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Introductory memorandum on the pressure losses in internal flow systems

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THE PREPARATION OF THIS DATA ITEM

This Memorandum was originally issued as ESDU 69016 in July 1969 and amended in October 1973 and October 1974. The Committee responsible for the preparation of the Memorandum was the Aerodynamics Committee of the Aeronautical Society and the Memorandum was accepted for inclusion in the Mechanical Engineering Series by the Fluid Mechanics and Thermodynamics Committee of the Institution of Mechanical Engineers. The inclusion of the Memorandum in the Chemical Engineering Series was approved by the Institution of Chemical Engineers.

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The Memorandum has been further amended and re-numbered FMI1. The work of the permanent professional staff of the Engineering Sciences Data Unit on this amendment was monitored and guided by the Internal Flow Panel which had the following constitution:

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INTRODUCTORY MEMORANDUM ON THE PRESSURE LOSSES IN INTERNAL FLOW SYSTEMS

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INTRODUCTORY MEMORANDUM ON THE PRESSURE LOSSES IN INTERNAL FLOW SYSTEMS

1. INTRODUCTION

The main aims of this Item are

- (i) to serve as an introduction to a series of Data Items, in which detailed numerical data are given,
- (ii) to provide guidance on simplified methods of calculating pressure losses,
- (iii) to discuss some of the more general aspects of the estimation of pressure losses in duct flow systems,
- (iv) to draw attention to the limitations that practical factors impose on the precision with which calculations can be made,
- (v) to present conversion factors for some of the quantities that appear in pressure loss calculations.

This Item covers a wide range of practical internal flow problems, but it is restricted to the steady, single-phase* flow of Newtonian fluids. See Section 9 for information on Items on non-Newtonian flow, fans, pumps, noise in air-conditioning systems, and fluid transients and their interaction with the pipeline.

Although variations of flow and fluid properties occur at all cross sections of a real flow, it is usual in practice to work in terms of averaged quantities. The equations given in this Item are based on the use of such quantities.

* A glossary of terms is provided in Section 10.

2. NOTATION AND UNITS

A coherent set of units is required for use in the equations quoted in this Memorandum. Three such sets are given below.

		Units		
		SI	British	
A	internal cross-sectional area of duct	m^2	ft^2	ft^2
a	speed of sound, $\sqrt{\gamma RT}$	m/s	ft/s	ft/s
c_p	specific heat capacity at constant pressure	J/kg K	$^*\text{ft pdl/lb}^\circ\text{F}$	$^\dagger\text{ft lbf/slug K}$
c_v	specific heat capacity at constant volume	J/kg K	$^*\text{ft pdl/lb}^\circ\text{F}$	$^\dagger\text{ft lbf/slug K}$
D	internal diameter of duct of circular cross section	m	ft	ft
D_E	equivalent or hydraulic diameter, $4A/P$	m	ft	ft
f	Fanning friction factor (see Equation (6.10))			
g	local acceleration due to gravity	m/s^2	ft/s^2	ft/s^2
H	total head of fluid flowing	m	ft	ft
ΔH	loss of total head of fluid flowing	m	ft	ft
K_H	pressure-loss coefficient or pressure-drop coefficient [‡] based on differences in total pressure			
K_s	pressure-drop coefficient [‡] based on differences in static pressure			
L	axial length of component	m	ft	ft
M	Mach number, V/a			
m	mass flow rate	kg/s	lb/s	$^\dagger\text{slug/s}$
P	internal wetted perimeter	m	ft	ft
p	static pressure (absolute)	N/m^2	$^*\text{pdl/ft}^2$	lbf/ft^2
p_H	total pressure (absolute)	N/m^2	$^*\text{pdl/ft}^2$	lbf/ft^2
Δp	static pressure drop	N/m^2	$^*\text{pdl/ft}^2$	lbf/ft^2

For footnotes see end of Notation.

