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A STUDY OF SWIRLING FLOW IN
AN ORIFICE PLATE FLOWMETER

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SUMMARY

This thesis presents a theoretical and experimental study of swirling flow in an orifice plate flowmeter.

Firstly, a solid body rotation was considered as the inlet velocity profile of an orifice plate. The beta ratio considered was 0.4. The Reynolds number was 7.82×10^4 . As a result, a theoretical value of the discharge coefficient was obtained.

Secondly, Laser doppler anemometry was used to determine the axial and tangential velocity profiles at 3 upstream and 2 downstream cross-sections of an orifice plate with a beta ratio of 0.4. A double right angle bend in two perpendicular planes was placed 6.5 diameters upstream of the flowmeter. The axial velocity profile was found to be slightly asymmetrical. The corresponding tangential profile was found to have a form of a solid body rotation. A 20" mercury manometer was used to measure the pressure difference across the orifice. The results showed that the discharge coefficient decreased.

Thirdly, a commercial computer program was used to calculate the swirling flow through an orifice plate. A solid body rotation was considered at the inlet of the orifice. With the pressure field calculated, the corresponding discharge coefficient was determined. An additional simulation was performed with a $1/7$ power law inlet profile. The discharge coefficient under swirling flow conditions was smaller than the one under non swirling flow conditions.

Based on the experimental and computational results, we can see that the swirling flow produced by a double bend greatly affects the performance of an orifice plate. It is recommended that the flowmeter must be located 70 diameters

downstream of a double bend. The BS 1042 does not provide a reasonable straigth length for this type of disturbance.

A mi madre

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SUMMARY

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NOTATION

Symbol	Description
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Roman: Upper case

A	cross-section area
D	diameter of the pipe
E	velocity approach factor
I	moment of inertia
L _S	bend-bend spacer length
Mang	angular momentum
Max	axial momentum
R/D	bend-radius ratio
R	pipe radius
Re	Reynolds number
Q	volumetric flowrate
S	swirl number
V	velocity
V _i	mean velocity
W	swirl parameter
X	body forces

Symbol	Description
--------	-------------

Roman: Lower case

a	radial distance
a/R	nondimensional radius
c_d	discharge coefficient
d	diameter of the orifice
f_T	frequency of the turbine meter output
k_M	meter factor
m	mass flowrate
p	pressure
r_0	radius of the inner wall
r	r-coordinate
v_i'	velocity fluctuation
z	z-coordinate

Symbol	Description
--------	-------------

Greek

β	momentum flux coefficient
β	ratio of diameters of orifice and pipe
δ_f	fringe separation
ϕ	coefficient of contraction
θ	θ -coordinate
θ	angle between the laser beams
λ	area ratio
λ_L	wavelength of light
μ	dynamic viscosity
μ_M	mean of normal distribution
ν	kinematic viscosity
ρ	density
ψ	swirl angle
Ω	swirl velocity
σ	standard deviation of normal distribution

Symbol	Description
--------	-------------

Superscripts

— mean value(time averaged)
z,r,θ cylindrical polar coordinates

Subscripts

a axis
av mean value(time averaged)
z,r,θ cylindrical polar coordinates
w wall

Mathematical symbols

Δ difference
∇ vector operator
∫ integral
∂ partial derivative

1. INTRODUCTION

In many applications a pipe fitting cannot be considered as isolated, since unless the pipe components are separated by many diameters of straight pipe, the flow leaving the first component can substantially affect the flow into , and away from the second component. One example of such interaction can be found in the combination of two 90 degree bends in different planes. A swirling flow is produced and may persist for over 100 diameters. Such a long length between the combined fitting and a flowmeter cannot be provided in many industrial pipeline layouts, and so it is important to know the errors that would be involved using a reduced length of straight pipe.

In this thesis a detailed picture of the swirling flow through an orifice plate is given . The discharge coefficient was evaluated and some conclusions are presented .A brief description of Chapters 2 to 6 is given below.

Chapter 2 presents a literature survey covering various areas of interest such as axisymmetric swirling flow through orifice plates and computer modelling of flow through orifices. Chapter 3 describes the theory involved with axisymmetric turbulent and swirling flow through orifice plates. Chapter 4 covers the experimental work as well as the results obtained. Chapter 5 comprises all the computational work required to get the numerical results. Finally, Chapter 6 presents discussions and conclusions on the results obtained as well as important recommendations for the installation of orifice plates.

2. LITERATURE SURVEY

A lot of research has been done to explain the nature of swirling flows and how they affect the discharge

coefficients of orifice plates. Murakami(1) says that according to his experiments with orifice plates, the swirl intensity and the area ratio governed the increase or decrease of the discharge coefficient. His experiments were carried out in the following way: He had a rotating impeller to create the level of swirl required in the water rig. A length of 40D downstream the swirl generator was used to ensure a fully developed swirling flow at the orifice entrance. The area ratios were 0.4, 0.6, and 0.8. It was found that the discharge coefficient varied in a range between +10% to -20%. He also measured the pressure distributions on the wall at different sections before and after the orifice plate. An increase of pressure recovery was detected. He explained this process by saying that the centrifugal forces reduce the separated region behind the orifice. Among some other interesting results he got were the uneven wall pressure and the velocity distributions in the flow. The former he said, was due to an eccentricity of the swirling center. Its maximum was located at sections 1D-3D downstream of the orifice plate. The latter was produced by the orifice plate itself and the level of swirl.

Rapier(2) studied the same process by using a high pressure gas circuit. The orifice plate was a corner tapped orifice with an area ratio of 0.627. It was located 37 diameters downstream of the disturbance. He also used a conditioner in order to get rid of the nonaxisymmetric flow created in the flow rig. When he had an empty pipe as a flow conditioner and a swirler as a disturbance, the change of discharge coefficient was +8%. Reynolds numbers up to 2.5×10^6 were used.

Kinghorn(3) determined that under different levels of swirl the change in discharge coefficient varied from 0 to +25%. Swirl was generated by a 305 mm vane damper control valve fitted with a scale and locking device. The vanes could be

set at any required angle to the line of flow. The pipework continued with a 305-203 mm reduction. Three 203 mm diameter orifice plates with area ratios of 0.25, 0.31, and 0.64 were used. These flowmeters were preceded by a variable length of 203 mm pipe.

Mottram and Rawat(4) analyzed the influence of two installation factors on orifice plates. These factors were pipe roughness and a 2x90 degrees offset bend combination. Air was drawn by a centrifugal fan. First it passed through a Borda inlet which was used as a reference flowmeter. Then, sections of seamless steel straight pipe took the flow to the orifice plate. During the rough pipe tests, this pipe was replaced by the same type of pipe but with different lining. The beta ratios of the orifice plates were 0.78, 0.58, and 0.44. The results show that the discharge coefficient is less affected by the rough pipe than by the smooth pipe. Minimum straight lengths upstream of the orifice plate are presented for zero additional uncertainty on the flow measurement. It should be noted that these lengths vary markedly according to the standard used.

Other researchers have studied the influence of bends on the performance of orifice plates. Irving(5) used a set of three concentric, square edged orifice plates. Area ratios were 0.25, 0.50, and 0.64. The bend geometries investigated were a 90 degree $R/D=1.5$ swept bend, and a combination of two 90 degree $R/D=1.5$ bends. He found that the isolated bend produced a maximum change of -7% in the discharge coefficient, and the bend combination resulted in a +3 to -3% change. He quoted the important aspect of the installation requirements indicated in the various flow metering standards. For the 90 degree bend and the 0.25, 0.50, and 0.64 plates the lengths specified were 7-14D, 13-30D, and 20-46D respectively. According to his experiments, these requirements are inadequate for bringing

the coefficient back to its standard value. Similarly, for the bend combination and the same plates, the standard requirements were 20-40D, 30-63D, and 40-82D. Once again, Irving's results show that these lengths are not enough to prevent the discharge coefficient from changing markedly.

He also compared his experiments with Nagel's(6). He said that the difference found was directly related to the Reynolds number used and the secondary flows generated by different fittings. In the case of a U bend, the secondary flows are almost non existent. With the isolated bend, there are relatively weak twin celled secondary flows, and with the bend combination there is a strong single centred secondary flow. These effects will be increased or decreased according to the Reynolds number used. Therefore, the Reynolds number is the factor that determines a positive or negative change in the discharge coefficient.

During the second stage of Irving's investigations, he studied the effect of some other combination of bends on the 0.25, 0.50, and 0.64 plates. The following aspects were covered:

- 1.- Bend-radius ratio, R/D : ratios of 0.5 (mitred bend), 1.5 and 4.75 were tested.
- 2.- Bend-bend spacer length, L_s : spacers of 0, 5, and 11 diameters were tested.
- 3.- Turning angle of the combination: angles of 0 degree (S bend), 90 degree (offset bend), and 180 degree (U bend) were tested.

He used the three pressure tapping locations (corner, flange and D and $D/2$) and a range of fitting-orifice spacer lengths. The Reynolds number range was 3.6×10^5 - 8.0×10^5 .

A comparison was made with other researchers, the main difference was the Reynolds numbers covered. He pointed out

that the orifice plate ratio is also a factor that defines the magnitude of the change in the discharge coefficient. In the case of a strong swirling flow, the presence of the swirl produces a secondary effect on the axial velocity distributions, he added. Finally, a set of results showed a change of +1.7, +2.5, and +3.1% for the 0.64, 0.50, and 0.25 plates respectively.

Other interesting aspects were the bend radius ratio and the bend-bend spacer. The results indicated that the more the bend radius is reduced the more negative is the change in the discharge coefficient. The graphs also showed a negative increase in the discharge coefficient for an increase in bend-bend separation.

The third stage of Irving's experiments included another study with isolated bends and a combination of bends. He highlighted the importance of the physics of the flow through an orifice plate. A flat velocity profile causes a bigger change in the discharge coefficient than a fully developed profile, he mentioned. The process involved is produced by two effects, a pressure rise at the upstream face of the plate and a further jet contraction at the downstream face of the plate. It follows that the discharge coefficient is decreased.

He also indicated that the area ratio of the plate is the factor which determines the effect dominating the flow field.

The effect of the disturbance decay in the pipe downstream of the fitting was studied. It was found that for tight-radius bends the change in the discharge coefficient was big but the disturbance was damped out more rapidly. On the other hand, it is important to evaluate the effect of the orifice plate ratio on the spacer length required to damp the disturbance out. Irving concluded that the interaction

between two bends was only present for spacer lengths of less than $2D$. It was also remarked that the installation requirements given in the standards take into account the orifice plate sensitivity to a flow disturbance. In the case of swirl produced by two 90 degree bends, the spacer lengths indicated by the standards were still too short to bring the discharge coefficient back to its standard value.

In the case of bends in combination, Irving explains that there is an interaction of pressure and centrifugal forces caused by the first bend and the amplification or attenuation of the secondary flows generated will be determined by the physical orientation of the second bend. The U bend produces very weak secondary flows while the S bend causes a strong twin-celled system of secondary flows.

The combination of two bends placed in different planes produces swirl. The intensity is a function of the turning angle the maximum being at around 60 to 90 degrees. Some other characteristics are the strength of the swirl and the modification of the axial velocity profile. Swirl decay can extend up to $100D$. This means that there is always a change in the discharge coefficient if the orifice-fitting length is short. The standards indicated the length required for different area ratio plates. Again, these requirements were not enough. The effect Irving said, can be understood by studying the physics of the flow. The condition of the fluid entering the orifice plate is very different from that for a fully developed flow. The swirling flow has a tangential momentum which must be conserved as it passes through the orifice plate. This means that the rotational velocity increases as the flow contracts, causing a rapid expansion of the jet. This in turn increases the pressure downstream of the orifice and reduces the differential pressure across it. At high levels of swirl, an increase in the discharge coefficient is present. The plate area ratio is the factor which determines the relative effect of axial

and rotational flow on the differential pressure. The smaller the area ratio the greater the rotational effect.

Irving also mentioned that there are other fittings which produce strong swirl. A combination of a bend and a partially open gate valve can be a good example. A way of predicting the change in the discharge coefficient has been suggested by Irving. He explained that a pressure correction factor can be used to assess the change. This pressure correction factor was based on three pressures, the pressure at $1D$ upstream tap, the pressure at a flange upstream tap, and the pressure at a $D/2$ downstream tap. The advantage of using this correction factor is that it requires only the static pressure measurements which can be obtained easily. This means that it can be used in industrial applications.

Researchers as Mottram and Spencer(7) studied the recent work developed by others and gave some recommendations for each case. They compared the results with the minimum length requirement established in the standards. In the case of a single 90 degrees bend they suggested that the maximum change in discharge coefficient might be 0.25% for the zero additional uncertainty length and 0.75% for the 0.5% additional uncertainty length. The minimum straight length of pipe between an $m=0.60$ orifice plate and a single bend should be increased from $18D$ to $31D$ for zero additional uncertainty and from $9D$ to $14D$ for 0.5% additional uncertainty. The recent data also showed that the minimum straight lengths of pipe between an orifice plate and two 90 degrees bends in the same plane are significantly less than those indicated in BS 1042.

On the other hand, the case of an orifice plate and two 90 degrees bends in different planes indicated that the revised lengths should be double those proposed for the 0.5% additional uncertainty. Besides, the authors proposed

to use a straightener instead of adding straight lengths of pipe. When a spacer is placed between the two bends in different planes upstream of the orifice plate, the minimum straight length can be reduced from that specified for bends with zero spacer.

Pabmanabhan(8) experimentally studied the dependence of the swirl decay parameter on Reynolds number and initial swirl level. He used a swirl generator made of 4 adjustable vanes. Vane angles were adjustable from 0 to 40 degrees with respect to the axis of the pipe. A vortimeter was provided for measuring the swirl angle. The orifice plate tested had a diameter $d=0.6D$. The range of Reynolds numbers used was from 1.0×10^5 to 2.5×10^5 . At the largest swirl angle tested(40 degrees), the coefficient of discharge was increased by approximately 5%. There seems to be a nonsignificant Reynolds number dependence for values greater than 1.8×10^5 . In other words, the swirl decay parameter decreased with increasing Reynolds number.

On the other hand, there is a significant dependence of the swirl parameter on the initial swirl levels. However, for higher initial swirl levels(values greater than 0.15) the dependence is less significant. According to other results, a significant wall pressure loss is present even for moderate swirl levels. The values obtained for a 30 degree swirl angle were twice as high as the corresponding losses for nonswirling flows.

Recently, experimental and computational work has been done by W. Keegans and M.J. Reader-Harris(9). The rig consisted of a double bend, two diameters of straight pipe, a perforated plate flow straightener, a further 8 diameters of straight pipe and an orifice plate of nominal diameter ratio 0.8. The double bend imparted swirl and asymmetry to the flow. The Reynolds number used was 1.0×10^6 . Laser doppler velocimetry was used to measure velocity and

turbulence profiles downstream and upstream of the orifice plate. The pressure difference across the orifice plate was measured so that discharge coefficients could be obtained. Flow through the orifice plate was also computed using the algebraic stress turbulence model, although upstream conditions were not identical in the computational and experimental work. The results showed that a perforated plate whose thickness is equal to the diameter of its holes is remarkably effective at removing swirl. The horizontal and vertical velocity profiles described perfectly well the swirling flow coming out from a double bend. A nearly zero vertical velocity was obtained downstream of the perforated plate. The horizontal velocity profile coming into the orifice plate was almost flat. Therefore, a nearly axisymmetric fully developed profile was found at the outlet of the flowmeter.]

G.E. Mattingly and R.W. Davis(10) also carried out computational work on turbulent flow through thin orifice plates. The flow fields modeled were axisymmetric. The orifice plates had beta ratios from 0.4 to 0.7. Reynolds numbers were in the approximate range 1.0×10^4 to 1.0×10^6 . The particular turbulence model used was the k-epsilon model developed at Imperial College, London. This study was concentrated on the beta ratios and inlet velocity profile effects. A series of plots were obtained for the radial and axial distributions of the fluid velocities and pressure anywhere in the flow field. The computed discharge coefficient was compared with published values. There was reasonable agreement between them. Some discrepancies were noted between computed and published values for the locations of salient features of the flow such as minimum pressure points and the vena contracta. For beta ratios greater than 0.6, the vena contracta lies downstream of the minimum wall pressure point while for beta less than 0.6 the vena contracta lies upstream of this point.

C.J. Bates(11) conducted a series of laser doppler anemometry measurements of the flow through an orifice plate. He used an orifice plate with a beta ratio of 0.5. His rig consisted of a pump which lifted water to a constant head reservoir 18.81 m above the supply basin. From the reservoir water flowed under gravity through a system of pipes into the test section, and then returned to the supply basin via a control valve. The optical system consisted of a 20 mW helium-neon laser as the coherent light source, a 30 cm focal length optical package incorporating dual Bragg cells to provide frequency shift facilities, together with a suitable photomultiplier and a frequency tracker. The Reynolds numbers covered were 67600, 87200, 136000, and 183000. Measurements were made at 5 different locations, two upstream and three downstream of the orifice. The axial velocity profiles encountered at the first upstream location were in good agreement with the fully developed profiles obtained from the one-seventh power law. The downstream profiles confirmed the existence of three regions namely the central core, the shear layer or mixing region and the negative flow adjacent to the wall. The upstream turbulence intensity profiles were not as significant as those found at the downstream locations. The peak values of turbulence velocity in the downstream section were from 40% to 70% of the mean velocity. These results were obtained in the shear layer.

3. THEORETICAL ASPECTS OF SWIRLING FLOWS THROUGH ORIFICE PLATES

3.1 Turbulent flow through an orifice plate.

3.1.1 General aspects of orifice plates.

An orifice plate is a device that is employed widely in industrial flow measurement. Basically, it is a thin flat plate with a single hole in the middle. This meter produces a large local variation of pressure. By using two pressure tappings the differences in pressure can be measured and a calibration curve relating differential pressure to flowrate can be established for the device.

The accuracy of the flowmeter, however, depends on the detailed specification of the geometry of an orifice plate. This means that the pressure difference generated is a function of several geometrical parameters of the orifice plate itself. The National and International Standards publish some of the geometries and their calibration curves. According to the BS 1042, the orifice plate has to comply with the following geometrical requirements (see Fig. 3.1)

- the orifice plate has to be circular and its centre has to be coincident with the duct axis.
- the upstream edge may be square, bevelled or conical and quarter-circle radiused.
- the internal face of the orifice parallel to the axis of flow has to be short. In order to achieve this specification the downstream edge may be bevelled.
- the diameter of the orifice has to be equal to or greater than 12.5 mm. The ratio $\beta = d/D$ has to be equal to or

greater than 0.2 or 0.23 and less than or equal to 0.75 or 0.80 according to the type of orifice plate used. Within these limits, the value of beta is chosen by the user to define an orifice plate of a given type.

