Chapter 7
Fluid Flow

A GUIDE
TO
CURRENT PRACTICE
1959

13.10 FLUID FLOW

Leonardo da Vinci (1452-1519) gave the first recorded sketches of the form of a liquid jet issuing from an orifice; he also proposed a design for an anemometer. Newton (1642-1727) measured the dimensions of the *vena contracta* and introduced the coefficient of discharge (which he set equal to $1/\sqrt{2}$) to make the theory accord with experimental observations.(54) In 1718, Polavi determined the coefficient experimentally, obtaining the value 0.62 — an improvement on Newton's value. Later determinations of the coefficient for a thin orifice were made, *inter alia*, by Girard in 1821 (0.725), Lagerhelm in 1822 (0.58), Aubuisson in 1826 (0.65) and Péclet (0.65). Flow through a mouthpiece was studied by Aubuisson, Eytelwein and Péclet.

The development of fluid machines and the understanding of fluid mechanics was given considerable impetus by Torricelli's invention of the mercury barometer (1644) and by Pascal's experiments with various liquids (1647), when he concluded that "in a fluid at rest the pressure is exerted equally in all directions". In 1686, Marriotte, from his experiments on water jets, first appreciated that the force exerted by a stream of water is proportional to the square of the velocity of flow. He also noted the resistance to flow in pipes and the increased resistance due to sudden changes of direction. Substantial contributions to the understanding of fluids were made by Sir Isaac Newton in his *Principia Mathematica*. Newton dealt with viscous shear in fluids, and flow around objects immersed in a moving fluid.

7. Fluid Properties.

Values of density and absolute viscosity for a selection of liquids and gases are tabulated in Section XIII. Details of the variation in kinematic viscosity due to temperature, for a limited selection of fluids, are shown in Fig. 2.

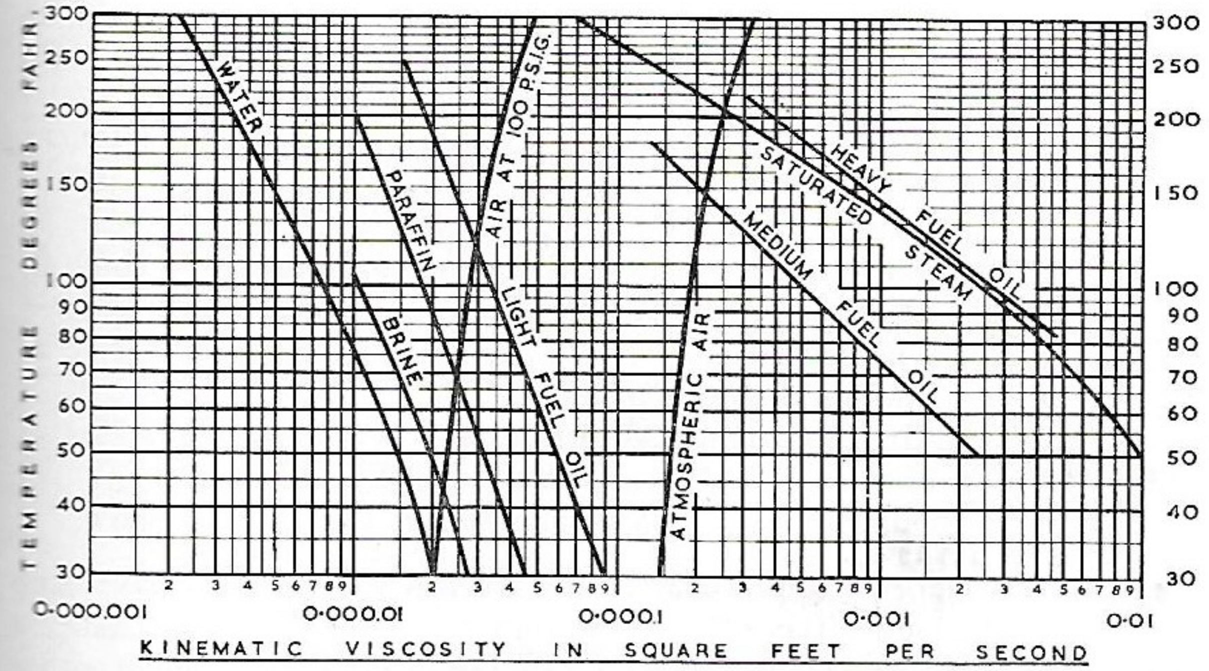

Fig. 2

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The pitot tube was invented by Pitot in 1732: he used two separate tubes, one being bent at 90°, to give the facing and static pressures in a fluid stream. In 1884, the Prussian Mining Commission investigated various methods of measuring air speed. The accuracy of the pitot tube was verified; and they found too that a thin plate orifice could be used, the volume being given by:

$$Q = C\,A\,\sqrt{gH}$$

where H = pressure drop (ft of air)
A = area of orifice
C = discharge coefficient, = 0.64 for a round orifice
= 0.61 for a square orifice.

