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Incompressible flow through orifice plates – a review of the data in the literature

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THE PREPARATION OF THIS TECHNICAL NOTE

The work on this particular Technical Note was monitored and guided by the Internal Flow Panel. This Panel first met in 1979 and now has the following membership:

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INCOMPRESSIBLE FLOW THROUGH ORIFICE PLATES – A REVIEW OF THE DATA IN THE LITERATURE

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B.1 PRESSURE LOSS COEFFICIENT CORRELATIONS	109

INCOMPRESSIBLE FLOW THROUGH ORIFICE PLATES – A REVIEW OF THE DATA IN THE LITERATURE

1. NOTATION AND UNITS

		SI Unit
A	length from upstream fitting to orifice plate, Table 5.3, Sketch 5.4, or factor in Equations (6.37) and (6.39) or factor in Equations (6.54), (6.56) and (6.57)	m – –
A_o	total open area for perforated plates and screens, Section 9.3.1	m ²
A_{vc}	flow area at the <i>vena contracta</i> plane, Sketch 3.2	m ²
A_1	pipe cross-sectional area, Sketch 3.2	m ²
A_2	orifice cross-sectional area, Sketch 3.2	m ²
a	diameter of a single tapping or width of annular slot, Sketch 5.3, or factor used in Equation (6.49)	m –
B	back-cut width, Sketch 3.1(i), or length downstream of orifice plate, Table 5.3, Sketch 5.4, or sloping term, Equation (6.44) or factor used in Equations (6.57) and (6.63)	m m – –
b	internal diameter of carrier ring, Sketch 5.3, or factor in Equations (6.41), (6.42) and (6.49)	m –
C	length from straightening vanes to orifice plate, Table 5.3, Sketch 5.4, or factor used in Equations (6.49), (6.57) and (6.75)	m –
C'	length from upstream fitting to straightening vanes, Table 5.3, Sketch 5.4	m
C_c	contraction coefficient, defined in Equation (3.3)	–
C_d	discharge coefficient, defined in Equation (5.1)	–
$C_{d,Re}$	discharge coefficient at Reynolds number Re , Equation (6.54)	–
$C_{d,r}$	discharge coefficient for round-edged orifice plates, Equation (6.54)	–
$C'_{d,r}$	factor used in Equation (6.75)	–
$C_{d,u}$	discharge coefficient at $Re_h = 2 \times 10^4$, defined in Equation (6.74)	–

$C_{d,\infty}$	discharge coefficient for $Re \rightarrow \infty$	—
C_f	flow coefficient, defined in Equation (5.6)	—
$C_{f,\infty}$	flow coefficient for $Re \rightarrow \infty$, Equations (6.36), (6.37), (6.39), (6.41) and (6.42)	—
C_v	velocity coefficient, defined in Equation (5.7)	—
D	internal pipe diameter	m
	or factor used in Equation (6.75)	—
d	orifice bore diameter	m
	or factor used in Equation (6.49)	—
Eu	Euler number, defined in Equation (4.3)	—
Eu_{lam}	Euler number in laminar flow, Equation (6.28)	—
Eu_{turb}	Euler number in turbulent flow, Equations (6.28) and (6.29)	—
e	factor used in Equation (6.49)	—
e_{cl}	longitudinal eccentricity, defined in Equation (8.2)	—
e_{cn}	perpendicular eccentricity, defined in Equation (8.3)	—
K''	constant used in Equation (6.25)	—
K_i	total pressure loss coefficient, defined in Equation (4.2)	—
K_{si}	static pressure loss coefficient, defined in Equation (4.2)	—
$K_{\Delta p}$	pressure loss coefficient associated to pressure drop Δp , used in Equation (6.32)	—
K_0	value of K_i for $t/d = 0$, Equations (6.7) and (6.21)	—
$K_{0.8}$	value of K_i for $t/d = 0.8$, Equation (6.22)	—
k	factor used in Equation (6.58)	—
k_1, k_2	constants used in Equation (6.43)	—
L_u	upstream pipe length, Sketch 4.1	m
L_d	downstream pipe length, Sketch 4.1	m
L_{s2x}	orifice wall reattachment length, Sketch 3.3(e)	m
L_{s3x}	downstream reattachment length, Sketch 3.2	m
L_1	factor defined as $L_1 = l_1/D$, used in Equations (6.48) and (6.56)	—

L_2	factor defined as $L_2 = l_2/D$, used in Equation (6.56)	–
L'_2	factor defined as $L'_2 = l'_2/D$, used in Equations (6.48) and (6.56)	–
l_e	edge chamfer length, Sketch 3.1(g)	m
l_1	distance between upstream pressure tap centreline and upstream orifice face, Sketch 5.2	m
l_2	distance between downstream pressure tap centreline and upstream orifice face, Sketch 5.2	m
l'_2	distance between downstream pressure tap centreline and downstream orifice face, Sketch 5.2	m
m	mass flow rate	kg/s
Ma	Mach Number	–
M_1	factor used in Equation (6.57)	–
M_2	factor defined as $M_2 = L_2/(1 - \beta)$, Equation (6.57)	–
M'_2	factor defined as $M'_2 = 2L'_2/(1 - \beta)$, Equations (6.48) and (6.56)	–
N_4	factor used in Equation (6.57)	–
p	static pressure	Pa
p_m	measured static pressure	Pa
p_t	total pressure	Pa
p_2/p_1	downstream to upstream tap static pressure ratio	–
Q	volume flow rate	m ³ /s
Q_{actual}	actual volume flow rate, Equation (5.1)	m ³ /s
$Q_{theoretical}$	theoretical volume flow rate, Equation (5.1)	m ³ /s
Re	Reynolds Number	–
Re_D	pipe Reynolds Number	–
Re_d	orifice Reynolds Number	–
Re_h	corner taps Reynolds Number, defined in Equation (6.60)	–
r	edge radius, Sketch 3.1(e)	m
S	solidity, defined as $S = 1 - \alpha$ or separation length between two bends, Table 5.2	– m
t	orifice plate bore thickness, Sketch 3.1	m
t^*	orifice plate thickness, Sketch 3.1	m

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t/d	orifice plate bore thickness to diameter ratio	—
U	uncertainty	—
u	axial local fluid velocity	m/s
u_{cl}	axial local centre-line velocity	m/s
V_1	pipe bulk velocity, defined as $V_1 = m/(\rho A_1)$	m/s
V_2	orifice bulk velocity, defined as $V_2 = m/(\rho A_2)$	m/s
$V_{2,actual}$	actual mean velocity at <i>vena contracta</i>	m/s
v	radial local fluid velocity	m/s
α	porosity or area ratio, defined for circular orifices in circular ducts as $\alpha = A_2/A_1 = d^2/D^2$	—
β	diameter ratio, defined as $\beta = d/D$	—
δ	boundary layer thickness	m
δ^*	boundary layer displacement thickness, defined as $\delta^* = \int_0^\delta (1 - u/u_{cl}) dy$	m
ϕ	edge chamfer angle, Sketch 3.1(g)	degree
φ	back bevel angle, Sketches 3.1(b) and (d)	degree
φ_i	cooling hole inclination angle, Sketch 9.1	degree
φ_o	cooling hole orientation angle, Sketch 9.1	degree
λ_a	correction factor for effect of plate thickness for $t/d < 0.8$, $\lambda_a = K_i/K_0$	—
λ_b	correction factor for effect of plate thickness for $t/d \geq 0.8$, $\lambda_b = K_i/K_{0.8}$	—
μ	fluid dynamic viscosity	Pa s
μ_r	ratio of actual to reference viscosity, Equation (6.26)	—
ρ	fluid density	kg/m ³
τ	parameter used in Equation (6.24)	—
ν	fluid kinematic viscosity	m ² /s
ξ	parameter used in Equation (6.15)	—
ζ	parameter used in Equation (6.14)	—
Δp	measured pressure drop for derivation of discharge coefficient	Pa
$\overline{\Delta p}$	time-mean value of pressure drop, Equation (8.1)	Pa
$\Delta p'$	fluctuating component of Δp , Equation (8.1)	Pa
Δp_l	pressure loss measured between 1D upstream and 6D downstream of orifice plate, Equations (6.10) and (6.11)	Pa

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$\Delta p'_{rms}$ root-mean-square value of $\Delta p'$, Equation (8.1) Pa

Subscripts

actual refers to real flow conditions with losses
lam refers to laminar flow regime
theoretical refers to theoretical (maximum) flow conditions with no losses
turb refers to turbulent flow regime
vc refers to conditions at *vena contracta* plane (Sketches 3.2 and 3.2 and 5.1)
1 refers to conditions at station upstream of orifice in region fully-developed flow (Sketches 3.2, 4.1, 5.1)
2 refers to conditions at entry to orifice (Sketches 3.2, 4.1 and 5.1)
3 refers to conditions at station downstream of recovery plane (Sketch 4.1)

2. INTRODUCTION

This Technical Note provides a review of the data in the literature for incompressible flow through orifice plates published prior to 2006. At that time ESDU 81039 provided correlations for the prediction of pressure loss across orifice plates and perforated plates for liquids in turbulent flow. These correlations did not cover a wide range of the geometry and flow parameters affecting the flow through orifice plates²⁴¹. This Technical Note was developed to aid the extension and improvement of those correlations. The reviewed data were used in the validation of the CFD predictions carried out by ESDU to assist the development of the new correlations.

The data found in the literature included measurements and numerical predictions of the flow field and pressure distribution over the orifice test pipe, pressure drops across the orifice plate and pressure loss and discharge coefficients data. A considerable part of these data was available from measurements in orifice flow-meters and in support of their international standards, *i.e.* ISO 5167²⁰¹, ASME MFC-3M²¹⁴ and ANSI/API 2530¹⁵⁷. These standards can be accessed by subscribers at the IHS Engineering Resource Center, <http://www.ihserc.com>, and are summarized in Note FI152²⁴³. However, the geometry and flow ranges of the data reviewed in this Technical Note are beyond the industry standards for flow measurement.

In this Technical Note, the geometry and flow characteristics in orifice plates are described in Section 3. Definitions for the pressure loss coefficient and discharge coefficient are given in Sections 4 and 5, respectively. The standard requirements for the discharge coefficient measurement in orifice flow-meters are summarized in Sections 5.2, 5.2.1 and 5.2.2.

The data for incompressible flow of liquids are reviewed in Section 6 and summarized in Table A.1 in Appendix A. These data include correlations for the pressure loss and discharge coefficients in Sections 6.3 and 6.5, which are also summarized in Tables B.1 to B.2 in Appendix B.

The consistency of the data in the literature for the downstream separation length and the pressure loss and discharge coefficients is discussed in Section 7. The effects of non-standard flow, operation and installation conditions are discussed in Section 8 and the supporting papers are summarized in Table A.2.

The main emphasis of this Technical Note is on data for

- circular, concentric orifices in circular pipes,
- ducts with identical downstream and upstream cross-sectional areas,
- laminar, transitional and turbulent flow regimes,
- fully-developed, swirl-free, steady flow conditions with no cross-flow upstream and downstream of the orifice plate and normal flow to the orifice plate, and
- single-phase, Newtonian liquids and gases ($Ma \leq 0.3$ at the *vena contracta* plane) in incompressible flow.

However, a brief review on data for cooling holes, annular and non-circular orifices, and perforated plates is presented in Section 9 and summarized in Tables A.3, A.4, and A.5 in Appendix A, respectively.

3. GEOMETRY AND FLOW CHARACTERISTICS OF ORIFICE PLATES

3.1 Orifice Plate Geometry

The geometry of the orifice plate is defined using the following parameters as illustrated in Sketch 3.1:

- orifice plate bore diameter, d ,
- pipe diameter, D ,
- orifice plate bore thickness, t ,
- orifice plate thickness, t^* ,
- diameter ratio, $\beta = d/D$,
- porosity, $\alpha = d^2/D^2$,
- plate bore thickness/diameter ratio, t/d ,
- **entry edge profile**, *i.e.* square (Sketches 3.1(a) and (c)), knife (Sketch 3.1(d)), bevelled (Sketch 3.1(h)), rounded with edge radius, r (Sketch 3.1(f)), quadrant (Sketch 3.1(e)), or chamfered edge with chamfer length, l_e , and chamfer angle, ϕ (Sketch 3.1 (g)),
- **exit edge profile**, *i.e.* square (Sketch 3.1(a)), square back-cut (Sketch (i)) or back bevel with back bevel angle, ϕ , (Sketch 3.1(b)).

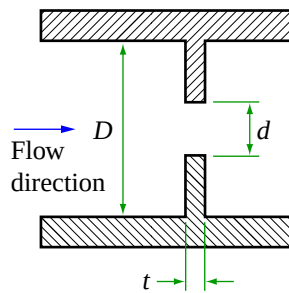
Thin (or short) orifice plates with square edges without and with back bevel angle, ϕ , (Sketch 3.1 (a) and (b)) are commonly used in standard flow measurement applications. Standard orifice plates typically have:

- $\phi = 45^\circ \pm 15^\circ$
- large diameter ratios, $0.2 < \beta < 0.75$,
- small plate bore thickness/diameter ratios, $t/d \leq 0.01$ (thin orifice),
- and edge radius, $r \leq 0.0004d$.

