Theoretical and Experimental Study on Fluid Flow in Valve Channels, Parts I and II

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SUMMARY
The object of this paper is to provide a basic understanding of the principal physical features of flow in valve channels. It will be demonstrated that incompressible jet flow theory is central to the flow phenomena under consideration. Influence of boundary layers on flow is discussed. In Part II flow loss data are presented in diagrams.

GENERAL DESCRIPTION OF FLOW IN COMPRESSOR VALVE CHANNELS.
As a first approach the flow in enlarged models of valve channels was observed in a smoke tunnel. Fig.1 gives sketches of typical observations. As the scale is 10:1 served. While these results were to be expected for a simple 90°-deflection flow it was somewhat surprising to find relatively clear jetlike conditions also for 2 x 90°-deflection flow.

The basic insights from these experiments are:
• Flow develops quite frictionless inside the valve channels similar to results predicted by jet theory of ideal fluids.
• Energy losses are produced behind the valve by diffusion of the emerging jets in a turbulent mixing process.

To prove these findings experiments were carried out with ring plate valves and pressure measurements inside the valve channels with small Pitot-probes (Ø 0.6 mm). The total pressure of the emerging jets was identical with the total pressure upstream the valve—or showed a slight depression only. Frequently the jets attached to a wall in the slot of the valve lift limiter (Coanda-effect, see Fig.2).

APPLICATION OF THE THEORY OF JETS
We discuss application of potential flow theory to valve flow using a special example. It is not the aim of this section to
to propose the broad use of detailed potential flow calculations in everyday design work but to deduce some useful information of general character which can be expressed in simple diagrams.

The channel was simplified that a two-dimensional velocity field was calculable. The flow field was composed of two potential flow solutions neglecting interference, Fig. 3. The valve channel entrance flow is approached by a flow through a cascade which forms jets. The free jet surfaces are areas of constant pressure and hence constant velocity. This problem has already been treated in literature by Betz and Petersohn[1]. Nothing is changed within the flow field if we imagine channel walls instead of the free jet surfaces. Stagnation points are formed in points marked A. From there velocity along the wall increases at first nearly linear, later very rapidly to reach the constant value at B. Then after short distance the side walls become nearly parallel. The calculated contour of the channel entrance may be regarded as somewhat like a "minimum rounded contour" for avoiding separation of flow in the entrance range. Constructed contours in valves should not remain under these curvatures. TABLE 1 next side gives some information for slotlike entrances and for a circular hole.

A second potential flow solution approaches the 90°-deflection flow forced by the valve plate. A jet emerging between parallel side walls and being deflected by a perpendicular plane is convenient for this part of valve flow. Some notes on this problem can be found in the book by Milne-Thomson[2]. The author himself has studied this problem in detail and made numerous calculations. The velocity along the side walls is nearly constant and shows rapid acceleration only very near to the edge opposite the perpendicular plane. For a particular example the velocity profile along the wall is plotted in Fig. 4. In the centre of the perpendicular plane a stagnation flow is established around stagnation point D. At C we find separation with constant pressure and constant velocity on the jet surface (downstream C). The velocity profile along the perpendicular plane is first nearly linear and later tends to the constant value which has already been established on the jet surface opposite. TABLE 2 gives a series of calculated jet contours. These curves may be used for the dimensioning of a chamfer at the sharp edge: The chamfer should be provided in such a way that the jet contour -now separating at the first edge- does not interfere with the seat plate. In special cases these curves may also be used for the construction of rounded off seat edges. As there is also constant pressure along these curves no separation will occur.

Van der Zanden[4] treated the more complex (plane) problem supposing finite width of the deflecting plate and also allowing varying angles of the plate, Fig. 5. It arises that for s/b ≈ 1 and e/b > 1.5 (ratios which often can be found in valves) there is little difference in the results when compared with infinite width of the deflecting plate. For smaller plates the final deflection angle of the jet is considerably smaller than 90° and loss coefficients become smaller. Numerical data including force coefficients are given in Part II.