- at least one upstream pressure tapping and one downstream tapping must be provided for the orifice plate, installed in one or other of the recommended standard positions.

Also, in order to obtain high accuracy, the orifice plate has to comply with the installation requirements listed below.

- the pipe must run full at the measuring section.

- the orifice plate must be installed in the pipe-line at a position such that the flow conditions immediately upstream approach those of a fully developed profile and are free from swirl.

- the orifice plate must be fitted between two sections of straight cylindrical pipe of constant cross-sectional area, in which there is no obstruction or branch connection.

- the required minimum straight lengths of pipe vary according to the nature of the fittings and the diameter ratio.

- the orifice plate must be kept clean at all times and free from accumulations of any type, either on the orifice plate or within the upstream and downstream runs.

Other geometries have been published by different researchers. Some alternative designs are shown in Fig. 3.2. These types of orifice plates are specially suitable for fluids with small particles, such as sand grains in suspension.

3.1.2 Incompressible turbulent flow through an orifice plate .

Figure 3.3 shows the different regions found in a turbulent flow through an orifice plate at high Reynolds numbers (above 4.0×10^3). The flow passing through the orifice separates away from the surface of the orifice plate and forms a jet which contracts to a minimum cross sectional area, A , downstream of the plate. The section at which this occurs is known as the vena contracta. Between the jet and the pipe a separated flow region is formed. Downstream from the vena contracta, transition takes place and the characteristics of the through-flow and the separated flow are lost in a region of turbulent mixing dominated by an adverse pressure gradient. Flow is entrained into the edge of the jet in the mixing region, giving rise to a circular motion within the separated flow. An amount of fluid equal to that previously entrained is returned to the separated flow in the region where the through-flow reattaches to the duct wall.

According to the features of the flow described above, the pressure distribution is shown in Fig. 3.4. On the axis, the flow accelerates continuously from some distance upstream of the orifice as far downstream as the vena contracta. This is evidenced by a corresponding favourable pressure gradient. Beyond the vena contracta, the velocity on the axis decreases as the area occupied by the through-flow increases to fill the available cross-section of the duct. In this region the pressure on the axis increases until it reaches a maximum at the recovery plane, which is approximately coincident with the plane at which reattachment occurs. At this stage the main effects of the orifice plate on the pressure distribution are essentially complete. However downstream of the recovery plane the

velocity profile continues to change until it finally reverts to the fully-developed form some fifty or more diameters downstream of the orifice plate. On the wall of the duct, there is a step change in pressure across the orifice plate. It is this large change in pressure which is used for flow metering purposes. Upstream of the orifice plate there is a tendency for the wall pressure to increase as the corner at the orifice plate is approached. Downstream of the plate there is a gradual decrease in pressure from the downstream corner to the vena contracta plane, just beyond the recovery plane. It will be noted that the pressure p is nearly constant over the cross section at the upstream reference plane, the vena contracta plane, and at the recovery plane. This is in marked contrast at the orifice plate itself. Just downstream of the orifice plate the streamlines of the flow exhibit appreciable curvature, and this flow pattern gives rise to an increase in pressure with distance from the centre of curvature. This is the reason for the wall pressure being significantly below the axial pressure in this region.

3.1.3 Discharge coefficient.

Now we have discussed the main features of an incompressible turbulent flow through an orifice plate, a simplified model will be used to derive the discharge coefficient(see Internal Fluid Flow, Ward-Smith).

Considering Fig. 3.3, section 1 refers to the conditions in the pipe upstream of the orifice plate, section 2 corresponds to the plane of the orifice plate, section 3 is located at the plane of the vena contracta, and section 4 defines the recovery plane. In order to simplify our model, the following assumptions are made:

- 1.- The flow is incompressible; ρ is constant.

2.- In the vena contracta region, a free jet and a separated flow are found.

3.- One dimensional relations are used to obtain the properties of the flow at sections 1, 3, and 4.

4.- The flow was considered inviscid between sections 1 and 3.

5.- The shear stress at the wall is neglected.

6.- The pressure was constant over the cross section at section 3.

7.- The separated flow at section 3 does not contribute to the momentum balance.

Using the energy equation for inviscid incompressible flow between sections 1 and 3, we have:

$$p_1 + (1/2) \rho V_{z1}^2 = p_{2j} + (1/2) \rho V_{z2}^2 = p_3 + (1/2) \rho V_{z3}^2. (1)$$

where V_{z3} is the mean velocity in the jet at section 3.

p_{2j} is the mean pressure in the jet at section 2.

Applying the continuity equation to the flow, we have:

$$\rho V_{z1} = \lambda \rho V_{z2j} = \phi \lambda \rho V_{z3j} = \rho V_{z4} = \rho Q/A_1 (2)$$

where λ and ϕ are defined as follows:

$$\lambda = A_2/A_1 = [D_2/D_1]^2 \quad \phi = A_v/A_2$$

The momentum equation applied to sections 3 and 4 is:

$$p_3 (1 - \phi \lambda) + \phi \lambda (p_3 + \rho V_{z3j}^2) = p_4 + \rho V_{z4}^2 (3)$$

Combining equations (1), (2), and (3) we obtain:

$$Q = (A_2 / (1 - \lambda^2)^{1/2}) \{2(p_1 - p_{2j}) / \rho\}^{1/2} \quad (4)$$

$$Q = (2(p_1 - p_3) / \rho)^{1/2} A_2 \phi / (1 - \lambda^2 \phi^2)^{1/2}$$

Equation (4) shows that the discharge coefficient is equal to one when the pressure difference is measured between sections 1 and the value in the jet at section 2. However, when the pressure difference is measured between sections 1 and 3, c_d is given as follows:

$$c_d = \{\phi^2 (1 - \lambda^2) / (1 - \phi^2 \lambda^2)\}^{1/2}$$

Considering that the wall pressure at section 2 is approximately the pressure at the vena contracta (p_3), the pressure difference $p_1 - p_3$ is normally used and equation (4) becomes

$$Q = A_2 E c_d \{2(p_1 - p_3) / \rho\}^{1/2} \quad (5)$$

where $E = 1 / (1 - \lambda^2)^{1/2}$ is the velocity approach factor

3.2 Swirling flow through an orifice plate.

3.2.1 Nature of swirling flows.

A swirling flow is a combination of vortex and axial motions. The particles in the fluid have tangential and axial components of velocity.

The swirling motion is usually produced by a combination of bends, valves or other fittings that cause a change in direction. Figure 3.5 shows the fluid flow passing two 90 bends (see Internal Flow Systems, Miller, D.S.). The pressure

and flow distributions are severely affected and the resulting swirling motion can persist for upwards of 100 pipe diameters.

If we compare the velocity profiles at the inlet and outlet of the two 90 bends, remarkable differences can be found. At the outlet, the axial velocity profile shows that the maximum velocity is located slightly away from the pipe axis. The tangential velocity profile reveals that close to the axis, there is a region where the velocity, V_θ , increases approximately linearly with r , as in a forced vortex. At some stage the maximum value of V_θ is reached and with further increase in r , V_θ decreases to zero at the wall. Some researchers have found that if the swirling motion is weak the forced vortex region with uniform axial velocities dominate the whole pipe section.

3.2.2 Swirl parameters.

A level of swirl developed in a fluid is specified by the swirl number, S . S is defined as the ratio of the angular momentum flux, M_{ang} , to the axial momentum flux, M_{ax} , at a given cross section. For an incompressible flow, we have.

$$S = M_{ang} / (a M_{ax}) = \int_0^a V_z V_\theta r^2 dr / (a \int_0^a V_z^2 r dr) \quad (6)$$

where V_z is the axial velocity,
 V_θ is the tangential velocity
 a is a radial distance

Sometimes the swirl parameter W is used to characterize the flow. W is defined as:

$$W = 4 M_{ang} / \pi \rho \bar{V}_z^2 D^3 \quad (7)$$

$\bar{V}_z$ being the mean axial velocity (flowrate divided by πa^2)

The parameters S and W are related by the momentum flux coefficient β . β is defined as:

$$\beta = (2/a^2) \int_0^a (v_z/\bar{v}_z)^2 r dr \quad (8)$$

and the resulting equation for S and W is:

$$S = 2 W/\beta \quad (9)$$

For turbulent flows with small enough swirl, ($S < 0.1$), β is almost one. Therefore a linear relationship is established between S and W . On the other hand, at high swirl levels the axial velocity profile is affected significantly and the linear relation is lost.

3.2.3 Incompressible swirling flow through an orifice plate.

Figure 3.6 shows the different regions found in a swirling flow through an orifice plate. The flow approaching the orifice plate is considered to be a forced vortex type motion. This swirling motion is amplified at the orifice plate plane. Due to the centrifugal forces created by the swirling flow, the jet formed at the outlet is expanded and the separated region is reduced. Some diameters away from the vena contracta, the separated flow is lost in a region of intense turbulent mixing. Finally, the flow reattaches the pipe wall and the swirling flow persists over many diameters downstream.

For the flow described above, the pressure distributions

are as follows:

Due to the favourable pressure gradient the flow is accelerated from some diameters upstream of the orifice plate. The centrifugal forces modify the pressure distribution over the cross sections upstream and downstream. A greater pressure is generated at the wall than at the axis of the pipe. Downstream of the vena contracta the velocity on the axis decreases and the flow tends to fill in the cross-section area of the pipe. The pressure over the cross section is increased until it reaches a maximum at the recovery plane. Some diameters downstream of the recovery plane, the velocity profile is still changing and eventually it reaches a point where a fully developed profile and a distorted tangential profile are found. On the wall of the pipe, the pressure difference across the orifice plate is affected by the swirling flow.

3.2.4 Discharge coefficient.

In previous paragraphs a description of the incompressible swirling flow through an orifice plate was given. Here, the discharge coefficient will be obtained using some simplifications.

The fluid flow is governed by the mass and momentum equations. These equations can be derived by applying the conservation of mass and momentum (Newton's second law of motion) to an infinitesimal fixed control volume.

Using cartesian tensor notation, the mass conservation equation or "continuity" equation for the mean flow may be written as:

$$\frac{\partial v_i}{\partial x_i} = 0 \quad (10)$$

and for the turbulent fluctuations:

$$\frac{\partial v_i'}{\partial x_i} = 0 \quad (11)$$

The corresponding momentum conservation or Navier-Stokes equation is:

$$\rho \left[\frac{\partial v_i}{\partial t} + v_j \frac{\partial v_i}{\partial x_j} \right] = - \frac{\partial p}{\partial x_i} + \rho x_i + \frac{\partial^2 v_i}{\partial x_j^2} - \frac{\partial (\rho \overline{v_i' v_j'})}{\partial x_j} \quad (12)$$

Using cylindrical polar coordinates z, r, θ with corresponding V_z, V_r, V_θ and instantaneous v_z', v_r', v_θ' the equations (10), (11), (12) become:

$$\frac{\partial V_z}{\partial z} + 1/r \frac{\partial (r V_r)}{\partial r} + 1/r \frac{\partial V_\theta}{\partial \theta} = 0 \quad (13)$$

$$\rho \left[V_z \frac{\partial V_z}{\partial z} + V_r \frac{\partial V_z}{\partial r} + V_\theta / r \frac{\partial V_z}{\partial \theta} \right] = - \frac{\partial p}{\partial z} - \rho \left[\frac{\partial \overline{v_z'^2}}{\partial z} + 1/r \frac{\partial (r \overline{v_z' v_r'})}{\partial r} + 1/r \frac{\partial (\overline{v_z' v_\theta'})}{\partial \theta} \right] + \mu \nabla^2 V_z \quad (14)$$

$$\rho \left[V_z \frac{\partial V_r}{\partial z} + V_r \frac{\partial V_r}{\partial r} - V_\theta^2 / r \right] = - \frac{\partial p}{\partial r} - \rho \left[\frac{\partial \overline{v_z' v_r'}}{\partial z} + 1/r \frac{\partial (r \overline{v_r'^2})}{\partial r} + 1/r \frac{\partial \overline{v_r' v_\theta'}}{\partial \theta} - \overline{v_\theta'^2} / r \right] + \mu \left[\nabla^2 V_r - V_r / r^2 - 2/r^2 \frac{\partial V_\theta}{\partial \theta} \right] \quad (15)$$

$$\rho \left[V_z \frac{\partial V_\theta}{\partial z} + V_r \frac{\partial V_\theta}{\partial r} + V_\theta / r \frac{\partial V_\theta}{\partial \theta} + V_r V_\theta / r \right] = - 1/r \frac{\partial p}{\partial \theta}$$

$$\begin{aligned}
& -\rho \left[\frac{\partial \overline{v_z' v_e'}}{\partial z} + \frac{\partial \overline{v_r' v_e'}}{\partial r} + \frac{1}{r} \frac{\partial \overline{v_e'^2}}{\partial \theta} - 2 \frac{\overline{v_r' v_e'}}{r} \right] \\
& + \mu \left[\nabla^2 v + \frac{2}{r^2} \frac{\partial v_e}{\partial \theta} - v_e / r^2 \right] \quad (16)
\end{aligned}$$

where $\nabla^2 = \frac{\partial^2}{\partial z^2} + \frac{\partial^2}{\partial r^2} + (1/r) \frac{\partial^2}{\partial \theta^2} + (1/r^2) \frac{\partial^2}{\partial \theta^2}$

If we assume that the flow is inviscid at high Reynolds numbers then eliminating small order terms and setting $\frac{\partial}{\partial \theta} = 0$ for the axisymmetric case, equations (13), (14), (15), and (16) become:

$$\frac{\partial v_z}{\partial z} + (1/r) \frac{\partial}{\partial r} (r v_r) = 0 \quad (17)$$

$$\rho \left[v_z \frac{\partial v_z}{\partial z} + v_r \frac{\partial v_z}{\partial r} \right] = -\frac{\partial p}{\partial z} \quad (18)$$

$$\rho \left[v_z \frac{\partial v_r}{\partial z} + v_r \frac{\partial v_r}{\partial r} - v_e^2 / r \right] = -\frac{\partial p}{\partial r} \quad (19)$$

$$\rho \left[v_z \frac{\partial v_e}{\partial z} + v_r \frac{\partial v_e}{\partial r} - v_r v_e / r \right] = 0 \quad (20)$$

Considering a swirling flow with the following velocity distributions:

$$\begin{array}{ll}
v_z = \text{const} & r < r_0 \\
v_z = 0 & r > r_0 \\
v_e = \Omega r & r < r_0 \\
v_e = 0 & r > r_0 \\
v_r = 0 &
\end{array}$$

Equations (17), (18), (19), and (20) are as follows:

Equations (17), (18), (19), and (20) are as follows:

$$0 = 0 \quad (21)$$

$$0 = -\frac{\partial p}{\partial z} \quad (22)$$

$$-\rho \Omega^2 r^2/r = -\frac{\partial p}{\partial r} \quad (23)$$

$$0 = 0 \quad (24)$$

The difference between the pressure p at any value of r and the pressure on the axis of a pipe p_a , is given by integrating (23):

$$p(r) - p_a = \rho \int_0^r \Omega^2 r dr = \rho \Omega^2 r^2/2 \quad (25)$$

Then the difference between the pressure p at the wall and the pressure on the axis of a pipe, is as follows:

$$p_w - p_a = \rho \Omega^2 r_0^2/2 \quad (26)$$

And considering Fig 3.6 we have:

$$p_{w1} - p_{a1} = \rho \Omega_1^2 r_{01}^2/2 \quad (27)$$

$$p_{w2} - p_{a2} = \rho \Omega_2^2 r_{02}^2/2 \quad (28)$$

where r_{01} is the pipe radius and r_{02} is taken as the radius of the orifice.

Finally, for an approximate calculation, the relative change of static pressures toward the pipe wall against the static pressure p_{w2} can be neglected and, $p_{w2} \approx p_{w3}$, so the

pressure difference across the orifice plate is:

$$\Delta p = p_{w1} - p_{w2} = (p_{a1} + \rho \Omega_1^2 r_{01}^2 / 2) - (p_{a2} + \rho \Omega_2^2 r_{02}^2 / 2) \quad (29)$$

Then, considering equation (5) and (29), we have:

$$c_d = (4Q / E d^2 \pi) (2((p_{a1} / \rho + \Omega_1^2 r_{01}^2 / 2) - (p_{a2} / \rho + \Omega_2^2 r_{02}^2 / 2)))^{-1/2} \quad (30)$$

This is the general formula. Henceforth we assume $\beta = 0.4$.

Therefore, if we take $V_{e1} = 0.27$ m/s at $r_{01} = 0.0385$ m, and if $r_{02} = 0.0151$ m ($= \beta \times r_{01}$).

$$\Omega_1 = V_{e1} / r_{01} = 0.27 / 0.0385 = 7.01 \text{ rad/s}$$

and, $M_{ang} = I \Omega$, where $I = m \frac{1}{2} r^2$

Taking $V_{z1} = 1.016$ m/s, the mass per second, m , at the inlet of the orifice plate is given by:

$$m = \rho V_{z1} A_1 = (1000)(1.016)(\pi / 4 (0.077)^2)$$

$$m = 4.7311 \text{ kg/s}$$

$$\text{Then, } M_{ang} = (4.7311) \left(\frac{1}{2} (0.0385)^2 \right) (7.01) = 0.02456 \text{ Nm}$$

$$\text{Considering: } M_{ang} = I_1 \Omega_1 = I_2 \Omega_2 \quad \Omega_2 = M_{ang} / I_2$$

$$\Omega_2 = 0.02456 / 4.7311 \left(\frac{1}{2} (0.0151)^2 \right) = 45.57 \text{ rad/s}$$

In the no swirl case, the pressures on the axis of the pipe at stations 1 and 2 (see Fig. 3.3) are:

$$p_{a1} = 7.0118 \text{ m} = 6.8757 \times 10^4 \text{ Pa}$$

$$p_{a2} = 1.1177 \text{ m} = 1.0960 \times 10^4 \text{ Pa}$$

These values are assumed to be the same as the measured

wall pressures at stations 1 and 2 respectively.

From equation (29):

$$\Delta p = (6.8757 \times 10^4 + (1000)((7.01)^2(0.0385)^2/2) - (1.0960 \times 10^4 + (1000)((45.57)^2(0.0151)^2/2)$$

$$\Delta p = 68793.41 - 11196.74 = 57596.66 \text{ Pa} = 5.873 \text{ m}_{\text{H}_2\text{O}}$$

$$\Delta p = 0.4328 \text{ m}_{\text{Hg}}$$

Then, taking $D = 0.077 \text{ m}$ and $d = 0.0308$, the velocity approach factor, E , is given by:

$$E = 0.077^2 / (0.077^4 - 0.0308^4) = 1.013$$

Finally, from equation (30) we calculate c_d .

$$c_d = 4(0.0047311) / 1.013 \pi (0.0308^2) (2(57596.66)/1000)^{1/2}$$

$$c_d = 0.5840$$

4. THE EXPERIMENTAL WORK

4.1 Nature, purpose and scope of the experiments.

The experimental work carried out consisted of the measurement of axial and tangential components of the velocity of water flowing through an orifice plate. This orifice plate was located 6.5 diameters downstream of a combination of two 90 degrees bends in two perpendicular planes. Also, a measurement of the pressure difference across the orifice plate was made.