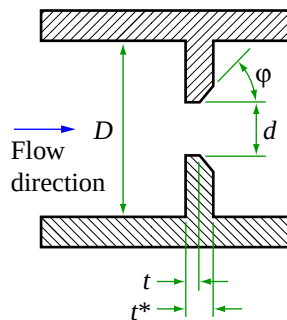
The orifice bore thickness, t , is specified by the ISO 5167²⁰¹, ASME MFC-3M²¹⁴ and ANSI/API 2530¹⁵⁷ international standards as follows:

- ISO 5167 and ASME MFC-3M: $0.005D \leq t \leq 0.02D$,
- ANSI/API 2530: largest of $t \geq 0.01d$ or $t > 0.127 \text{ mm}$;
smallest of $t \leq 0.02D$ or $t \leq 0.125d$.

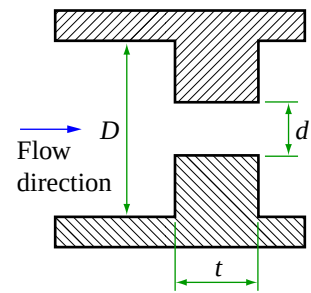
The ISO 5167 and ASME-MFC-3M standards specify a plate thickness within $t \leq t^* \leq 0.05D$. However, for $50 \text{ mm} \leq D \leq 64 \text{ mm}$, a plate thickness $t^* = 3.2 \text{ mm}$ is acceptable.



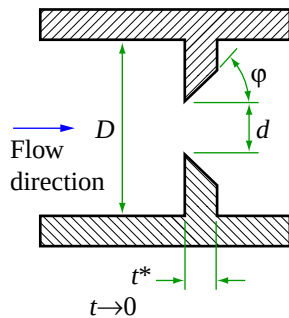
(a) short (or thin) square-edged orifice



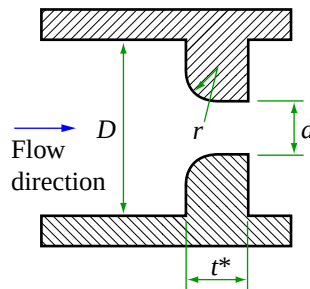
(b) short square-edged orifice with back bevel angle, ϕ



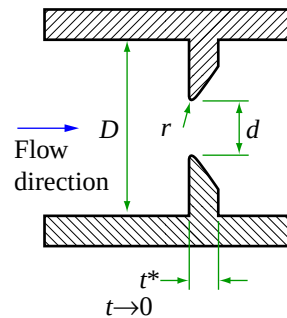
(c) long square-edged orifice



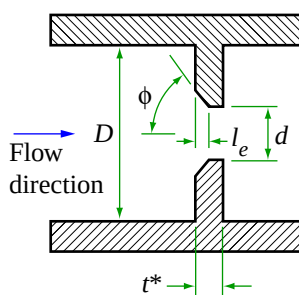
(d) knife-edged orifice with back bevel angle, ϕ



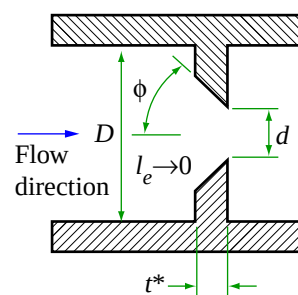
(e) quadrant-edged orifice



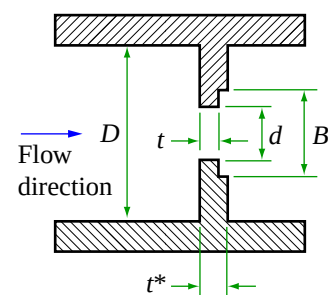
(f) short round-edged orifice



(g) chamfer-edged orifice with chamfer angle, ϕ , and length, l_e



(h) short orifice with inlet bevelled edges



(i) square-edged orifice with square back-cut

Sketch 3.1 Orifice plate geometry

3.2 Flow Characteristics

The majority of the data reviewed in this Technical Note are for an idealized flow condition: axisymmetric, fully-developed, swirl-free, steady flow upstream and downstream of the orifice plate, with no interference with other components. This condition, referred here as the “reference flow condition”, is difficult to achieve because of the various pipe fittings, such as bends, tees, and valves, upstream and downstream of the orifice plate. These fittings produce distortions of the flow velocity profiles which affect the pressure gradient upstream of the orifice plate and flow characteristics through the orifice plate and, to a smaller extent, downstream of the orifice plate.

In the “reference flow condition”, the characteristics of the flow through an orifice plate depend on the orifice to pipe diameter ratio, β , thickness to diameter ratio, t/d , orifice edge geometry, and pipe Reynolds number,

$$Re_D = \frac{DV_1\rho}{\mu}, \quad (3.1)$$

or the orifice Reynolds number, defined as

$$Re_d = \frac{dV_2\rho}{\mu} = \frac{Re_D}{\beta} \quad (3.2)$$

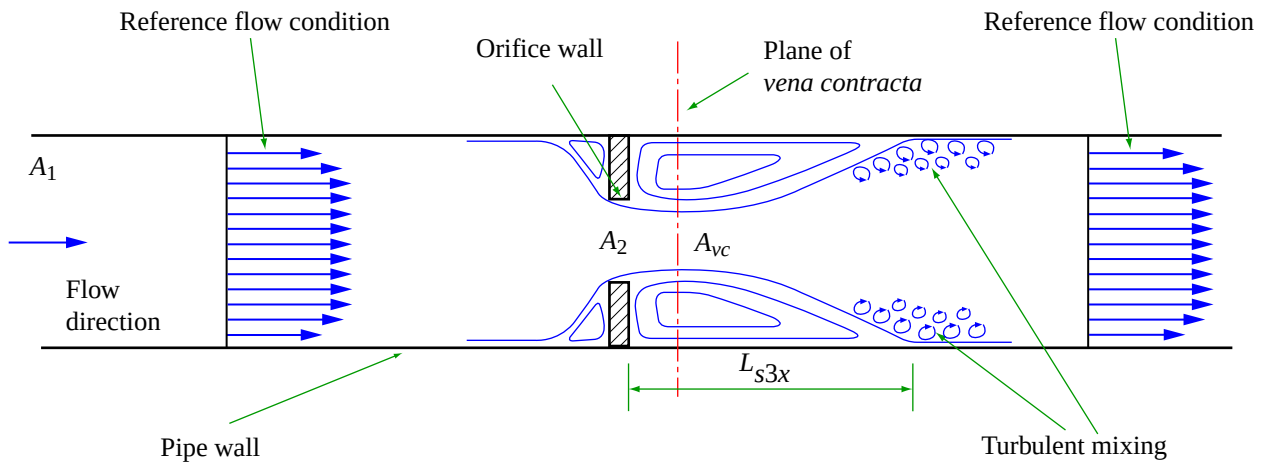
An important parameter in determining the flow characteristics is also the contraction coefficient, C_c , defined as

$$C_c = \frac{A_{vc}}{A_2} \quad (3.3)$$

where A_{vc} is the jet cross-sectional area at *vena contracta* plane and A_2 is the orifice cross-sectional area (Sketch 3.2).

The typical flow characteristics in turbulent flow ($Re_D \geq 10^4$) are illustrated in Sketch 3.2. The change of cross-sectional area through the orifice causes flow velocity gradients and, hence, pressure gradients. As the flow approaches the orifice, it accelerates as it contracts through the orifice opening and causes pressure to fall. At the entry edge of the orifice, the flow separates. The inward radial velocity component causes the jet to continue to contract to a minimum cross-sectional area downstream the orifice entry edge. The *vena contracta* plane is identified by the minimum cross-sectional area of the jet, and it has been observed to occur at an axial location typically about $0.5D$ from the orifice upstream face.

The fluid entrainment caused by the jet continues downstream of the *vena contracta*. The jet then expands gradually until the flow reattaches downstream. The entrainment process drives a recirculation zone immediately behind the orifice with some fluid moving counter to the main flow. The expansion and reattachment are accompanied by a corresponding deceleration. As the flow re-expands, there is an intensive mixing, resulting in an irrecoverable energy loss and a flatter velocity profile with high turbulence levels.



Sketch 3.2 Schematic of typical flow characteristics for $Re_d \geq 10^4$

Different flow regimes establish depending on whether the jet, which is formed downstream of the orifice entry, remains separated from the orifice wall and reattaches on the pipe wall or whether it reattaches on the orifice wall and on the pipe wall (see Sketch 3.3). The condition at which the flow changes from one regime to another is influenced by geometry parameters such as:

- orifice thickness/diameter ratio, t/d ,
- porosity, α , or diameter ratio, β , and
- entry edge profile,

and flow parameters such as:

- Orifice Reynolds number, Re_d , and
- Mach number, Ma .

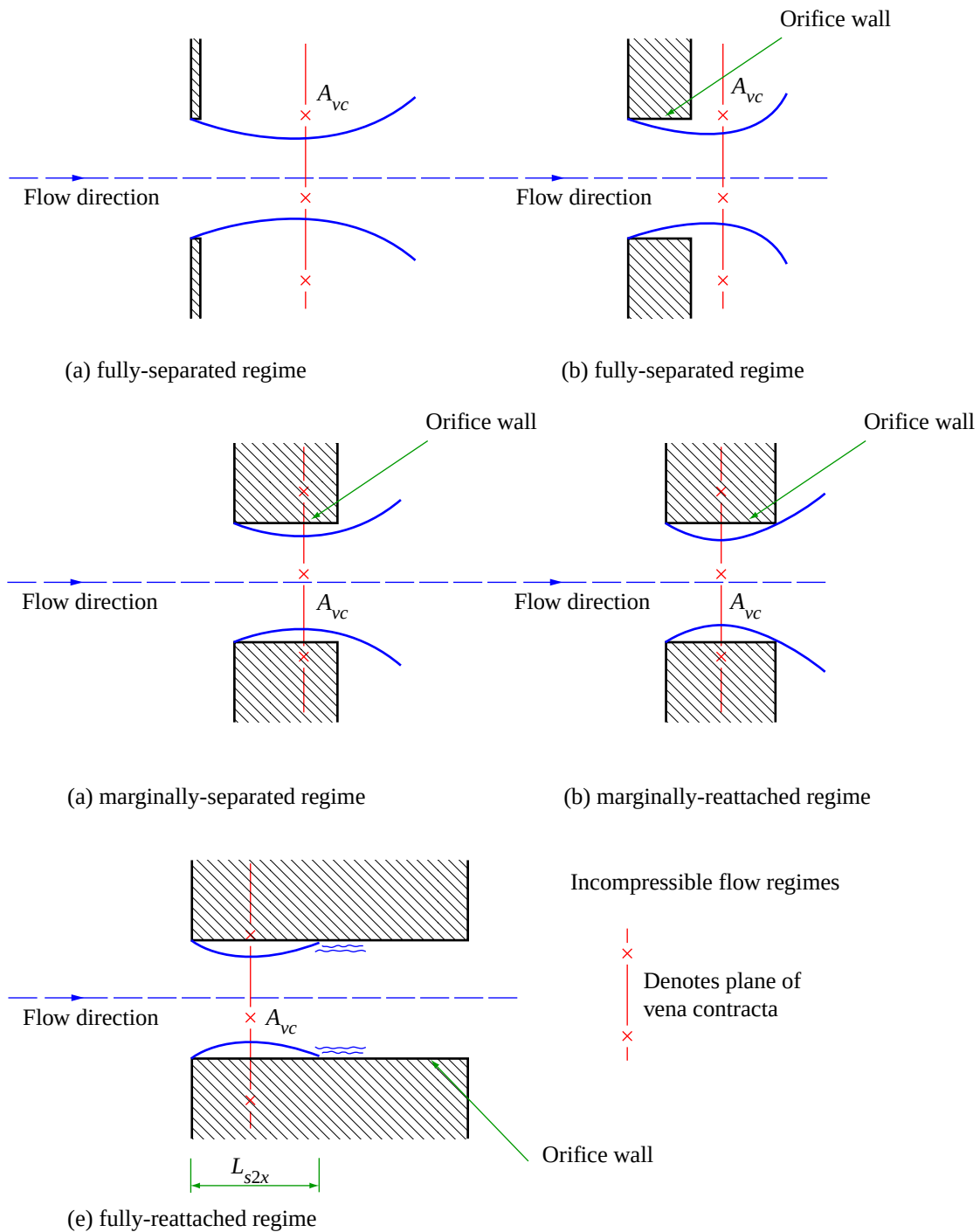
3.2.1 Incompressible flow

The characteristics flow regimes for a square-edged orifice occurring in incompressible turbulent flow are illustrated in Sketch 3.3. These are based on Ward-Smith^{41,77} and are summarized as follows.