Δp_H	total pressure loss	N/m ²	*pdl/ft ²	lbf/ft ²
Q	volume flow rate, m/ρ	m ³ /s	ft ³ /s	ft ³ /s
R	gas constant**	J/kg K	*ft pdl/lb°F	†ft lbf/slug K
Re	Reynolds number, $\rho VD/\mu$			
T	static temperature	K	°R	K
T_H	total temperature	K	°R	K
u	local absolute velocity	m/s	ft/s	ft/s
V	mean flow velocity, Q/A	m/s	ft/s	ft/s
x	distance along axis of straight duct of constant cross section	m	ft	ft
z	vertical distance above arbitrary datum plane	m	ft	ft
α	kinetic energy coefficient, $\alpha = \left(\frac{1}{m} \int_A \frac{1}{2} \rho u^2 dm \right) / \left(\frac{1}{2} \rho V^2 \right),$ see Equation (6.3)			
γ	ratio of specific heat capacities, c_p/c_v			
ε	effective roughness height of surface	m	ft	ft
Φ	heat flow rate	J/s	*ft pdl/s	ft lbf/s
ρ	density	kg/m ³	lb/ft ³	†slug/ft ³
μ	dynamic viscosity	N s/m ² or kg/m s	*pdl s/ft ² or lb/ft s	lbf s/ft ² or †slug/ft s
λ	friction factor, $4f$			

* 1 pdl = 31.08×10^{-3} lbf, 1 pdl/ft² = 215.8×10^{-6} lbf/in².

† 1 slug = 32.17 lb.

‡ The symbol C_K is also used instead of K .

** R = universal gas constant/molecular weight. Universal gas constant = 8314 J/kg-mol K = 1545 ft lbf/lb-mol °F.

Suffixes

1	appropriate to upstream reference conditions
2	appropriate to downstream reference conditions
<i>N</i>	net value
<i>G</i>	gross value
<i>L</i>	appropriate to axial length of component

3. CONVERSION FACTORS

Conversion factors for pressure, volume flow rate, specific heat capacity, density and dynamic viscosity are given in Tables 13.1 to 13.5. These Tables allow conversion between the sets of units given in Section 2 and between those units and other commonly used units.

4. DIMENSIONAL CONSIDERATIONS

Dimensional analysis shows that, for a pipe system of given geometry running full, the pressure drop, Δp , between any two points is related to the pipe geometry, flow properties and fluid properties in the following way

$$\frac{\Delta p}{\frac{1}{2}\rho V^2} = \text{function} \left(\frac{\varepsilon}{D}, \frac{\rho V D}{\mu}, \frac{V}{a}, \gamma, \text{geometry} \right). \quad (4.1)$$

The quantity $\frac{1}{2}\rho V^2$ is known as the dynamic pressure and the ratio $\Delta p/(\frac{1}{2}\rho V^2)$ is known as the pressure loss coefficient or pressure drop coefficient, denoted by the symbol K^* . The term ε/D is a measure of the roughness of the duct surface. The ratio $\rho V D/\mu$ is called the Reynolds number, Re . It has been found in tests on pipes of constant cross section that, for fully-developed flow, if $Re < 2000$ the flow is laminar and if $Re > 4000$ the flow is usually turbulent[†]. Transition from laminar to turbulent flow takes place over the range $2000 < Re < 4000$.

For those gas flows in which the effects of compressibility are important (see Section 8), two additional non-dimensional parameters have special significance, namely the Mach number, M , and the ratio of the specific heat capacities, γ .

When considering the flow in ducts of non-circular cross section it is convenient to use the equivalent or hydraulic diameter given by

$$D_E = \frac{4A}{P}. \quad (4.2)$$

For ducts of circular section, $D_E = D$.

* C_K is sometimes used.

† Under laboratory conditions laminar flow can be maintained to Reynolds numbers much higher than 4000.

5. THE CONTINUITY EQUATION

The rate of mass flow through a pipe is simply related to the mean flow velocity, V , and density, ρ , by the expression

$$m = \rho AV. \quad (5.1)$$

This relation applies both to incompressible and compressible flows.

In incompressible flow the continuity equation can be written in the simple form

$$Q = AV, \quad (5.2)$$

where Q is the volumetric flow rate.