For axially symmetric flows (e.g., circular holes etc.) methods using complex variables and conformal mapping are not available. For this reason very little information about potential flows can be found in literature. Experimental results gained at sufficient thin boundary layers are used to close this gap. Here we use results of Schrenk[5] gained with water as working fluid and with large models having diameters of about 50 mm. To which extent experimental results approach the results of potential flow depends on effects of viscosity which is discussed in the next section.

BOUNDARY LAYER CONSIDERATIONS

Beginning at the stagnation point A a...
TABLE 1 Valve channel entrance contours with constant pressure and velocity

a) for series of parallel slots with distance \( t \):

\[
y = \frac{b}{2+\sqrt{1+4(b/t)^2}} \left[ \ln \left( \frac{1+4(b/t)^2}{1-4(b/t)^2} \right) \right]^{1/2} - \frac{1}{2} (t/2b + 2b/t) \ln \left( \frac{1+4(b/t)^2}{1-4(b/t)^2} \right)
\]

\[
x = \frac{b}{2+\sqrt{1+4(b/t)^2}} \left[ \tan^{-1} \left( \frac{4b/t}{1-(2b/t)^2} \right) - \tan^{-1} \left( \frac{4b/t}{1-2b/t} \right) \right] \sin y
\]

\( \gamma \rightarrow \) parameter, \( \tan \gamma = y \)

b) for a single slot:

\[
y = \frac{a}{2+\sqrt{1+4(b/t)^2}} \left[ \ln \tan \left( \frac{\pi}{4} - \frac{\gamma}{2} \right) + \sin \frac{\gamma}{2} \right] \quad x = \frac{a}{2+\sqrt{1+4(b/t)^2}} \left( \cos \frac{\gamma}{2} - 0.5 \right)
\]

\( b = 0.611a; \gamma \) parameter \( \tan \gamma = y' \)

<table>
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**Fig. 4** a Potential flow velocities along walls and laminar boundary layers.

b Velocities along valve plate (e=\( \infty \))
boundary layer is formed along the valve channel walls (except at walls in contact with wakes). The thickness σ of the boundary layer is usually defined as the distance from the wall at which the velocity reaches 99% of that velocity obtained in the free stream (unaffected by frictional effects). To judge the influence of the boundary layer in our case we better use the displacement thickness δ_d of the boundary layer. δ_d is defined as the distance by which the wall must be displaced in order to compensate for the presence of the boundary layer with respect to flow rate when compared with frictionless flow. If e.g. a valve plate carries a boundary layer of constant displacement thickness δ_d the valve behaves in some aspects like a valve with an effective lift (s-δ_d). Boundary layers may flow laminar or turbulent.

The author has studied boundary layer problems in valves. He found out that from an engineering point of view it is not necessary to carry out detailed boundary layer calculations but just to apply existing solutions from boundary layer theory in an adequate manner. The 2 solutions which are most useful for this are: a) stagnation flow b) flow along a flat plate. Fig. 6 reviews some results from literature. Generally δ_d increase along the wall except if there is a rapid acceleration in the region outside the boundary layer. In this case the thickness may decrease. Knowledge of the velocity distribution along walls according to potential flow is necessary for boundary layer calculations. In Fig. 4 for a special example potential flow velocity along walls is plotted. Along the left wall of the central part the corresponding laminar boundary layer thickness δ is plotted for air as a fluid (1 bar, 20°C, kinematic viscosity ν = 15.10^-6 m^2/s).