- When the flow remains separated from the orifice wall, the flow regime is known as fully-separated (Sketch 3.3(a) and (b)). Typically, this flow occurs in the range $0 < t/d < 0.75$, where the contraction coefficient is increasing monotonically from $C_c = 0.61$ at $t/d = 0$ to $C_c = 0.8$ at $t/d = 0.75$ (Sketch 3.4).
- As the thickness of the orifice is increased, the jet tends to converge on the inner wall of the orifice (Sketch 3.3(c)). This regime is known as marginally-separated and typically occurs over a small but finite range around $t/d = 0.75$.
- With further increase in t , the jet just reattaches to the orifice and immediately separates again (Sketch 3.3(d)). This is known as the marginally-reattached flow regime.
- For larger t and higher Re , the flow reattaches in the form of a turbulent boundary layer to the orifice wall. This regime is known as fully-reattached flow regime (Sketch 3.3(e)) and occurs for $t/d > 0.75$, where C_c is constant at $C_c \approx 0.61$.

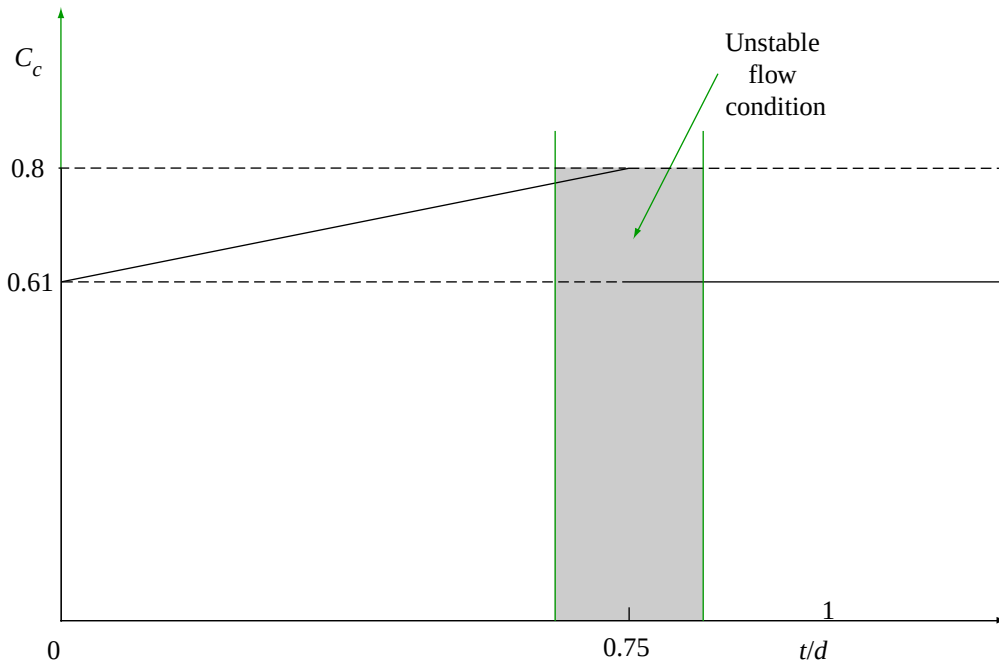
The two intermediate stages of marginally-separated and marginally-reattached flow regimes represent an uncertain region where either reattached or separated flow can occur.

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Sketch 3.3 Orifice flow regimes in incompressible flow

The characteristic flow regimes described above are based on Ward-Smith observations^{41,77}. The ESDU validated CFD calculations of the flow in sudden contractions have shown that the contraction coefficient (*i.e.* the downstream separation height) is independent of the Reynolds number but it varies with contraction ratio (*i.e.* diameter ratio). This is in contradiction with Ward-Smith findings for $t/d > 0.75$ where C_c is constant at $C_c \approx 0.61$. The ESDU CFD calculations of the flow through orifices provide further details of the flow characteristics for the above flow regimes.



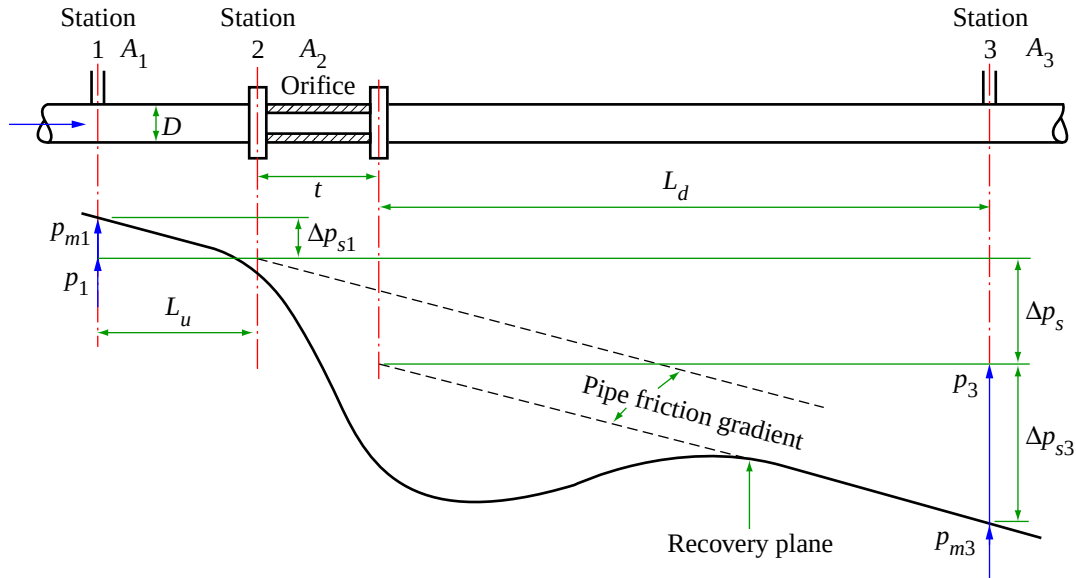
Sketch 3.4 Variation of contraction coefficient with orifice thickness to diameter ratio

3.2.1.1 Reynolds Number effect

At very low values of Reynolds number ($Re_D < 10$), the flow remains attached to the orifice wall. As the Reynolds number increases, the flow separates at entry of the orifice wall to form a jet downstream of the orifice. At high laminar flow Reynolds number ($Re_D \approx 2000$), transition from laminar to turbulent flow takes place in the jet, but re-laminarization may occur downstream of the orifice plate depending on the diameter ratio, β . At higher Reynolds number, transition to turbulence is well established and the nature and geometry of the flow are then largely insensitive to further increases in Reynolds number. For practical purposes, a value of $Re_d = 10^4$ can be taken as the lower limit for a single-holed orifice. For long orifices where the reattached flow loss is dominant, no significant variation has been observed for $Re_d \geq 10^4$.

4. PRESSURE LOSS COEFFICIENT

Pressure loss coefficient data are based on the following definition of pressure drop across the orifice.



Sketch 4.1 Definition of pressure loss

Sketch 4.1 shows a typical static-pressure distribution along the axis of a duct containing an orifice plate. The excess static-pressure drop is defined by

$$\Delta p = (p_{m1} - \Delta p_1) - (p_{m3} - \Delta p_3) = p_1 - p_3 \quad (4.1)$$

where p_{m1} and p_{m3} are static pressures measured in the system at stations 1 and 3 respectively, and p_1 and p_3 are static pressures at the same points but adjusted for pipe friction losses up to the entry and beyond the exit of the orifice. The quantities Δp_1 and Δp_3 are the friction losses in the straight pipes upstream and downstream of the orifice. The total-pressure loss, Δp_t , is defined similarly. Stations 1 and 3 are in regions of fully-developed flow.

For incompressible flow and $A_3 = A_1$, $\Delta p = \Delta p_t$, and the total and static pressure loss coefficient can be expressed as a function of the pipe bulk-mean velocity, V_1 , as

$$K_i = \frac{\Delta p_t}{\frac{1}{2} \rho V_1^2} = K_{si} = \frac{\Delta p}{\frac{1}{2} \rho V_1^2} \quad (4.2)$$

Alternatively the total and static pressure loss coefficient can be expressed as a function of the orifice bulk-mean velocity, V_2 , which is also referred to as the Euler number:

$$Eu = \frac{\Delta p}{\frac{1}{2} \rho V_2^2} = \beta^4 \frac{\Delta p}{\frac{1}{2} \rho V_1^2} = \beta^4 K_i \quad (4.3)$$

4.1 Pressure Measurements

Pressure measurements for the determination of the pressure loss coefficient need to be made sufficiently far from the orifice plate to be independent of the measurement location. It is also important to determine correctly the frictional pressure loss in the pipe. This can be determined from pressure measurements in the pipe with the orifice plate removed or in the upstream and downstream sections where fully-developed flow conditions occur.

The ISO-5167²⁰¹ and ASME MFC-3M²¹⁴ standards correlations for the pressure loss across the orifice plate are based on measurements of the wall static pressure at approximately $1D$ upstream and $6D$ downstream of the orifice plate, where the static pressure is considered recovered (fully-developed flow). These “standard” conditions were reported as “reference” conditions by Ward-Smith⁴¹ who indicated that the static pressure recovery plane is about $6D$ downstream of the orifice plate. The ESDU CFD analysis provide an insight into the downstream pipe length required to achieve “reference” conditions and allow to quantify the effect of deviations of the “standard” conditions from the “reference” conditions.

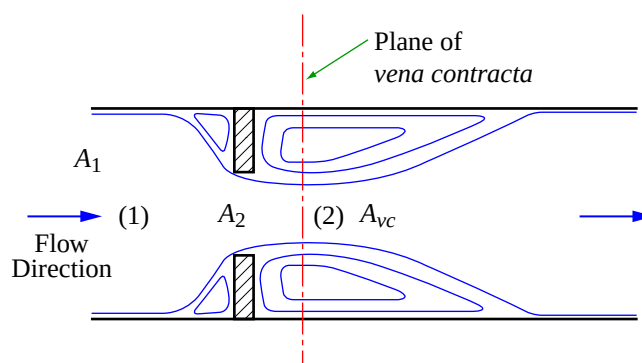
5. DISCHARGE COEFFICIENT

The discharge coefficient data discussed in this Technical Note are based on the following definitions used in incompressible flow. The discharge coefficient for the orifice plate is defined as the ratio of the actual volume flow rate, Q_{actual} , to the (maximum) theoretical volume flow rate, $Q_{theoretical}$ (see also Freeman²¹⁸):

$$C_d = \frac{Q_{actual}}{Q_{theoretical}} \quad (5.1)$$

The (maximum) theoretical flow rate can be calculated from the Bernoulli Equation assuming one-dimensional axial flow and constant static pressure at section (1) (Sketch 5.1), upstream the orifice plate, and at section (2), at the plane of *vena contracta* (minimum jet area) (Sketch 5.1):

$$p_1 + \frac{1}{2}\rho V_1^2 = p_2 + \frac{1}{2}\rho V_2^2 \quad (5.2)$$



Sketch 5.1 Definition of discharge coefficient

There are different ways of defining $Q_{theoretical}$. If Equation (5.2) is used and the continuity equation,

$$Q_{theoretical} = V_1 A_1 = V_2 A_2, \quad (5.3)$$

then,

$$Q_{theoretical} = \frac{\frac{\pi}{4}d^2}{\sqrt{1-\beta^4}} \sqrt{\frac{2\Delta p}{\rho}} \quad (5.4)$$

where $\Delta p = p_1 - p_2$ is the maximum static pressure drop across the orifice plate.

Substitution of Equation (5.4) in Equation (5.1) gives:

$$C_d = \frac{Q_{actual}}{\frac{\pi}{4}d^2} \sqrt{1-\beta^4} \sqrt{\frac{\rho}{2\Delta p}} \quad (5.5)$$

If the pressure drop $\Delta p = p_{t1} - p_2$ across the orifice is used to define $Q_{theoretical}$, a different equation is derived which coincides with Equation (5.5) for small values of β ($\beta \approx 0$).

The term $\sqrt{1-\beta^4}$ is also known as velocity of approach factor, and

$$C_f = C_d \sqrt{1-\beta^4} \quad (5.6)$$

is known as the flow coefficient.

The discharge coefficient can also be defined as a function of the contraction coefficient, $C_c = \frac{A_{vc}}{A_2}$, and the velocity coefficient,

$$C_v = \frac{V_{2,actual}}{V_2} \quad (5.7)$$

where A_2 is the orifice area, $V_{2,actual}$ is the actual mean velocity at *vena contracta* and V_2 is the bulk orifice velocity.

As $Q_{actual} = A_{vc} V_{2,actual}$ and $Q_{theoretical} = A_2 V_2$, from Equation (5.1) the discharge coefficient is given by:

$$C_d = C_v C_c \quad (5.8)$$

5.1 Tap Arrangements

In order to determine the discharge coefficient, different pressure tap arrangements are used to measure the pressure drop across the orifice:

- (a) flange taps,
- (b) corner taps,
- (c) D and $D/2$ taps,
- (d) *vena contracta* taps, and
- (e) pipe taps.

Flange, corner and D and $D/2$ tap arrangements are used in the ISO 5167²⁰¹, ASME MFC-3M²¹⁴ and

ANSI/API 2530¹⁵⁷ standards and are described in detail in Section 5.2.1. In the *vena contracta* taps, the upstream pressure tap is placed $1D$ from the upstream face of the orifice plate and the downstream tap is placed at the plane of minimum pressure (*vena contracta* plane)⁹⁴. In the pipe taps, the upstream pressure tap is placed $2\frac{1}{2}D$ from the upstream face of the orifice plate and the downstream pressure tap is placed $8D$ from downstream face of the orifice plate⁹⁴.