In incompressible flow in a pipe of given dimensions these equations enable V to be calculated immediately if ρ and m or Q are known; hence the dynamic pressure $\frac{1}{2}\rho V^2$ can be evaluated directly. For compressible flows, the local density is related to the flow velocity, so to evaluate $\frac{1}{2}\rho V^2$ further relationships are required. These are given in Section 8.

6. PRESSURE LOSS CALCULATIONS IN INCOMPRESSIBLE FLOW

6.1 Conditions for Which the Flow May Be Assumed Incompressible

For the purposes of calculating pressure losses, it can be assumed that, under conditions of steady flow, liquids are incompressible, *i.e.* the density is constant.

For gases, over wide ranges of temperature and pressure, density is given by

$$\rho = \frac{p}{RT}. \quad (6.1)$$

From this relation, it is clear that, if pressure and temperature changes within the flow are small, the density changes are also small, and the flow may be assumed to be incompressible. In the absence of heat transfer, for the low-speed flow of gases this condition will, in practice, usually be satisfied if the Mach number is everywhere less than about 0.2. If the maximum changes of temperature or pressure within the flow exceed $0.1T$ or $0.1p$ respectively, the errors introduced by assuming an incompressible flow analysis are likely to be significant and the compressible flow analysis of Section 8 should be used.

6.2 Basic Procedures for Pressure Loss Calculations

6.2.1 Coefficients based on differences in total pressure

The total-pressure loss (or total-pressure drop, as it is also known) between any two stations 1 and 2 along a pipe is proportional to $\frac{1}{2}\rho V^2$ and can be written in the form

$$p_{H1} - p_{H2} = \Delta p_H = \frac{1}{2}\rho V^2 \times K_H, \quad (6.2)$$

where K_H is a non-dimensional parameter known as the total-pressure loss coefficient and p_H is the total pressure, given by the relation

$$p_H = p + \alpha \frac{1}{2} \rho V^2 + \rho g z. \quad (6.3)$$

The term, α , in Equation (6.3) allows for the difference between V^2 and the integral $(1/m) \int_A u^2 dm$ which is representative of the flow's kinetic energy. For most flow conditions, α may be approximated to unity and in practice, Equation (6.3) is nearly always expressed as

$$p_H = p + \frac{1}{2} \rho V^2 + \rho g z \quad (6.4)$$

because information on α is seldom available.

Equation (6.2) may be expressed in the alternative form

$$H_1 - H_2 = \Delta H = K_H V^2 / 2g, \quad (6.5)$$

where the total head, H , is given by

$$H = \frac{p}{\rho g} + \frac{V^2}{2g} + z. \quad (6.6)$$

For the flow of gases, the term in z is generally negligible and can be omitted from the equations.

When the cross-sectional areas at the two points between which the pressure loss, Δp_H , is quoted are different, the pressure loss coefficient may be expressed in terms of the value of $\frac{1}{2} \rho V^2$ at either point. Thus, the pressure loss coefficient may be referred to either station 1, *i.e.*

$$(K_H)_1 = \frac{\Delta p_H}{\frac{1}{2} \rho V_1^2} = \frac{\Delta H}{V_1^2 / 2g}, \quad (6.7)$$

or to station 2, *i.e.*

$$(K_H)_2 = \frac{\Delta p_H}{\frac{1}{2} \rho V_2^2} = \frac{\Delta H}{V_2^2 / 2g}. \quad (6.8)$$

For pressure losses between points of different areas, care must be exercised to ensure that the correct reference value of $\frac{1}{2} \rho V^2$ is used when interpreting the pressure loss coefficient data. When the areas are equal no such problem of interpretation arises.