For the boundary layer near stagnation point A eq(1) may be applied and yields the very small constant thickness of δ = 0.12 mm, δ_d = 0.03 mm. In point B δ decreases to about half the initial value due to rapid acceleration. Now a constant value of 40 m/s along the wall is achieved. The boundary layer develops just in the same way as with a flat plate as potential flow pressure and velocity are constant and curvature of wall is unimportant for boundary layer. δ grows continuously to a position about 0.01 mm before seat edge is reached to a maximum thickness of δ = 0.35 mm, δ_d = 0.117 mm. From this position to the seat edge poten-
potential flow: \( w_x = ax \), \( w_y = ay \)

streamlines

potential flow: \( w_x = ax \) \( w_y = ay \), streamlines

laminar boundary layer thickness:

\[
\delta = 2.4 \sqrt{\frac{v}{a}} = \text{const.}; \quad \delta_1 = 0.27 \cdot \delta 
\]

(1)

Fig. 6 Some boundary layer theory results

a) stagnation flow

total flow velocity increases rapidly from 40 to 76.1m/s and causes a decrease of boundary layer thickness. Unfortunately usual boundary layer theory does not cover extreme rapid accelerations of flow as found here. We may estimate that \( \delta \) decreases somewhere between a factor 40/76.1 = 0.53 and 1 to about 0.25mm. Downstream the seat edge the fluid within the boundary layer partly joins the wake and partly is entrained by the jet.

At the valve plate the boundary layer thickness is calculated by eq(1) using velocity distributions of potential flow (Fig.4b). As the velocity develops nearly linear beginning from stagnation point D \( \delta \) is nearly constant along the whole valve plate. Only for small lift ratios \( s/b \) \( \delta \) slightly decreases. The author has prepared a diagram which allows to find \( \delta \) in point D, Fig.7.

To use this diagram we have to form Reynolds number based on slot width \( 2b = 0.004m \) and the according velocity \( w = 40 \cdot 0.004/15 \cdot 10^{-6} = 10700 \). For \( s/b = 1 \) we find \( \delta_1 / \delta = 0.92 \) from the diagram and finally \( \delta = 0.018mm, \delta_1 = 0.27 \cdot 0.067mm \).

From this example we can learn that the boundary layers normally remain thin. We may neglect the very thin boundary layer caused by the impact of the fluid onto the seat plate. Also the boundary layer on the valve plate remains thin. What first merits attention when the flow velocities decrease in a valve and therefore boundary layer thicknesses increase is the boundary layer along the side walls of the seat plate.

We have not yet discussed under which conditions boundary layers remain laminar. Thin layers normally remain laminar (valve plate). To judge the layer on the seat channel side walls we have to form a Reynolds number based on velocity in the channel \( w_1 \) and thickness \( v \) of seat plate:

\[
Re_v = \frac{w_1 v}{\nu} = 40 \cdot 0.014/15 \cdot 10^{-6} = 37000
\]

Transition between laminar and turbulent boundary layer occurs at \( Re_{v, \text{crit}} = 1.2 \cdot 10^5 \) depending on general turbulence level in oncoming flow. An edge at point B produced by machining the seat plate) may further reduce \( Re_{v, \text{crit}} \). Surface roughness of a precisely cast seat plate was measured by the author to about 0.025mm. This value should not affect \( Re_{v, \text{crit}} \).

It arises that at high pressures \( Re_v \) becomes very large because kinematic viscosity becomes small and turbulent boundary layers are to be expected. Table 3 gives an
TABLE 3 Typical maximal boundary layer thicknesses in channels of seat plate

| Fluid | p | $\theta_c$ | $\nu$ | $w_1$ | $v$ | $Re_v$ | $\delta_{1}$ | $\delta_{1}$ | $\delta_{1}/b$ % |
|-------|---|-----------|-------|-------|-----|--------|-----------|-----------|----------------|----------------|
| R 22 | 20 | 10^-5     | 25    | 10    | 25000 | 0.10   | 0.034     | 1.7        | 1.9            |                |
| air   | 50 | 3.2.10^-5 | 15    | 15    | 7000  | 0.90   | 0.31      | 16         |                |                |
| air   | 100| 10^-5     | 40    | 12    | 32000 | 0.34   | 0.115     | 5.8        |                |                |
| air   | 100| 0.172.10^-5 | 25 | 40 | 5.8.10^5 | 0.66 | 0.08 | 4 |            |                |
| $H_2$ | 50 | 126.10^-6 | 120   | 14    | 13300 | 0.61   | 0.21      | 10.5       |                |                |
| $H_2$ | 100| 1.26.10^-6 | 60    | 40    | 1.9.10^6 | 0.82 | 0.093 | 4.7 |            |                |

impression of boundary layer thicknesses under various operating conditions. For a first judgement of the influence of boundary layer on flow losses the ratio $\delta_{1}/b$ is of interest. From table 3 one can see that $\delta_{1}/b$ usually is below 10%. Care should be taken when using loss coefficients derived from experiments with low pressure fans. The influence of compressibility on boundary layer thickness in valves is very small and can be neglected.