It is important to note that, with the exception of pipe taps, pressure tap arrangements for the determination of the discharge coefficient are not appropriate for the determination of the pressure loss coefficient. This is because they measure the largest pressure drop and do not allow sufficient length downstream of the orifice plate for pressure recovery occurring in the return of the flow to fully-developed conditions.

Discharge coefficient data are mainly determined empirically and depend on a number of factors, such as the geometry of the orifice plate, the pipe Reynolds number, tap location, pipe diameter, diameter ratio, and, for compressible flow, on specific heat ratio. As the discharge coefficient is dependent on the Reynolds number, which is itself dependent on Q_{actual} , it has to be obtained by iteration.

A considerable amount of data has been generated to develop discharge coefficient correlations for the orifice flow-meter standards ISO 5167²⁰¹, ASME MFC-3M²¹⁴ and ANSI/API 2530¹⁵⁷ (Sections 6.5.2.1 and 6.5.2.2). The applicability limits of such correlations (Equations (6.56) and (6.57)) are described in the following section.

5.2 ISO 5167, ASME MFC-3M and ANSI/API 2530 Standards

Since the 1930s, a significant amount of work has been devoted to the generation of discharge coefficient data for orifice flow-meters. A detailed review of the ISO 5167²⁰¹, ASME MFC-3M²¹⁴ and ANSI/API 2530¹⁵⁷ standards can be found in the Note FI 152²⁴³. The ISO 5167 and ASME MFC-3M standards are largely based on the BS 1042 standard²⁶.

In the ISO 5167, ASME MFC-3M and ANSI/API 2530 standards, the geometry, and installation and operating conditions of orifice flow-meters are specified. The standard correlations for the discharge coefficient are presented in Equations (6.56) and (6.57) apply to:

- Newtonian, single-phase, physically and thermally homogenous, fluid (liquids and gases) with no phase change,
- subsonic, steady-state, non-pulsating (steady) flows,
- axisymmetric, swirl-free, fully-developed upstream velocity profile,
- $Re_D \geq 5000$,
- gases with downstream to upstream tap pressure ratio $0.8 \leq p_2/p_1 \leq 1.0$ (ASME MFC-3M, ANSI/API 2530), or $0.75 \leq p_2/p_1 \leq 1$ (ISO 5167).

The effects of conditions departing from those specified in the standards on the accuracy of flow measurement are discussed in Section 8. The tap arrangements and pipe length requirements are described in Sections 5.2.1 and 5.2.2, respectively.

5.2.1 Standard pressure tapping arrangements

The tap arrangements for the ISO 5167²⁰¹, ASME MFC-3M²¹⁴ and ANSI/API 2530¹⁵⁷ orifice flow-meters are:

- (a) flange pressure taps,
- (b) corner pressure taps, and
- (c) D and $D/2$ pressure taps.

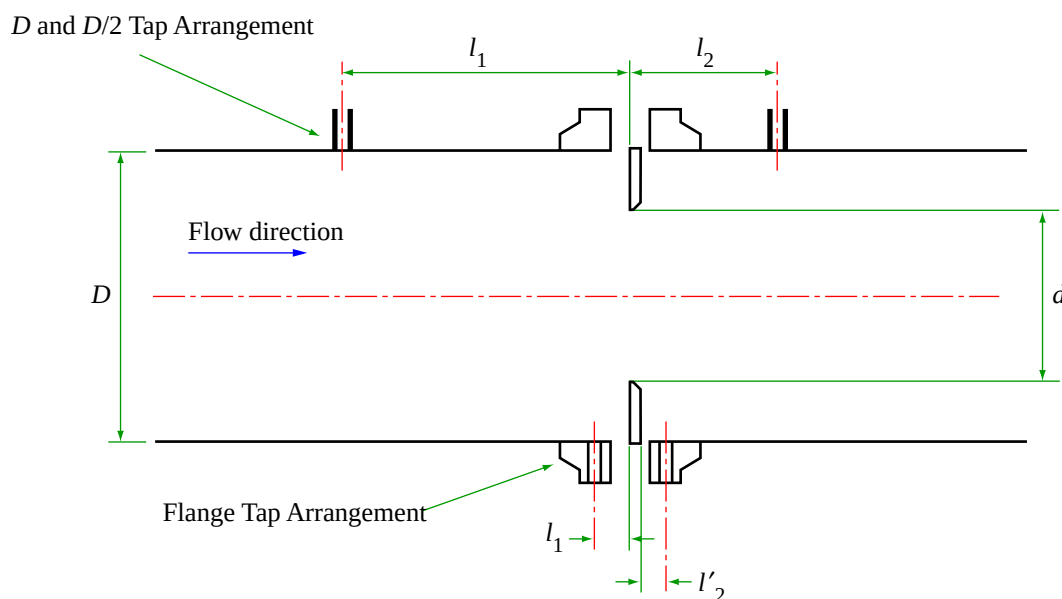
The flange and D and $D/2$ tap arrangements are shown in Sketch 5.2. The corner tap arrangement is shown in Sketch 5.3. Data in the ISO 5167 and ASME MFC-3M standards are for the three above tap arrangements. Data in the ANSI/API 2530 are for flange pressure tapplings, although correlations for pipe taps are given for natural gas only in an Appendix of the standard²⁴³.

Flange and corner tapping arrangements may produce errors in the pressure drop measurements across the orifice as they do not take into account variation of the orifice meter geometry and measurements may be taken at locations where static pressure gradients are significant (see Section 8.6). The dimensions for the D and $D/2$ tap arrangement do not depend on the geometry of the flow-meter. Therefore, this tapping arrangement is considered the most reliable⁹⁴.

Details for the **flange and D and $D/2$ tap arrangements** are in Sketch 5.2 and Table 5.1. Note that in the D and $D/2$ tap arrangement, the spacing l_2 is the distance between the centreline of the downstream pressure tapping and the upstream face of the orifice. However, in the flange tap arrangement, the spacing l'_2 of the downstream pressure tapping is measured from the downstream orifice face.

Details for the **corner tap arrangement** in the ISO 5167 and ASME MFC-3M standards are in Sketch 5.3 and Table 5.1. The pressure tapplings may be either single taps or annular slots. Both types of tapplings may be located either in the pipe or its flanges or in carrier rings as shown in Sketch 5.3. The diameter of a single tapping or the width of the annular slot, a , is specified to prevent accidental blockage and give satisfactory dynamic performance. The internal diameter of the carrier ring, b , has to ensure that no protrusion occurs into the pipe.

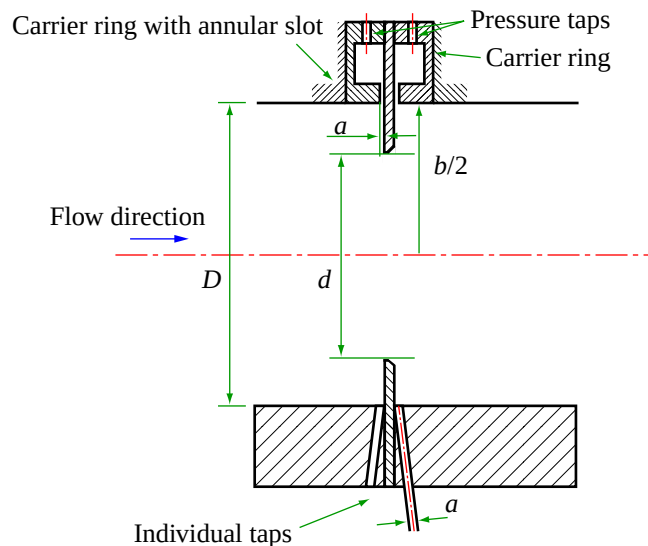
In all tap arrangements, the edges of the tap hole are flush to the internal surface of the pipe wall, free from burrs and can be slightly rounded.



Sketch 5.2 Flange tap and D and $D/2$ tap arrangements

TABLE 5.1 Details of Standard Tap Locations

Tap arrangement	ISO 5167	ASME MFC-3M	ANSI/API 2530
Flange	$l_1 = l_2'$ $l_2' = 25.4 \text{ mm} \pm 0.5 \text{ mm}$ for $\beta > 0.6$ and $D < 150 \text{ mm}$ $l_2' = 25.4 \text{ mm} \pm 1 \text{ mm}$ for $\beta \leq 0.6$ $l_2' = 25.4 \text{ mm} \pm 1 \text{ mm}$ for $\beta > 0.6$ and $150 \text{ mm} \leq D \leq 1000 \text{ mm}$	$l_1 = l_2'$ $l_2' = 25.4 \text{ mm} \pm 0.5 \text{ mm}$ for $\beta > 0.6$ and $D < 150 \text{ mm}$ $l_2' = 25.4 \text{ mm} \pm 1 \text{ mm}$ for $\beta \leq 0.6$ $l_2' = 25.4 \text{ mm} \pm 1 \text{ mm}$ for $\beta > 0.6$ and $150 \text{ mm} \leq D \leq 1000 \text{ mm}$	$l_1 = l_2'$ For $D \leq 101.6 \text{ mm}$, $l_2' = 25.4 \text{ mm} \pm 1.588$ for $\beta \leq 0.6$, $l_2' = 25.4 \text{ mm} \pm (2.966 - 3.447\beta) \text{ mm}$ for $0.4 < \beta \leq 0.75$ For $D \leq 101.6 \text{ mm}$, $l_2' = 25.4 \text{ mm} \pm 3.175 \text{ mm}$ for $\beta \leq 0.4$, $l_2' = 25.4 \text{ mm} \pm (5.956 - 6.951\beta) \text{ mm}$ for $0.4 < \beta \leq 0.75$
D and D/2	$l_1 = D \pm 0.1D$ $l_2 = 0.5D \pm 0.02D$ for $\beta \leq 0.6$ $l_2 = 0.5D \pm 0.01D$ for $\beta > 0.6$	$l_1 = D \pm 0.1D$ $l_2 = 0.5D \pm 0.02D$ for $\beta \leq 0.6$ $l_2 = 0.5D \pm 0.01D$ for $\beta > 0.6$	—
Corner	Clean fluids and vapours: — $\beta \leq 0.65$: $0.005D \leq a \leq 0.03D$ — $\beta > 0.65$: $0.01D \leq a \leq 0.02D$ — $D < 100 \text{ mm}$: $a \leq 2 \text{ mm}$ for any β Any values of β : — clean fluids: $1 \text{ mm} \leq a \leq 10 \text{ mm}$ — vapours and annular chambers: $1 \text{ mm} \leq a \leq 10 \text{ mm}$; — vapours, liquefied gases, single tappings: $4 \text{ mm} \leq a \leq 10 \text{ mm}$; $D \leq b \leq 1.04D$	Clean fluids and vapours: — $\beta \leq 0.65$: $0.005D \leq a \leq 0.03D$ — $\beta > 0.65$: $0.01D \leq a \leq 0.02D$ — $D < 100 \text{ mm}$: $a \leq 2 \text{ mm}$ for any β Any values of β : — clean fluids: $1 \text{ mm} \leq a \leq 10 \text{ mm}$ — vapours and annular chambers: $1 \text{ mm} \leq a \leq 10 \text{ mm}$; — vapours, liquefied gases, single tappings: $4 \text{ mm} \leq a \leq 10 \text{ mm}$; $D \leq b \leq 1.04D$	—



Sketch 5.3 Corner tap arrangement

5.2.2 Pipe length requirements

To ensure that flow measurements are within the accuracy stated by the standards, the fluid entering the orifice has to have an axisymmetric, swirl-free, fully-developed flow profile. These conditions are achieved using adequate lengths of upstream and downstream pipes and flow conditioners. In the collection of the empirical database for the ANSI/API 2530¹⁵⁷ standards, these flow conditions were achieved using a pipe length of 45 pipe diameter downstream of a Sprinkle flow conditioner.

TABLE 5.2 Required straight lengths between orifice plates and fittings without flow conditioners.

