosity of fluid, μ / ρ	m ² /s
ρ	density of fluid	kg/m ³
ϕ	angle between diffuser wall and diffuser axis (total divergence angle = 2ϕ for symmetrical diffusers)	degree
ψ	angle between local flow direction and axial direction (swirl angle)	degree

* 1 Pa is 1 Pascal = 1 N/m².

Suffixes

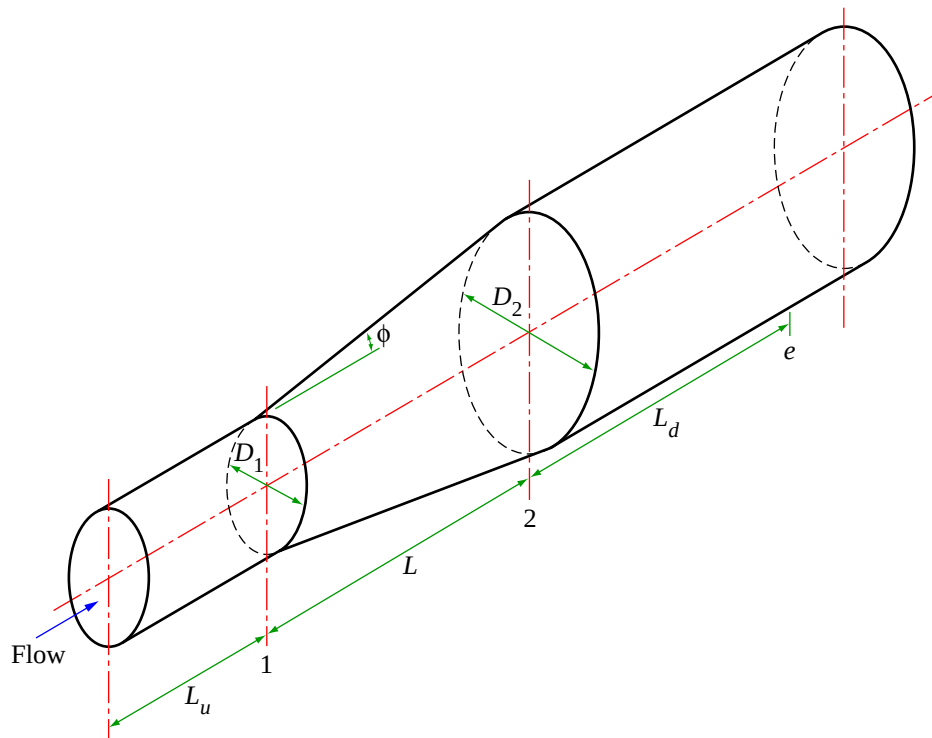
e	denotes exit plane for system considered
i	denotes inner wall of annular duct
max	denotes maximum value at a cross section
o	denotes outer wall of annular duct
x	value in axial direction
y	value in radial direction (axisymmetric sections) or in plane of dimension “ b ” (rectangular sections)
z	value in circumferential direction (axisymmetric sections) or in plane of dimension “ w ” (rectangular sections)
1	denotes inlet plane of diffuser
2	denotes exit plane of diffuser

Mean values are denoted by a bar over a symbol, *e.g.* $\bar{p}_t$. With the exception of the mean velocity, $\bar{V}$, the averaging method is taken to be a mass flow average, *i.e.*

