An experimental determination of settling length for turbulent flow of air in smooth annuli with square or bellmouth entrances.

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AN EXPERIMENTAL DETERMINATION OF SETTLING LENGTH FOR TURBULENT FLOW OF AIR IN SMOOTH ANNULI WITH SQUARE OR BELLMOUTH ENTRANCES

A THESIS

Submitted to the Faculty of Graduate Studies Through the Department of Mechanical Engineering in Partial Fulfillment of the Requirements for the Degree of Master of Applied Science at the University of Windsor

by

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Windsor, Ontario, Canada

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Settling lengths for turbulent isothermal flow of air in three smooth concentric annuli of diameter ratios 0.306, 0.527, and 0.842 were determined. Both square-edged and bellmouth entrances were investigated for Reynolds number ranging from 7,000 to 47,500.

Flow separation caused by the abrupt change in area of the square-edged entrance resulted in skewed (distorted) velocity profiles near the entrance. This skewness disappeared further downstream and fully developed mean velocity profiles were established. The settling lengths for a square-edged entrance varied from 25 to 35 diameters.

With a bellmouth entrance, the development of the velocity profile was conventional with a uniform profile near the inlet. The value of the settling lengths for the bellmouth entrance were 10 to 15 diameters more than the corresponding one for the square-edged entrance.
ACKNOWLEDGEMENTS

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**NOTATION**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
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<tbody>
<tr>
<td>$D$</td>
<td>Pipe diameter</td>
</tr>
<tr>
<td>$D_1$</td>
<td>Outside diameter of inner pipe</td>
</tr>
<tr>
<td>$D_2$</td>
<td>Inside diameter of outer pipe</td>
</tr>
<tr>
<td>$D_e$</td>
<td>Equivalent diameter, $D_2 - D_1$</td>
</tr>
<tr>
<td>$r$</td>
<td>Radial coordinate</td>
</tr>
<tr>
<td>$r_1$</td>
<td>Outside radius of the inner pipe</td>
</tr>
<tr>
<td>$r_2$</td>
<td>Inside radius of the outer pipe</td>
</tr>
<tr>
<td>$x$</td>
<td>Coordinate in axial direction with origin at beginning of constant-area section</td>
</tr>
<tr>
<td>$L_e$</td>
<td>Settling length</td>
</tr>
<tr>
<td>$u$</td>
<td>Velocity at radius $r$ in the axial direction</td>
</tr>
<tr>
<td>$(U_{av})_x$</td>
<td>Local average velocity at location $x$</td>
</tr>
<tr>
<td>$U_{av}$</td>
<td>Average velocity</td>
</tr>
<tr>
<td>$(U_{max})_x$</td>
<td>Local maximum velocity at location $x$</td>
</tr>
<tr>
<td>$U_{max}$</td>
<td>Maximum velocity in the fully developed flow</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Mass density</td>
</tr>
<tr>
<td>$v$</td>
<td>Kinematic viscosity</td>
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</tbody>
</table>
\( P_t \) \quad \text{Total pressure}  \\
\( P \) \quad \text{Static pressure}  \\
\( Re \) \quad \text{Reynolds number, } Re = \frac{U_{av} (D_2 - D_1)}{v}
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CHAPTER I

INTRODUCTION

It is well known that when a fluid enters a duct from a plenum chamber the velocity distribution at the entrance is practically uniform over the cross-section. The velocity at the wall is, however, zero, so that an infinitely thin boundary layer is formed round the walls of the pipe, the thickness of which increases further downstream. As the mass flow remains constant, the flow in the central core must accelerate to compensate for retardation of flow near the wall. The velocity distribution changes, as the thickness of the boundary layer increases along the length of the duct. When the boundary layer reaches the centre of the duct, the velocity distribution establishes a pattern which remains invariant thereafter. The portion of the duct in which the velocity development occurs is called the hydrodynamic entrance length or the settling length, and the flow in this region is called the developing flow or the entrance region flow.

Numerous theoretical and experimental investigations have been made to determine the settling length in the flow through pipes and between parallel plates.