Laser Doppler Anemometry was used for the velocity measurements because of its non-intrusive character and its high accuracy. Although some information concerning the turbulent intensities in the flow was collected as a result of using this measurement system (and are presented), this was not the prime concern of the work.

A 20" manometer was used to determine the pressure difference across the orifice plate.

The purpose of the experiments was to obtain a description of the flow field through the orifice plate, together with a measurement of the pressure difference. Then, based on the pressure difference given in the experimental work and BS 1042, the discharge coefficient was calculated.

Thus, some indication of the influence of swirling flows on orifice plates was determined.

Velocities were measured on 3 sections preceding the orifice plate to establish the inlet profile, and on 2 sections following the orifice plate. Sufficient measurements were made to define the nature of the flow.

4.2 A description of the test facility and the measurements made.

4.2.1 An overall description of the flow loop.

Figure 4.1 to 4.3 shows the test facility designed for the velocity measurements mentioned in Section 4.1. The fiberglass tank made with 4 ft. square panels contained about 1900 gallons (7 m³). The water was drawn out of one side of the tank close to one end through a bellmouth and was returned (eventually) to the other end. In the middle of the tank a perforated screen, dividing the tank in two, ensured that no air was entrained into the pump and that adequate mixing took place to provide steady and uniform flow conditions at the pump inlet. A drain was fitted at the end of the tank furthest from the pump inlet. Periodically the tank was drained, cleaned and refilled in order to provide a sufficiently clean water supply for the measurements. To lengthen the life of a tankful of water wooden covers were placed over the tank to reduce the amount of dust entering the water. They also supported the returning pipelines. A dipping thermometer capable of measuring temperature to within 0.5 °C was hung in the tank at the end to which water returned.

The flow loop started at the tank. Water left the tank through a 6" diameter steel pipe. Then, it passed through a 90° bend following a fully open gate valve and a 30 hp centrifugal pump whose duty point was a flowrate of 40000 gal/hour and a total head of 95 ft of water. A steel reducer was used to go from the 5" diameter outlet of the pump to a 4" diameter steel pipe. Then, an ABS tee junction divided the flow into two loops. This tee junction supplied water to another loop. It was shut off with a gate valve throughout the course of the experiments. Our loop continued with another 4" to 3" diameter ABS reducer and

another tee junction providing one line which went through a ball valve and on into the test section and another which passed through a 5 μ m filter and a ball valve before rejoining the first line before the test section. A bypass line controlled with a diaphragm valve returning into the far end of the tank was built off the line before the filter. This line was used to reduce the pressure in the loop. The filter was not used during the experiments. The inlet to the test sections started with a 90° mitred elbow preceded by a section of 10 diameters(approx.) straight pipe. This arrangement avoided the interaction of bends which in turn can create an undesired level of swirl(see BS 1042). After the elbow, 36 diameters of straight ABS pipe preceded a double ABS bend. According to the standards for orifice plate measurement(BS 1042), the length above mentioned was sufficient to provide the double bend with a fully developed flow.

The double bend had an effective spacer length of 1 diameter. The radius ratio of each of the component bend was 2.3.

Following the double bend was a test section of 6.5 diameters perpex pipe and the orifice plate under test. The water was passed through another test section of 6.5 diameters perpex pipe. Details of the construction of the test sections are given in Section 4.2.2. The loop continued with a straight section of 53 diameters(approx.) perpex pipe, a turbine meter, and a 180° bend. Downstream the bend, an arrangement of 6.5 diameters ABS pipe, a tube bundle straightener of 6.5 diameters in length, and 59 diameters(approx.) straight length of perpex pipe formed the outlet of the test section. During the course of the experiments, another orifice plate with the same beta ratio was installed 53 diameters downstream of the tube bundle straightener. In this way, a measurement of the pressure difference across the orifice plate was made under

nonswirling flow conditions.

Finally, water came back through a section consisting of 26 diameters of ABS pipe, a 90 mitred elbow, 10 diameters of ABS pipe, a 90 mitred elbow, 32 diameters of ABS pipe, a butterfly valve, and 14 diameters of ABS pipe. The butterfly valve was used to increase the pressure at the downstream side of the orifice plate. The pipeline then ran along the top of the tank and down into it at the nearest end.

The turbine meter had been calibrated at the Central Electricity Generating Board and was provided with its own calibration curve. There were 58 diameters of straight pipe preceding it, ensuring a reading as accurate as the calibration made possible. The meter factor (k_M) was 19015 $\pm$ 15 for the flowrate used in the experiments, the flowrate being f_t/k_M in m^3/s where f_t is the frequency of the turbine meter output.

Measurements were taken with the flow loop operating at one flowrate. The frequency output of the turbine meter was $f_t=90$ Hz. This flowrate was maintained to within an accuracy of ± 1 Hz at all times which is an error of 1.1%. The flowrate was achieved by opening the diaphragm valve in the bypass line fully and throttling the flow with the ball valve located at the inlet of the test section. The pressure at the inlet to the double bend was 10 psi. Then, the pressure at the inlet and outlet of the orifice plate was 9.97 psi and 1.59 psi respectively. This ensured that there was no cavitation in the loop. The reading of the turbine meter was steady after an initial period lasting half an hour during which it read up to 1% high. It was not measurably sensitive to the temperature range involved (11 °C to 20 °C) although the Reynolds number was a function of the kinematic viscosity of water, ν .

The average axial velocity at the flowrate used, was 1.016 m/s based on the formula.

$$V_{zav} = 4f_t / D^2 k_M$$

and an average internal pipe diameter of 77 mm. The Reynolds number (based on a nominal value of $1 \times 10^{-6} \text{ m}^2/\text{s}$ for ν) was 7.82×10^4 .

4.2.2 The test sections.

The test sections on either side of the orifice plate were constructed out of sections of extruded perpex pipe. Each section was flanged with PVC flanges (see Fig. 4.4). One of the ends of the upstream section was concentrically machined to position the orifice plate center on the axis of the pipe. The downstream section was flat at both ends. Each test section was 0.50 m long. A special tool was used to join the sections together. Because the outer rims of the flanges were aligned by the annulus, the inner rims were also aligned. Therefore the pipes were aligned. The gasket between the flanges (3 mm) and the non-circularity of the pipe itself were the only imperfections in the geometry of the test sections. The average was 77 mm for the internal diameter.

A couple of pressure tapings were installed on the outer sleeve of the flanges. This facilitated the measurement of the pressure difference across the orifice plate. According to the standards the tapings were positioned 1" away from the orifice plate itself and were 3.2 mm in diameter. By using 1/4" flexible plastic tubing, the copper tapings were connected to a 20" U manometer (see Fig. 4.4). The tapings were positioned at a clockwise angle (looking downstream) of $90^\circ \pm 1^\circ$ to the horizontal.

The upstream section had an additional tapping positioned 2

diameters away from the orifice plate. This tapping was used for flow visualization at the downstream side of the plate(see Fig.4.4).

4.2.3 The pressure measurements.

A Bourdon gauge was attached to the pressure tapping just before the double bend. This ensured that the same flow conditions existed at all times and no excessive pressure was exerted on the test sections.

The measurement of differential pressure across the orifice plate was carried out with a mercury manometer. This instrument had an accuracy of $\pm 1.0\%$.

4.2.4 The velocity measurements.

The velocity measurements were carried out using Laser Doppler Anemometry(see Section 4.3). Axial velocities were measured on the horizontal diameter. The locations of the sections where the measurements were made, are shown in Fig.4.5. Based on a visualization of the flow(see Section 4.5) coming out from the orifice plate, the locations of the downstream sections were fixed .

Tangential velocities were measured on the horizontal diameter. All the measurements were taken at the nominal Reynolds number of 7.82×10^4 .

4.3 Laser doppler anemometry.

4.3.1 Introduction.

Laser Doppler Anemometry(LDA) is based on the fact that if two laser beams cross in a flow, forming an interference pattern, then a dust particle following the flow which passes through these fringes provides a photomultiplier with an oscillating signal. The frequency of this signal is

proportional to the velocity of the fluid in the direction perpendicular to the fringes. The LDA system used consisted of a 15 mW He-Ne laser, beam splitter, Bragg cell frequency shifter, focusing lens(focal length 300 mm) to produce the intersecting beams and a photomultiplier(focal length 150 mm) situated on the other side of the pipe to receive forward scattered light providing the signal for analysis. Natural seeding was used.

The Bragg cell shifts the frequency of one of the beams by 40 Mhz in order to make the fringe pattern a moving one. This reduces the errors introduced by low velocity particles or particles crossing the intersection volume at shallow angles to the fringes. It also eliminates the possibility of sign ambiguity.

The LDA system is shown in Figure 4.6 to 4.9. All the optical components producing the beams are supported on a metal frame with three screws. These screws are used to position the laser beams.

4.3.2 Signal processing.

The signal from the photomultiplier is an oscillating signal with a 40 MHz frequency shift. This signal is passed through a mixer that removes the 40 MHz shift and adds a small shift of 1 MHz or 2 MHz. Then, electronic filters isolate the flow signal and remove most of the noise. Finally, every "doppler burst" contained in the signal is analyzed and validated by the counter processor. The accepted bursts generate a validation rate as a fraction of the total data rate. Frequency values are transferred to an Apple computer which calculates the mean and standard deviation of the data collected. The user gives as input to the computer the following data:

- Frequency shift

- Fringe separation
- Velocities range
- Number of samples required

4.3.3 The alignment procedure.

The positioning of the beam intersection had to be carefully carried out. The laser was positioned on a free standing traversing table. This table consisted of a squared metal frame, two traversing mechanisms and four adjustable legs. The legs were positioned in such a way that the frame was horizontal in the direction perpendicular to the pipe and aligned with the pipe axis in the other direction. This was accomplished by using a spirit level to within $\pm 0.1^\circ$. The first adjustment ensured that measurements would be made on a horizontal line in the cross pipe direction. The second adjustment was the basis for the aligning of the laser beam perpendicular to the pipe. The traversing mechanism consisted of a screw thread of pitch 2 mm in the cross pipe direction and a similar thread in the flow direction. The photomultiplier was positioned on a similar table placed on the other side of the pipe. This table had no screw mechanisms but had slideways enabling continuous movement of the optics in both directions in the horizontal plane.

The alignment of the laser was carried out in the following steps. The table was levelled as described in previous paragraphs. The laser was positioned on a big piece of flat ABS which in turn rested on the traversing table. The laser's weight was taken by the side and rear screws located at the base plate. The laser beams were then positioned in the horizontal plane (they could be rotated about the centreline to an accuracy of $\pm 0.1^\circ$ by lining up a marked spot on the rotating part with a particular position on a scale in degrees marked in the stationary part). The laser's position was then altered until the returning beams

coincided exactly with the outgoing ones. This was determined by holding a piece of paper in front of one of the beams and establishing that the spots due to the incoming and outgoing beams coincided. This ensured that the laser's axis was perpendicular to the face of the pipe to within $\pm 0.1^\circ$. Further adjustments were made using the laser's supports until the incident beams and the beams reflected off the nearest wall of the pipe were all in a common plane (containing the pipe axis) together with the outgoing beams. This ensured that the beams' intersection was on the horizontal diameter rather than a displaced chord.

Measurements of axial velocity were taken with the beams in this horizontal position and measurements of tangential velocity on the horizontal diameter were taken after simply rotating the beams around without further alignment of the laser.

4.3.4 Calibration and allowance for the effects of refraction.

The fringe separation used for the velocity measurements is defined as follows:

$$\delta_f = \lambda_L / 2 \sin \theta$$

where:

δ_f = fringe separation

λ_L = wavelength of the light

2θ = angle between the laser beams

The angle θ was found by measuring the distance from the laser beams intersection and the screen, and the distance between the spots where the beams hit the screen. These distances were 217 cm and 36 cm respectively. This gives

$\Theta = 5.5^\circ$. Then, taking $\lambda_L = 632.8 \text{ nm}$ and $\Theta = 5.5^\circ$ we find $\zeta_f = 3.3 \text{ m}$. These measurements refer to values in air. There is no need to change the value of ζ_f because of refraction because although $\sin$ changes by the ratio of the refractive indices, the wavelength of the light changes similarly.

Now, we have to consider the effects of refraction at the pipe walls on the linearity of the traverse and upon the calibration factor.

In the case of the measurement of the axial velocity on the horizontal diameter, the position of the beams' intersection is a linear function of the position of the traversing table. The beams always intersect at the same angle with two parallel surfaces and so the calibration factor ζ_f does not change across the traverse (see Fig.4.10).

In the case of the measurement of the tangential velocity, the position of the beams' intersection is not a linear function of the traversing table. This position was calculated with a computer program. The deviation in the calibration factor was about 1.5%. The alterations are then in fact well within limits for the tangential velocity measurements and are ignored. The calibration constant was not changed.

4.3.5 The measurement procedure.

The procedure for taking the measurements was as follows. The table was traversed so that the beams intersected on the inside wall of the pipe and was then traversed whole numbers of turns of the thread across the pipe. In the case of the axial velocity measurements, a measurement was taken every 2 turns. In the case of the tangential velocity measurements, a measurement was taken every 3 or 4 turns. When measuring on the horizontal diameter it took 29 turns

to traverse the pipe in the horizontal plane, and 54 turns when in the vertical plane. 29 turns outside the pipe corresponds to 77 mm inside.

The errors in positioning the laser for measurements consisted of the error in positioning the beams intersection on the wall of the pipe at the start of a traverse.

At each measurement point the photomultiplier was positioned symmetrically between the beams and to one side of the plane containing them in such a way as to be focused onto the beams' intersection. The angle between the axis of the beams and that of the photomultiplier was always kept small (less than 10°) in order to obtain a strong signal.

The voltage applied was first increased until a signal was obtained, and the focusing and precise positioning of the photomultiplier were adjusted to maximize the data rate obtained. The applied voltage was then reduced until the data rate was only 200 Hz.

In the case of the upstream test section, the frequency shift used was 900 Hz for axial and tangential velocity measurements, the high and low pass filters being fixed at 256 kHz and 2 MHz respectively. In the case of the downstream test section, the frequency shift used was 3 MHz for axial and tangential measurements, the high and low pass filters being fixed at 256 kHz and 8 MHz respectively.

The computer was provided with upper and lower limits of velocity and then was asked to take 1500 individual velocity measurements. All the measurements lying outside the specified limits were eliminated and the mean (μ) and standard deviation (σ) of the accepted measurements were calculated. The number of measurements accepted out of the 1500 was 1400-1490 for the axial and tangential velocity

measurements. Based on the histograms provided by the computer and the velocities expected in the test sections, the velocity limits were selected.

The temperature of the water in the tank was recorded at regular intervals of half an hour. It usually rose by about 0.75°C in that time. Measurements were not taken outside the range 10°C to 25°C . Tests were carried out to determine whether the change in Reynolds number due to the change in with temperature were significant. It was found that they were not, the change in Reynolds number being no more than 10% .

4.4 The flow visualization procedure.

The procedure for visualizing the flow through the orifice plate was as follows. The upstream section was provided with a tapping located 1 diameter away from the orifice plate(see Fig. 4.4). A syringe containing purple dye was connected to a plastic tubing. This tubing was 20 cm in length. A needle made of steel pipe was connected to the other end of the plastic tubing. Then, the other end of the needle was passed through a nut with a hole in the middle and connected to the tapping in the upstream section(see Fig.4.11).

Once the device described above was set in the upstream section, the purple dye was injected and the behaviour of the flow downstream of the orifice plate was observed. The Reynolds number was 7.82×10^4 . The pipe was traversed across the vertical diameter. The needle tip was located at the axis of the pipe and at $0.3D$ above and below the horizontal diameter.

Additionally, a visualization of the flow in the upstream section was done using 2 cm long cotton threads glued on

the inner surface of the pipe.

Having done the visualization at the orifice plate located downstream of the double bend, another orifice plate with the same beta ratio was installed 53 diameters downstream of the tube bundle straightener. Again, the needle was set in the upstream section and the dye was injected. In this way, the flow visualization was completed.

4.5 The results of the flow visualization.

Figures 4.12 to 4.15 show the results of the flow visualization downstream of the orifice plate. In the case of Figures 4.12 and 4.13, the injection point was located at 15.4 mm below the axis of the pipe. Figures 4.14 and 4.15 show the flow pattern when the injection point was at the axis of the pipe. The orifice plate was located downstream of the double bend. In the upstream section a swirling flow rotating clockwise was observed. In the downstream section, a jet coming out from the orifice plate was identified. However, this jet had a downward inclination due to the swirl imparted by the double bend. Also, a large recirculation region extending over 2 diameters away from the orifice plate was observed. A region of intense turbulent mixing was found at 3 diameters downstream of the orifice plate.

Figure 4.16 to 4.17 show the results of the flow visualization downstream of the orifice plate located 53 diameters away from the tube bundle straightener. A jet coming out from the orifice plate was also found. This jet was centered with the axis of the pipe. A large recirculation region was distinguished downstream of the orifice plate.

The inlet profile was found to effect the outlet profile downstream of an orifice plate.

4.6 The results of the velocity measurements.

4.6.1 Axial velocity measurements.

In the case of the upstream section, the axial velocity profiles show that there is a slight deviation from the fully developed flow(see Figures 4.18 to 4.20). In the case of the downstream sections, the axial velocity profiles give a picture of the jet coming out from the orifice plate(see Figures 4.24 to 4.25). It can be shown that the maximum velocity decreases as the jet moves further away from the orifice.

4.6.2 Tangential velocity measurements.

Figures 4.21 to 4.23 and 4.26 to 4.27 show the tangential velocity measurements on the test sections. The upstream measurements are presented in Fig. 4.21 to 4.23. The downstream measurements are shown in Fig. 4.26 to 4.27. Negative radial positions are on the right hand side of the pipe looking downstream. The upstream swirl levels shown were calculated using the trapezium rule, integrating across the pipe. The formula used was as follows. From equation (6) with $V_z = \bar{V}_z = \text{const.}$, we have:

$$2S = (2/V_{zav}) \int_{-1}^1 x^2 V_{\theta} dx$$

where x is the distance from the axis of the pipe. The downstream measurements show that the swirl level is amplified at the orifice plate.

4.7 The results of the pressure measurements.

The location of the pressure measurements are shown in Fig.

4.4 . According to BS 1042, the flange tappings were located 1" away from the orifice plate. For the orifice plate downstream of the double bend, the pressure difference reading was $\Delta p = 20''$ hg. Then, the discharge coefficient, c_d , was calculated from equation (5).