In the same year that the pitot tube was invented, Couplet made a study of the flow of water in the pipe system serving the fountains at Versailles. Martin[41] regards these as the first useful experimental data; from them, Couplet concluded that the loss of head was proportional to the square of the velocity.

Antoine Chézy experimented on water flow in pipes over a period of some years, and in 1775 gave the first rational statement on pipe friction. His expression for turbulent flow was:

$$v = C\,\sqrt{m_l\,i}$$

where v = fluid velocity
m_l = hydraulic mean depth
i = hydraulic gradient, head loss per unit length.

The expression was modified by Prony in 1794, who suggested that the resistance depended on both the first and second powers of the velocity.[54] Neither Chézy nor Prony took any account of the surface characteristics of the pipe.

In 1738, Daniel Bernoulli published his *Hydrodynamica*, containing the broad outline of the now famous Bernoulli equation. The principles were later used by the Italian physicist Venturi in 1797 in the construction of the flow meter which bears his name.

Rather earlier, in 1766, Borda determined the energy loss which occurs at a sudden contraction or other change of section. He seems to have been the first to include the factor $2g$ explicitly in a flow equation.

The law for laminar flow in a tube was deduced by Poiseuille (1841) from experimental work; the expression was derived theoretically by Neumann and Hagenbach in 1858-60.

Aubuisson put forward in 1834 some principles of fluid flow, which led to a formula of the type:

$$H = a\,L \cdot \frac{v^2}{g m_1}$$

In 1854 Hagen gave, for the resistance in turbulent flow:

$$H = \rho\,\frac{L\,v^{1\cdot 75}}{D^{1\cdot 25}}$$

thus predating some of Osborne Reynolds' discoveries. Hagen, considering his own experiments and those of earlier investigators, concluded that frictional losses could be simply represented by using fractional indices. However, some years earlier (1845) Weisbach had proposed an expression of the form:

$$h = 4\zeta \cdot \frac{L}{D} \cdot \frac{v^2}{2g}$$

with the friction coefficient ζ varying with $1/\sqrt{v}$, thus allowing frictional losses to be expressed in terms of velocity head. This was akin to Girard's formula for the flow of gases, and to Aubuisson's.

This form of expression is perhaps more usually associated with D'Arcy (and in fact often bears his name), though Rouse and Ince state that D'Arcy's formula, published in 1857, included terms in both v and v^2 as well as a dimensional coefficient involving roughness. The ASHRAE Guide attributes the expression to Aubuisson.

Experiments by Weisbach, Ledoux, Rietschel, Unwin and others showed that the friction factor ζ varied with both velocity v and pipe diameter d.

Péclet, about 1860, carried out an extensive series of tests to determine the resistance of various pipe fittings (i.e. bends, valves, tees etc.) to the flow of water. Rietschel(52) expressed his own results in terms of velocity head loss ($v^2/2g$):

sharp elbow	1.0 velocity head
round elbow	0.5
return bend	0.8
sudden enlargement	1.0
open cock	0.1-0.3
open valve, ordinary seat	0.5-1.0

Although the second half of the 19th century was the age of steam power, remarkable advances were still made in the understanding of fluid flow and the development of fluid machinery. Stokes produced the law which bears his name and relates to the rate of fall of a solid sphere in a viscous fluid. Horace Lamb published his classic textbook on *Hydrodynamics* in 1879. But the outstanding investigator of this period was Osborne Reynolds.

In 1883, Reynolds discovered the two modes of motion in fluids known as "streamline" and "turbulent", and he went on to explain how the transition takes place at a "critical velocity". He was the first to show a definite relationship to exist between the frictional loss due to water flowing in a pipe and certain physical factors which can be expressed in the form of the dimensionless number vd/ν, the Reynolds Number.

His work enabled some of the apparent discrepancies in the earlier work of Poiseulle (1846) and Darch (1857) to be explained. It was around this period that William Froude and later his son Robert, carried out their experiments on the resistance of surfaces of different shapes and finishes being drawn through the water at different speeds.

For the flow of air, Girard (1821) demonstrated that:

$$v^2 = \frac{2g\ D\rho}{k\ l}$$

where k is a coefficient of friction. Péclet doubted Girard's suggestion that k depended on the material: he believed it to be a constant (0.024) for a range of pressures, pipe diameters and length, for both air and coal gas. Péclet's book[46] contains a very detailed account of his experiments on the flow of gases through orifices, changes of section, a succession of bends and so on. The trials, which were carefully executed with good accuracy, confirmed and extended the work of Aubuisson and of Eytelwein.

Apart from the relatively small quantity of data acquired prior to 1850, our knowledge of the friction of air in pipes and ducts may be said to date from Weisbach's experiments. The theory had been established by Montgolfier and Bernoulli. Box quoted the pressure loss for air flowing through a pipe as:

$$H = \frac{v^2 l}{(3.7d)^5}$$

where H = pressure loss, in w.g.
v = volume, ft^3/min
l = length of pipe, yards
d = pipe diameter, in.