The influence of boundary layers on flow losses in valves may be attributed to two separate mechanisms:

- a "rounding effect" - which principally is favourable - reduces the velocity increase due to sharp seat edges. This effect does not occur when rounded off seat edges are used.

- a "displacement effect" which is unfavourable and acts like a reduction of actual valve lift.

ESTIMATION OF THE INFLUENCE OF VISCOSITY AND COMPRESSIBILITY

As previously has been pointed out incompressible jet flow theory together with basic geometric parameters are of primary importance for valve flow. The additional effects of viscosity and compressibility may be judged in a diagram Mach number $Ma = w_1/a$, Fig. 8. $\delta_{1}/b$ is a measure of viscous effects. $Ma$ is a nominal Mach number which can be calculated in an easy way from the basic operating parameters of the compressor:

$$Ma = w_1/a$$

(4)

where "a" is velocity of sound related to stagnant condition in the space upstream of the valve. The actual maximal velocities in a valve are higher than the calculated value $w_1$ due to jet contraction. Therefore critical conditions and choked flow already occur at a nominal Mach number of about 0.3.

The examples of table 3 are marked in Fig. 8. A valve operating in a compressor covers a certain range in this diagram due to piston velocity (and speed) variations. This corresponds to an inclined line in Fig. 8. The same is true for loss experiments with valves.

When adopting maximal errors of roughly 5% in the effective flow area (≈ 10% in loss coefficient), a horizontal and a vertical line can divide the whole field of the diagram Fig. 8 in 4 sections: 1, 2, 3, 4:

1. Effects of viscosity and compressibility are small. Constant values of loss coefficients or flow areas may be used.

2. Equations for compressible fluids must be

Fig. 8 Survey on influence of viscosity and compressibility
Fig. 9 Experimental results μ (see eq(5)) for a multi-ring-plate valve (5 circular slots, free area in the seat plate A = 26 cm², ²b = 3.5 mm). For this valve μ depends on a rather slightly and therefore was replaced by its mean value 0.706. - a) μ = μ(s) for values p₂/p₁ = 0.15 - 0.8 b) μ = μ(p₂/p₁) c) μ = μ(Reᵣ).

With respect to compressibility valves behave similar to nozzles in isentropic flow including choked flow condition. Choked flow condition is established at lower pressure ratios than theoretically calculated for nozzles (for the tested valve choked flow condition occurred at p₂/p₁ = 0.33 ± 0.04). This phenomenon is qualitatively understood and quantitatively expressed by 2 experimentally gained coefficients μ, λ:

\[ \dot{m} = \mu A \left[ \frac{2k}{k-1} p₁ p₂ \left( \frac{p₂}{p₁} - \frac{p(k+1)}{k} \right) \right] \]

with P = 1 - λ(1 - p₂/p₁)

μ = μ(s); λ = λ(s) or constant

\dot{m} mass flow rate
k ratio of specific heat
Eq(5) with constant values of $\mu, \lambda$ for a fixed lift $s$ approach the experimental values $\bar{m}$ within about $\pm 4\%$ for a wide range of pressure ratios $p_2/p_1 = 0$ to 0.8. For greater pressure ratios --which also belong to common operating conditions of compressor valves-- viscous effects become predominant. To express the experimental values of $\bar{m}$ exactly we now have to allow the coefficient $\mu$ to be a function not only of lift $s$ but also of Reynolds number $Re$:

$$\mu(s, Re)$$

(6)

Fig. 9 shows diagrams for $\mu$ and $\mu(Re)$ with $s$ as a parameter for the tested multi-ring-plate valve. It can be seen that viscous effects become important for $Re < 70,000$. The parameter $\lambda$ was roughly independent from lift ($\lambda = 0.706$) for the tested valve.