In industrial heat exchangers and atomic reactors there are many cases where heat transfer occurs in the entrance region of annular ducts. In the theoretical analysis of this heat transfer the velocity distribution and the settling length are important factors. No detailed investigation has so far been made to determine the settling length.

In the limiting case for annular flow where the ratio of the
diameter of the inner pipe to the diameter of the outer pipe approaches 1.0, the flow is the same as that between parallel walls. Also, when the ratio of the diameters approaches zero the flow is comparable to pipe flow except for the condition of zero velocity on the axis. Thus the study of annular flow over a range of diameter ratios presents a link between pipe flow and the flow between parallel walls.

The determination of settling length is more important for turbulent flow as most flows that occur in nature are turbulent. If the fully developed flow is turbulent, the entrance configuration is of prime importance as it affects the flow downstream. If the entrance is abrupt (square-edged) the boundary layer would probably be turbulent from the beginning and if it is rounded (bellmouth) the boundary layer may be laminar immediately beyond the entrance and then become turbulent some distance downstream.

The aim of this investigation is to determine the settling lengths for annuli of different diameter ratios over a range of Reynolds number using square-edged and bellmouth entrances.
CHAPTER 2

LITERATURE SURVEY

A review of past literature reveals that although a considerable amount of work has been done on annular flow, most investigators have been concerned with the structure of fully developed velocity profiles, and little work has been done on developing flow in the entrance region of annular ducts.

As flow through tubes and between parallel plates are limiting cases of the flow through an annulus, this literature is also reviewed.

2.1 Theoretical Analyses

Boussinesq (Ref. 1) employed an approximate form of the Navier-Stokes equation to obtain a solution for the development of the velocity profile for laminar flow in round tubes. His results led to a predicted length-diameter ratio for a purely laminar inlet zone of

$$\frac{L_e}{D} = 0.065 \text{ Re}$$

Schiller (Ref. 2) investigated the laminar inlet zone with the aid of Karman's momentum theory for the boundary layer. After assuming that the typical velocity profile near the inlet was composed of a straight-line segment terminated by parabolic arcs, he applied the momentum equation to the entire cross section and the Bernoulli equation to the central, frictionless core of the fluid. The rate of development of the velocity profile and the pressure drop from the entrance were predicted. For the length of a purely laminar inlet zone he obtained

$$\frac{L_e}{D} = 0.029 \text{ Re}$$

(2-2)
Atkinson and Goldstein (Ref. 3), also using an approximate form of the Navier-Stokes equation, improved the results of Boussinesq near the entrance.

Langhaar (Ref. 4) presented the most complete analysis for laminar flow. He retained more terms of the Navier-Stokes equation than had been retained by other investigators and obtained a solution by a linearizing procedure. His results are in the form of a table from which velocity profiles and pressure drops could be computed. Unlike the previous work in this field, no definite inlet length was found, but instead the velocity profile was found to approach the parabolic form in asymptotic fashion. The length in which the center-line velocity reaches 99 percent of its asymptotic value was predicted to be

\[
\frac{X}{D} \text{99 percent inlet} = 0.115 \Re_e \tag{2-3}
\]

Latzko (Ref. 5) analyzed the development of a turbulent velocity profile in a tube by a method analogous to that of Schiller (Ref. 2). The basic assumption of Latzko's analysis was that a typical velocity profile in the inlet length was composed of a straight-line segment terminated by arcs, the velocity distribution of which followed the one-seventh-power law. For the total inlet length of a purely turbulent flow he obtained

\[
\frac{L}{D} = 0.69 \Re_e^{0.25} \tag{2-4}
\]

Sparrow (Ref. 6) studied laminar flow in the inlet section of two parallel planes, and obtained the relation

\[
\frac{L_e}{2b} = 0.00648 \Re_e \tag{2-5}
\]
where \( b \) is the distance between the parallel planes.

Sparrow and Lin (Ref. 7) have obtained a solution for the developing laminar flow in annular ducts, by linearizing the inertia terms appearing in the equations of motion. The solution indicates that even for a radius ratio substantially less than unity (0.4) the development is similar to that of flow between parallel plates. On the other hand, for radius ratios as small as 0.001 the results depart significantly from those for a circular tube. Murakawa (Ref. 8) has also obtained an approximate settling length for the laminar flow in an annulus by using a series type solution.