From Table 3 in BS EN ISO 5167-2:2003(E)*

Values expressed as multiples of pipe internal diameter, D

β	Upstream (inlet) side of orifice plate																							Downstream (outlet) side of the orifice plate		
	Single 90° bend; two 90° bends in any plane ($S > 30D$) ^a		Two 90° bends in the same plane: S-configuration ($30D \geq S > 10D$) ^a		Two 90° bends in the same plane: S-configuration ($10D \geq S$) ^a		Two 90° bends in perpendicular planes ($30D \geq S \geq 5D$) ^a		Two 90° bends in perpendicular planes ($5D > S$) ^{a,b}		Single 90° tee with or without an extension Mitre 90° bend		Single 45° bends Two 45° bends in the same plane: S-configuration ($S \geq 2D$) ^a		Concentric reducer $2D$ to D over a length of $1.5D$ to $3D$		Concentric expander $0.5D$ to D over a length of D to $2D$		Full bore ball valve or gate valve fully open		Abrupt symmetrical reduction		Thermometer pocket or well ^c of diameter $\leq 0.03D$ ^d		Fittings (columns 2 to 11) and the densitometer pocket	
1	2		3		4		5		6		7		8		9		10		11		12		13		14	
—	A ^e	B ^f	A ^e	B ^f	A ^e	B ^f	A ^e	B ^f	A ^e	B ^f	A ^e	B ^f	A ^e	B ^f	A ^e	B ^f	A ^e	B ^f	A ^e	B ^f	A ^e	B ^f	A ^e	B ^f	A ^e	B ^f
≤ 0.20	6	3	10	g	10	g	19	18	34	17	3	g	7	g	5	g	6	g	12	6	30	15	5	3	4	2
0.40	16	3	10	g	10	g	44	18	50	25	9	3	30	g	5	g	12	8	12	6	30	15	5	3	6	3
0.50	22	9	18	10	22	10	44	18	75	34	19	9	30	18	8	5	20	9	12	6	30	15	5	3	6	3
0.60	42	13	30	18	42	18	44	18	65 ^h	25	29	18	30	18	9	5	20	11	14	7	30	15	5	3	7	3.5
0.67	44	20	44	18	44	20	44	20	60	18	36	18	44	18	12	6	28	14	18	9	30	15	5	3	7	3.5
0.75	44	20	44	18	44	22	44	20	75	18	44	18	44	18	13	8	36	18	24	12	30	15	5	3	8	4
NOTE 1 The minimum straight lengths required are the lengths between various fittings located upstream or downstream of the orifice plate and the orifice plate and the plate itself. Straight lengths shall be measured from the downstream end of the curve portion of the nearest (or only) bend or the tee or the downstream end of the curved or conical portion of the reducer or the expander.																										
NOTE 2 Most of the bends on which the lengths in this table are based had a radius of curvature to $1.5D$.																										
a S is the separation between the two bends measured from the downstream end of the curved portion of the upstream bend to the upstream end of the curved portion of the downstream bend.																										
b This is not a good upstream installation: a flow conditioner should be used where possible.																										
c The installation of thermometer pockets or wells will not alter the required minimum upstream straight lengths for the other fittings.																										
d A thermometer pocket or well of diameter between $0.03D$ and $0.13D$ may be installed provided that the values in Columns A and b are increased to 20 and 10 respectively. Such an installation is not, however, recommended.																										
e Column A for each fitting gives lengths corresponding to “zero additional uncertainty” values.																										
f Column B for each fitting gives lengths corresponding to “0.5% additional uncertainty” values.																										
g The straight length in Column A gives zero additional uncertainty: data are not available for shorter straight lengths which could be used to give the required straight lengths for Column B.																										
h $95D$ is required for $Re_D > 2 \times 10^6$ if $S < 2D$.																										

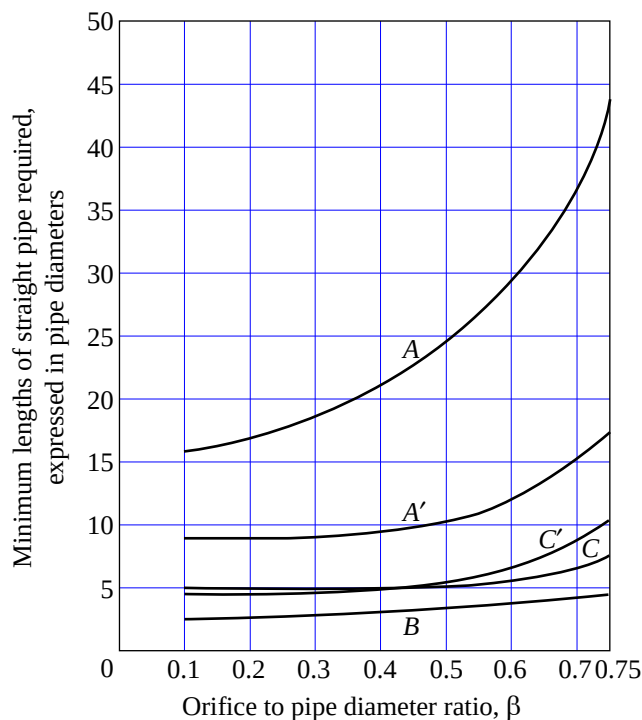
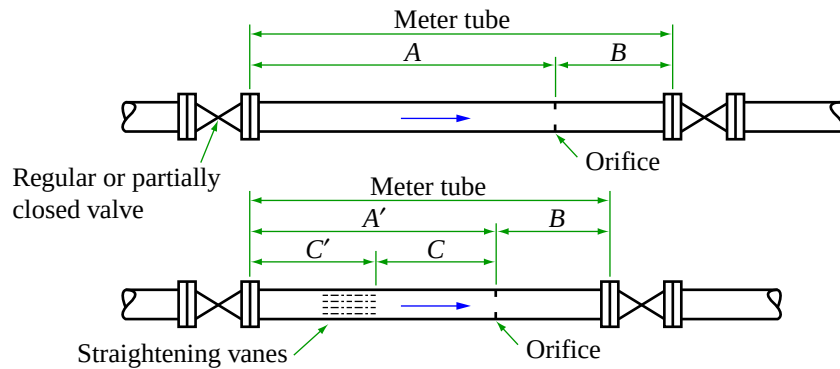
* Permission to reproduce extracts of BS EN ISO 5167-2:2003(E)²⁰¹ is granted by BSI. British Standards can be obtained from BSI Customer Services, 389 Chiswick High Road, London, W4 4AL. Tel: +44 (0)20 8996 9001. email: cservices@bsi-global.com

In Table 5.2, the minimum straight lengths upstream and downstream of the orifice plate without flow conditioners are given for the ISO 5167²⁰¹ standard for various installation fittings. If the straight lengths used are equal to or longer than the values specified in Columns A of Table 5.2, it is not necessary to increase the uncertainty in discharge coefficient to take into account the effect of the specific installation. If the upstream or downstream straight length is shorter than the value in Columns A and either equal to or greater than the value shown in Columns B for a given fitting, an additional uncertainty of 0.5 % has to be added to the uncertainty in the discharge coefficient.

According to the ANSI/API 2530¹⁵⁷ standard, the minimum pipe lengths depend on β and increase with it. A typical example for a partly closed valve upstream of the pipe is given in Sketch 5.4. The minimum upstream and downstream pipe lengths for $\beta = 0.75$ are given in Table 5.3 for various upstream pipe fitting.

TABLE 5.3 ANSI/API 2530¹⁵⁷ minimum pipe lengths upstream and downstream of orifice plate for $\beta = 0.75$

Minimum pipe length	Partly closed valve upstream	two non-planar 90° bends less than 10D apart	two planar 90° bends less than 10D apart	reducer or expander upstream
Upstream of orifice plate (A)	54D	35D	21.5D	13.5 D
Downstream of orifice plate (B)	4.5D	4.5D	4.5D	4.5D
Up to flow straightening vanes (C')	10.3D	8D	7D	–
From flow straightening vanes to orifice plate (C)	7.8D	7D	6.5D	7D



Sketch 5.4 Upstream and downstream pipe lengths for a upstream partly closed valve.

From ANSI/API 2530¹⁵⁷

6. INCOMPRESSIBLE FLOW OF LIQUIDS

Johansen² initiated detailed studies on the flow through square-edged orifices with 45° back bevel. Flow visualization and measurement of the discharge coefficient were conducted. Many experimental studies have been carried out since then by organizations, universities, and companies for the determination of discharge coefficient, and, to a smaller extent, of pressure loss coefficient. A substantial part of this work has been devoted to orifice flow-meters, *i.e.* square-edged short orifices in turbulent flow. In more recent years, experimental and numerical methods have been used to study the flow through orifices in various engineering applications such as cooling holes, fuel lines, hydraulic systems, air conditioning systems, and water pipe systems. Data for pressure loss and discharge coefficients are still actively generated in these areas.

The data available in the literature for incompressible flow of liquids include measurements and numerical predictions of the flow field and pressure distribution along the orifice test pipe, pressure drops across the orifice plate and pressure loss and discharge coefficients data. Details of the geometry and flow conditions of these data are summarized in Table A.1 in Appendix A. The reliability of these data has been assessed from the reported information and their consistency is discussed in Section 7.

The reviewed data are presented in the following subsections and their geometry and flow coverage is illustrated in Sketches 6.1 to 6.6 and 6.8 to 6.9. The data for flow field and pressure distributions are used in the ESDU CFD validation studies. These data are marked with (*) in Table A.1.

6.1 Flow Field Data

The ranges of diameter ratio, β , and pipe Reynolds number, Re_D , covered by the selected experimental (solid line) and numerical (dotted line) flow field data available in the literature for short (knife- and square-edged) and long orifices are illustrated in Sketches 6.1 and 6.2, respectively.

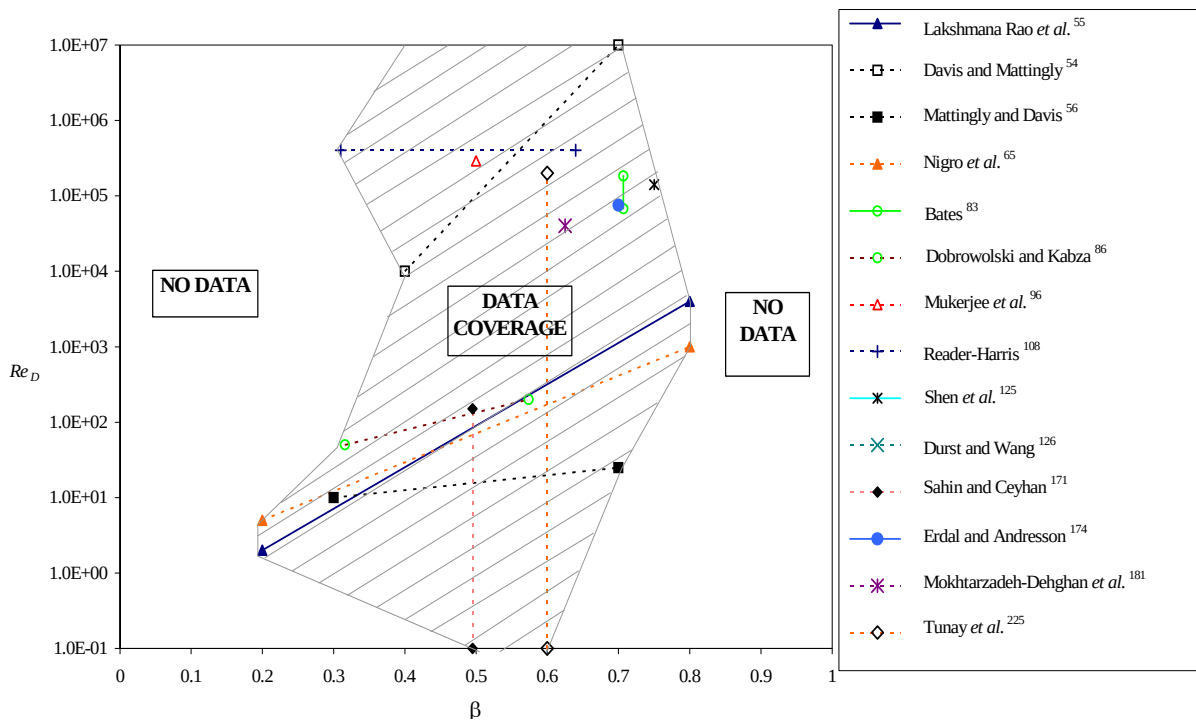
6.1.1 Measurements

Two of the earliest attempts at measuring turbulent flow details in the vicinity of an orifice were made by Dyban and Epik³⁵ and Teyssandier and Wilson⁴⁶ using hot wire anemometry. Pressure recovery (at flow reattachment) lengths were measured by Lakshmana Rao *et al.*⁵⁵. Recovery lengths for fully-developed flow downstream of the orifice plate were measured by Alvi *et al.*⁶⁰ in laminar to turbulent flow. 1-D LDV measurements were used by Bates⁸³ to investigate the flow through square-edged orifice plates.