Let us now try to give a brief interpretation of the experimental results. In Fig. 9b we may divide the whole range of pressure ratios into 3 sections:

section 1: $p_2/p_1 = 0 \div 0.8$. The characteristics are: choked flow for very small pressure ratios; due to jet expansion a completely reattached supersonic jet emerges from the channel in the valve lift limiter, see Fig. 10. For greater pressure ratios the jet becomes subsonic.

section 2: $p_2/p_1 = 0.8 \div 0.95$. The jet contracts more because jet expansion due to compressibility effects is smaller. Flow is attached on both walls of the valve lift limiter. Pressure recovery takes place and this increases the values of $\mu$ --especially for large values of lift $s$. For small lifts the area ratio is unfavourable for pressure recovery.

section 3: $p_2/p_1 = 0.95 \div 1$. Completely incompressible flow conditions; no jet expansion. The channel in the valve lift limiter is too short for reattachment on both walls; Coanda-effect; no pressure recovery (see Fig. 2). Viscous effects become predominant. The inclination of the curves in Fig. 9c indicate a power of about 0.5 for $Re$, $Re^{0.5}$. Laminar boundary layers govern the flow characteristics.

A detailed discussion could include the influences of the various geometric parameters on flow characteristics. This may be of interest especially for the design process.

**CONCLUSIONS**

- Valve channel flow can be understood in detail when applying existing concepts of fluid flow theory.
- Incompressible jet flow theory is essential to flow phenomena in valve channels and allows to calculate the influence of geometric parameters.
- With respect to compressibility valves behave like nozzles in isentropic flow including choked flow conditions. Choked flow conditions are established at lower pressure ratios than those theoretically calculated for nozzles.
- Effects of viscosity can be understood by the concept of boundary layers. Among the occurring boundary layers those along the side walls of the seat plate are of primary importance.
- Better understanding of the valve channel flow gives a new background to:
  - meaningful correlations for loss data
  - planning and interpretation of experiments
  - optimization procedures
  - new valve concepts

**REFERENCES**


**Fig. 10** Interpretation of experimental results
THEORETICAL AND EXPERIMENTAL STUDY ON FLUID FLOW IN VALVE CHANNELS

PART II PRESSURE LOSS DATA

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INTRODUCTION

As previously pointed out frictionless incompressible jet flow (with separation at sharp edges) is essential to the present issues, appropriate data have been collected and produced by the author. These data are partly based on theoretical jet flow solutions partly on experimental results which have been gained with sufficiently thin boundary layers. In some cases pressure recovery according to momentum theorem (sudden enlargement) is included in these data. All the data taken from literature were checked carefully and had to be intensely processed to fit into the required context. Data based on theoretical jet flow solutions are marked by "Th", data based on experiments by "Ex". The latter are expected to be within a ±10% margin to the theoretical jet flow solution (not yet available in literature to the knowledge of the author). Data which include calculated pressure recovery (by momentum theorem) is marked by "Mo". In special cases data on force coefficients are also presented.

Pressure loss data here usually are given by the loss coefficient $f_s$ based on the equation

$$\Delta P = f_s \cdot \frac{1}{2} \rho \dot{V}_s^2 = f_s \cdot \frac{1}{2} \rho \dot{V}_s^2$$

where $\dot{V}_s$ is the volumetric flow rate (m$^3$/s) and $A_s$ is the area formed by the valve plate and the seat edge.

The discharge coefficient $C_{DI}$ used in incompressible flow equations is connected with the loss coefficient $f_s$ by

$$C_{DI} = \frac{1}{f_s}$$

where $C_{DI}$ is related to $A_s$.