The discharge coefficient under swirling conditions was found to be 0.5711. For nonswirling conditions, the discharge coefficient was 0.6027, 0.6026, and 0.6039 for the flange, corner, and D and D/2 tappings (from BS 1042). Then, the difference between the coefficients calculated was - 5.15%.

5. THE COMPUTATIONAL WORK

5.1 Scope of the computational work.

The aim of the computational work was to obtain a description of the flow field through an orifice plate subjected to a swirling flow. The swirling flow considered was an axisymmetric swirl of solid body rotation type. According to the data of Mc Manus(12), the flow profile coming out from a double bend is of this type.

A commercial computer package named PHOENICS provided by CHAM ltd. was used for the simulation of the flow field through the orifice plate. A concentric orifice with a beta ratio of 0.4 and zero thickness was specified.

Based on the pressure difference obtained from the simulation, the discharge coefficient was calculated.

Additionally, a computer simulation was performed using a fully developed profile without swirl as inlet condition for the orifice plate. The discharge coefficient was calculated and it was compared with the coefficient determined for the swirling flow conditions.

5.2 The computational model.

Figure 5.1 shows the computational test section. This section was a three-dimensional circular pipe of 3" diameter. The orifice plate was located 4 diameters downstream of the inlet.

5.3 The computer code.

The PHOENICS code solves the discretized versions of the hydrodynamic equations and the equations for heat and mass transfer by a variant of the SIMPLE procedure. The method

is denoted as a "fully implicit finite domain" method. There are two built-in grid geometries; cartesian and cylindrical polar coordinates. PHOENICS embodies the k-epsilon model of turbulence to close the momentum equations for turbulent flow. This model consists of two differential equations, one for the turbulent kinetic energy (k) and one for its dissipation rate (epsilon). The set of constants used in these equations throughout the simulations is the so called "universal" set of constants.

5.3.1 Components of PHOENICS.

Figure 5.2 shows the components of PHOENICS. It comprises two different computer programs. EARTH is the main program and SATELLITE is a small program in which the user can define Fortran statements or transmit data-input files to EARTH.

EARTH embodies all the main flow-simulating software. It includes coding sequences which represent the laws of physics applied to elements of space and time. SATELLITE is considered as an interpreter. It translates all the PHOENICS Input Language (PIL) into commands to which the computer responds. The PIL commands are specified in the library or created by means of the text editor of the computer system.

Additionally, the user can send data to SATELLITE interactively. The SATELLITE also contains a Fortran subroutine, named SATLIT. In this way, the user can make changes to the subroutine and transmit them to EARTH.

EARTH also has a component called GROUND in which the user can define specific problem information. All this data is taken into account by EARTH while carrying out its flow-simulating function. GREX1 is another subroutine which has embodied standard commonly-required features for different

flow problems.

The PHOTON program (PHOENICS output option) is used to plot graphical displays on the VDU screen. Options are provided to plot the grid, the flow vectors and pressure contours, etc..

5.4 Boundary Conditions.

Figure 5.3 shows the boundary regions for the test section. At the inlet the profiles for the velocity and the two turbulence model parameters were prescribed. A solid body rotation superimposed on the fully developed profile was used for modelling the swirling flow. Constant values of k and epsilon were specified at the inlet. A constant pressure was prescribed at the outlet. Wall functions were applied to the pipe wall and to the orifice plate.

5.5 Grid selection.

In the final simulation a model with 65 cells in the axial direction, 10 cells in the radial direction, and 16 cells in the tangential direction has been used. The grid-lines in the vicinity of the orifice plate are shown in Figures 5.4 and 5.5. Grids with cells of varying sizes were used. A fine grid was implemented in regions with large velocity gradients; i.e. close to the plate in the axial direction and close to the wall and to the orifice lip in the radial direction. The developed grid was optimized for an orifice meter with a diameter ratio close to the value used in the flow rig.

Numerical experiments proved that small cells should also be used in the estimated position of vena contracta.

5.6 Calculation of the discharge coefficient.

The discharge coefficient was calculated from equation (5)

From the simulated pressure field, Δp was evaluated for corner, flange, and D and D/2 tappings. Considering the Reynolds number, the volumetric flowrate was calculated and c_d was determined.

5.7 Results of the numerical experiments.

5.7.1 The effect of a fully developed profile with swirl.

In Figure 5.6 we can see a velocity vector diagram of the computed swirling flow through an orifice plate. Clearly, the overall features can be identified. A forced vortex is approaching the orifice. Three regions are formed at the outlet, namely a central jet, a shear layer and a recirculation flow. Reattachment of the flow was located at 2.46 D downstream of the orifice plate.

It can also be observed from the computed pressure field that the maximum pressure is located at the wall of the pipe. This confirms the effect of the centrifugal forces acting in the fluid, as a result the separated region behind the orifice is reduced and the upstream and downstream pressures are increased.

The ratio of the maximum inlet tangential velocity to mean axial velocity was 0.3.

The computed discharge coefficient was found to be 0.7175, 0.7152, and 0.7161, for the flange, corner, and D and D/2 tappings respectively.

5.7.2 The effect of a fully developed profile without swirl.

A 1/7 power law profile was specified at the inlet of the orifice plate. Figure 5.7 shows the corresponding velocity

vector diagram. Downstream of the orifice, a central core , a shear layer, and a recirculation region were identified. The reattachment of the flow was located at 2.28 D downstream of the orifice.

Computations were performed for two Reynolds numbers. The accuracy was reduced when increasing the Reynolds number. Possible explanations are that the correctness of the turbulence model is Reynolds-dependent or the streamline pattern changes so that the grid becomes inappropriate when varying the Reynolds number.

The computed discharge coefficient was found to be 0.7177, 0.7153, 0.7256, for the flange, corner, and D and D/2 tappings.

6. DISCUSSIONS, CONCLUSIONS, AND RECOMMENDATIONS.

6.1 Discussion.

6.1.1 The discharge coefficient.

In Chapter 3 the discharge coefficient was calculated using a simplified model of a swirling flow through an orifice plate. For the level of swirl considered, the results show that there is an increase on the upstream and downstream wall pressure. The downstream increase is greater than the upstream increase and as a consequence the discharge coefficient is decreased.

With the measured wall pressures upstream and downstream of the orifice plate, the value of the discharge coefficient was obtained in Chapter 4 ($c_d = 0.5711$). This value is lower than the one obtained for non swirling conditions.

In Chapter 5 a computer simulation of the swirling flow through a zero thickness orifice plate was made. The discharge coefficient found was 0.7175. The results demonstrate that the c_d is reduced.

The simplified model used in Chapter 3 gives a good approximation of the discharge coefficient under swirling flow. The theoretical c_d determined was 0.5840. The experimental discharge coefficient agrees well with the theoretical discharge coefficient.

From the results given above, it can be seen that a reasonable agreement was found between the computed and the experimental discharge coefficient. The inaccuracy of the computed value may be due to the k-epsilon turbulence model giving a poor description of swirling flow.

The literature on the effect of swirl on the discharge

coefficient shows wide differences in the reported results. Mottram and Rawat (4) found no more than 2% increase in the discharge coefficient. Brennan (13) reported 8% increase. Murakami (1) obtained a range between +10% to -20%. Irving (5) showed a +3 to -3% change. Mc Hugh, Dyet and Kinghorn (3) have also reported some discrepancies between their work and the work of Irving (5), and Lugt(14). The disagreement between the published data available and the data obtained in Chapter 3 and 4 was found to be the Reynolds number used and the swirl generating system.

Other researchers computed the discharge coefficient under swirling flow conditions. M. J. Reader-Harris (9) found a +1.5 to -1.3% change. B. R. Patel (15) reported a maximum of +2.4% change. The swirl number's range (S value) considered in Patel's work was 0.1 to 2.4. Some discrepancies were found between the reported values and the computed data determined in Chapter 5. The reason was that the Reynolds and swirl numbers used in the published data were different from those specified in the simulation.

6.1.2 Pressure distribution on walls.

The pressure distribution across the orifice plate is severely affected by a swirling flow. Due to the action of the centrifugal forces, the upstream and downstream wall pressures are increased. According to Murakami (1), this effect becomes more important for orifices with small beta ratios. He also says that as the swirl is increased and the beta ratio decreased, the vena contracta approach the downstream side of the orifice plate.

In the present experiments with the double bend, a deviation of the swirl center from the pipe axis can be observed at about 1D to 3D downstream of the orifice (see Fig. 4.12 to 4.14). This must have an effect on the wall pressure distribution.

6.1.3 Velocity profiles.

The velocity profiles shown in Figures 4.18 to 4.27 are in good agreement with the corresponding ones given by other researchers. Mottram and Rawat (4) found similar profiles in the downstream section of a 2x90 degrees double bend. Murakami's velocity profiles (1) are also similar to those presented in Chapter 4.

Basically, the experimental profiles obtained correspond to swirling flow having a form of a solid body rotation. Also, it is clearly seen that the swirling motion is amplified by the orifice plate. The swirl number (S) 5.5 diameters downstream of the double bend was about 0.077. This swirl number increases to a value of 0.099 at a section located 6.0 diameters downstream of the double bend.

According to the BS 1042, 2 degrees is the maximum swirl level angle allowed for orifice plate measurements. This figure seems to be a good measure of the point at which the axial velocity profile begins to change. For higher values, it has been observed that the axial velocity profile differs from the fully developed profile (see Fig. 4.18 to 4.20).

6.2 Conclusions.

1.- The flow coming out from a double bend is a swirling flow. Its shape resembles a solid body rotation.

2.- An orifice plate placed 6.5 diameters downstream of the double bend used in the above work is subjected to a swirling flow with a swirl number of $S = 0.099$ (about a swirl angle of 8 degrees).

3.- Taking the experimental value of 0.5711 and the

corresponding published value of 0.6027 for the discharge coefficient under swirling and non swirling flow conditions respectively, the calculated change in c_d is 5.15% (see Section 4.7). This is for the value $\beta = 0.4$.

4.- The PHOENICS program proved to be a powerful tool to calculate the swirling flow field through an orifice plate.

5.- According to Halsey's work (16), the minimum straight length between the double bend and an orifice plate flowmeter should be 70 diameters. He considered a swirl angle less than 2 degrees. BS 1042 suggests a value of 36 diameters. Mottram and Spencer (7) propose 76 diameters.

6.3 Recommendations for installation of orifice plate flowmeters.

1.- An orifice plate must be placed away from double bends.

2.- Users of orifice plate flowmeters have to define the type of measurement needed and the most appropriate installation.

3.- If an orifice plate has to be placed downstream of a double bend, a straight length of 70 diameters has to be specified. In some piping layouts, the use of a flow straightener may be considered as an alternative.

4.- The minimum straight lengths provided in standards have to be reviewed. Otherwise, significant errors will be produced by the orifice plate flowmeters.

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APPENDIX A.

THE RESULTS OF THE VELOCITY MEASUREMENTS.

Tables 1 to 5 show the results of the velocity measurements. The positive values indicated in the traverse positions, correspond to the direction from right to left as an observer looks downstream. The distances are non-dimensional radius values.

The velocity measurements are given in m/s. The turbulence intensity shown is the root mean square of the velocity fluctuations in the flow direction considered. A positive value of the tangential velocity corresponds to a flow going upwards.

Table 1

Velocity measurements on section 1 at $Re = 7.82 \times 10^4$.

Traverse Position	Axial Velocity	Turbulence Intensity
-1.0	Pipewall	
-0.86	1.052	0.107
-0.79	1.103	0.105
-0.72	1.130	0.087
-0.59	1.142	0.084
-0.45	1.158	0.085
-0.31	1.132	0.089
-0.17	1.114	0.083
-0.03	1.106	0.077
0.10	1.047	0.087
0.24	1.036	0.085
0.38	1.025	0.081
0.52	1.031	0.092
0.66	1.016	0.092
0.79	1.000	0.100
0.86	0.980	0.109
0.93	0.910	0.123
1.0	Pipewall	

Traverse Position	Tangential Velocity	Turbulence Intensity
-1.0	Pipewall	
-0.78	-0.103	0.074
-0.72	-0.088	0.075
-0.66	-0.062	0.077
-0.62	-0.060	0.078
-0.56	-0.049	0.079
-0.46	-0.036	0.079
-0.32	-0.025	0.075
-0.17	-0.022	0.079
-0.01	-0.016	0.089
0.06	-0.009	0.098
0.17	-0.003	0.104
0.27	0.009	0.106
0.40	0.014	0.112
0.52	0.051	0.123
0.63	0.093	0.132
0.74	0.153	0.143
0.83	0.223	0.139
0.92	0.259	0.159
1.0	Pipewall	

Table 2
Velocity measurements on section 2 at $Re = 7.82 \times 10^4$.

Traverse Position	Axial Velocity	Turbulence Intensity
-1.0	Pipewall	
-0.86	1.062	0.099
-0.79	1.095	0.086
-0.72	1.111	0.084
-0.59	1.135	0.078
-0.45	1.133	0.070
-0.31	1.121	0.082
-0.17	1.080	0.083
-0.03	1.063	0.084
0.10	1.064	0.080
0.24	1.054	0.081
0.38	1.041	0.085
0.52	1.029	0.090
0.66	1.014	0.103
0.79	1.026	0.103
0.86	0.972	0.115
0.93	0.916	0.124
1.0	Pipewall	

Traverse Position	Tangential Velocity	Turbulence Intensity
-1.0	Pipewall	
-0.95	-0.119	0.072
-0.89	-0.106	0.067
-0.83	-0.094	0.071
-0.78	-0.086	0.071
-0.72	-0.066	0.071
-0.66	-0.058	0.072
-0.62	-0.050	0.074
-0.56	-0.037	0.074
-0.46	-0.030	0.079
-0.36	-0.023	0.079
-0.22	-0.018	0.084
-0.01	-0.012	0.086
0.02	-0.009	0.091
0.13	0.003	0.097
0.23	0.007	0.105
0.37	0.037	0.115
0.49	0.087	0.125
0.57	0.127	0.131
0.71	0.212	0.138
0.81	0.250	0.140
0.90	0.272	0.158
1.0	Pipewall	

Table 3
Velocity measurements on section 3 at $Re = 7.82 \times 10^4$.

Traverse Position	Axial Velocity	Turbulence Intensity
-1.0	Pipewall	
-0.86	1.026	0.103
-0.79	1.079	0.085
-0.72	1.089	0.083
-0.59	1.111	0.074
-0.45	1.125	0.070
-0.31	1.123	0.071
-0.17	1.115	0.072
-0.03	1.102	0.080
0.10	1.076	0.079
0.24	1.065	0.082
0.38	1.044	0.076
0.52	1.049	0.077
0.66	1.015	0.084
0.79	1.040	0.096
0.86	1.000	0.105
0.93	0.922	0.119
1.0	Pipewall	

Traverse Position	Tangential Velocity	Turbulence Intensity
-1.0	Pipewall	
-0.95	-0.122	0.073
-0.89	-0.116	0.069
-0.83	-0.103	0.082
-0.72	-0.080	0.075
-0.61	-0.048	0.068
-0.46	-0.034	0.076
-0.31	-0.012	0.084
-0.17	-0.010	0.079
-0.06	-0.009	0.090
0.06	-0.006	0.100
0.20	0.012	0.104
0.34	0.044	0.113
0.46	0.077	0.115
0.58	0.138	0.129
0.68	0.162	0.124
0.78	0.189	0.143
1.0	Pipewall	

Table 4
Velocity measurements on section 4 at $Re = 7.82 \times 10^4$.

Traverse Position	Axial Velocity	Turbulence Intensity
----------------------	-------------------	-------------------------

-1.0	Pipewall	
-0.45	0.699	1.811
-0.31	3.245	1.888
-0.17	6.978	1.449
-0.03	7.886	0.501
0.10	7.836	0.730
0.24	7.117	1.506
0.38	3.344	1.957
0.52	1.655	2.683
1.0	Pipewall	

Traverse Position	Tangential Velocity	Turbulence Intensity
----------------------	------------------------	-------------------------

-1.0	Pipewall	
-0.31	0.048	2.238
-0.17	0.101	1.295
-0.06	0.209	1.233
0.06	0.307	1.195
0.20	0.649	2.383
0.34	1.205	3.029
1.0	Pipewall	

Table 5
Velocity measurements on section 5 at $Re = 7.82 \times 10^4$.