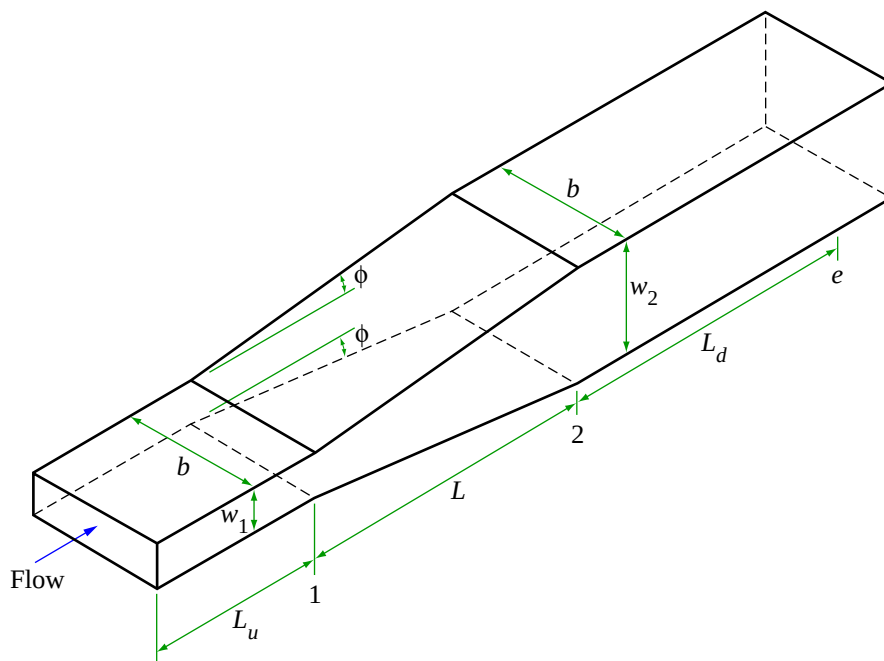
$$\bar{F} = \frac{1}{m} \int_A F dm ,$$

where F is any variable.