The source of the data for each individual case is stated but it must be noted that often complicated calculations had to be done before arriving at the data presented.

In practice the entrance in the seat plate is often not rounded (as it is always supposed for the loss data presented) but sharp (rectangular). If the seat channel is of constant cross section and has a length $l > d$, reattachment of flow occurs within the seat channel. The additional loss coefficient can be estimated by the following formula

$$\Delta f_s = \delta (s/d)^2$$

For a valve comprising different channels with different individual loss coefficients $f_{si}$ a simple mathematical operation yields for the integral loss coefficient based on total area $A = \sum A_i$ (see Fig. 11)

$$f = \frac{1}{\left( \sum A_i \cdot \frac{1}{A \sqrt{f_i}} \right)^{1/2}} ; C_{DI} = \sum \frac{A_i}{A} f_{DIi}$$

Fig. 11 Integral loss coefficient

To demonstrate the application of this formula and the data presented we calculate the integral loss coefficient of the multi-ring-plate valve for which loss data have been measured and presented in [7].

This valve is based on a system of circular slots. We may approach this design by a series of long rectangular slots of type 8.1 and 8.3 (see data sheets next pages). The dimensions of these slots are given in Fig. 12. There are struts in the seat plate and in the valve lift limiter but flow is not disturbed by such struts in the space.
between valve plate and seat plate. The outer ring of the valve corresponds to case S.1, the remaining slots to case S.3. The table of Fig.2 makes clear the theoretical calculation of the discharge coefficient CDI.

On the other hand one can calculate an incompressible discharge coefficient C_D_I from the experimental results, Fig.9a.

For small pressure differences $p_2-p_1$ eq(5) in Part I yields

$$C_D = \mu \sqrt{\lambda}$$  \hspace{1cm} (10)

For the calculation of the experimental value $C_D$, $\mu$ and $\lambda$ are used from the range where viscous effects are not predominant ($p_2/p_1$=0 to 0.8). The results are compared in the diagram in Fig.2. The experimental values are larger for all lifts $s$ by a factor 1.40-0.03. This may be attributed to the following facts:

- Geometrical differences between a long rectangular slot (cases S.1; S.3) and the multi-ring-plate valve.
- While the slots in the seat plate are interrupted by struts, the area between the seat plate and the valve is free for flow throughout (area blocked by struts: 9.5 cm², free area in seat plate: 26 cm²).

-A rounding effect of the boundary layer may increase the effective flow area.

-Pressure recovery may occur in some slots.

It can be seen that besides a constant factor the jet flow theory can express valve loss behaviour adequately.

With some experience and feedback the results of jet flow theory may be very useful for the design process, especially for optimization procedures (where constant factors are not important).
PRESSURE LOSS DATA FOR LONG RECTANGULAR SLOTS

1 x 90°-deflection flow

S.1 Th [4]

area $A$

$F = A \cdot c_p (p_1 - p_2)$

$\alpha^\circ = 1$

$\alpha^\circ = 1.25$

$\alpha^\circ = 2$

$\alpha^\circ = 1.5$

$\alpha^\circ = 20$

$\alpha^\circ = 1$

$\alpha^\circ = 1.25$

$\alpha^\circ = 2$

$\alpha^\circ = 1.5$

$\alpha^\circ = 20$

S.2 Th Mo

area $A_s$

$F = A \cdot c_p (p_1 - p_2)$

$\alpha^\circ = 1$

$\alpha^\circ = 1.25$

$\alpha^\circ = 2$

$\alpha^\circ = 1.5$

$\alpha^\circ = 20$

$\alpha^\circ = 1$

$\alpha^\circ = 1.25$

$\alpha^\circ = 2$

$\alpha^\circ = 1.5$

$\alpha^\circ = 20$
Calculated values (Th, No)


--- Extrapolated curve
PRESSURE LOSS DATA FOR LONG RECTANGULAR SLOTS  2 x 90°-deflection flow