The implicit assumption is that the friction factor is constant in all circumstances.

Weisbach's contribution was that of the experimental determination of the friction loss. His book on *Fluid Mechanics* was published in Germany in 1855. The pressure drop in a circular duct was:

$$H = 4\zeta \cdot \frac{l\ v^2}{2g\ d}$$

as for water in pipes, and he found that the value of 4ζ ranged from 0.015 to 0.026 for straight pipes of different kinds. A similar value was found by Ledoux. For 90° elbows, Weisbach found $4\zeta = 1.41$ to 1.61; and for long 90° bends, $4\zeta = 0.47$. Unwin found the coefficient 4ζ to vary with the diameter and the roughness. It will be noticed that there is as yet no realisation of the fact that 4ζ depends on the velocity or the Reynolds Number.

The Prussian Mining Commission found the resistance to air flow in cast iron pipe to be proportional to:

$$d^{-1.37} v^2\ \rho^{2/3}$$

while Rietschel's tests at Charlottenburg yielded:

$$R = 0.058\ v^{1.85}\ d^{-1.26}$$

where R = resistance, in wg/ft run
v = velocity, ft/s
d = diameter, in

and Fritzsche's work in 1907 gave:

$$R = \frac{b\ \rho^{0.852}\ v^{1.924}}{d^{1.281}} \quad (\rho \text{ in lb/ft}^3)$$

These expressions represent a return to the earlier empirical form.

Prandtl and his students were responsible for important advances in the theory of fluid flow and its experimental verification. Blasius (1908) published an analytical solution which put Prandtl's qualitative theory into quantitative terms; and this in turn was fully verified by later experiment. In his 1911 paper, on similarity laws in fluid flow, Blasius showed that the resistance coefficient for smooth pipes was a unique function of the quantity vd/ν (the Reynolds number) - $f = 0.0791\ Re^{-0.25}$. A subsequent paper, in 1913, contained a correlation plot based on data from Saph and Schoder at Cornell as well as Blasius' own on water and air.(54) A year later, Stanton and Pannell at the National Physical Laboratory extended the correlation to include data on oil.

Later work by Lees (1915), Nikuradse (1932), Colebrook and White (1937) and Moody (1944), and theoretical work by Prandtl and Karman (1930-2) has enabled f to be determined for smooth, commercial and fully rough pipes. For smooth pipes, the friction coefficient is related to the flow conditions by the expression:

$$\sqrt{1/f} = 4\ \log_{10}\left(\frac{Re \cdot \sqrt{f}}{1.255}\right)$$

while for fully rough pipes it is given by:

$$\sqrt{1/f} = 4\ \log_{10}\left(\frac{3.7d}{k}\right)$$

where k is the absolute roughness. The latter coefficient is thus independent of Reynolds number. The transition from smooth to fully rough pipes was studied by Colebrook and White, who proposed the equation:

$$\sqrt{1/f} = 4\ \log_{10}\left(\frac{1.255}{Re \cdot \sqrt{f}} + \frac{k}{3.7d}\right)$$

to cover the range of commercial pipes. These equations form the basis of most current fluid flow tables.

SECTION 3 THE FLOW OF FLUIDS IN PIPES AND DUCTS

3.1. NOTES AND FORMULAE

3.1.1. Straight Pipes and Ducts

The head lost due to friction may best be determined by the use of a rational formula:

$$H = \frac{4 f L u^2}{2 g D} \quad \text{......3.1}$$

where

H = Head lost, feet of fluid flowing	..	ft
f = Coefficient of friction, dimensionless.		
L = Length of pipe	..	ft
u = Velocity	..	ft/sec
g = Acceleration due to gravity ..	..	ft/sec^2
D = Internal diameter of pipe ..	..	ft

The coefficient of friction is a variable dependent upon:

(*a*) The physical characteristics of the fluid flowing, the velocity of flow and the internal diameter of the pipe, which three components may be combined for consideration in terms of Reynolds Number, a dimensionless quantity:

$$R_N = \frac{u D \rho}{\mu} = \frac{u D}{\nu} \quad \text{......3.2}$$

where

R_N = Reynolds Number.	
ρ = Density of fluid	.. lb/ft^3
μ = Absolute viscosity of fluid ..	.. lb/ft sec
ν = Kinematic viscosity of fluid	.. ft^2/sec

(*b*) The roughness of the pipe wall relative to the internal diameter, which is expressed in terms of a dimensionless ratio, k_s/D, where k_s is a lineal measure of absolute roughness having the same dimensional units as the diameter.

The relation between the coefficient of friction and these components involves the use of the following expressions:

$R_N < 2000.$

Here flow is streamline or laminar in character, the roughness of the pipe walls is not a significant factor and the coefficient of friction may be calculated from the formula of Poiseuille:

$$f = \frac{16}{R_N} \quad \text{......3.3}$$

$R_N > 3000.$

Here flow is turbulent and all the components previously discussed have value. The formula of Colebrook and White has now been generally accepted as being the best theoretical approach:

$$\frac{1}{\sqrt{f}} = -4 \log_{10}\left(\frac{k_s}{3{\cdot}7D} + \frac{1{\cdot}255}{R_N\sqrt{f}}\right) \quad \text{......3.4}$$

The advantages accruing from the use of these two expressions for evaluation of the coefficient of friction are that problems involving the flow of any fluid in any type of pipe or duct may be solved with accuracy.