Traverse Position	Axial Velocity	Turbulence Intensity
----------------------	-------------------	-------------------------

-1.0	Pipewall	
-0.59	0.178	0.175
-0.45	1.387	1.365
-0.31	3.279	3.227
-0.17	5.663	5.574
0.03	6.583	6.479
0.10	7.280	7.165
0.24	5.857	5.764
0.38	3.964	3.902
0.52	1.819	1.790
1.0	Pipewall	

Traverse Position	Tangential Velocity	Turbulence Intensity
----------------------	------------------------	-------------------------

-1.0	Pipewall	
-0.06	0.106	1.340
0.06	0.184	1.631
0.20	0.509	2.637
0.34	0.689	2.790
1.0	Pipewall	

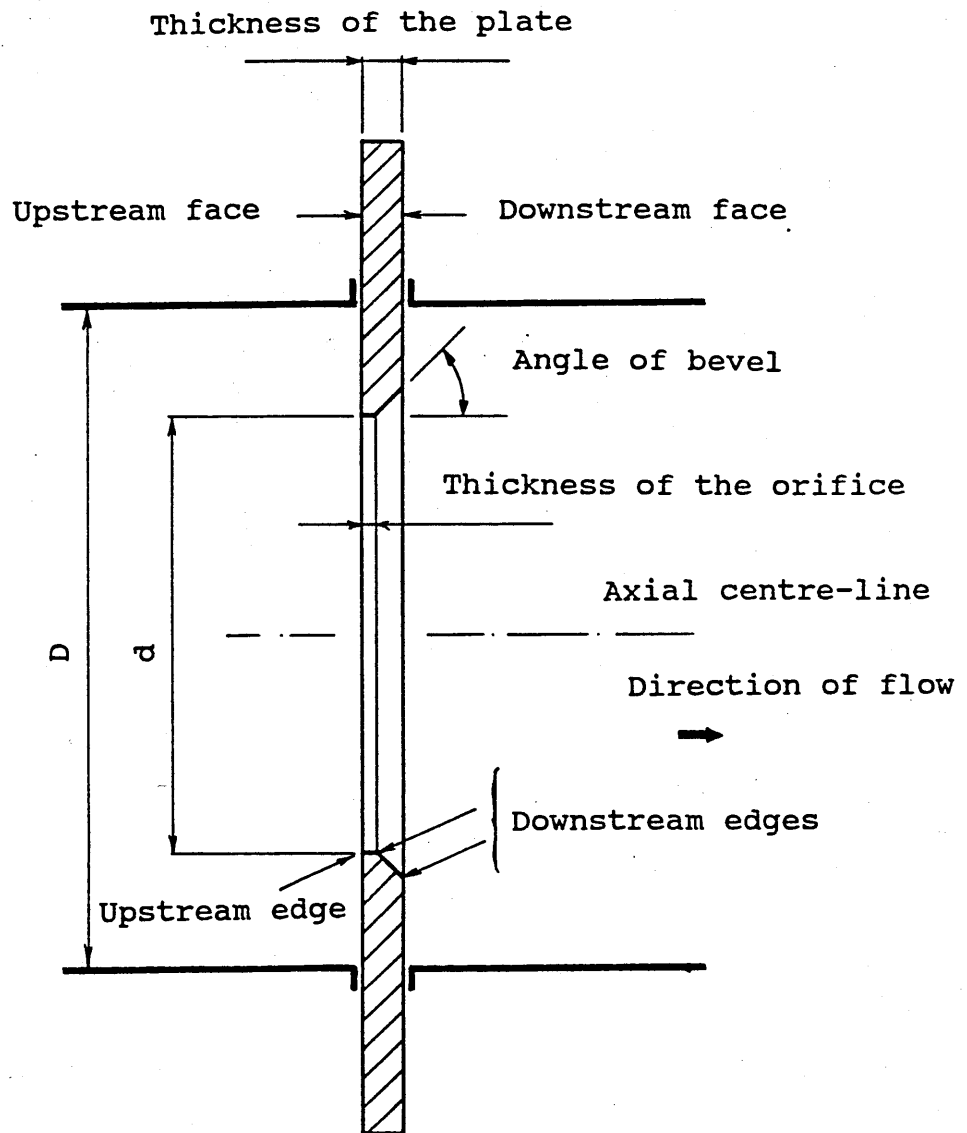
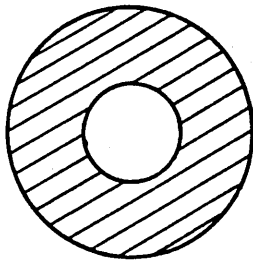
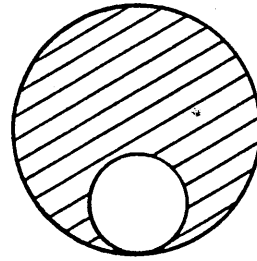


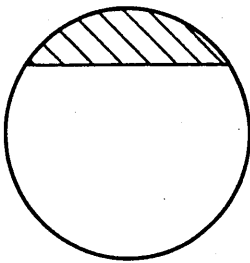
Figure 3.1 Orifice plate geometry



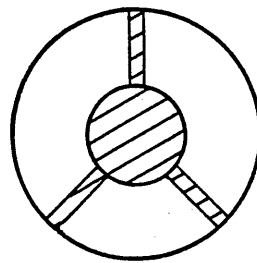
(a)



(b)



(c)



(d)

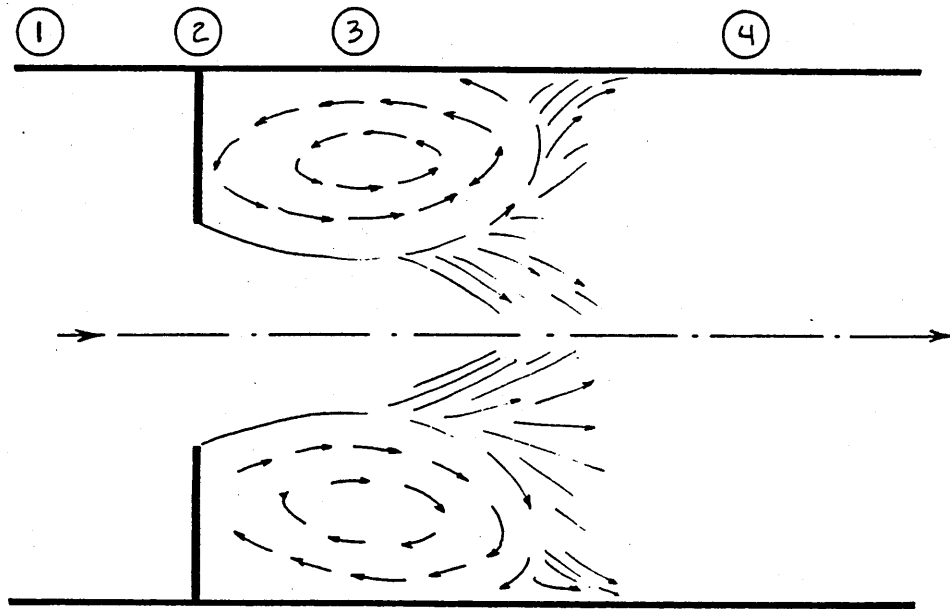
(a) Concentric

(b) Eccentric

(c) Segmental

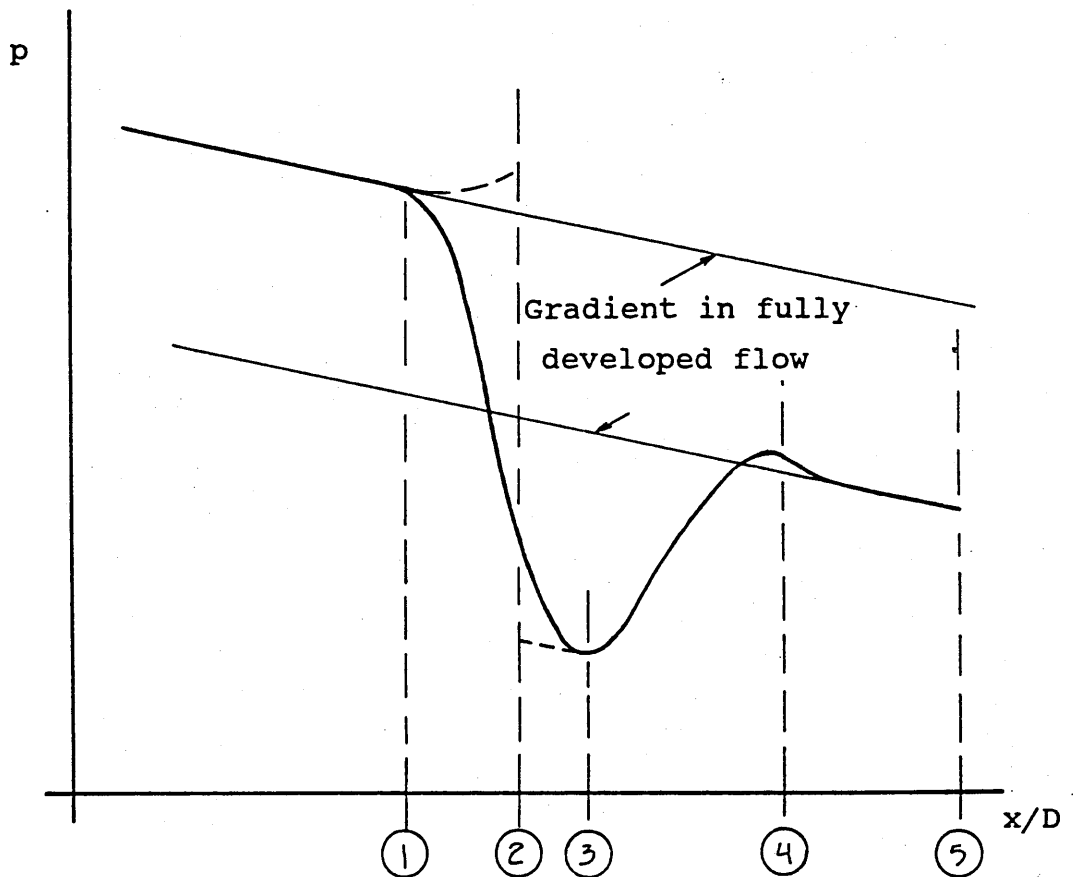
(d) Annular

Figure 3.2 Alternative designs of orifice plates



- ① Upstream conditions
- ② Plane of orifice plate
- ③ Vena contracta plane
- ④ Recovery plane

Figure 3.3 Turbulent flow pattern



- ① Upstream conditions
- ② Plane of orifice plate
- ③ Vena contracta plane
- ④ Recovery plane
- ⑤ Reestablishment of fully developed flow

--- Wall
 ——— Axis

Figure 3.4 Pressure distributions in an orifice plate

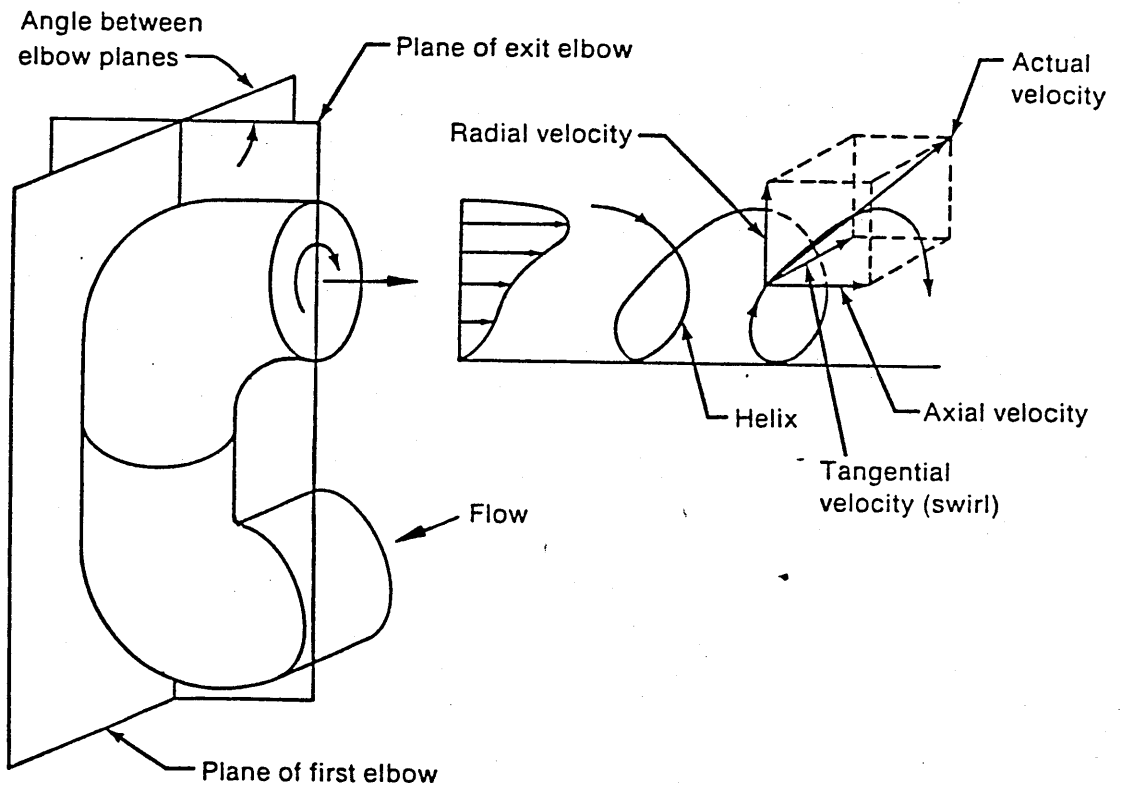
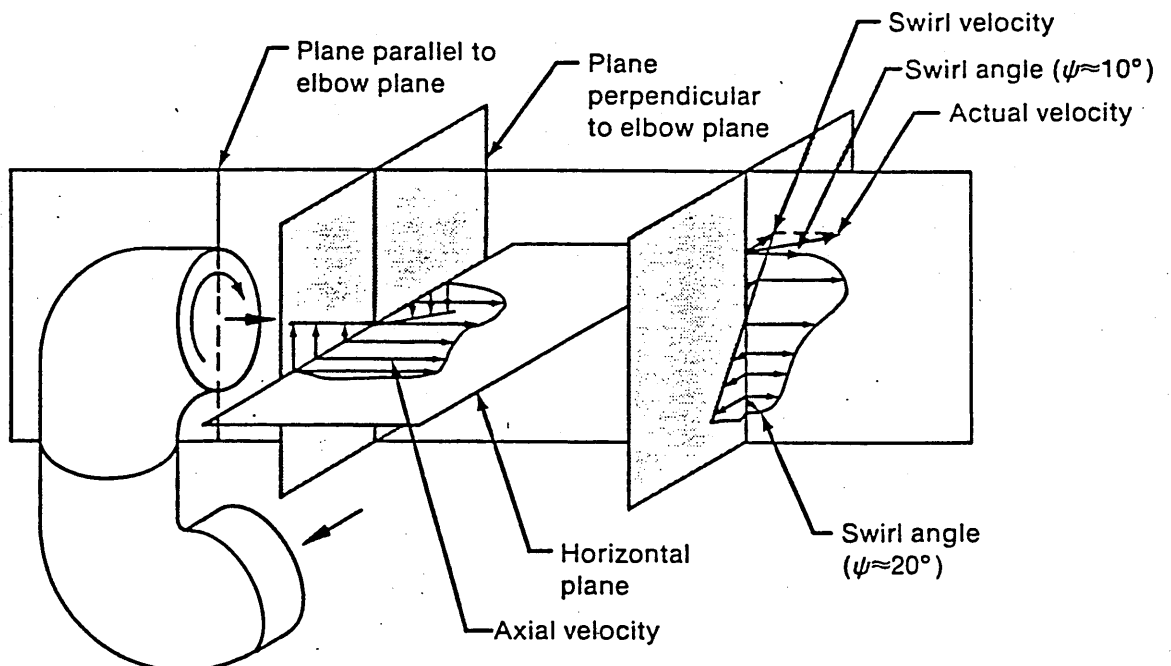


Figure 3.5 Velocity profiles in swirling flow following close-coupled elbows in different planes