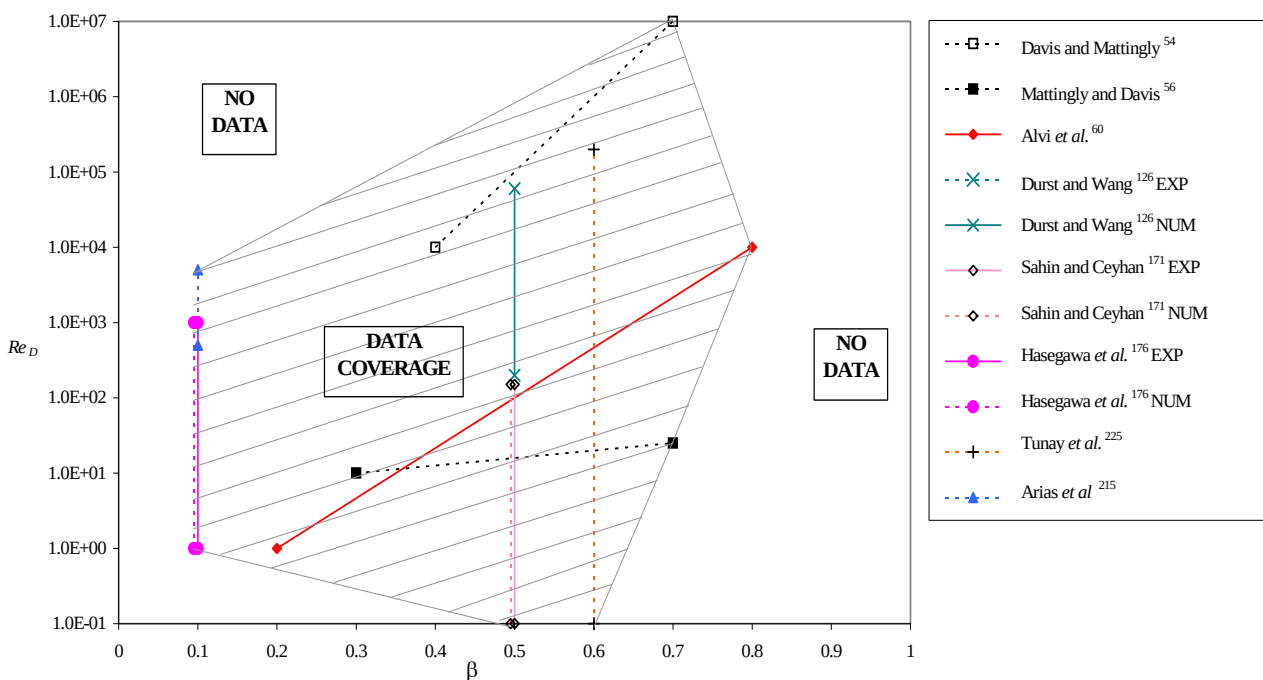
Durst *et al.*^{115,127} and Durst and Wang¹²⁶ used 1-D LDV and numerical studies to investigate the flow through square-edged orifice plates in laminar and turbulent flows. Their findings can be summarized as follows:

- small recirculation observed upstream of the orifice plate,
- main downstream recirculation with reattachment length varying with Re (minimum at $Re_d = 2000$ of $L_{s3x} = 7d/2$ and two maximum at $Re_d = 850$ ($L_{s3x} = 10d/2$) and $Re_d = 22000$ ($L_{s3x} = 11.6d/2$),
- secondary downstream recirculation bounded by main recirculation with extension $1.3d/2$,
- overshoot of axial velocity in the orifice.

Shen *et al.*¹²⁵ used 2-D LDV to measure mean velocity and turbulent intensity in a knife-edged with a 30° back bevel orifice. No recirculation was detected upstream of the orifice plate. The downstream recirculation extended up to $L_{s3x} = 5d/2$.



Sketch 6.1 Flow field experimental (solid line) and numerical data (dotted line) data for short orifices



Sketch 6.2 Flow field experimental (solid line) and numerical data (dotted line) data for long orifices

6.1.2 Numerical studies

Early CFD simulations were carried out by Mattingly and Davis⁵⁶ in laminar flow, and Davis and Mattingly⁵⁴ in turbulent flow using TEACH-T. For the laminar flow simulations, different orifice plate thicknesses were considered. For the turbulent flow simulations, the $\kappa - \varepsilon$ turbulence model was used.

Laminar flows were studied by Nigro *et al.*⁶⁵ and Dobrowolski and Kabza⁸⁶. Nigro *et al.*⁶⁵ considered knife- and square-edged short orifices, Dobrowolski and Kabza⁸⁶ considered square-edged with 45° back bevel. Mukerjee *et al.*⁹⁶ used PHOENICS to simulate turbulent flow through square-edged orifices. The effect of velocity profile swirl and skewing on the discharge coefficient due to bends upstream of the orifice plate was investigated.

Reattachment lengths were calculated by McGreehan *et al.*¹⁰⁵ in laminar flow, and by Reader-Harris¹⁰⁸ in turbulent flow. Reader-Harris¹⁰⁸ used TEACH-T to simulate the flow through square-edged orifices with 45° back bevel. The same finite-difference CFD software was used by Durst and Wang^{126,146} to simulate turbulent flows through square-edged short and long orifices. Comparisons with the experimental data showed differences within 2%.

In laminar flow, square-edged short and long orifices were studied by Sahin and Ceyhan¹⁷¹ and Hasegawa *et al.*¹⁷⁶. The latter used FIDAP to simulate flow through small diameter orifices. The same CFD software was used by Mokhtarzadeh-Dehghan *et al.*¹⁸¹ to simulate turbulent flows in knife-edged 45° -bevel orifices. Results were compared with annular orifices calculations. In turbulent flow, Erdal and Andersson¹⁷⁴ reported on the predictions of the axial velocity and turbulent kinetic energy in square-edged short orifices using PHOENICS.

The majority of the recent numerical studies have been carried out using the CFD software FLUENT. Arias *et al.*²¹⁵ used FLUENT to simulate laminar and transitional flow through square-edged long orifices. Tunay *et al.*²²⁵ used the same CFD software to simulate laminar to turbulent flows in square-edged short and long orifices. Jocksch and Gravett²³⁰ used FLUENT for orifices with square edge and 45° back bevel in turbulent flow and Kulkarni and Moeykens²³¹ for square-edged short orifices in laminar to turbulent flow.

All turbulent flow CFD calculations reported here have been carried out using the $\kappa - \varepsilon$ turbulence model with logarithmic wall function. It has been reported that effect of higher order differencing scheme has proved beneficial to simulate flow separation details. Some calculations did not predict the presence of the upstream recirculation region observed in some experiments. Downstream reattachment lengths were, in general, over-predicted. Velocity predictions were within 5% of the measured values.

6.2 Pressure Loss Data

Pressure loss data summarized in the next subsections include:

- measurements and numerical predictions of the static pressure distribution over the length of the test pipe and orifice walls,
- pressure drop across the orifice, and
- pressure loss coefficient data.

Pressure loss coefficient correlations are presented in Section 6.3.

The ranges of diameter ratio, β , and pipe Reynolds number, Re_D , covered by the selected experimental (solid line) and numerical (dotted line) pressure profile and pressure drop data available in the literature for short (knife- and square-edged) and long orifices are illustrated in Sketches 6.3 and 6.4, respectively.

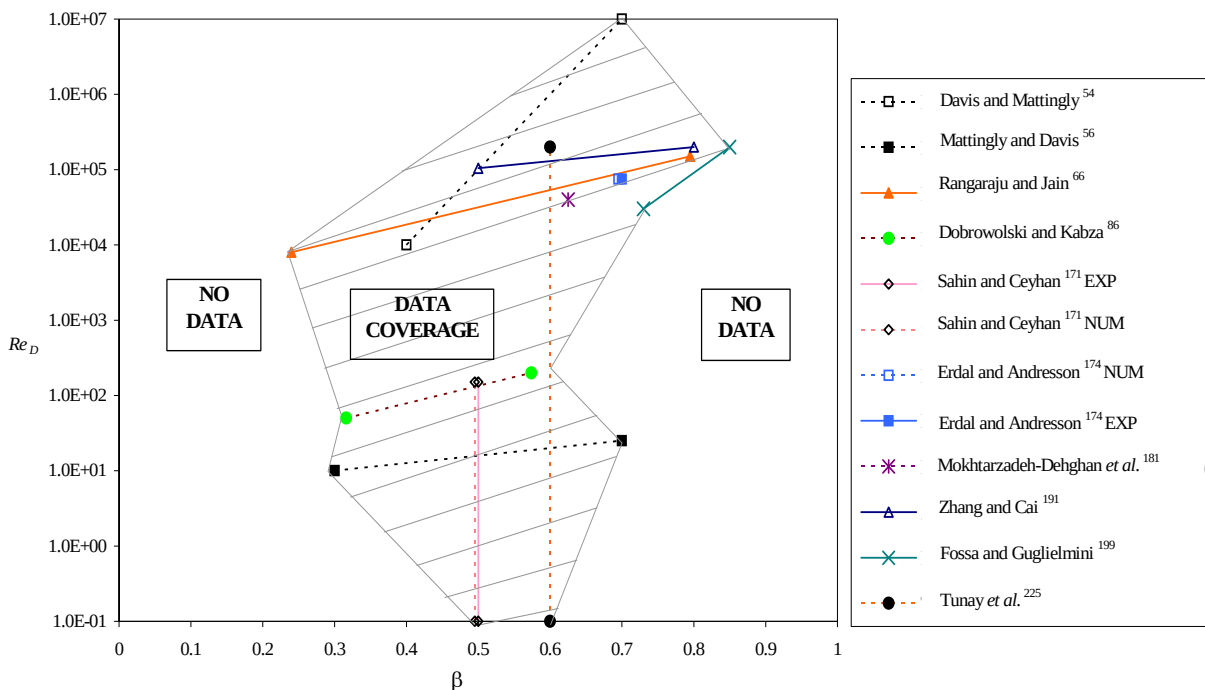
Ranges covered by the selected pressure loss coefficient data and correlations for short and long orifices are illustrated in Sketches 6.5 and 6.6, respectively. Most data provide information for $0 < t/d < 5$.

6.2.1 Wall static pressure distribution

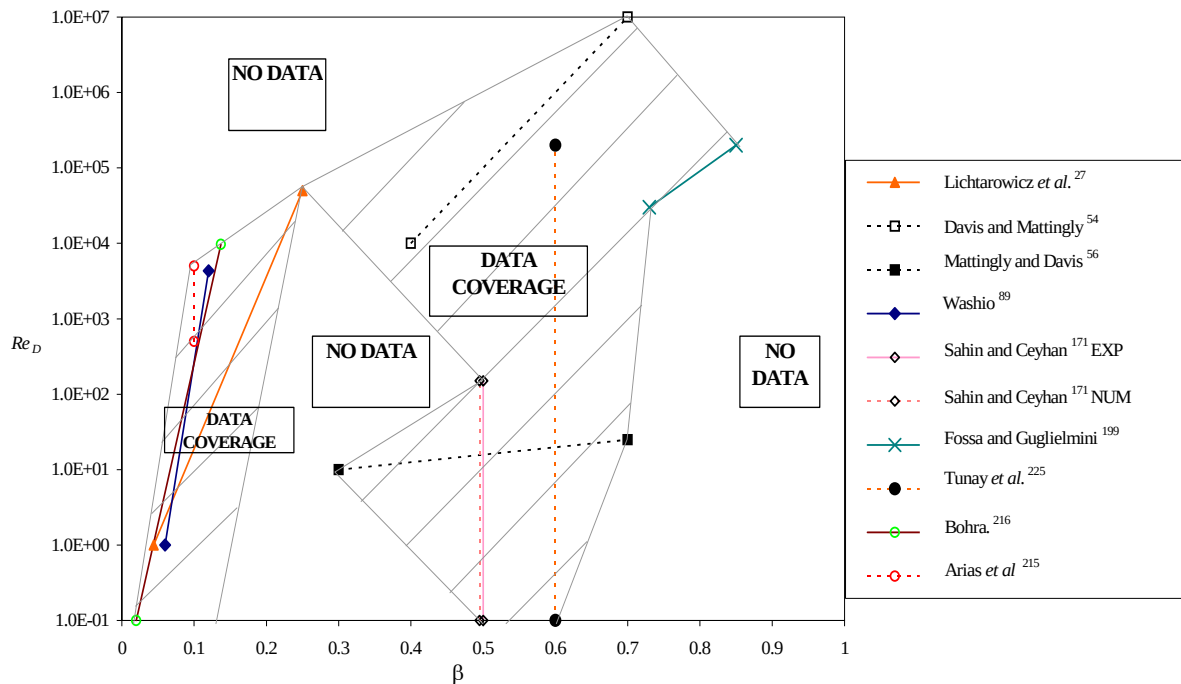
6.2.1.1 Measurements

Early measurements of the pressure distributions were carried out by Lichtarowicz *et al.*²⁷ in square-edged long orifices. For square-edged short orifices, Shima⁹⁷ carried out measurements in transitional flow, and Rangaraju and Jain⁶⁶ carried out measurements in turbulent flow. For square-edged long orifices, Washio⁸⁹ carried out measurements in laminar and transitional flow.

In turbulent flow, Zhang and Cai¹⁹⁰ measured pressure distributions for square-, sharp-, and quadrant-edged short orifices, while Chen *et al.*²¹⁷ measured pressure distributions for square-edged long orifices.



Sketch 6.3 Experimental (solid line) and numerical (dotted line) pressure data for short orifices



Sketch 6.4 Experimental (solid line) and numerical pressure data (dotted line) data for long orifices

6.2.1.2 Numerical studies

Early numerical predictions of the pressure distribution were carried out for square-edged orifices by Mattingly and Davis⁵⁶ in laminar flow, and by Davis and Mattingly⁵⁴ in turbulent flow.

In laminar flow, Dobrowolski and Kabza⁸⁶ carried out predictions of the pressure distribution for a square-edged orifice with 45° back bevel. Sahin and Ceyhan¹⁷¹ carried out predictions for square-edged short and long orifices. In turbulent flow, Mokhtarzadeh-Dehghan *et al.*¹⁸¹ carried out predictions for a square-edged orifice with 45° bevel.