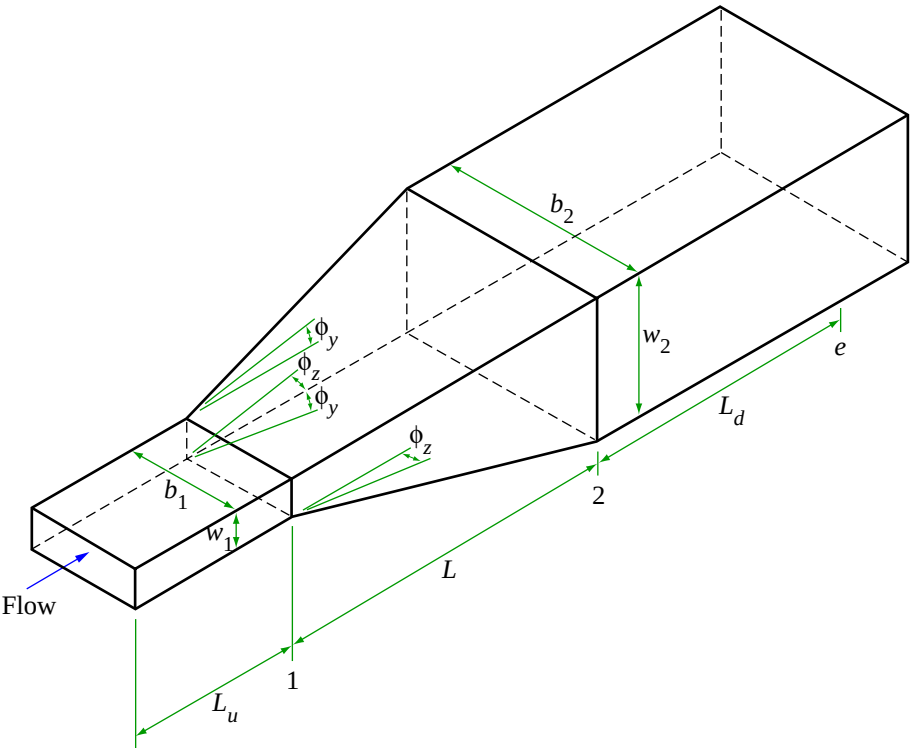
2.2 Diffuser Geometry



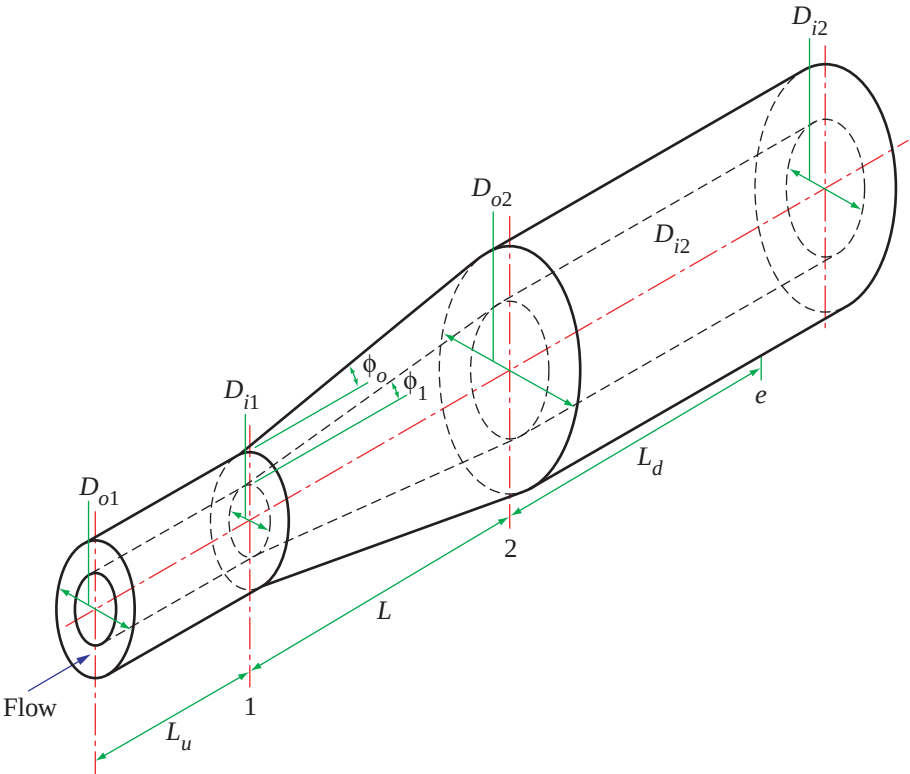
Sketch 2.1 Conical (ESDU 73024)



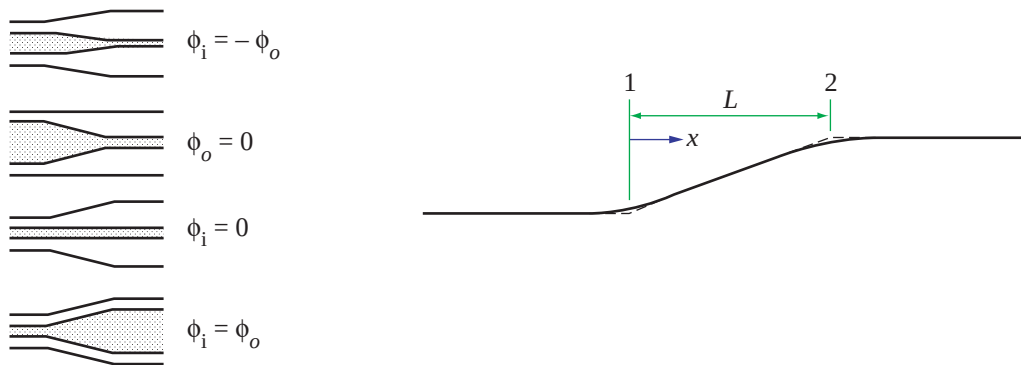
Sketch 2.2 Plane-walled, single-plane expansion (ESDU 74015)



Sketch 2.3 Plane-walled, two-plane expansion (Reference 6, square section only)



Sketch 2.4 Annular (ESDU 75026)



Sketch 2.5 Some special annular types (ESDU 75026) and definition of inlet and outlet planes for diffusers with rounded junction

2.3 Diffuser Performance Parameters

Diffuser performance is usually expressed in terms of the static-pressure rise across the diffuser with, less often, information on the total-pressure loss and the outlet flow conditions.