Heaton, Reynolds and Kays (Ref. 9) have used an integral method similar to the one used by Langhaar (Ref. 4) for tubes and obtained the entrance length for laminar flow in an annulus. The final results are in the form of tables from which velocity profiles could be determined.

2.2 Experimental Investigations

Schiller and Kirsten (Ref. 10) studied the development of the velocity profile for turbulent flow in pipes for both rounded and sharp edged entrances. For a Reynolds number ranging from 10,000 to 50,000, they observed that an entrance length of 50 to 100 equivalent diameters was necessary for the formation of a fully developed velocity profile. They also noted that for a rounded entrance, the settling length decreased with an increase of Reynolds number, but for a sharp edged entrance the opposite was true.

Nikuradse (Ref. 11) measured the velocity profiles for a laminar inlet region of a pipe. A comparison with the theories (Refs. 1, 2, 3 and 4) indicates, that on the basis of center-line velocities Langhaar's treatment is the best over the entire inlet region. The method of
Schiller predicts the center-line velocities accurately at points close to the entrance and poorly at points near the end of the inlet zone. The methods of Boussinesq and of Atkinson-Goldstein, on the other hand, compare well with Nikuradse's measurements near the end of the inlet region and compare poorly near the entry to the pipe.

Deissler (Ref. 12) measured velocity profiles at various distances from rounded and right-angle-edge entrances for the flow of air in a tube over a range of Reynolds number from 48,000 to 580,000. His results indicated that the flow development was more rapid for the right-angle entrance than for the rounded entrance. With the right-angle entrance, fully developed flow was obtained after about 4.5 tube diameters from the entrance, but with the rounded entrance the flow was still developing slightly at 100 tube diameters from the entrance. For both entrances, however, the flow close to the wall developed in a much shorter distance than did the flow in the center of the tube. The flow close to the wall determines the shear stress, so that the effect of entrance on the shear stress at the wall is slight except for very small tubes.

Brighton and Jones (Ref. 13) have reported on settling lengths for turbulent flow of air in annuli. They obtained a settling length of 34.5 diameters with the aid of inlet roughness elements and screen. Without these, the inlet length was much greater. Hart and Lawther (Ref. 14) in the discussion of the paper by Brighton and Jones (Ref. 13) pointed out that this was contrary to their experience for pipe flow. Sand-roughening the pipe wall at the inlet had no effect on the development length and screens were similarly ineffective.

Rothfus, Monrad, Sikchi, and Heideger (Ref. 15) reported that the
outer wall shear stress was still approaching its asymptotic value at 250 equivalent pipe diameters from the entrance for the annulus tested. Further, it was found that neither the level of turbulence at the inlet nor the Reynolds number (at least for $5,000 < R_e < 45,000$) had much effect away from the entrance.

Quarmby (Ref. 16) has reported that his settling length results agreed with the results of Brighton and Jones (Ref. 13), that is, it was of the order of 30 to 40 equivalent diameters. His settling length investigations were not detailed enough to determine the effects of Reynolds number and radius ratio. The inside surface of the tube was honed and the Reynolds number ranged from 10,000 to 90,000.

Henseler and Howard (Ref. 17) during their investigations on annular diffusers found that the settling length was of the same order as reported by Brighton and Jones (Ref. 13). These workers investigated two different annuli of diameter ratios 0.51 and 0.76 in the turbulent flow range. The annulus had a bellmouth entrance and there were two fly screens with paper tubes in between, in the plenum chamber before the test section.

Goel (Ref. 18) investigated the settling length for turbulent flow of air in four, square entrance annuli. The Reynolds number ranged from 5,000 to 50,000 and diameter ratios from 0.2 to 0.7. He obtained correlations relating the settling length, equivalent diameter, Reynolds number, and diameter ratios.