3.1.2. Approximate Data

In the case of compressible fluids flowing under conditions where the pressure drop is considerable in proportion to the initial pressure, the change in density between the initial and final conditions must be taken into account. For a given pipe diameter and a given rate of mass flow, velocity is inversely proportional to density ($u\rho$ = constant) and, since absolute viscosity varies only slightly with pressure change, the Reynolds Number and the corresponding friction coefficient may be evaluated in the same manner as for non-compressible flow, using the initial or final fluid characteristics as most convenient.

In practical cases, however, changes in temperature or in state will normally accompany the pressure drop $\triangle P$ and the use of a rational solution would suggest an accuracy beyond the validity of the data available. Approximate formulae are thus commonly used, of the following type:

$$\triangle P = A \frac{u^x}{D^y} \rho \triangle L \quad \text{......3.5}$$

where A, x and y are determined experimentally for a specific case or are deduced as approximations to the rational approach.

For any one set of circumstances, it is possible to simplify this type of expression in order to provide a convenient basis for tabular presentation by:

(*a*) Rearrangement to incorporate a term for mass flow W in lieu of that for velocity:

$$\triangle P = B \frac{W^x}{D^{(2x+y)}\rho^{(x-1)}} \triangle L \quad \text{......3.6}$$

where B is determined experimentally.

(*b*) Consideration of some relation between density and pressure for the fluid considered:

$$\rho = CP^n \quad \text{......3.7}$$

where C is determined experimentally.

(*c*) Substitution of 3.7 in 3.6 and integration with respect to P:

$$P_1^m - P_2^m = E \frac{W^x}{D^{2x+y}} L \quad \text{......3.8}$$

where E is determined experimentally.

The accuracy of such approximate expressions is necessarily limited to a relatively narrow range of values of Reynolds Number and one value only of absolute roughness. Extrapolation beyond these limits will result in an increasing error.

3.1.3. Pipe and Duct Fittings

The head lost due to the existence of bends, expansions, junctions and other local items is most correctly expressed in terms of fractions of the velocity head:

$$H = k\frac{u^2}{2g} \quad \ldots\ldots 3.9$$

where k is a factor for the local item concerned, as determined by experiment.

The importance of the effect of such losses varies greatly with the problem considered and the manner in which they are evaluated depends, to some extent, upon their importance relative to the associated "straight pipe" loss. In ventilation work, where a large proportion of the total head lost is due to such fittings, it is usual to make a separate calculation in connection with each item employing a table or chart based upon the formula quoted above. For problems concerned with pipework, as distinct from ductwork, however, where the fittings component of the total loss is of less importance, it is convenient to use data which express local losses in terms of the length of straight pipe which would produce a numerically similar loss of head. This "equivalent length" (EL) is evaluated by equating the two fundamental formulae, 3.1 and 3.9, to produce:

$$\text{EL} = k\frac{D}{4f} \quad \ldots\ldots 3.10$$

3.2. BASIC DATA FOR ALL FLUIDS

3.2.1. Straight Pipes, Solution of Problems by Calculation

The rational flow formula (3.1) and the two complementary expressions for the coefficient of friction (3.3 and 3.4) may be combined and transposed into convenient units for the solution of problems involving the flow of any liquid:

$R_N < 2000$

$$G = 0{\cdot}000\ 142\ 6\ \frac{d^4H}{v} \quad \ldots\ldots 3.11$$

$R_N > 3000$

$$G = -4\,(N_1Hd^5)^{\frac{1}{2}}\log_{10}\left[\frac{k_s}{3{\cdot}7d} + \frac{N_2d}{(N_1Hd^5)^{\frac{1}{2}}}\right] \quad \ldots 3.12$$

where

G = Rate of flow .. .. gal/min
H = Head lost per 100 foot run, feet of fluid flowing .. .. ft
v = Kinematic viscosity of fluid .. .. ft²/sec
k_s = Absolute roughness of pipe wall .. in
d = Internal diameter of pipe .. .. in
$N_1 = 0{\cdot}055\ 85$
$N_2 = 30{\cdot}73\ v$

Values of density and absolute viscosity for a selection of liquids and gases are tabulated in Section 21. Details of the variation in kinematic viscosity due to temperature, for a limited selection of fluids, are shown in Fig. 3.2.