Some simple diffuser performance parameters are defined in this Section. They are used in the Data Items (see References 3, 4 and 5) as they can be derived and applied when few details of the inlet flow are known. If more detail were available these definitions could still be conveniently used but the performance parameters could then be correlated in more detail with the inlet flow variables.

Other more complex parameters have been proposed, often with the aim of defining a measure of efficiency of diffusion, generally by using a nearer approximation to the specific kinetic energy of the flow than the product $\frac{1}{2}\rho\bar{V}_1^2$. Some examples are given in References 7, 31, 45, 61 and 68.

Definitions of performance parameters for diffusers with swirling inlet flow or with auxiliary flows (suction or blowing) may or may not debit the diffuser performance with the power required to generate these flows. Since the swirl may be a by-product of other processes and the auxiliary flows may serve useful functions in the overall duct system, it appears preferable to present diffuser performance alone and make separate allowance in the system calculation for these other considerations.

2.3.1 Static-pressure rise, incompressible flow

The static-pressure rise between the inlet section 1 and an outlet section e (at the diffuser exit or some specified point along the tailpipe) is usually non-dimensionalised with reference to a mean inlet kinetic pressure, $\frac{1}{2}\rho\bar{V}_1^2$, to give the static-pressure recovery coefficient, C_{pr} , as

$$C_{pr} = \frac{\bar{p}_e - \bar{p}_1}{\frac{1}{2}\rho\bar{V}_1^2}. \quad (2.1)$$

One definition of the diffuser effectiveness, η , compares the actual static-pressure rise with that which could be achieved with the same entry conditions and the same geometry if the flow were frictionless and the exit flow completely uniform. Then η can be expressed as C_{pr}/C_{prI} where

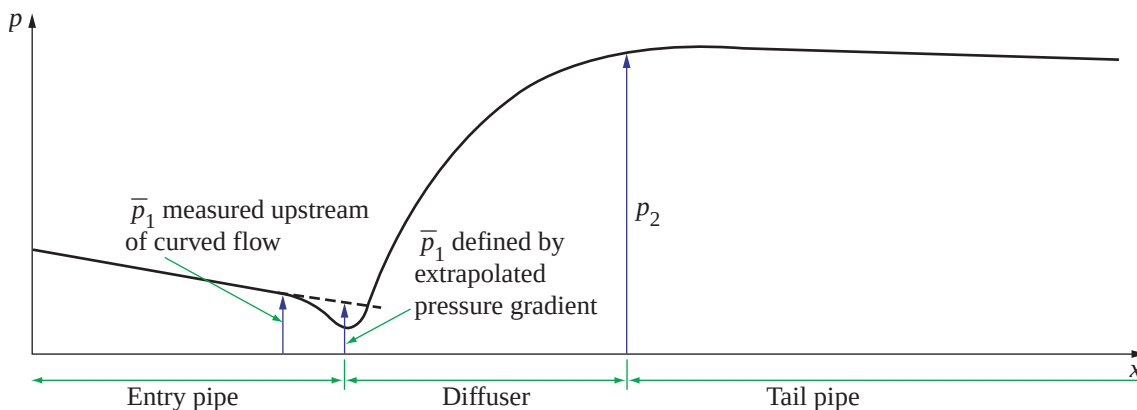
$$C_{prI} = \frac{\alpha_1(\frac{1}{2}\rho \bar{V}_1^2) - \frac{1}{2}\rho \bar{V}_2^2}{\frac{1}{2}\rho \bar{V}_1^2} = \alpha_1 - \frac{1}{(A_2/A_1)^2}, \quad (2.2)$$

where α_1 is the inlet kinetic energy coefficient defined by Equation (2.7).