For Reynolds number lower than 22,000 he obtained

$$L_e/D_e = 0.795 R_e^{0.374} (D_1/D_2)^{-0.60}$$

and for Reynolds number greater than this the relation was
\[
L_e/D_e = 15.96 \, Re^{0.077} \, (D_1/D_2)^{-0.624}
\]  \hspace{1cm} (2-7)

From the above paragraphs, it is clear that the value of settling length for annular flow is uncertain.
CHAPTER 3

THE EXPERIMENTAL APPARATUS

The apparatus used for the present investigation is shown in Fig. 1. Details of the equipment are discussed in the following sections.

3.1 Flow System

3.1.1 Fan and Plenum Chamber

The air was supplied by an American Blower Corporation, type v2, industrial centrifugal fan driven by a 208 volt, 3 phase, 3 h.p. induction motor. The fan gave a very satisfactory performance during the experimental investigation. Flow of air from the fan to the annulus was controlled by a butterfly valve placed at the delivery side of the fan.

The plenum chamber was a 3' x 3' x 3' sheet metal box. One of the sides of the box was removable, to provide easy access to the test section entrance. A rubber gasket was used to prevent leakage of air. The air inlet and outlet to the plenum chamber were directly opposite to one another. A baffle 8 inches in diameter was located 8 inches from the entrance to the chamber to help distribute the flow. At 10 inches and 12.75 inches from the baffle 40 mesh screens were installed. The baffle and screens were removable.

3.1.2 Test Section

The annuli were made from 12 foot lengths of aluminum tubing. The details of the sizes are given in Table 1. Measurement of the sizes of the pipe at various locations on the periphery and along the length
indicated a maximum deviation of 0.004-inch.

The inner tube was supported concentrically in the outer one by sets of two pins, made from 0.060-in. drill rod, located at four foot intervals along the annulus. The pins passed through predrilled holes in the inner and outer tubes which were 0.50-in. apart and at right angles to each other. When the pins were inserted the inner tube had to align itself with the outer one. The maximum eccentricity was found to be about 3% based on passage width.

The bellmouth entrances were made from wood and the inside surface was smooth and flush with the inside of the outer pipe. Figure 2 shows the dimensions of the bellmouth.

Finely machined wooden plugs of hemispherical shape (Fig. 2) were pushed into the two open ends to prevent air passing through the inner tube. The same core plugs were used for both the entrance configurations. Geometrically similar core plugs were used for different annuli.

3.2 Pressure Measurements

Wall taps were used for the static pressure measurement. Holes of 0.040-in. diameter were drilled in the outer tubes and internal burrs were removed. One inch sections of 0.25-in. tubing were fastened to the outer pipes around the 0.040-in. diameter holes.

Diametrically opposite to the static pressure holes small slots were machined in the outer tube to provide access for the total pressure probe. The pitot tube was made from stainless steel hypodermic needle tubing having a 0.028-in. outside diameter and 0.014-in. opening. A two inch micrometer head was mounted to a housing which contained the pitot tube assembly under spring tension (Fig. 4). This arrangement permitted the
pitot tube to be positioned with an accuracy of 0.001-in. The micro-
meter housing with the pitot tube was mounted on a heavy base plate. A
threaded shaft between the housing and the base plate provided the
vertical movement of the pitot tube. The pitot tube and the pipes were
connected to a dry battery and a bulb to indicate the contact between
the probe and wall.

The dynamic pressure was measured using a micromanometer. Tygon
tubes of 0.125-in. inside diameter were used to connect the manometer to
the pitot tube and wall tap.
Before starting test runs each annulus was tested for eccentricity using a depth micrometer. The maximum eccentricity was 3% based on the passage width for the different annuli investigated.

The flow rate through the annulus was varied by operating the butterfly valve on the delivery side of the fan. After a desired flow rate had been established, the system was allowed to run for about 20 to 25 minutes before recording the data.

The initial position of the pitot tube was fixed as being the point where the pitot tube initially came into contact with the inner surface of the outer tube. This was done by establishing an electrical contact between the probe and the wall of the annulus. The pitot tube was moved away from the outer wall with the traversing mechanism and positioned at seventeen to twenty-five points in the passage width to obtain the velocity distribution. At the completion of a test run, the dynamic pressure at the initial position was checked for a change in flow rate. The room temperature and barometric pressure were measured before and after each individual test run.