Numerical predictions for square-edged short orifices were compared with experimental data by Erdal and Andersson¹⁷⁴ in turbulent flow. Recent numerical predictions of the wall static pressure for square-edged short and long orifices were reported by Arias *et al.*²¹⁵ in laminar and transitional flow and by Tunay *et al.*²²⁵ in laminar to turbulent flow.

6.2.2 Pressure drop and pressure loss coefficient data

6.2.2.1 Pressure drop data

Pressure drop measurements have been carried out by Zhang and Cai¹⁹⁰ in square-, knife-, and quadrant-edged short orifices in turbulent flow; by Fossa and Guglielmini¹⁹⁹ for square-edged long orifices in turbulent flow; and by Bohra²¹⁶ for square-edged long orifices with square back-cut in laminar to turbulent flow.

6.2.2.2 Pressure loss coefficient

Measurements for pressure loss coefficient data have been extensively reviewed by Ward-Smith⁴¹. Lakshmana-Rao and Shridharan⁴³ carried out measurements of the pressure loss coefficients for sharp- and square-edged short and long orifices in laminar flow. Data for square-edged short orifices were published

by Edwards *et al.*⁹⁹ in laminar flow, by Washio⁸⁹ in laminar and transitional flow, by Alvi *et al.*^{59,60} in laminar to turbulent flow, and by Rangaraju and Jain⁶⁶ in turbulent flow. Experimental data for square-edged thin orifices with 45° back bevel were published by Shima⁹⁷ in transitional to turbulent flow, and Humpherys^{103,118} and Reader-Harris¹⁰⁸ in turbulent flow.

Experimental data for square-edged long orifices were published by Hasegawa *et al.*¹⁷⁶ in laminar flow, by Minks²⁰² in laminar and transitional flow, and by Fox and Stark¹²⁸ in laminar to turbulent flow. Bohra²¹⁶ carried out measurements for square-edged long orifices with square back-cut in laminar to turbulent flow. Numerical data for square-edged long orifices were carried out by Arias *et al.*²¹⁵ in laminar and transitional flow.

In round- and chamfer-edged orifices in turbulent flow, Broach *et al.*²⁰⁹ carried out measurements and Yeh²²⁶ performed numerical predictions.

6.3 Pressure Loss Coefficient Correlations

Commonly used engineering references for pressure loss calculations, such as Miller¹⁶⁹ and Idel'chik¹⁹³, provide correlations based on experimental data for different orifice geometries. A review of these data is presented in the Note FI 149²⁴², where comparisons with the correlations in ESDU 81039⁸⁴ are also discussed. A brief review of such correlations is given in the next subsections.

The ranges of diameter ratio, β , and pipe Reynolds number, Re_D , covered by the selected experimental (solid line) and numerical (dotted line) pressure loss coefficient data and correlations available in the literature for short (knife- and square-edged) and long orifices are illustrated in Sketches 6.5 and 6.6, respectively. Most data provide information for $0 < t/d < 5$. Summaries of the equations described in Sections 6.3.1 and 6.3.2 are in Table B.1 and B.2 in Appendix B.

6.3.1 Short orifices

It is common use to express the correlations for pressure loss coefficient, K_i , or Euler Number, $Eu = \beta^4 K_i$, as a function of the porosity, α , so that they can be applied to non-circular pipes. In this Technical Note, pressure loss or Euler Number correlations are mainly expressed as a function of the diameter ratio, β . Alternative expressions as a function of α are also provided for some of these correlations.

In **turbulent flow**, the pressure loss through a short (or thin) orifice can be estimated from the loss due to the irreversible expansion process downstream of the *vena contracta* to the pipe wall applying the **Carnot-Borda** formula. If we assume that the loss upstream of the *vena contracta* is negligible, the pressure loss coefficient for one single expansion (short orifice) is given, as a function of $\beta = d/D$ or $\alpha = d^2/D^2$, by

$$K_i = \left(\frac{1}{C_c \beta^2} - 1 \right)^2 = \left(\frac{1}{C_c \alpha} - 1 \right)^2 \quad (6.1)$$

where C_c is the contraction coefficient, $C_c = A_{vc}/A_2$. A correlation for the contraction coefficient as a function of the diameter ratio, β , is recommended by Weisbach¹. This is presented in Equation (6.34).

From the Bernoulli energy, the continuity and the momentum balance equations, Ward-Smith⁸¹ derived

the following equation:

$$C_d = \sqrt{\frac{C_c^2(1-\beta^4)}{1-C_c^2\beta^4}} = \sqrt{\frac{C_c^2(1-\alpha^2)}{1-C_c^2\alpha^2}} \quad (6.2)$$

which substituted in Equation (6.1) gives the pressure loss coefficient in terms of the discharge coefficient valid for the loss at the pressure recovery point (flow reattachment point):

$$K_i = \frac{1-\beta^4}{C_d^2\beta^4} + 2\left(1 - \sqrt{\frac{1-\beta^4}{C_d^2\beta^4} + 1}\right) = \frac{1-\alpha^2}{C_d^2\alpha^2} + 2\left(1 - \sqrt{\frac{1-\alpha^2}{C_d^2\alpha^2} + 1}\right) \quad (6.3)$$

This equation is based on the one-dimensional flow analysis where the upstream loss to the *vena contracta*, estimated using the Bernoulli equation, is neglected and the downstream loss from the *vena contracta* to the flow reattachment is estimated using the Carnot-Borda formula.

Based on the continuity and energy flow equations (Carnot-Borda) and Equation (6.2), **Benedict**⁵³ derived an equation for the Euler Number across short square-edged orifices for the loss at the pressure recovery point (flow reattachment point) in turbulent flow:

$$Eu = \beta^4 K_i = \left(\frac{1-\beta^4}{C_d^2}\right) - 2\beta^2\left(\frac{1}{C_c} - \beta^2\right) = \left(\frac{1-\alpha^2}{C_d^2}\right) - 2\alpha\left(\frac{1}{C_c} - \alpha\right) \quad (6.4)$$

A correlation for the contraction coefficient recommended by Weisbach¹ is presented in Equation (6.34).

Benedict⁵³ reported that Equation (6.4) compares well with **Ginsburg**²⁴ correlation:

$$Eu = \beta^4 K_i = 1/2(1-\beta^2) + (1-\beta^2)^2 + 1.41\sqrt{1-\beta^2}(1-\beta^2) \quad (6.5)$$

This is derived considering the following terms for pressure loss: (1) an entry term, $0.5(1-\beta^2)$, (2) an exit term, $(1-\beta^2)^2$, and (3) an interaction term, $2\sqrt{0.5(1-\beta^2)}(1-\beta^2)$.

Based on reviewed experimental data from various sources, **Ward-Smith**⁴¹ provides the following correlations valid for $0 < \beta < 0.87$ ($0 < \alpha < 0.75$) and $Re_D \geq 10^4$:

$$K_i = \left[\frac{1}{0.608\beta^2(1-\beta^{5.2}) + \beta^{7.2}} - 1 \right]^2 = \left[\frac{1}{0.608\alpha(1-\alpha^{2.6}) + \alpha^{3.6}} - 1 \right]^2 \quad (6.6)$$

This equation has a root-mean-square error of 9.5 %.

The **ESDU** correlations in ESDU 81039⁸⁴ are largely based on the data reported in Ward-Smith⁴¹. Pressure loss coefficients correlations are provided for Reynolds numbers $Re_d > 10^4$ (turbulent flow) in sharp-edged (knife- and square-edged), thin and thick orifice plates with edges free from any chamfering, rounding or burrs. The pressure loss coefficient, K_i , is calculated as follows:

$$K_i = \frac{K'_0}{\beta^4} \lambda_a = \frac{K'_0}{\alpha^2} \lambda_a = K_0 \lambda_a \quad (6.7)$$

This page Amendment B

The basic coefficient K_0 are calculated from the corresponding value of K'_0 . This is provided in graphical format.

Humpherys¹¹⁸ developed the following correlation for square-edged short orifices in turbulent flow based on measurements in water irrigation pipes with $0.38 \leq \beta \leq 0.82$:

$$Eu = \beta^4 K_i = \frac{1 - 0.9\beta^{1.7}}{C_d^2}, \quad (6.8)$$

from which the discharge coefficient can be derived as

$$C_d = \frac{1 - 0.9\beta^{1.7}}{\sqrt{Eu}}, \quad (6.9)$$

The **ISO-5167**²⁰¹ and **ASME MFC-3M**²¹⁴ standards provide identical correlations for the pressure loss and pressure loss coefficients in thin square-edged orifices. A pressure loss across the orifice plate, Δp_l , is defined as the wall static pressure between the pressure measured at approximately $1D$ upstream of the orifice plate and that measured about $6D$ downstream of the orifice plate, where the static pressure is considered recovered to fully developed flow conditions. The following two equations are provided for square-edged thin orifices with $0.2 \leq \beta \leq 0.75$ and for $Re_D > 5000$:

$$\bullet \quad \Delta p_l = \frac{\sqrt{1 - \beta^4(1 - C_d^2)} - C_d \beta^2}{\sqrt{1 - \beta^4(1 - C_d^2)} + C_d \beta^2} \Delta p \quad (6.10)$$

$$\bullet \quad \Delta p_l = \left(1 - \beta^{1.9}\right) \Delta p \quad (6.11)$$

where Δp is the differential pressure measured across the orifice plate defined in Equation (5.4).

A correlation for the pressure loss coefficient K_i is also given in the **ISO-5167**²⁰¹ and **ASME MFC-3M**²¹⁴ standards based on **Urnner**¹⁷⁹ theoretical analysis:

$$K_i = \left[\frac{\sqrt{1 - \beta^4(1 - C_d^2)}}{C_d \beta^2} - 1 \right]^2 = \left[\frac{\sqrt{1 - \alpha^2(1 - C_d^2)}}{C_d \alpha} - 1 \right]^2 \quad (6.12)$$

This Equation is based on the Carnot-Borda Equation (Equation (6.1) and Equation (6.2) and is valid for the loss at the pressure recovery point (flow reattachment point).

Idel'chik et al.¹⁵³ provide the following pressure loss correlations for:

- short ($t/d \leq 0.015$) **knife-edged** orifices for $Re_d > 10^5$:

$$K_i = \left(1 + 0.707 \sqrt{1 - \beta^2} - \beta^2 \right)^2 \left(\frac{1}{\beta^2} \right)^2 = \left(1 + 0.707 \sqrt{1 - \alpha} - \alpha \right)^2 \left(\frac{1}{\alpha} \right)^2 \quad (6.13)$$

- short ($t/d \leq 0.015$) **round-edged** orifices for $Re_d > 10^4$:

$$K_i = \left(1 + \sqrt{\zeta(1 - \beta^2)} - \beta^2 \right)^2 \left(\frac{1}{\beta^2} \right)^2 = \left(1 + \sqrt{\zeta(1 - \alpha)} - \alpha \right)^2 \left(\frac{1}{\alpha} \right)^2 \quad (6.14)$$

where ζ is an empirical parameter that depends on t/d and varies between 0.06 and 0.44;

- short ($0.01 < t/d < 0.016$) orifices with **entry bevel** (40° to 60°) for $Re_d > 10^4$:

$$K_i = \left(1 + \sqrt{\xi(1 - \beta^2)} - \beta^2 \right)^2 \left(\frac{1}{\beta^2} \right)^2 = \left(1 + \sqrt{\xi(1 - \alpha)} - \alpha \right)^2 \left(\frac{1}{\alpha} \right)^2 \quad (6.15)$$

where ξ is an empirical parameter that depends on t/d and varies between 0.13 and 0.46.

Equation (6.13) is based on the entry, exit and interaction loss terms as that of Ginsburg²⁴ (Equation (6.5)). Equations (6.14) and (6.15) are empirical.

Miller¹⁶⁹ provides plots (not equations) of the pressure loss coefficient correlations for short ($t/d < 0.1$) sharp- and **round-edged** orifices for

- $Re_d > 10^3$, **transitional** and turbulent flow (Class 2), and
- $1 < Re_d < 10^4$, **laminar** and **transitional** flow (Class 3).

Based on Miller's classification, Class 2 data are based on experimental data "where no detailed cross checking is possible against other sources". Class 3 data are "experimentally derived values from less reliable sources".

In **laminar flow**, **McNeil and Stuart**²³² provide the following correlation for square-edge short orifices with $\beta = 0.5$ and $10 \leq Re_D \leq 200$:

$$K_i = 91.1 Re_D^{-0.384} = 91.1 \left(\frac{Re_d}{\beta} \right)^{-0.384} \quad (6.16)$$

6.3.2 Long orifices

In **turbulent flow**, the pressure loss through a long orifice can be estimated applying the **Carnot-Borda** formula to the losses due to the irreversible expansion process downstream of the *vena contracta* to the orifice wall (theoretical entry loss) and then to the pipe wall (theoretical exit loss). If the loss upstream the *vena contracta* is assumed to be negligible, the Euler Number for a double expansion is given by:

$$E_u = \left(\frac{1}{C_c} - 1 \right) + (1 - \beta^2)^2 = \left(\frac{1}{C_c} - 1 \right) + (1 - \alpha)^2 \quad (6.17)$$

Dividing both sides of Equation (6.17) by β^4 , the pressure loss coefficient is given by:

$$K_i = \left(\frac{1}{C_c \beta^2} \right)^2 - 1 - \frac{2}{\beta^4} \left(\frac{1}{C_c} - 1 \right) - 2 \left(\frac{1}{\beta^2} - 1 \right) = \left(\frac{1}{C_c \alpha} \right)^2 - 1 - \frac{2}{\alpha^2} \left(\frac{1}{C_c} - 1 \right) - 2 \left(\frac{1}{\alpha} - 1 \right) \quad (6.18)$$

where C_c is the contraction coefficient, $C_c = A_{vc}/A_2$. A correlation for the contraction coefficient recommended by Weisbach¹ is presented in Equation (6.34).