An alternative version simplifies this by assuming a uniform inlet velocity of $\bar{V}_1$ so that $\alpha_1 = 1$ and

$$C_{prI} = 1 - \frac{1}{(A_2/A_1)^2}. \quad (2.3)$$

In most cases it is assumed that the static pressures are constant across sections 1 and 2. However, because of flow curvature in the region of the geometrical throat, the static pressure measured at the wall at section 1 is below the mean to an extent that depends on the throat shape and diffuser angle. It is usual therefore to define $\bar{p}_1$ as the static pressure at the wall a short distance upstream of the geometrical throat or sometimes as the static pressure obtained at section 1 by extrapolating the general trend of the static pressure gradient in the upstream duct. Sketch 2.6 shows an example of the wall static-pressure distribution along a duct-diffuser system which illustrates these points. If another duct component is placed very close to the diffuser exit this may impose a particular exit pressure distribution.



Sketch 2.6 Variation of wall static pressure along a diffuser

In the case of swirling inlet flows or curved inlet flow (as at a bend exit for example) the static pressure will vary across the section and mean values must be determined from a static pressure profile.

2.3.2 Total-pressure loss, incompressible flow

The total-pressure loss may be similarly referenced to the mean inlet kinetic pressure to give the total-pressure loss coefficient, K_t , as

$$K_t = \frac{\bar{p}_{t1} - \bar{p}_{te}}{\frac{1}{2}\rho \bar{V}_1^2}. \quad (2.4)$$

The mean total pressure is taken as the mass-averaged value given by

$$\bar{p}_t = \frac{1}{m} \int_A p_t dm. \quad (2.5)$$

In cases where the static pressure is sensibly constant across the section this can be evaluated as

$$\bar{p}_t = p + \alpha(\frac{1}{2}\rho \bar{V}^2), \quad (2.6)$$

in which α , the kinetic energy coefficient, relates the mass-averaged kinetic pressure to the product $\frac{1}{2}\rho \bar{V}^2$, i.e.,

$$\alpha = \left(\frac{1}{m} \int_A \frac{1}{2}\rho V^2 dm \right) / (\frac{1}{2}\rho \bar{V}^2). \quad (2.7)$$

For axial incompressible flows Equation (2.7) reduces to

$$\alpha = \frac{1}{A \bar{V}^3} \int_A V^3 dA. \quad (2.8)$$

The total-pressure loss between the inlet and exit planes of a diffuser or diffuser/tailpipe system can also be expressed in terms of the change in kinetic energy and the change in static pressure. Thus, using mean values,

$$\bar{p}_{t1} - \bar{p}_{te} = \alpha_1(\frac{1}{2}\rho \bar{V}_1^2) - \alpha_e(\frac{1}{2}\rho \bar{V}_2^2) - (\bar{p}_e - \bar{p}_1), \quad (2.9)$$

from which, using Equations (2.1) and (2.4),

$$K_t = \alpha_1 - \frac{\alpha_e}{(A_2/A_1)^2} - C_{pr}. \quad (2.10)$$

The term $\alpha_e/(A_2/A_1)^2$ represents the kinetic energy remaining in the flow at the exit plane. If the diffuser or diffuser/tailpipe system discharges into a large space (*e.g.* the atmosphere), so that this kinetic energy is not utilised in another duct component, the overall total-pressure loss coefficient between the inlet plane and the large space will be given by

$$K_t = \alpha_1 - C_{pr}. \quad (2.11)$$

When the diffuser is followed by another duct component a value of α_e is required to evaluate Equation (2.10). If a long enough tailpipe is fitted (of the order of $4D_{h2}$ long, but dependent on diffuser geometry and inlet flow conditions), the distorted flow at the diffuser exit will decay to a fully-developed flow condition at the tailpipe exit and a value of α_e of about 1.05 may be taken for the diffuser/tailpipe combination. For shorter tailpipes the flow at the tailpipe exit will remain distorted and higher values of α_e (in extreme cases as high as 4⁹²) will apply. Exceptionally, if the diffuser is of small area ratio and wall angle and the inlet flow is near uniform, the flow may still be developing at the diffuser exit and in the tailpipe. A value of α_e of about 1.04 will then be a reasonable approximation.