Where compressible fluids are involved, an alternative transposition will be found to be more convenient and may be used without substantial error *provided that the head lost is small compared to the initial pressure and that no change of state is anticipated:*

$R_N > 3000$

$$Q = -4\,(N_1Hd^5)^{\frac{1}{2}}\log_{10}\left[\frac{k_s}{3{\cdot}7d} + \frac{N_2d}{(N_1Hd^5)^{\frac{1}{2}}}\right] \quad \ldots 3.13$$

where

Q = Rate of flow .. .. ft³/min
H = Head lost per 100 foot run, inches w.g. at 62°F .. .. in
ρ = Mean density of fluid .. .. .. lb/ft³
$N_1 = 0{\cdot}007\ 464\ \rho^{-1}$
$N_2 = 4{\cdot}928\ v$

Table 3.1. Surface roughness factors for piping, etc. (Absolute Roughness)

Material	k_s inches
Non-ferrous drawn tubing, including plastics ..	0·000 06
Asbestos cement piping	0·000 5
Black steel piping	0·001 8
Aluminium ducting	0·002
Galvanised steel piping or ducting	0·006
Cast-iron piping	0·008
Cement or plaster ducting	0·01
Fair-faced brickwork or concrete ducting ..	0·05
Rough brickwork ducting	0·2

Table 3.2. Relative roughness values for piping

Nominal Bore inches	Values of Relative Roughness $k_s/d \times 10^{-3}$ (dimensionless)							
	Black Steel Piping			Galvanized Steel Piping			C.I.	Copper
	B.S. 1387		No. Standard	B.S. 1387		No. Standard	B.S. 1211	B.S. 659
	Medium	Heavy		Medium	Heavy			
$\frac{3}{8}$	3·69	4·08	—	12·50	13·90	—	—	0·15
$\frac{1}{2}$	2·83	3·06	—	9·59	10·30	—	—	0·12
$\frac{3}{4}$	2·12	2·24	—	7·13	7·57	—	—	0·08
1	1·68	1·78	—	5·62	6·00	—	—	0·06
$1\frac{1}{4}$	1·27	1·33	—	4·26	4·47	—	—	0·05
$1\frac{1}{2}$	1·09	1·14	—	3·67	3·81	—	—	0·04
2	0·86	0·89	—	2·89	2·98	—	—	0·03
$2\frac{1}{2}$	0·67	0·68	—	2·23	2·28	—	—	0·02
3	0·57	0·58	—	1·89	1·99	—	2·52	0·02
$3\frac{1}{2}$	0·51	0·50	—	1·63	1·67	—	—	0·02
4	0·44	0·44	—	1·45	1·48	—	1·91	0·02
5	0·35	0·36	—	1·17	1·20	—	1·53	0·01
6	0·30	0·30	—	0·98	0·99	—	1·28	0·01
7	—	—	0·25	—	—	0·83	1·10	—
8	—	—	0·22	—	—	0·73	0·97	—
9	—	—	0·20	—	—	0·66	0·86	—
10	—	—	0·18	—	—	0·59	0·78	—
12	—	—	0·15	—	—	0·49	0·64	—

3.2.2. Straight Pipes, Solution of Problems by Semi-graphical Means

An alternative method of solution involves the use of the Moody chart of Poiseuille and Colebrook-White data (Fig. 3.1) and the following routine:

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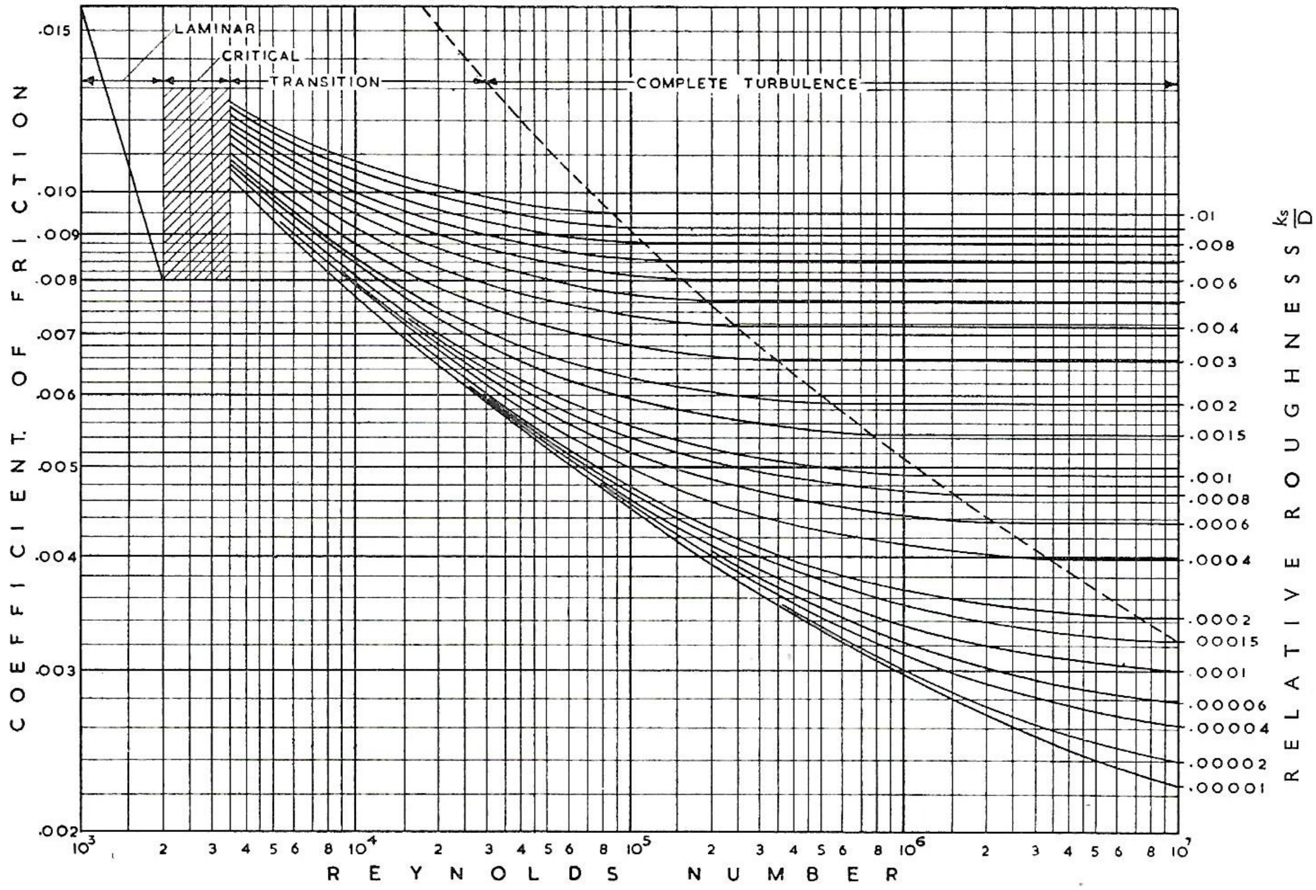