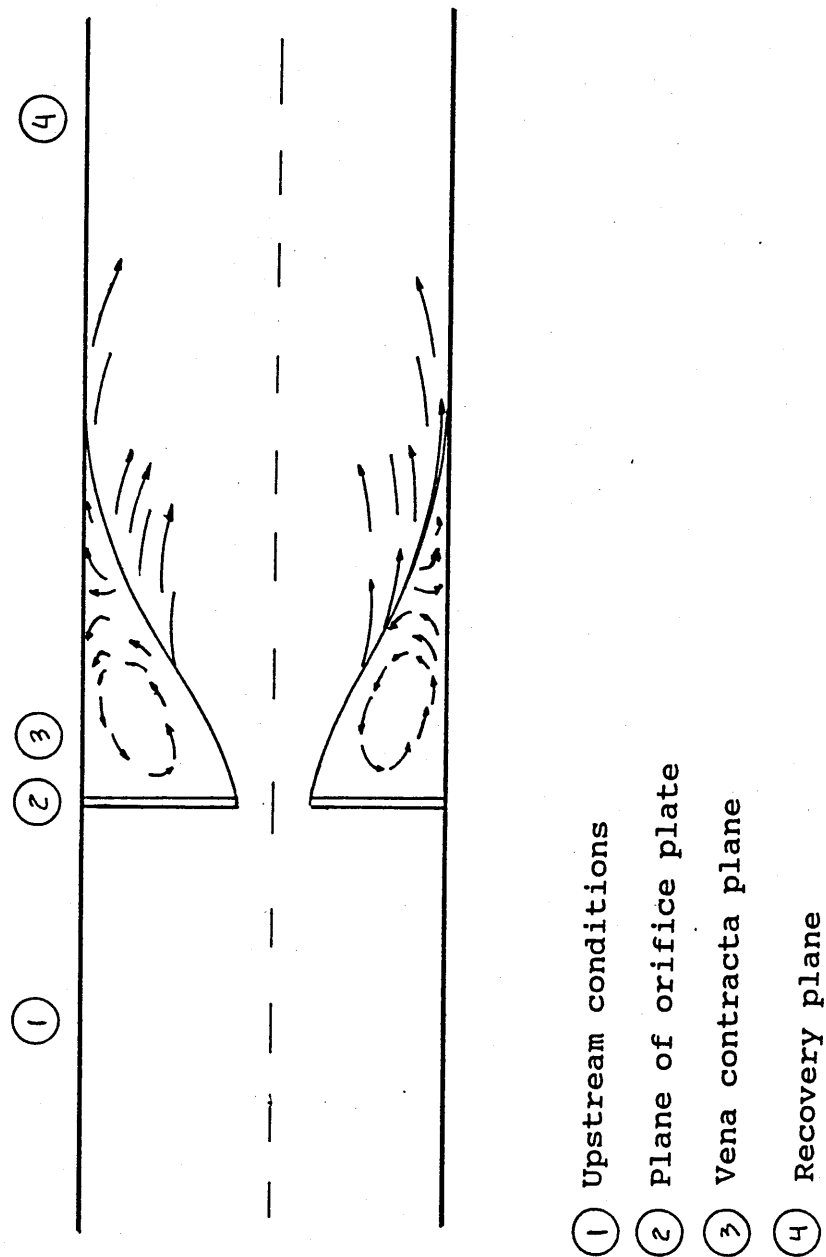


Figure 3.6 Swirling flow pattern

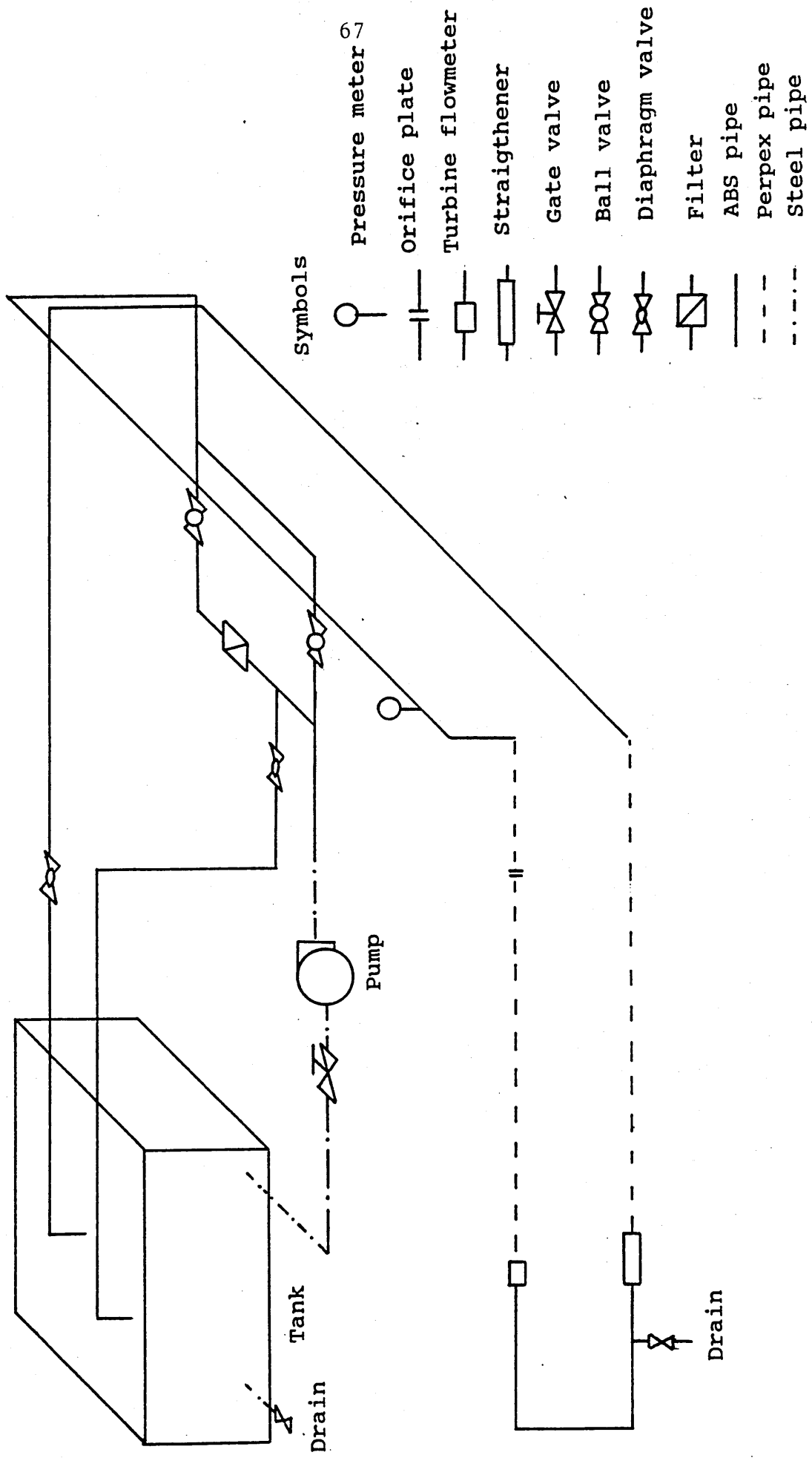


Figure 4.1 Schematic diagram of the flow loop

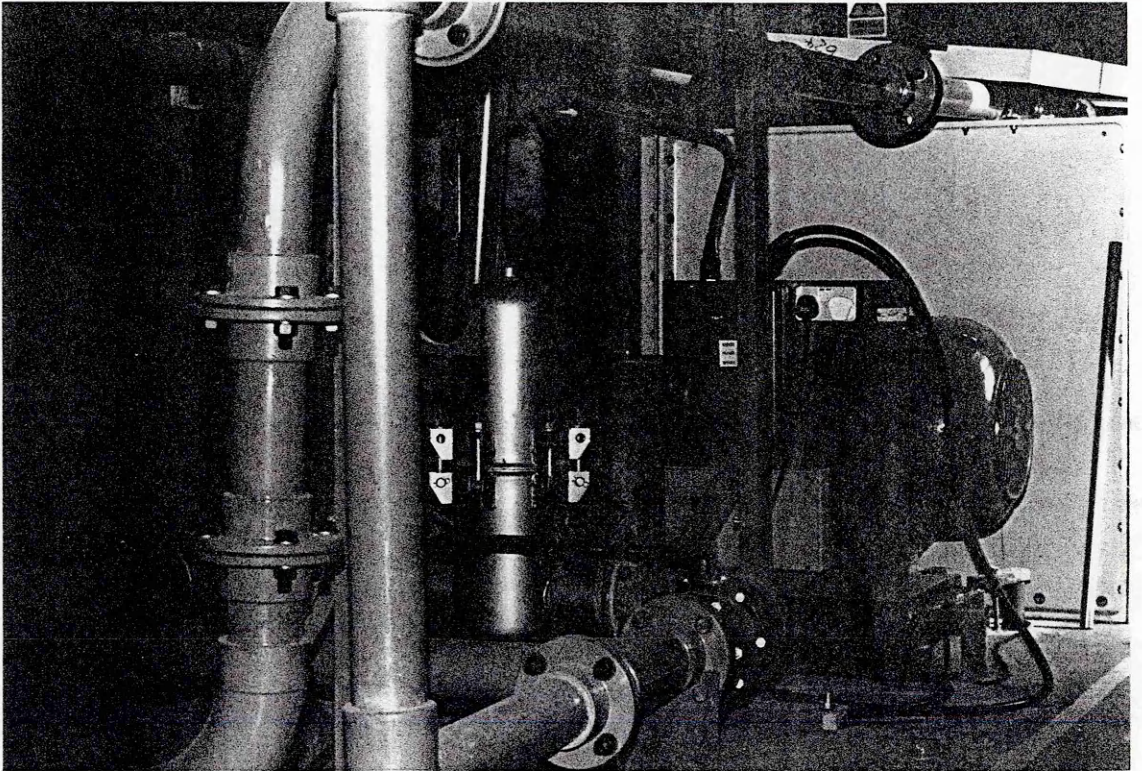


Figure 4.2 The flow loop.

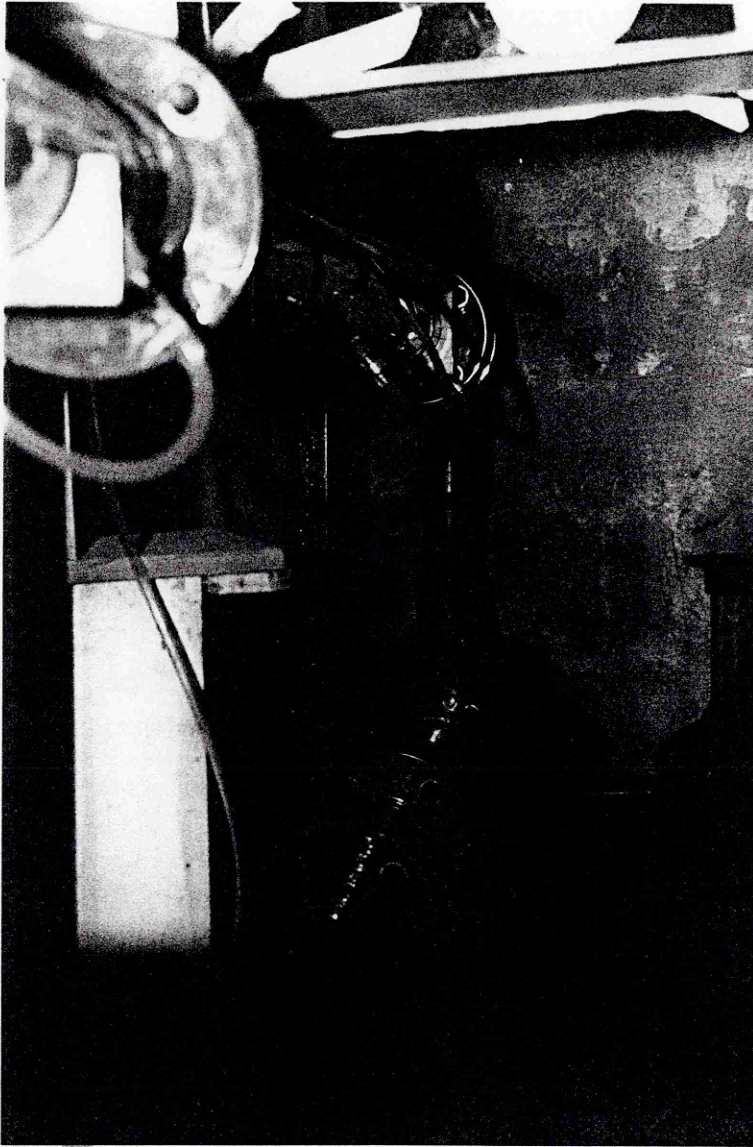


Figure 4.3 The flow loop.

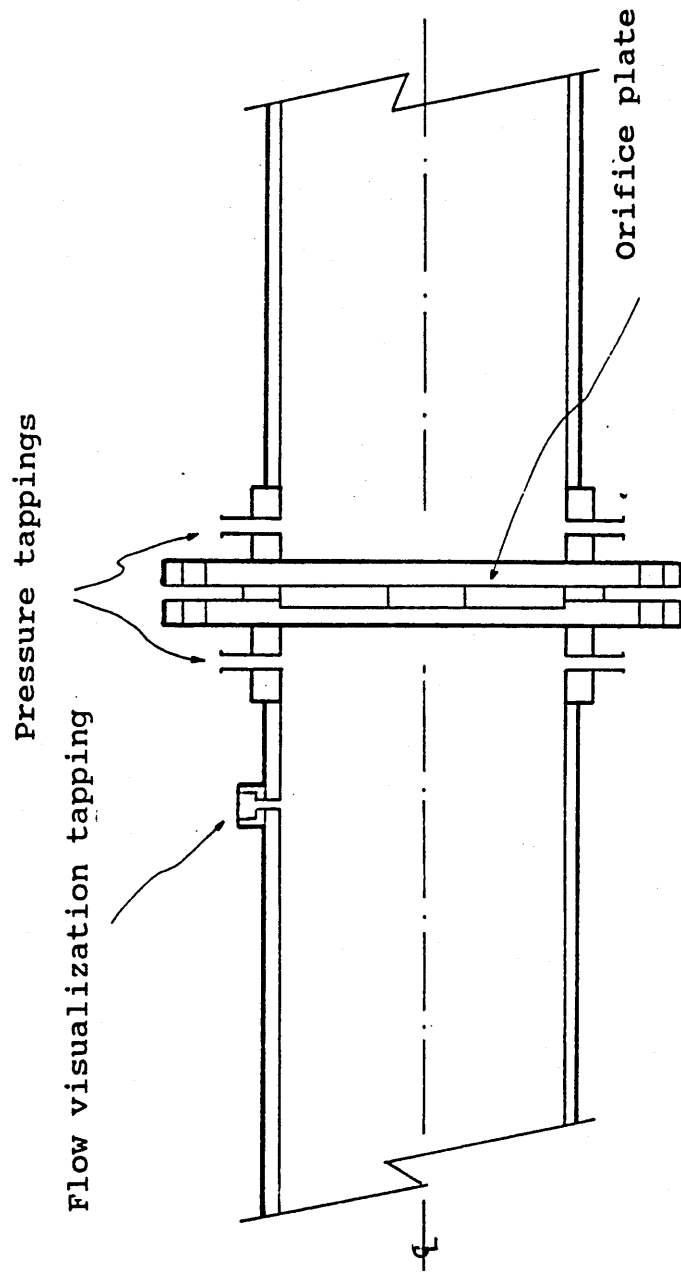
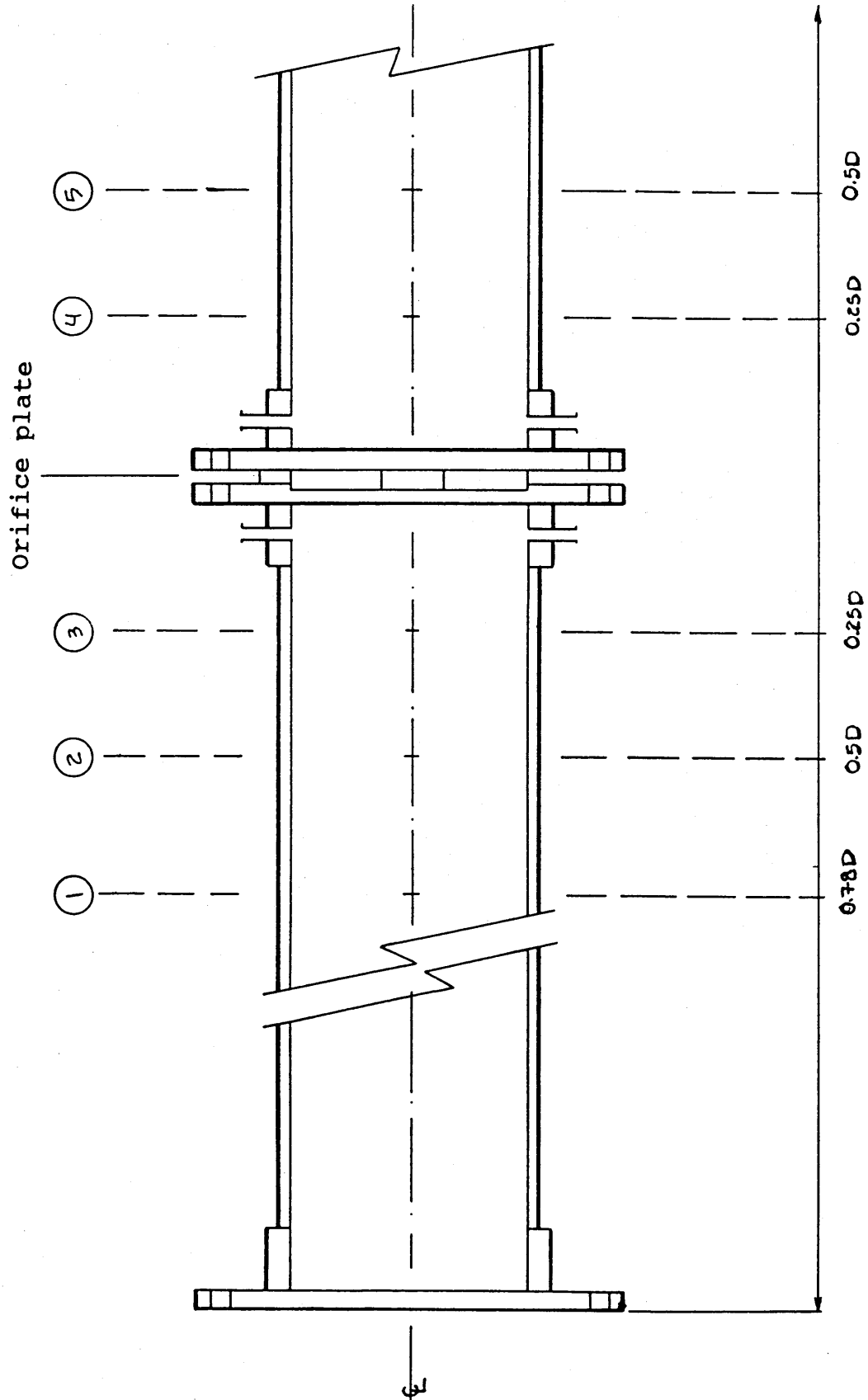


Figure 4.4 Test sections



○ Measuring section number Figure 4.5 Locations of the velocity measurements expressed as so many pipe diameters away from the orifice plate

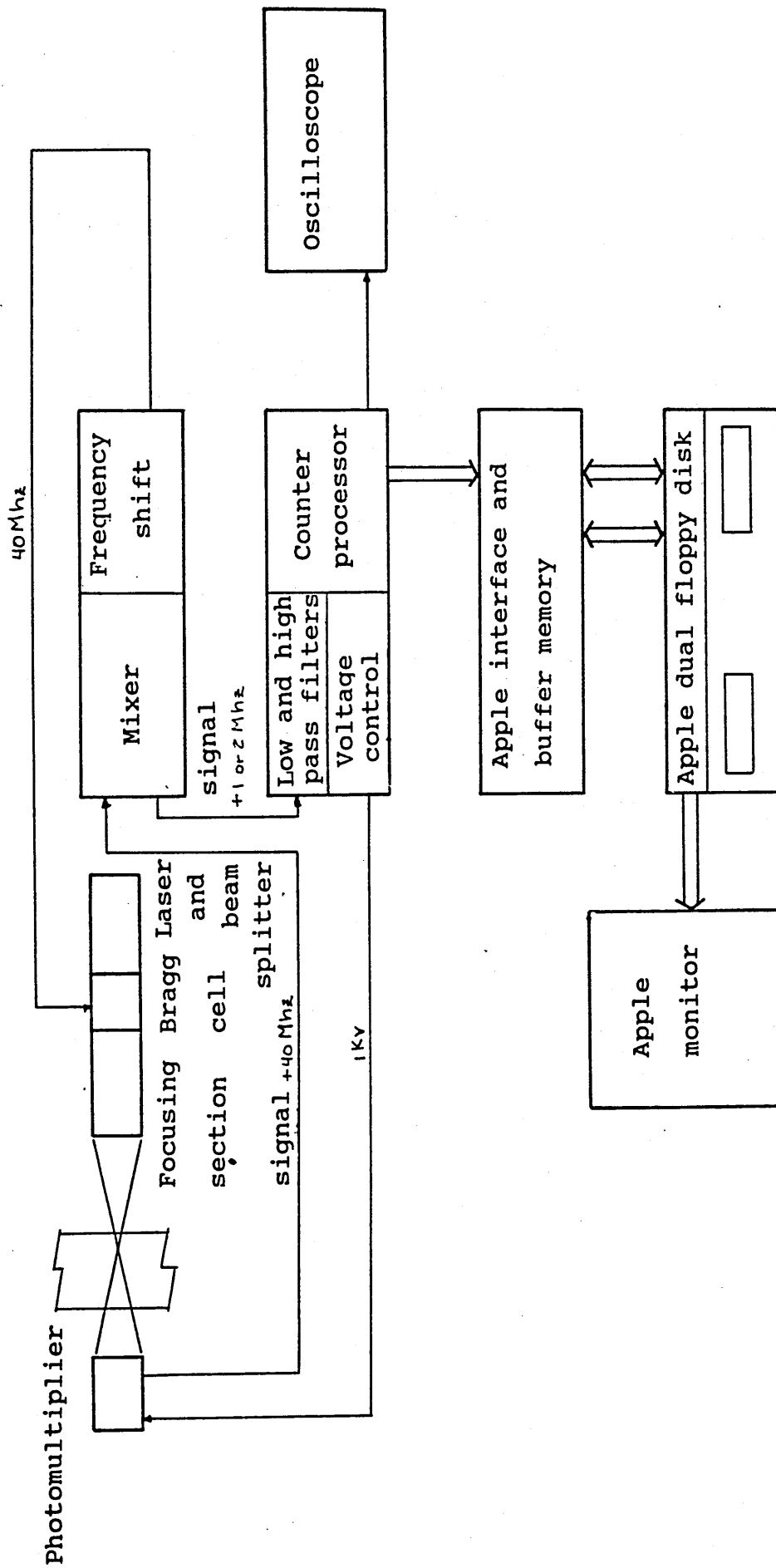


Figure 4.6 LDA system

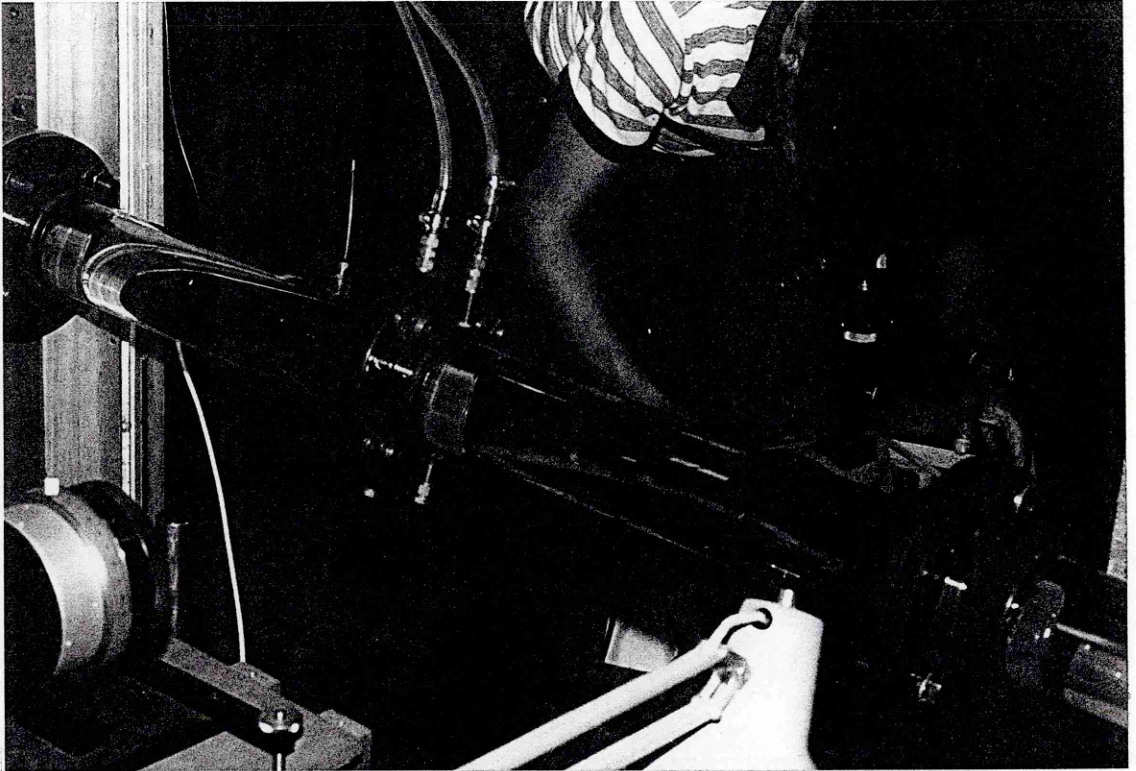


Figure 4.7 The photomultiplier.