Equation (2.10) might suggest that a high value of α_e gives a low total-pressure loss but in practice diffusers producing highly distorted flows also give low static-pressure recovery. A high value of α_e is also undesirable because it implies a very non-uniform exit flow which is likely to have an adverse effect on the performance of the next duct component downstream.

2.3.3 Outlet flow specification

The information available on the outlet flow from a diffuser or diffuser-duct combination is often restricted to general comments on flow stability and symmetry. These may be correlated in the form of a series of overall “flow regimes” in which the diffuser geometry lies, as discussed in Section 4.1.

Detailed outlet flow measurements are of value in providing a data base from which typical values of outlet flow parameters can be derived. In the practical design situation difficulties in ensuring an exact correspondence of a diffuser in service conditions with the design condition would not allow the detailed measurements to be applied with certainty. However, information on some gross parameters of the flow is of use in design of downstream duct components when these are in close proximity to the diffuser (say closer than $10D_{h2}$) and in evaluating diffuser performance. Particularly useful for this latter purpose (see Sections 2.3.2 and 2.3.4) is the kinetic energy coefficient, α , defined by Equation (2.7).

Some other useful parameters might be the blockage parameter (defined in Section 2.3.3.1), maximum/mean velocity and location of maximum velocity (for asymmetry definition), momentum coefficient, axial turbulence intensity and mean swirl angle. The flow at a diffuser exit often fluctuates with time so that these values will then be averages and a measure of the fluctuation frequency will then also be required.

2.3.3.1 The blockage parameter, B

An effective flow area, A_E , of a duct cross section can be defined as the area that could pass the actual mass flow rate of the system if the velocity were uniform and equal to the actual maximum velocity in the cross section. Thus

$$A_E = m / (\rho V_{max}). \quad (2.12)$$

The effective area will be equal to the actual duct area only if the velocity is actually uniform across the section. For all other cases A_E is less than A and the difference, $A - A_E$, is often termed the “blocked area”. A parameter characterising the non-uniformity of the velocity profile, the “blockage parameter” B , is then defined as the ratio of the blocked area to the actual duct area, so that,

$$B = \frac{A - A_E}{A} = 1 - \frac{m_{actual}}{\rho A V_{max}} = 1 - \frac{\bar{V}}{V_{max}}. \quad (2.13)$$

The blockage ratio can also be expressed in terms of the boundary-layer displacement thickness (using an axisymmetric or three-dimensional definition, as for example in Reference 45, for axisymmetric sections). The exact relation depends on the cross section.

2.3.4 Insufficient and inefficient diffusion

Equation (2.10) can be rearranged to give the static-pressure recovery coefficient as

$$C_{pr} = \alpha_1 - \frac{\alpha_e}{(A_2/A_1)^2} - K_t, \quad (2.14)$$

while Equation (2.2) expresses the “ideal” pressure recovery coefficient as

$$C_{prI} = \alpha_1 - \frac{1}{(A_2/A_1)^2};$$

so that

$$C_{prI} - C_{pr} = \frac{(\alpha_e - 1)}{(A_2/A_1)^2} + K_t. \quad (2.15)$$

From this it can be seen that the actual pressure recovery is less than the ideal value because of a lack of uniformity (since $\alpha_e > 1$) in the exit velocity profile and a loss of total pressure. Reference 45 terms these “insufficiency of diffusion” and “inefficiency of diffusion” respectively. Reference 45 also derives an equation similar to Equation (2.15) but in terms of the total-pressure loss along the streamline of maximum velocity, showing that for diffusers in which an inviscid core exists throughout a diffuser the reduction in pressure recovery below the ideal value is predominantly due to insufficient diffusion rather than inefficient diffusion.