Investigations were made at different Reynolds number for each of the configurations. It was not always possible to repeat the runs at the same Reynolds number for the different entrance conditions, because the butterfly valve controlling the flow was sensitive at part openings. The Reynolds number ranged from 7,000 to 47,500. The maximum Reynolds number was limited by the capacity of the fan.
Average velocity in each case was obtained from velocities at ten points representing ten equal area elements of the annular cross section. These ten points were included in the traverse.

The configuration and the flow conditions investigated are given in Table 3.

The dynamic pressure was measured to an accuracy of 0.001-inch of water. Ambient air temperature was recorded to the nearest 0.5°F. Barometric pressure was obtained to the nearest 0.005-inch of mercury.
CHAPTER 5

EXPERIMENTAL RESULTS

5.1 Reduction of Data

The fluid was assumed to be incompressible and velocities were calculated from the equation

\[ P_t - P = \frac{1}{2} \rho \ u^2 \] (5.1)

where \( \rho \) was determined from the ambient conditions. The static pressure was assumed to be uniform across the annulus.

The velocity was assumed to represent axial point velocity at the geometrical center of the pitot-tube opening.

In order to obtain the ratios of local to average velocity, the average velocity \( U_{av} \) was obtained by averaging the ten values obtained at equal area elements. This value was compared with that obtained by the area integration method. The difference between the two values was found to be negligible.

The experimental data was reduced using an IBM 1620 computer. The input to the computer were ambient temperature, barometric pressure, and dynamic pressure readings. The computer printed the point velocities, average velocity, and the velocity ratios \( u/(U_{av}) \).

5.2 Discussion of Results

5.2.1 Square-Edged Entrance

Figure 5 shows the mean velocity profiles obtained for annulus A.
at Reynolds number 44,700. In this Fig. the velocity profiles at
different distances from the entrance are superimposed on one another.
It is seen from the figure that the velocity profile initially undergoes
a development and finally establishes a fixed pattern.

The velocity profile close to the entrance is skewed (distorted)
towards the core, as seen from Fig. 5. The skewness disappears further
downstream before the fully developed profile is established. This
particular phenomenon was also observed for the other two annuli with
square-edged entrances (see Figs. 6 and 7) and at all Reynolds numbers
investigated. The skewness was also observed by Okiishi and Serovy
(Ref. 25) and they explained this as due to flow separation at the
entrance caused by abrupt change in the flow area. Goel (Ref. 18) had
not observed in his investigations, the skewness of the velocity profiles
near the inlet.

The variation of local maximum velocity along the annulus length
for different Reynolds number is plotted non-dimensionally using the
local average velocity and the equivalent diameter (Fig. 8). Settling
lengths were obtained with this type of plot by determining the point
where \( \left( \frac{U_{\text{max}}}{U_{\text{av}}} \right)_x \) reached a constant value. It can be seen from
Fig. 8 that the Reynolds number has little effect on the settling length.
Further, for the fully developed flow the Reynolds number has practically
no effect on the velocity distribution (Fig. 9), and hence the ratio of
maximum to average velocity is almost a constant. Plots similar to
Fig. 8 are shown in Figs. 10 and 11 for annuli B and C respectively.
The vena-ccntracta which appears to have been formed in annulus A is
due to flow separation caused by the abrupt change in area.
Effects of Reynolds number and diameter ratio on settling length are shown in Fig. 12. The settling length is increasing with an increase in diameter ratio. This is in agreement with the findings of Okiishi and Serovy (Ref. 25) and in disagreement with that reported by Goel (Ref. 18).

The settling length is increasing initially with Reynolds number but at values higher than 22,000 it is almost constant.

To get a non-dimensional velocity term, Goel (Ref. 18) used the fully developed maximum velocity instead of the average velocity. He obtained the settling lengths from plots of \( \left( \frac{U_{\text{max}}}{U_{\text{max}}} \right)_x \) versus \( x/D_e \). Lack of flow symmetry about the axis and eccentricity of the annulus could lead to a continuous increase in the local maximum velocity, apparently leading to larger values of settling length. This may be partly overcome by using local value of the average velocity instead of the fully developed maximum velocity.