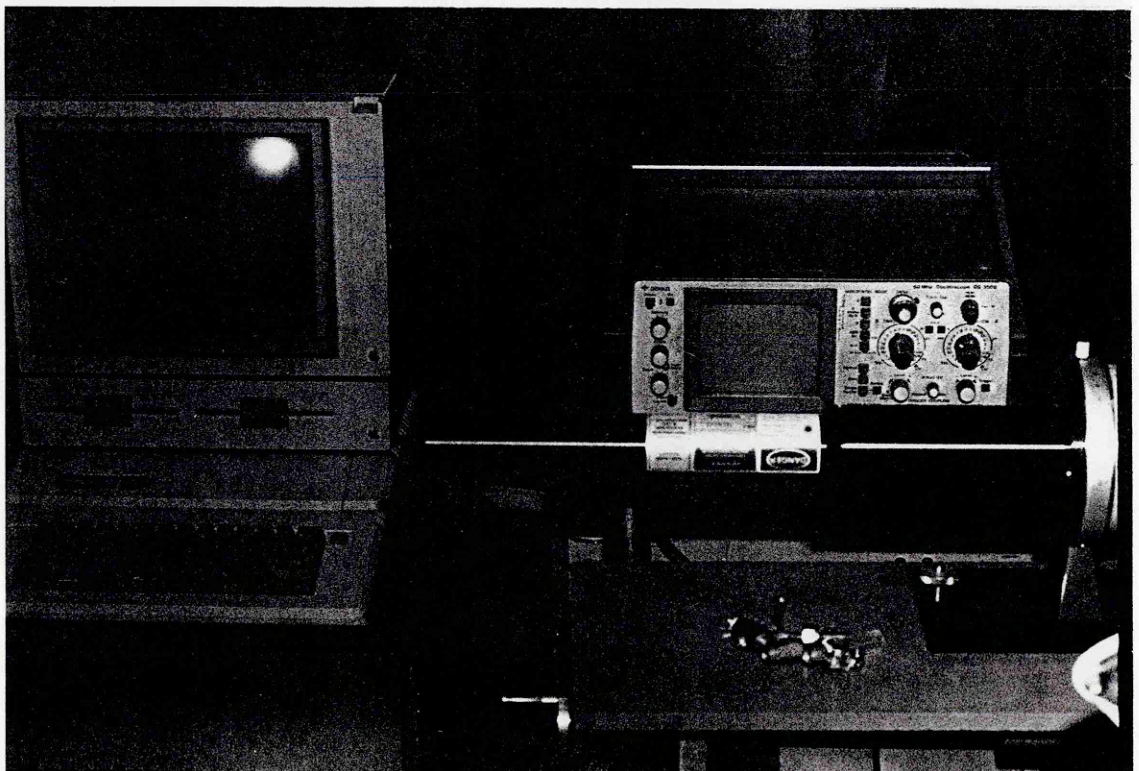


Figure 4.8 The signal processing equipment.

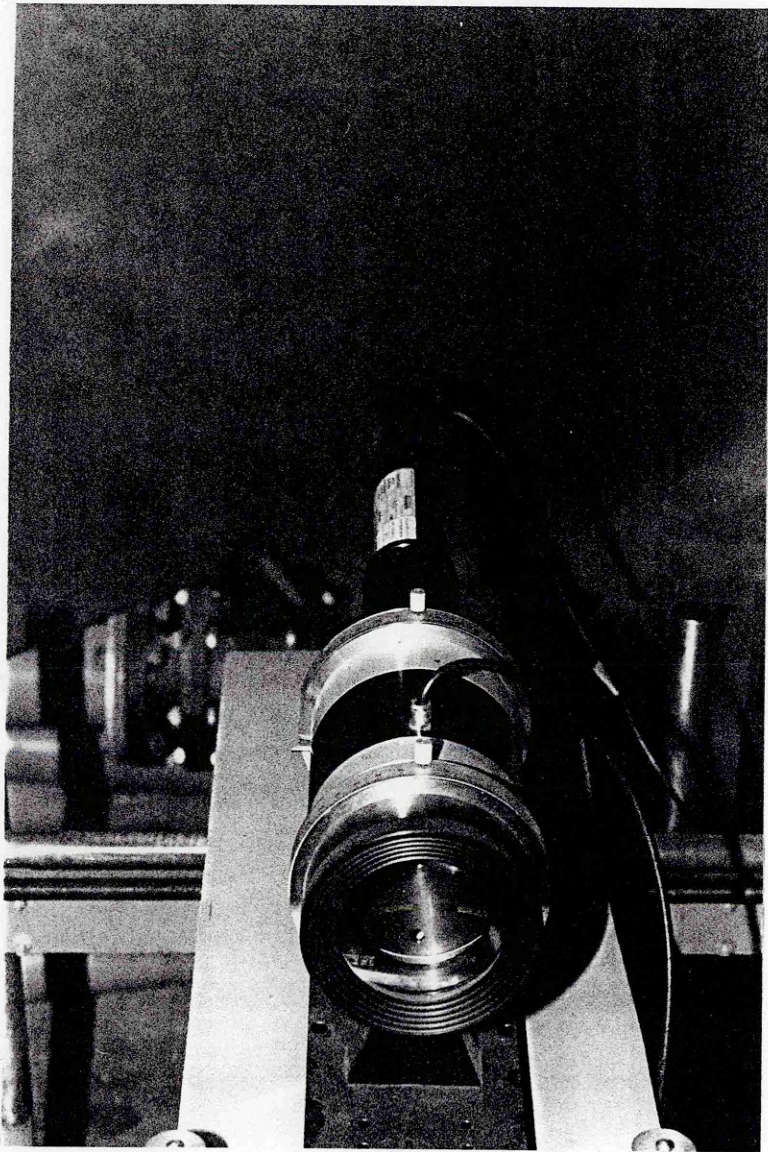


Figure 4.9 The laser, beam splitter , Bragg cell , and focusing lens.

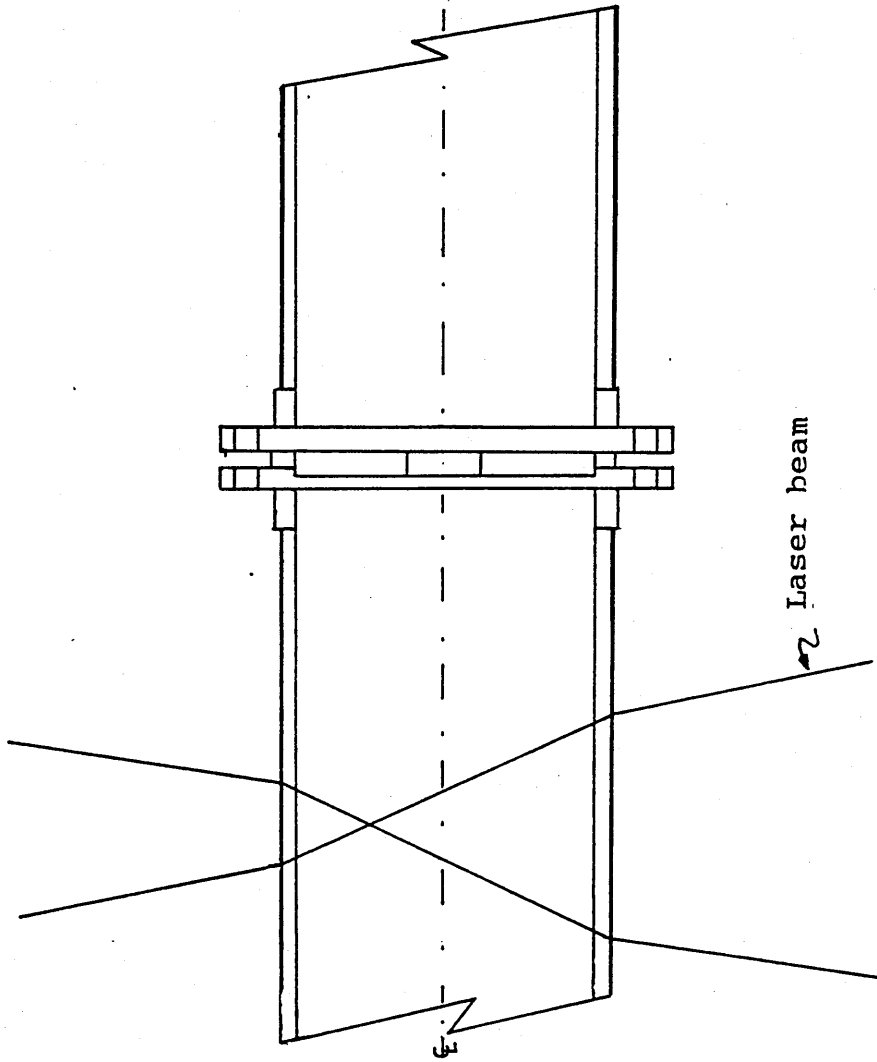


Figure 4.10 Laser beam's intersection for the axial velocity measurements

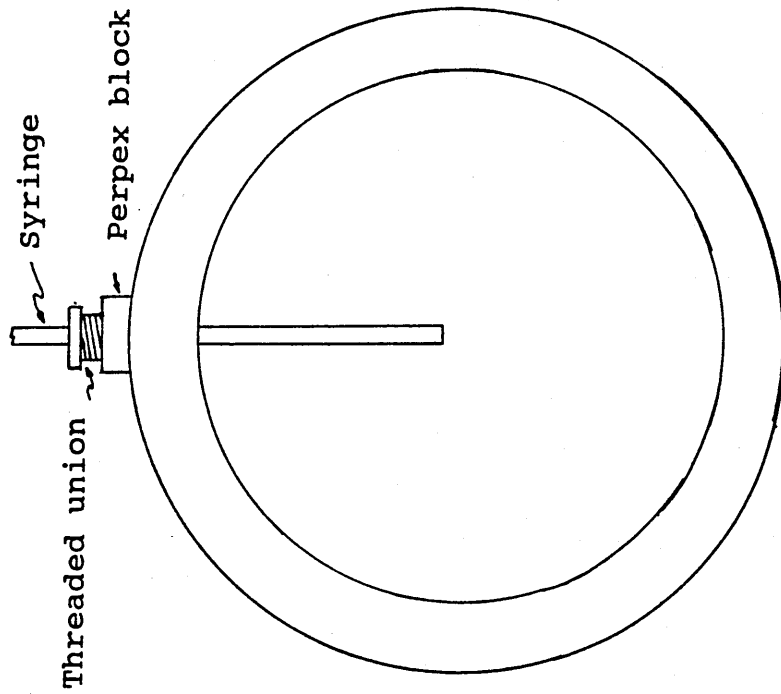


Figure 4.11 Detail of the flow visualization tapping

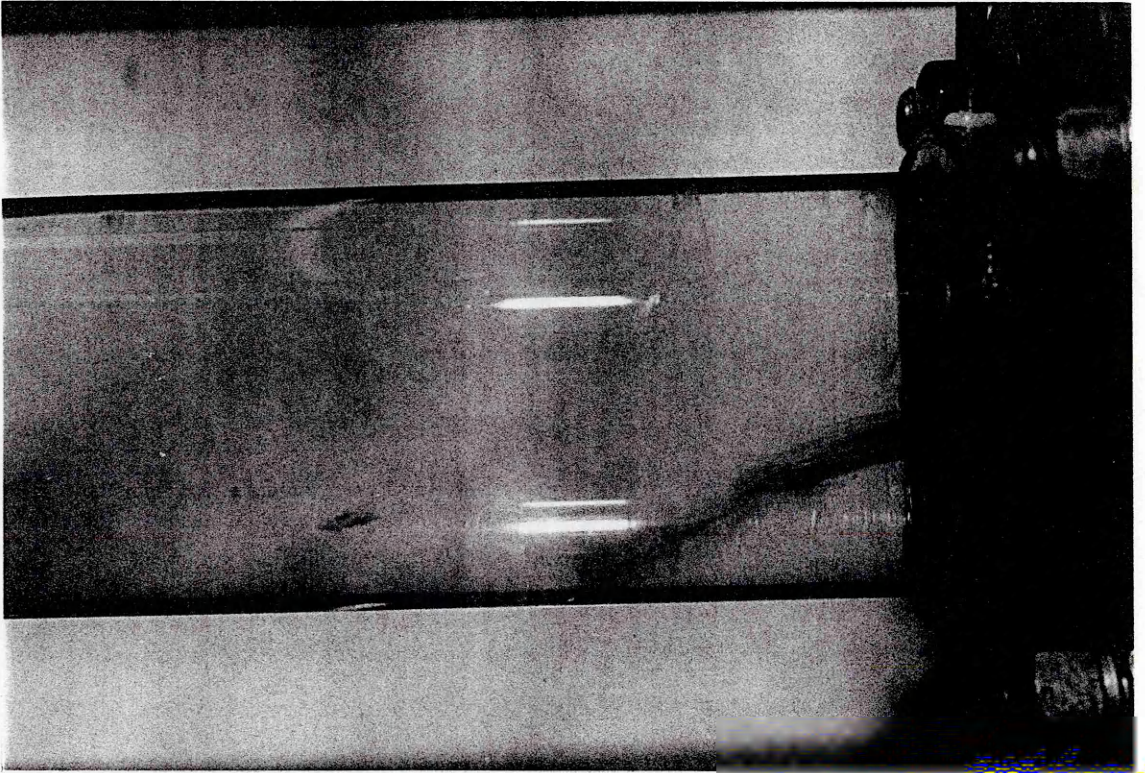


Figure 4.12 Swirling flow coming out of the orifice plate.

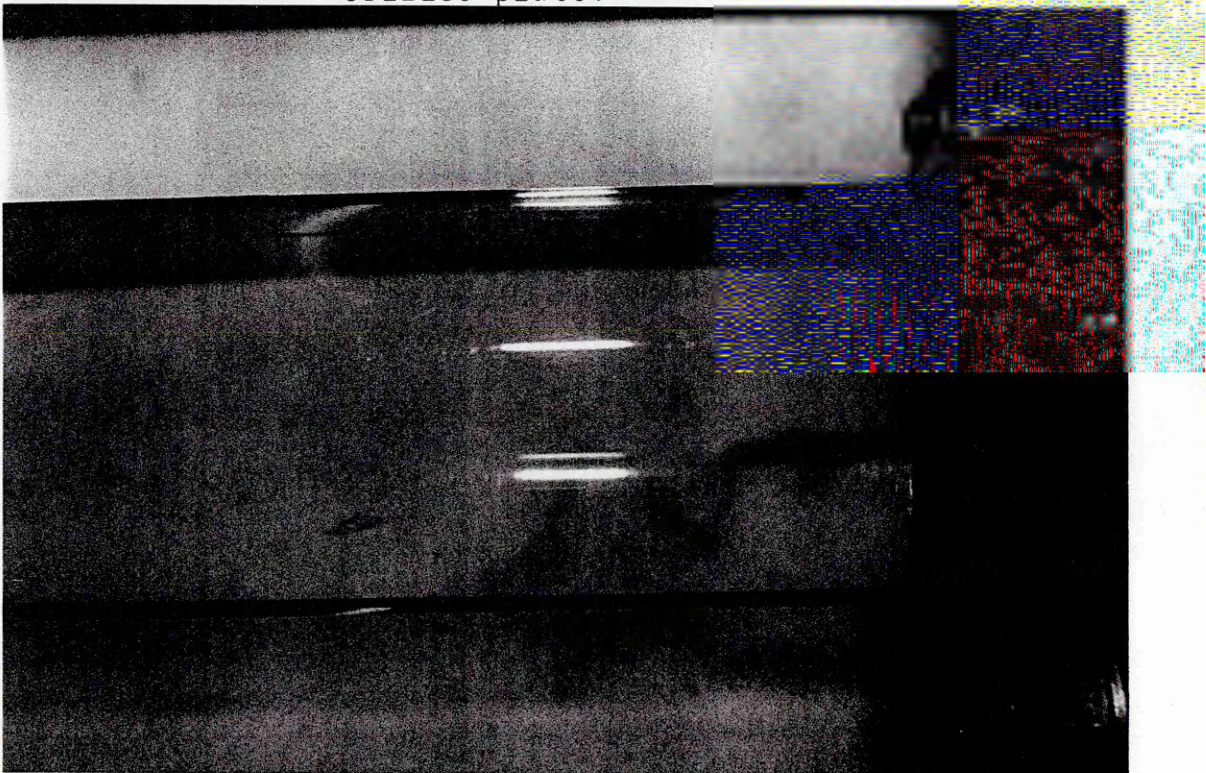


Figure 4.13 Swirling flow coming out of the orifice plate.