**S.3 Th* Mo**

No potential flow solution available in literature (to the knowledge of the author). An approach was used to calculate \( \frac{f_s}{f_b} \). In principle the 2 x 90°-deflection flow was composed by 2 independant 90°-deflection flows. A certain correction was used to account for the fact that the second 90°-deflection flow occurs without a stagnation point.

\[
\frac{f_s}{f_b} = 0.8
\]

\[
\frac{e}{b} = 15 \div 2
\]

\[
\begin{array}{c|c|c|c|c|c|c}
S & 2 & 3 & 4 & 5 & 6 & 7 \\
\hline
\frac{f_s}{f_b} & 0.5 & 0.7 & 1 & 1.2 & 1.4 & 1.5 & 2 \\
\end{array}
\]

**S.4 Th* Mo**

If \( l_2 > 2f \) reattachment of flow and pressure recovery occurs. For \( l_2 > 4f \) pressure recovery is finished. Application of momentum theorem results in the following formula:

\[
\frac{f_s,s.4}{f_s,s.3} = \frac{1}{1 - \frac{2s}{f_b} (\sqrt{f_s,s.3} - \frac{s}{f_b})}
\]

The amount of pressure recovery depends on width of slot 2f. Optimum width: two times jet width 2d (according to case S.3; 2d = 2a \( \sqrt{f_s,s.3} \)). In this special case the formula above is simplified to:

\[
\frac{f_s,s.4}{f_s,s.3} = \frac{1}{2}
\]

**S.5 Th* Mo**

No reattachment and pressure recovery occurs for \( l_2 < 1.5f \). Pressure recovery in the first 90°-deflection flow is finished at \( l_1 \approx 3 \div 4s \).

\[
f_b = 0.8 \quad 1 \quad 1.2
\]

**S.6 Th* Mo**

Pressure recovery in both 90°-deflection flows

\[
\frac{f_s}{f_b} = 0.8
\]

\[
\begin{array}{c|c|c|c|c|c|c}
S & 2 & 3 & 4 & 5 & 6 & 7 \\
\hline
\frac{f_s}{f_b} & 0.5 & 0.6 & 0.7 & 0.8 & 0.9 & 1 \\
\end{array}
\]
Loss data for circular configurations are derived from experimental results published in great detail by Schrenk. Experiments were carried out with large models (diameters of hole about 50mm) and water as working fluid. Data as loss coefficients, force coefficient, Reynolds number boundary layer thickness etc. were derived by the author from published data. Reynolds number effects seem to be weak in the concerned range ($\delta/d \approx 1\%$).

Conditions of transition:

\[ \frac{\delta}{d} \approx 1\%; \text{ influence of Reynolds number small} \]

\[ \frac{d_1}{d} > 1 \]

\[ \frac{d_2}{d} > 1.6 \]

\[ \frac{d_3}{d} > 1 \]

\[ \frac{d_4}{d} > 1.2 \]

\[ \delta_1/d \approx 0.9\% \]

\[ \delta_2/d \approx 0.1\% \]

\[ \delta_3/d \approx 0.3\% \]

\[ \delta_4/d \approx 0.7\% \]

\[ \delta_5/d \approx 1\% \]

Mean values for $\frac{d_1}{d} > 1.1$ and $\frac{1}{s} \leq 2$

(for $\frac{1}{s} > 2$ : pressure recovery $\rightarrow$ C.3!)
PRESSURE LOSS DATA FOR CIRCULAR CONFIGURATIONS

C.3 Ex

roughly linear decrease due to pressure recovery

C.4 Ex [5]

\[ \frac{d_1}{d} = 1.2 \]
\[ \frac{D}{d} = 1.4 - 3 \]

\[ \text{Re}_d = 33,000 \div 120,000 \]

C.5/C.6

Ex [5]

for:
\[ \frac{d_1}{d} = 0.70 \]
\[ \frac{d_2}{d} = 1.20 \]
\[ \frac{r}{d} = 0.15 \]

\[ \text{Re}_{d_1} = 71,500 = \text{const.} \]

\[ \delta_t / d_1 \approx 1\% \]