It was originally thought that the difference in the settling lengths obtained by Goel (Ref. 18) and those in the present investigation was due to difference in entrance conditions. His entrance condition may be described as a re-entrant type rather than a square-edged one (Ref. 26). No significant difference in results was noted between the re-entrant and square-edged entrances (Fig. 12a).

The following correlation (Fig. 13) has been obtained relating the settling length, equivalent diameter, Reynolds number and diameter ratio, for a square-edged entrance. For Reynolds number ranging from 12,000 to 45,000

\[
\frac{L_e}{D_e} = 27.9 \quad (R_e)^{0.015} \quad \left( \frac{D_1}{D_2} \right)^{0.21} \quad (5-2)
\]
It can be stated that the settling length for annuli with a square-edged entrance generally lies between 25 and 35 diameters. This is shorter than that for pipe flow.

The maximum error in fixing the values of the settling length is about 5 to 6 diameters based on the spacing of the stations and the rate of development.

5.2.2 Bellmouth Entrance

The mean velocity profiles obtained at different distances from the entrance, superimposed on one another are shown in Fig. 14 for annulus A. The velocity distribution very near the entrance is uniform and it develops as the flow proceeds downstream. The development length obtained with the bellmouth entrance was generally a few diameters more than the corresponding one with a square-edged entrance. This is due to the initial part of the boundary layer remaining laminar for a few diameters from the entrance. The distance which the boundary layer remains laminar would depend to a large extent on the initial level of turbulence. In the present investigation no measurement of turbulence level was made. Figures 15 and 16 are respectively for annuli B and C.

Figures 17, 18, and 19 show the variation of \( \frac{U_{\text{max}}}{U_{\text{av}}} \) versus non-dimensional distance along the annulus at different Reynolds numbers for annuli A, B, and C respectively. It can be seen from Figs. 17-19 that the settling length is comparatively shorter at lower values of the Reynolds number, although the difference is very small over the range of the investigation.

Figure 20 is a plot of settling length versus Reynolds number for different diameter ratios. The general trends regarding the variation
of settling length with diameter ratios and Reynolds number are the same as for the square-edged entrance. However, the value of the settling length are 10 to 15 diameters larger (Figs. 21, 22 and 23).

The following correlation (Fig. 24) for the bellmouth entrance has been obtained for Reynolds number greater than 22,000

\[ \frac{L_e}{D_e} = 42.6 \ (R_e)^{0.006} \ (D_1/D_2)^{0.20} \]  

(5-3)

5.2.3 Effect of Screens in the Plenum Chamber on the Settling Length

In the beginning of the investigation the air was blown directly from the blower through the plenum chamber into the test section. It was felt that the short settling lengths obtained might be due to the jet effect caused by the direct short circuiting of the air from inlet to outlet of the plenum chamber. A baffle to divert the air and two 40 mesh screens spaced 2.75 inches apart were installed in the plenum chamber and runs were made. Figure 25 shows the velocity profile near the inlet with and without screens for annulus A with bellmouth entrance. It is clear from the Fig. that the presence of baffle and screens has reduced the jet effect, as the velocity profile near the inlet is flatter, than without them. The final development length was, however, found to be the same. The presence of screens did, however, reduce the fluctuations partially. Figure 26 shows the variation of maximum velocity along the length of the annulus A with and without screens at Reynolds number 13,700.

Figures 27 and 28 are the same plots as Figs. 25 and 26 respectively for the annulus A with square-edged entrance. No appreciable change in velocity profile is noticed.
1. The settling or entrance length based on equivalent diameter for the turbulent flow of air in an annulus is shorter than that in a pipe flow.

2. The settling length increases with the annulus diameter ratio.

3. The settling length initially increases with Reynolds number but remains constant at values higher than 23,000.

4. Correlation for settling length in terms of equivalent diameter, Reynolds number, and diameter ratios are obtained:
   - With a square-edged entrance for $R_e > 12,000$
     \[
     \frac{L_e}{D_e} = 27.9 \ (R_e)^{0.015} \ (D_1/D_2)^{0.21}
     \]
   - With a bellmouth entrance for $R_e > 22,000$
     \[
     \frac{L_e}{D_e} = 12.6 \ (R_e)^{0.006} \ (D_1/D_2)^{0.20}
     \]

5. An annulus with a bellmouth entrance has a settling length, a few diameters larger than the one with a square-edged entrance.