Fig. 3.1

(*a*) Calculate the value of Reynolds Number for the flow conditions obtaining, using either of the following simplifications if required:

$$R_N = \frac{0{\cdot}255Q}{dv} = \frac{0{\cdot}0407G}{dv} \quad \ldots\ldots\ldots\ldots\ldots\ldots 3.14$$

(*b*) Calculate the pipe roughness ratio k_s/d or read from Table 3.2 where appropriate.

(*c*) Determine the value of the coefficient of friction from Fig. 3.1 by reading vertically from the scale of Reynolds Number to the point of intersection with the appropriate curve of k_s/d, and thence to the left-hand scale.

(*d*) Calculate the head lost, per 100 foot run, in terms of feet of the fluid flowing using either of the following transpositions of the rational formula if required:

$$H = \frac{696fQ^2}{d^5} = \frac{17{\cdot}9fG^2}{d^5} \quad \ldots\ldots\ldots\ldots\ldots\ldots 3.15$$

3.2.3. Equivalent Length of Pipe in Feet corresponding to the Velocity Head

The appropriate value of the equivalent length may be determined either by calculation where:

$$EL = 0{\cdot}0208\frac{G^2d}{(N_1Hd^5)} \quad \ldots\ldots\ldots\ldots\ldots\ldots 3.16$$

or by use of Fig. 3.1. In the latter case, the coefficient of friction may be inserted in the following fundamental formula:

$$EL = \frac{d}{48f} \quad \ldots\ldots\ldots\ldots\ldots\ldots 3.17$$

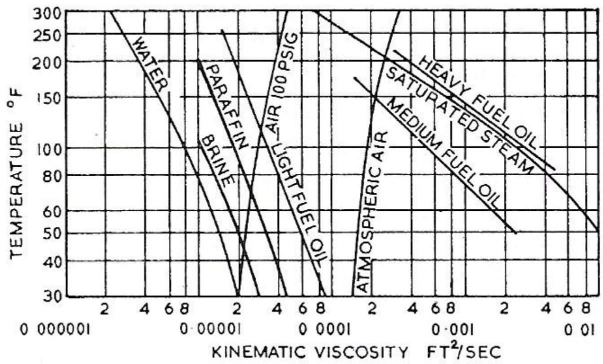

Fig. 3.2

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3.3. THE FLOW OF WATER IN PIPES

3.3.1. Straight Pipes

The following tables have been based upon the rational formula (3.1), combined with the Colebrook–White expression for the coefficient of friction (3.4). In convenient terms for water flow, these two items may be combined as follows:

$$W = -4\,(N_1Hd^5)^{\frac{1}{2}} \log_{10}\left[\frac{k_s}{3{\cdot}7d} + \frac{N_2 d}{(N_1Hd^5)^{\frac{1}{2}}}\right] \quad \ldots .3.18$$

where

W = Rate of flow .. .. lb/h

H = Head lost per 100 foot run, inches w.g. at 62°F .. .. in

ρ = Density of water at temperature considered .. .. lb/cu ft

μ = Absolute viscosity of water at temperature considered .. .. lb/ft h

d = Internal diameter of pipe .. .. in

k_s = Absolute roughness of pipe wall .. in

N_1 = 26·87 ρ (1645 for water at 150°F)

N_2 = 0·082 13 μ (0·0854 for water at 150°F)

It will be noted that the tabulated data for low rates of flow have been printed in a distinctive type. This is an indication that the flow condition then obtaining is either streamline or in the critical zone where either streamline or turbulent flow may exist.* The rates of flow so indicated should be used with caution since instability exists, and pipe sizes adjusted if possible to avoid such conditions.