Based on reviewed experimental data, **Ward-Smith**⁴¹ provides the following correlations for square-edged long orifices, valid for $Re_D \geq 10^4$ and for:

- (a) $t/d < 0.8$ and $0 < \beta < 0.75$ ($0 < \alpha < 0.87$):

$$K_i = \left[\frac{1}{0.608 \beta^2 (1 - \beta^{5.2}) (1 + (t/d)^{3.5}) + \beta^{7.2}} - 1 \right]^2 \quad (6.19)$$

with a root-mean-square error of 9.5%.

- (b) $0.8 < t/d < 7.1$ and $0 < \beta < 0.69$ ($0 < \alpha < 0.48$):

$$K_i = \left[\frac{1}{\beta^2 \left(0.872 - 0.015 \frac{t}{d} - 0.08 \frac{d}{t} \right) (1 - \beta^{6.6}) + \beta^{8.6} \left(1 + 0.134 \left(\frac{t}{d} \right)^{1/2} \right)^{-1}} - 1 \right]^2 \quad (6.20)$$

with a root-mean-square error of 3.8 %.

The **ESDU** correlations in ESDU 81039⁸⁴ are largely based on the data reported in Ward-Smith⁴¹. Pressure loss coefficients correlations are provided for Reynolds numbers $Re_d > 10^4$ (turbulent flow) in sharp-edged (knife- and square-edged), thin and thick orifice plates with edges free from any chamfering, rounding or burrs. The greater scatter of the experimental data in the region of $0.6 < t/d < 1$ suggests that this represents a probable overlap region between the separated- and reattached-flow regimes. It is not recommended to design orifices in this range because of the uncertain nature of the flow.

The pressure loss coefficient, K_i , is calculated as follows:

$$(a) \quad \text{for } t/d < 0.8 \quad K_i = \frac{K'_0}{\beta^4} \lambda_a = \frac{K'_0}{\alpha^2} \lambda_a = K_0 \lambda_a, \quad (6.21)$$

$$(b) \quad \text{for } t/d \geq 0.8 \quad K_i = \frac{K'_{0.8}}{\beta^4} \lambda_a = \frac{K'_{0.8}}{\alpha^2} \lambda_a = K_{0.8} \lambda_b. \quad (6.22)$$

The basic coefficients K_0 and $K_{0.8}$ are calculated from the corresponding values for K'_0 and $K'_{0.8}$, which are provided in graphical format. The correction factors, λ_a and λ_b , for the effect of t/d are also provided in graphical format.

For higher values of t/d , i.e. $0.98 < t/d < 7.1$, and for $0 < \beta < 0.69$, the following empirical correlation is given:

$$K_i = \left\{ \left[\beta^2 \left(0.872 - 0.015 \frac{t}{d} - 0.08 \frac{d}{t} \right) (1 - \beta^{6.6}) + \beta^{8.6} \left(1 + 0.134 \sqrt{\frac{t}{d}} \right)^{-1} \right]^{-1} - 1 \right\}^2 \quad (6.23)$$

Based on experimental data, **Idel'chik et al.**¹⁵³ provide the following semi-empirical correlation in turbulent flow for long ($0.015 < t/d < 10$) square-edged orifices and for $Re_d > 10^5$:

$$K_i = \left[\left((0.5 + \tau \sqrt{1 - \beta^2}) (1 - \beta^2)^2 + (1 - \beta^2)^2 + 0.02 \frac{t}{d} \right) \left(\frac{1}{\beta^2} \right)^2 \right] \quad (6.24)$$

where τ is an empirical parameter that depends on t/d and varies between 0 and 1.35.

Miller¹⁶⁹ provides plots (not equations) of the pressure loss coefficient correlations for long ($0 < t/d < 5$) square-edged orifices for $Re_d > 10^5$:

- $Re_d > 10^4$, turbulent flow (Class 3), and
- $1 < Re_d < 10^4$ and $\beta < 0.3$, **laminar** and **transitional** flow (Class 3).

For $t/d > 5$, Miller suggests estimating pressure loss by adding friction losses in the 5 diameter length. If then $\beta < 0.632$, it is recommended to sum the losses for a sudden contraction, sudden expansion, and friction losses over the orifice length. Based on Miller's classification, Class 2 data are based on experimental data "where no detailed cross checking is possible against other sources". Class 3 data are "experimentally derived values from less reliable sources".

In **laminar flow**, **Hasegawa et al.**¹⁷⁶ developed the following correlation for square-edged orifices with $\beta = 0.1$, $0 < t/d < 1.2$ and for $1 \leq Re_D \leq 1000$ based on their measurements and numerical studies, and reviewed experimental data:

$$Eu = 2.2 + \frac{K''}{Re_D} = 2.2 + \frac{K''}{\beta Re_d} \quad (6.25)$$

Values for K'' are provided from experimental data as $K'' = 37.7$, and from the numerical predictions as $K'' = 27$.

In **laminar** and **transitional** flow, Bohra²¹⁶ developed the following equations for square-edged orifices with square back-cut based on measurements for $0.32 < t/d < 5.72$, $0.02 < \beta < 0.137$ and $0.085 < Re_d < 9677$. The correlations are given in terms of the Euler number.

For $Re_d \left(= \frac{Re_D}{\beta} \right) < 6$, the following correlation is provided:

$$Eu = \left(\frac{1}{Re_d^{1.203}} \right) \left[64 \left(\frac{t}{d} \right)^{1.502} \left(\mu_r^{-0.47} + 36\pi \right) \right] \quad (6.26)$$

where μ_r is the ratio of the actual viscosity to the reference viscosity of 0.1 kg/(m s).

For $Re_d \left(= \frac{Re_D}{\beta} \right) < 6$, Euler Numbers for the laminar and turbulent flow regimes are combined to give:

$$Eu = \left[Eu_{lam}^3 + Eu_{turb}^3 \right]^{1/3} \quad (6.27)$$

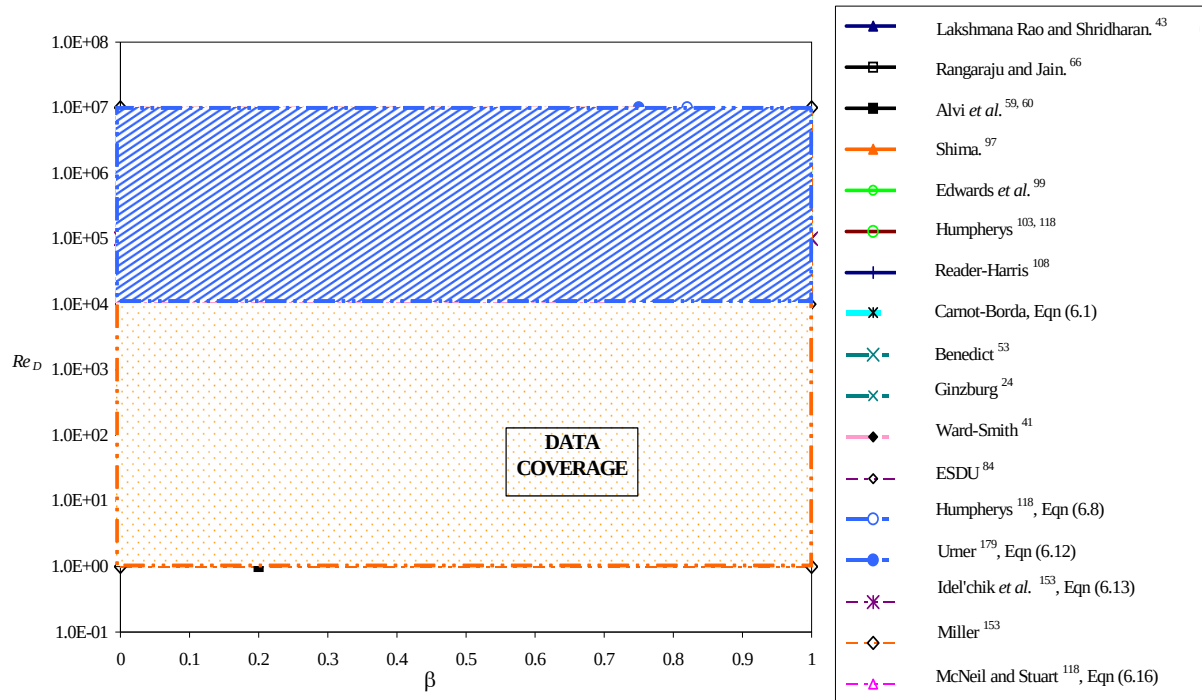
where the laminar flow Eu_{lam} has a similar form to that of Equation (6.26) for $Re_d < 6$, and Eu_{turb} is derived from the high Reynolds number expression for the flow through an expansion:

$$Eu_{turb} = \frac{1 - \beta^4}{C_d^2} \quad (6.28)$$

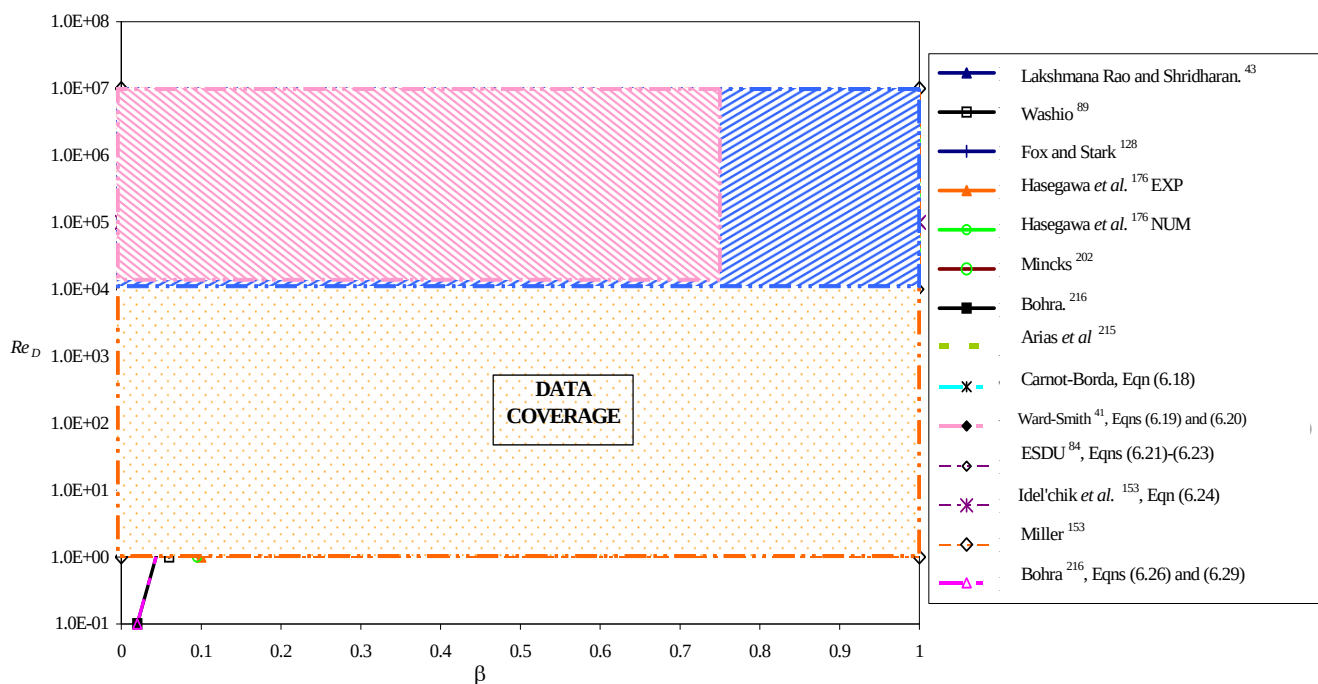
The combined expression for Eu is given as:

$$Eu = \left[\left\{ \left(\frac{1}{Re_d^{0.882}} \right) \left[64 \left(\frac{t}{d} \right)^{1.159} \beta^{0.075} \mu_r^{-0.334} + 17.16\pi \right] \right\}^3 + \left\{ \frac{1 - \beta^4}{C_d^2} \right\}^3 \right]^{1/3} \quad (6.29)$$

Equations (6.26) and (6.29) fit Bohra's experimental data within $\pm 25\%$. The values of C_d that fit Bohra's experimental data are given in Equations (6.64) to (6.66) by Morris and Garimella¹⁸².



Sketch 6.5 Pressure loss coefficient data (solid line) and correlations (dashed line) for short orifices



Sketch 6.6 Pressure loss coefficient data (solid and dotted line) and correlations (dashed line) for long orifices