6. The velocity profile near the inlet depends on the entrance geometry but after a few diameters downstream the profile becomes independent of entrance geometry.
7. In the case of square-edged entrance the velocity profile close to the entrance is skewed towards the core.

8. For fully developed flow, the Reynolds number has practically no effect on the velocity distribution and hence the ratio of maximum to average velocity is a constant.
BIBLIOGRAPHY


FIG. 5 DEVELOPMENT OF VELOCITY PROFILE FOR ANNUlus A
WITH SQUARE-EDGED ENTRANCE ($Re = 44,700$)
FIG. 6. DEVELOPMENT OF VELOCITY PROFILE FOR ANNULUS B WITH SQUARE-EDGED ENTRANCE \((R_e = 22,100)\)
FIG. 7: DEVELOPMENT OF VELOCITY PROFILE FOR ANNULUS C
WITH SQUARE-EDGED ENTRANCE ($R_e = 23,500$)
FIG. 3: PLOT OF \( \frac{(U_{max})_x}{(U_{av})_x} \) VERSUS \( X/d_e \) AT DIFFERENT REYNOLDS NUMBERS FOR ANNULUS A WITH SQUARE-EDGED ENTRANCE.
FIG. 9  FULLY DEVELOPED VELOCITY PROFILE FOR ANNULUS B
WITH BELLMOUTH ENTRANCE AT DIFFERENT REYNOLDS NUMBER
FIG. 10  PLOT OF \( \frac{(U_{\text{max}} x)}{(U_{\text{av}} x)} \) VERSUS \( \frac{x}{D_e} \) AT DIFFERENT REYNOLDS NUMBER FOR ANNULUS B WITH SQUARE-EDGED ENTRANCE
FIG. 11  PLOT OF \( \frac{(U_{\text{max}})_x}{(U_{\text{av}})_x} \) VERSUS \( \frac{x}{D_e} \) AT DIFFERENT REYNOLDS NUMBER FOR

ANNULUS C WITH SQUARE-EDGED ENTRANCE
FIG. 12 VARIATION OF SETTLING LENGTH WITH REYNOLDS NUMBER 
AND DIAMETER RATIOS (SQUARE-EDGED ENTRANCE)
FIG. 12a. PLOT OF \( \frac{(U_{\text{max}})_x}{(U_{av})_x} \) VERSUS \( \frac{x}{D_e} \) FOR ANNULUS B WITH RE-ENTRANT AND SQUARE-EDGED ENTRANCES (Re = 32,000)
FIG. 13 EXPERIMENTAL DATA FOR SQUARE-EDGED ENTRANCE SHOWN ON REGRESSION LINE, 12,000 < \( \rho_e < 15,000 \)

\[ 27.9 \left( \frac{R_e}{D_e} \right)^{0.015} \left( \frac{D_1}{D_2} \right)^{0.21} \]

(Correlation Coefficient 0.9663)
FIG. 14. DEVELOPMENT OF VELOCITY PROFILE FOR ANNULUS A WITH BELLMOUTH ENTRANCE ($R_e = 47,000$)
FIG. 15: DEVELOPMENT OF VELOCITY PROFILE FOR ANNULUS B

WITH BELLMOUTH ENTRANCE ($Re = 42,500$)
FIG. 16 DEVELOPMENT OF VELOCITY PROFILE FOR ANNULUS C
WITH BELLMOUTH ENTRANCE \( (R_e = 23,500) \)
FIG. 17  PLOT OF $\frac{(U_{\text{max}})_x}{(U_{\text{av}})_x}$ VERSUS $\frac{x}{D_e}$ AT DIFFERENT REYNOLDS NUMBER FOR ANNULUS A WITH BELLMOUTH ENTRANCE
FIG. 10 PLOT OF \( \frac{u_{\text{max}}}{u_{v_{x}}} \) VS. \( x/D_{e} \) AT DIFFERENT REYNOLDS NUMBER FOR ANNULUS B WITH HILLCOURT ENTRANCE
FIG. 20. VARIATION OF SETTLING LENGTH WITH REYNOLDS NUMBER AND DIAMETER RATIOS (BELLMOUTH ENTRANCE)