3.3.2. Equivalent Lengths

Adjacent to each flow quantity, an additional column printed in italic type lists the lengths of pipe in feet which will produce a pressure loss equivalent to one velocity head. These figures are provided for use in conjunction with the tables of multiplying factors (Table 3.28) and have been derived from an expression, complementary to 3.18, as follows:

$$\mathrm{EL} = \frac{dW^2}{48(N_1Hd^5)} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots 3.19$$

3.3.3. Auxiliary Data for Water Flow in Heating Installations

In order that loss of head and equivalent length in feet may be estimated for alternative conditions of water flow, sets of correction data are presented in Tables 3.3 and 3.4. These take the form of multiplying factors by which the data read from the main tables may be corrected to suit the alternative conditions.

The units of head loss, after correction, remain those used for the main tables, i.e. inches w.g. at 62°F. Any specification of pump head should be phrased in such a manner that a manufacturer is made aware of the units used.

* In the calculation of the figures for low rates of flow the turbulent condition has been assumed.

Table 3.3. Flow of water at 300°F in steel tubes. Correction factors for Table 3.5

Φ_H = Correction factor for application to Head Loss read from Main Table.
Φ_L = Correction factor for application to Equivalent Length read from Main Table.

Loss of Head per 100 ft as read from Main Table	½		1		2		4		8		12		Approx. Velocity in ft/sec
	Φ_H	Φ_L	Φ_H	Φ_L	Φ_H	Φ_L	Φ_H	Φ_L	Φ_H	Φ_L	Φ_H	Φ_L	
0·2	0·80	*1·3*	0·84	*1·3*	0·87	*1·2*	0·90	*1·2*	0·92	*1·2*	0·94	*1·2*	1·0
0·5	0·83	*1·3*	0·86	*1·2*	0·89	*1·2*	0·92	*1·2*	0·94	*1·2*	0·96	*1·1*	
1·0	0·85	*1·2*	0·88	*1·2*	0·91	*1·2*	0·94	*1·2*	0·96	*1·1*	0·97	*1·1*	
2·0	0·87	*1·2*	0·90	*1·2*	0·93	*1·2*	0·96	*1·1*	0·98	*1·1*	0·99	*1·1*	3·0
5·0	0·90	*1·2*	0·92	*1·2*	0·95	*1·1*	0·98	*1·1*	0·99	*1·1*	1·01	*1·1*	
10	0·92	*1·2*	0·94	*1·2*	0·97	*1·1*	0·99	*1·1*	1·01	*1·1*	1·01	*1·1*	
20	0·94	*1·2*	0·96	*1·1*	0·98	*1·1*	1·00	*1·1*	1·02	*1·0*	1·02	*1·0*	12·0
50	0·97	*1·1*	0·98	*1·1*	0·99	*1·1*	1·02	*1·0*	1·02	*1·0*			
100	0·98	*1·1*	1·00	*1·1*	1·01	*1·1*	1·02	*1·0*					
150	1·00	*1·1*	1·01	*1·1*	1·02	*1·0*							

Table 3.4. Flow of water at 150°F in medium grade tubes to B.S. 1387. Correction factors for Table 3.5

Φ_H = Correction factor for application to Head Loss as read from Main Table.
Φ_L = Correction factor for application to Equivalent Length as read from Main Table.

Nominal bore inches	Φ_H	Φ_L	Nominal bore inches	Φ_H	Φ_L
½	0·67	1·08	2½	0·87	1·03
¾	0·75	1·06	3	0·89	1·02
1	0·74	1·06	3½	0·90	1·02
1¼	0·80	1·05	4	0·92	1·02
1½	0·82	1·04	5	0·97	1·01
2	0·85	1·03	6	0·97	1·01

For practical purposes, correction factors for the flow of water at 150°F in medium grade tubes may be considered to be constant for any one size over the range of head loss conditions shown in the main tables.

3.3.4. Auxiliary Data for Water Flow in Domestic Service Installations

In order that loss of head and equivalent length in feet may be read conveniently from quantities usually to hand during the design of domestic-supply plants, Tables 3.7, 3.8, 3.9, 3.10 and 3.11 have been prepared. These give rates of flow in gal/min against an argument of feet w.g. at 62°F.