6.2.2 Losses in straight pipes

For the flow in straight pipes of constant cross section, the pressure loss is directly proportional to the length, x , of the pipe. Hence data for straight pipes are usually presented using the Fanning friction factor, f , which is defined by the relation

$$f = \frac{\Delta p_H}{\frac{1}{2}\rho V^2} \times \frac{D_E}{4x} = \frac{\Delta H}{V^2/2g} \times \frac{D_E}{4x}. \quad (6.9)$$

The friction factor, λ , is often used instead of f . These parameters are related by the expression

$$\lambda = 4f. \quad (6.10)$$

6.2.3 Losses in systems with no interference between components

For given reference values of $\frac{1}{2}\rho V^2$ and Re , the loss in total pressure between two points along a pipe can, as a working approximation, be considered to be a function of the duct geometry and surface finish between the two points only, *i.e.* it can be assumed that conditions upstream and downstream of the region of interest can be ignored, and that velocity profile effects are unimportant. This approximation leads to a particularly simple means of estimating the pressure loss in a system, since it follows that a component of given geometry and surface roughness has a particular loss coefficient. Thus the overall loss through a system can be evaluated by summing the losses through the individual components.

This procedure is the one that is conventionally used in design work, but it is stressed that implicit in such a calculation procedure there are certain assumptions that are discussed in more detail in Section 6.3.

6.3 Other Factors Affecting Pressure Loss Calculations

The method of calculating pressure loss given in the previous Section serves for many practical purposes, but in some circumstances that approach is not adequate. Certain additional factors are important and these are discussed below.

6.3.1 The region of influence of a component: interference-free flow

Consider first of all the flow in a straight pipe of constant cross section. If the pipe is long enough, the velocity profile downstream becomes independent of x and the flow is described as fully-developed. The static pressure is constant at a cross section (ignoring the variations due to gravity which can occur in liquid flows) and decreases linearly with increasing x . Thus the region of influence of a component can be determined experimentally by attaching straight pipe to the exit and entry of the component of sufficient length for fully-developed conditions to be attained upstream and downstream. Consider, for example, the pressure distribution along a pipe line containing a bend for incompressible turbulent flow (Figure 1, taken from Reference 3). The variations in static pressure that are present across a section within the bend extend for a diameter or two into the straight pipes upstream and downstream. The constant pressure gradient associated with fully-developed flow in a straight pipe is not re-established until 50 or more diameters downstream from the bend, although after 10 diameters a constant pressure gradient may usually be assumed with no significant loss in accuracy.

When the flow upstream and downstream of a component is fully-developed, the component is said to be free of interference effects, since its performance is independent of the flow beyond the regions of fully-developed flow.

6.3.2 Coefficients based on differences in static pressure

In real flows the total pressure varies over the cross section and it is, in practice, often simpler to define unambiguously the mean static pressure rather than the mean total pressure at a section. Also there are advantages in using this approach when the problem of interference is considered (see later). Factors such as these have led to the use of loss coefficients based on changes in static pressure.

There are certain points that have to be observed when using these coefficients. Firstly, it is important in experimental work to ensure that the correct static pressure is measured and that it is not influenced by local curvature of the flow. Secondly, in calculating the performance of a duct system, it is usually the total-pressure loss that is required and hence it is necessary to convert from static to total pressures. This can be done very simply by calculating the total pressures as below.

For liquid flows, the sum $(p + \rho gz)$ is constant at a section in a straight pipe away from the influence of area change and bends. Since the contribution of the term ρgz depends on the orientation of the pipework, it is convenient to quote data in terms of static pressure differences between two points assuming the two points are in the same horizontal plane. Loss coefficient data (based on static pressure differences) can then be quoted in an identical form for both liquids and gases and the contribution of ρgz in liquid flows accounted for independently.

The static-pressure drop coefficient is defined as

$$(K_s)_1 = \frac{p_1 - p_2}{\frac{1}{2}\rho V_1^2}, \quad (6.11)$$

based on upstream conditions, or

$$(K_s)_2 = \frac{p_1 - p_2}{\frac{1}{2}\rho V_2^2}, \quad (6.12)$$

based on downstream conditions.

A useful way of defining a corresponding total-pressure loss coefficient is as follows. The local total pressure is defined as

$$p_H = p + \frac{1}{2}\rho V^2. \quad (6.13)$$

Since, from the continuity principle,

$$\frac{V_1}{V_2} = \frac{A_2}{A_1}, \quad (6.14)$$