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FIG. 21: PLOT OF $(U_{\text{max}}/x)/(U_{\text{av}}/x)$ VERSUS $x/D_e$ FOR ANNULUS A WITH SQUARE-EDGED AND BELLMOUTH ENTRANCES ($P_e = 22,350$)
FIG. 22. PLOT OF $\frac{U_{\text{max}}}{U_{\text{av}}}$ VERSUS $\frac{x}{D_e}$ FOR ANNULUS B WITH SQUARE-FDGD AND BELL-MOUTH ENTRANCES ($Re = 32,000$)
FIG. 23 - PLOT OF $\frac{(U_{\text{max}})_x}{(U_{av})_x}$ VERSUS $x/D_e$ FOR ANNULUS C WITH SQUARE-EDGED AND BELLMOUTH ENTRANCES ($r_e = 23,500$)
FIG. 24. EXPERIMENTAL DATA FOR BELLMOUTH ENTRANCE SHOWN ON REGRESSION LINE, \(22,000 \leq R_e \leq 47,500\)

\[ (R_e^{0.006}) (D_1/D_2)^{0.20} \]

(Diameter Ratios)
- 0.306
- 0.527
- 0.642
FIG. 25  EFFECT OF SCREENS & BAFFLE ON VELOCITY DISTRIBUTION
NEAR THE ENTRANCE FOR ANNULUS A WITH BELLMOUTH ENTRANCE

\( \frac{r_2-r}{r_2-r_1} \)

\( \frac{u}{(U_{av})_x} \)

With Screens & Baffle
Without Screens & Baffle

\( (R_e = 13,700, x/D_c = 5.3) \)
FIG. 26: PLOT OF $\frac{U_{\text{max}}}{U_{\text{av}}} \times \frac{x}{D_e}$ VERSUS $x/D_e$ FOR ANNULUS A WITH AND WITHOUT SCREENS AND BAFFLE (Bellmouth Entrance, $Re = 13,700$)
FIG. 27  EFFECT OF SCREENS AND BAFFLE ON VELOCITY DISTRIBUTION
NEAR THE ENTRANCE FOR ANNULUS A WITH SQUARE-EDGED ENTRANCE

\( \frac{r_2 - r}{r_2 - r_1} \)

\( u \left( \frac{u}{u_{av}} \right) \)

With Screens & Baffle
Without Screens & Baffle

\( (Re = 13,420, x/D_e = 5.3) \)
FIG. 28. PLOT OF \( \frac{u_{\text{max}}}{u_{\text{av}}^x} \) VERSUS \( x/D_e \) FOR ANNULUS A WITH AND WITHOUT SCREENS AND BAFFLE (Square-edged Entrance, \( D_e = 13,420 \))
Table 1. Pertinent dimensions of annuli

<table>
<thead>
<tr>
<th>Annulus designation</th>
<th>Inner-tube outside diameter in.</th>
<th>Outer-tube inside diameter in.</th>
<th>Diameter ratio</th>
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<tbody>
<tr>
<td>A</td>
<td>0.500</td>
<td>1.634</td>
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<tr>
<td>B</td>
<td>1.250</td>
<td>2.370</td>
<td>0.527</td>
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<tr>
<td>C</td>
<td>4.000</td>
<td>4.750</td>
<td>0.842</td>
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Table 2. Traverse-station location

<table>
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<tr>
<th>Axial location number</th>
<th>Distance in inches from the beginning of constant area section, x</th>
<th>( \frac{x}{D_2-D_1} )</th>
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<td>Annulus A</td>
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Table 3. A summary of the test runs

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<th>Annulus D designation</th>
<th>Reynolds number</th>
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<td></td>
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<td>A</td>
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</table>
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1939 Born in Mysore, India.

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