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3.6. THE FLOW OF AIR IN DUCTS

3.6.1. Pressure Drop in Sheet Metal Ducts

The air flow chart (Fig. 3.3) has been plotted from calculations based upon the rational formula (3.1), combined with the Colebrook-White expression for the coefficient of friction (3.4). In convenient terms for ventilation data, these two items may be combined as follows:

$$Q = -4(N_1Hd^5)^{\frac{1}{2}} \log_{10}\left[\frac{k}{3{\cdot}7d} + \frac{N_2d}{(N_1Hd^5)^{\frac{1}{2}}}\right] \quad \ldots\ldots 3.30$$

where

Q = Rate of flow .. .. ft³/min
H = Pressure lost per 100 ft run, inches w.g. at 62°F .. .. .. in
ρ = Density of air .. .. lb/ft³
μ = Absolute viscosity of air .. .. lb/ft h
d = Internal diameter of duct .. .. in
k_s = Absolute roughness of duct wall .. in
$N_1 = 0{\cdot}007\,465\rho^{-1}$
$N_2 = 0{\cdot}001\,369\mu\rho^{-1}$

The chart represents the flow of dry air at 61 °F, 1000 mb barometer, in ducts constructed from clean galvanized sheet steel with joints made in accordance with good commercial practice. The consequent values for the various constants are listed in Table 3.20.

Table 3.20

Constant	Value
Density .. .. lb/ft³	0·075
Absolute viscosity .. .. .. lb/ft h	0·042 7
Absolute roughness of duct wall .. in	0·006
N_1	0·099 527
N_2	0·000 779

3.6.2. Air at Other Temperatures

For air at other temperatures, the pressure loss read from the chart may be corrected by use of the expression:

$$H_2 = H_1 \,.\, C\left(\frac{521}{460 + t}\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots 3.31$$

where t is the alternative air temperature, deg.F, and values for the viscosity correction factor C are read from Table 3.21.

Table 3.21

Alternative Temperature, °F	100	150	200	250	300
Correction Factor C ..	1·01	1·03	1·06	1·08	1·10

3.6.3. Ducts of Other Materials

For ducts of materials other than galvanized sheet steel, the correction factors listed in Tables 3.22, 3.23 and 3.24 may be used to correct the rates of pressure loss read from the air flow chart. Where the alternative duct is of rectangular section, the correction factors may be used only in conjunction with the equivalent diameter as read from Table 3.25 and not with the data read from Table 3.26.

The roughness values quoted for the alternative materials have been taken from standard authorities.

Table 3.22. Pressure drop in rough brickwork ducts ($k_s = 0{\cdot}2$ in)
Correction factors for Fig. 3.3

Loss of Head per 100 feet as read from Chart	Correction factors for ducts having the following diameters					
	5 in	10 in	24 in	30 in	50 in	80 in
0·01	—	1·85	1·85	1·85	1·84	1·83
0·02	—	1·99	1·95	1·95	1·92	1·89
0·05	—	2·14	2·07	2·05	2·01	1·95
0·1	2·39	2·26	2·15	2·13	2·08	2·03
0·2	2·51	2·37	2·23	2·19	2·13	—
0·5	2·67	2·48	2·30	2·26	2·18	—
1·0	2·79	2·56	2·34	2·30	2·22	—
2·0	2·86	2·62	2·38	2·33	—	—
5·0	2·95	2·67	2·41	2·36	—	—
10	3·01	2·90	2·43	2·39	—	—

Table 3.23. Pressure drop in fair faced brickwork or concrete ducts ($k_s = 0{\cdot}05$ in)
Correction factors for Fig. 3.3

Loss of Head per 100 feet as read from Chart	Correction factors for ducts having the following diameters					
	5 in	10 in	24 in	30 in	50 in	80 in
0·01	—	1·32	1·32	1·32	1·32	1·32
0·02	—	1·34	1·34	1·33	1·33	1·33
0·05	—	1·41	1·41	1·41	1·41	1·41
0·10	1·47	1·46	1·46	1·46	1·44	1·44
0·2	1·53	1·51	1·50	1·49	1·47	—
0·5	1·61	1·57	1·54	1·53	1·53	—
1·0	1·66	1·61	1·57	1·56	1·54	—
2·0	1·71	1·65	1·59	1·57	—	—
5·0	1·76	1·68	1·60	1·59	—	—
10	1·78	1·69	1·61	1·60	—	—

Table 3.24. Pressure drop in (*a*) neat cement or plaster ($k_s = 0{\cdot}01$ in); (*b*) sheet aluminium ($k_s = 0{\cdot}002$ in); (*c*) spiral wound galvanized ($k_s = 0{\cdot}003$ in) ducts
Correction factors for Fig. 3.3

Loss of Head per 100 feet as read from Chart	Correction factors for ducts of all diameters		
	Plastered Ducts	Aluminium Ducts	Spiral Ducts
0·01	1·03	0·96	0·97
0·02	1·04	0·95	0·96
0·05	1·06	0·93	0·95
0·1	1·08	0·92	0·94
0·2	1·08	0·90	0·93
0·5	1·09	0·88	0·92
1·0	1·09	0·86	0·92
2·0	1·10	0·85	0·90
5·0	1·10	0·83	0·88
10	1·11	0·80	0·87

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