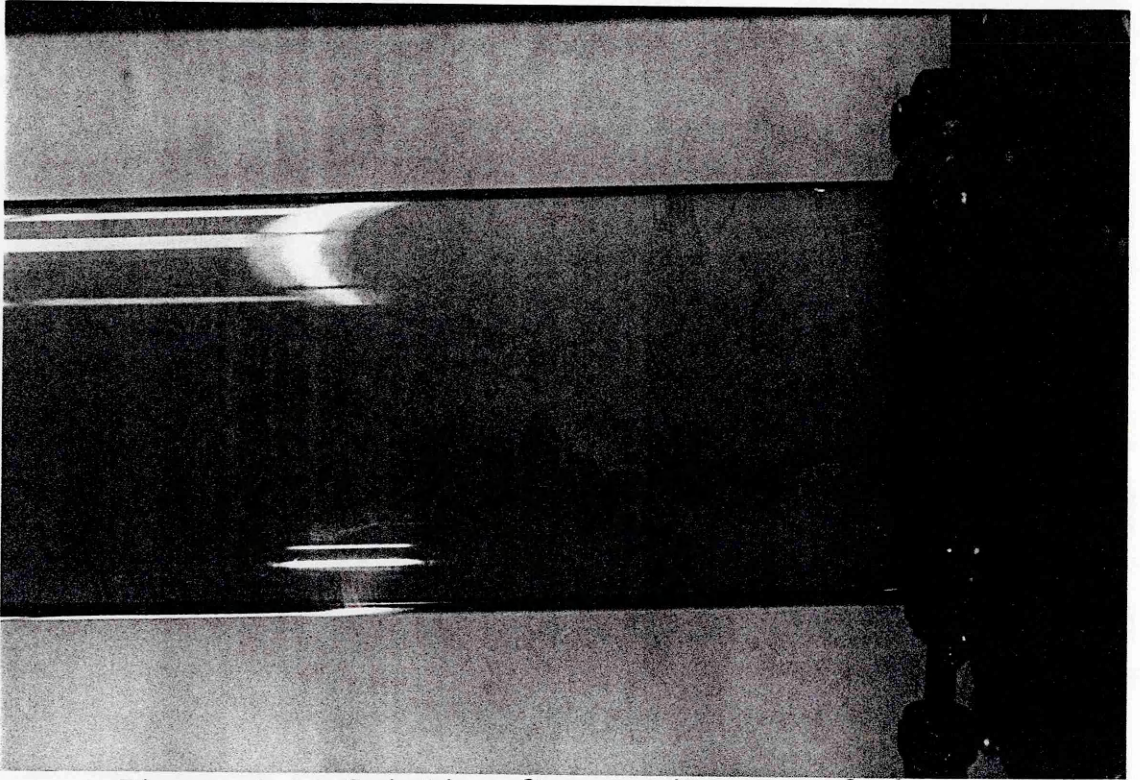


Figure 4.14 Swirling flow coming out of the orifice plate.

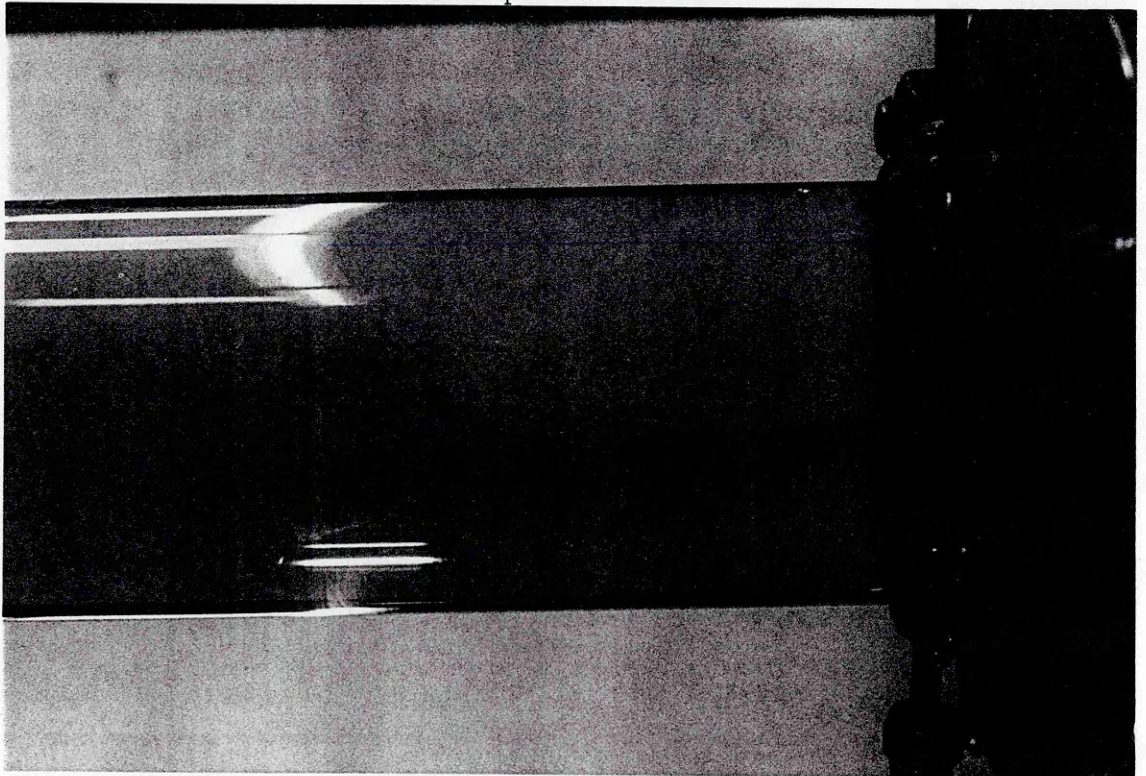


Figure 4.15 Swirling flow coming out of the orifice plate.

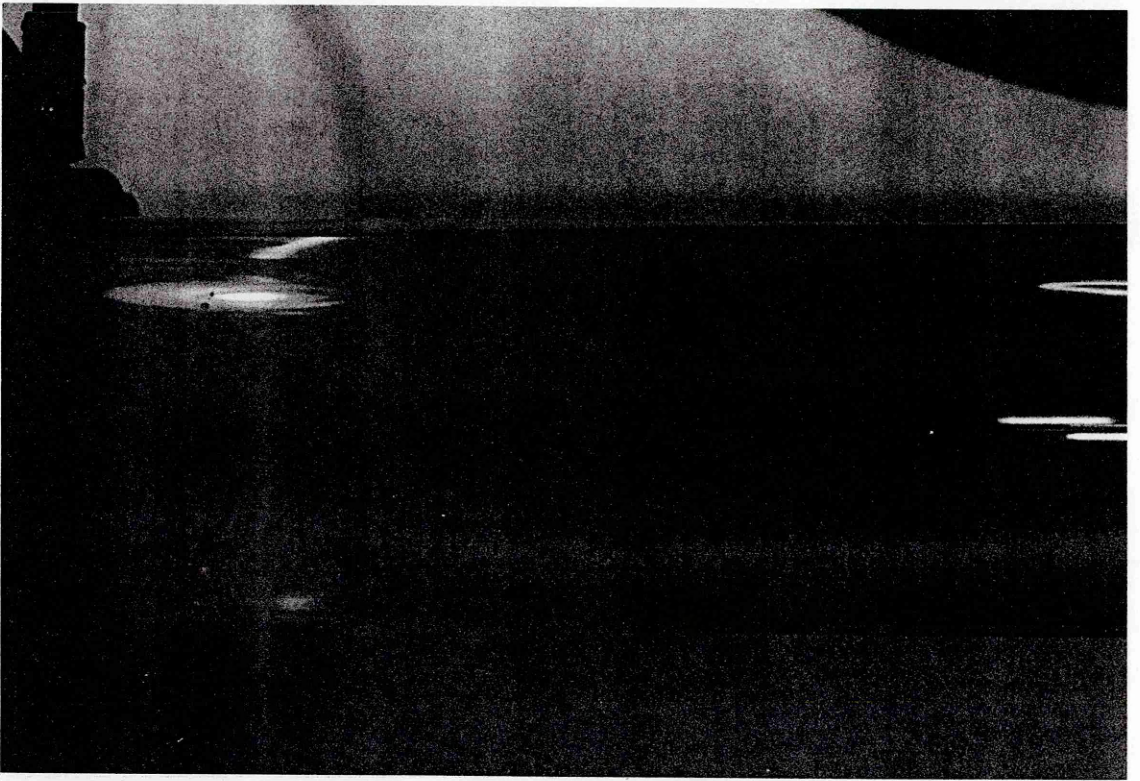


Figure 4.16 Turbulent flow coming out of the orifice plate.

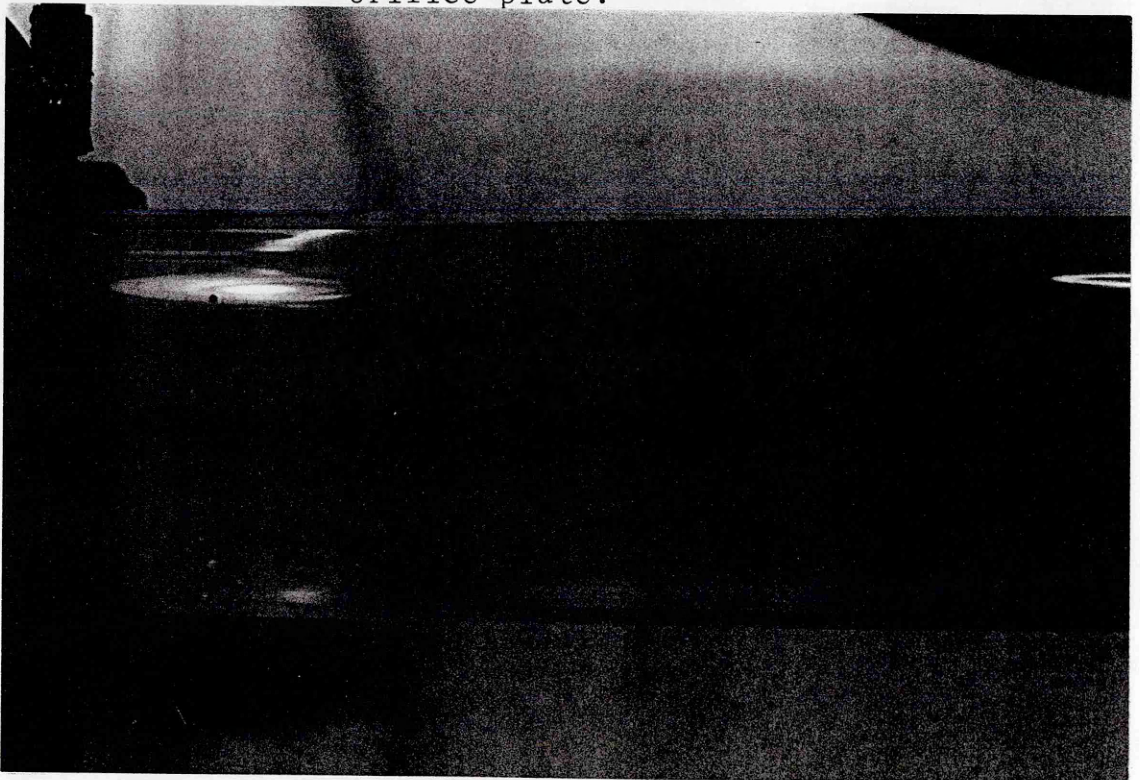


Figure 4.17 Turbulent flow coming out of the orifice plate.

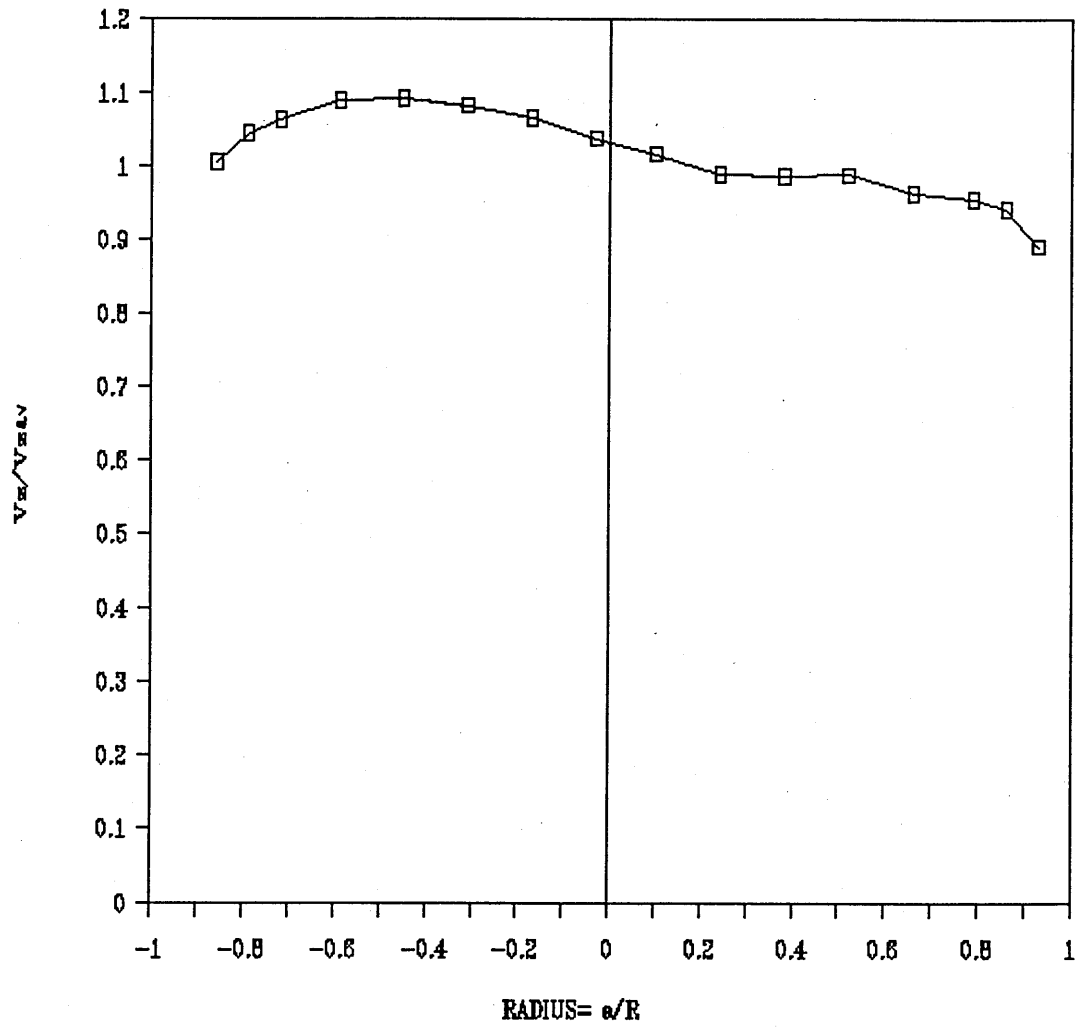


Figure 4.18 Axial velocity measurements on section 1.

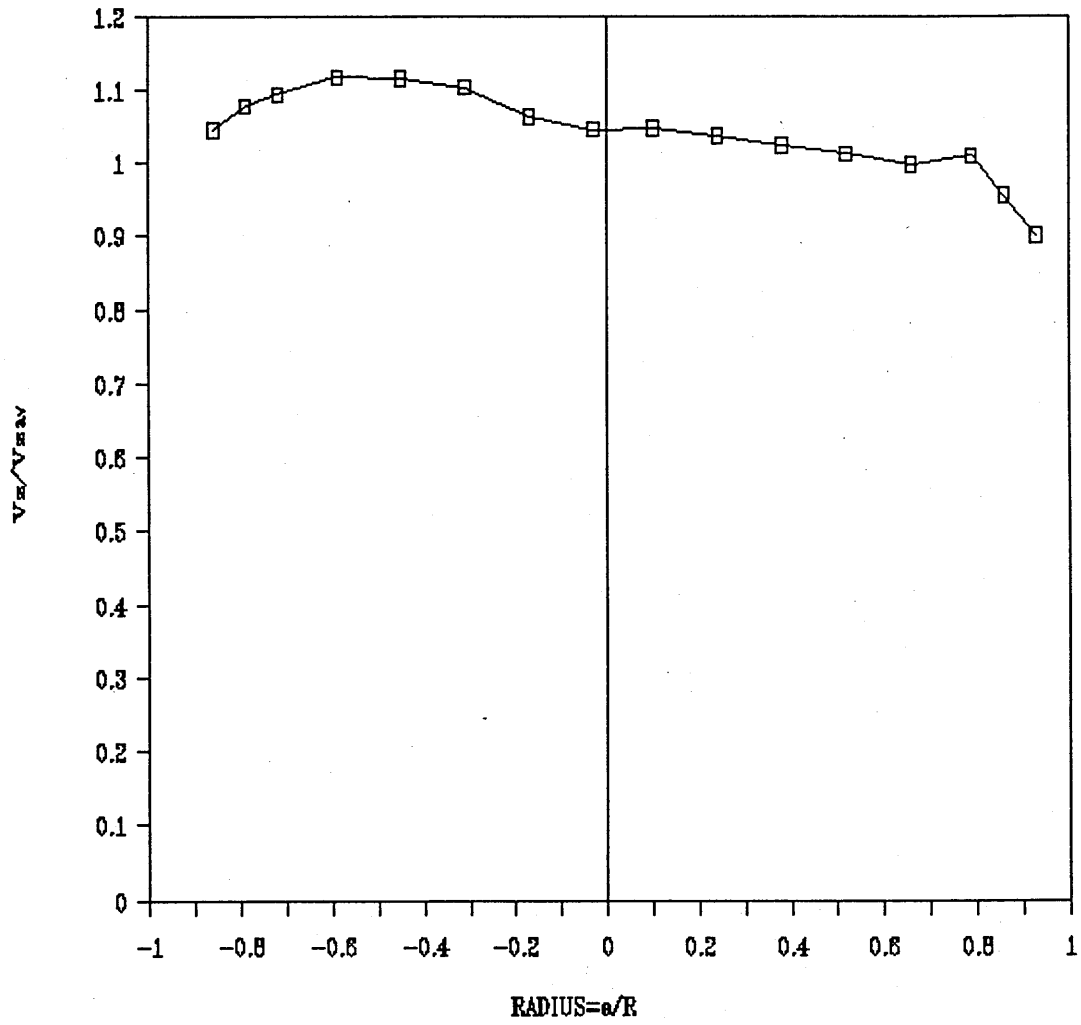


Figure 4.19 Axial velocity measurements on section 2 .

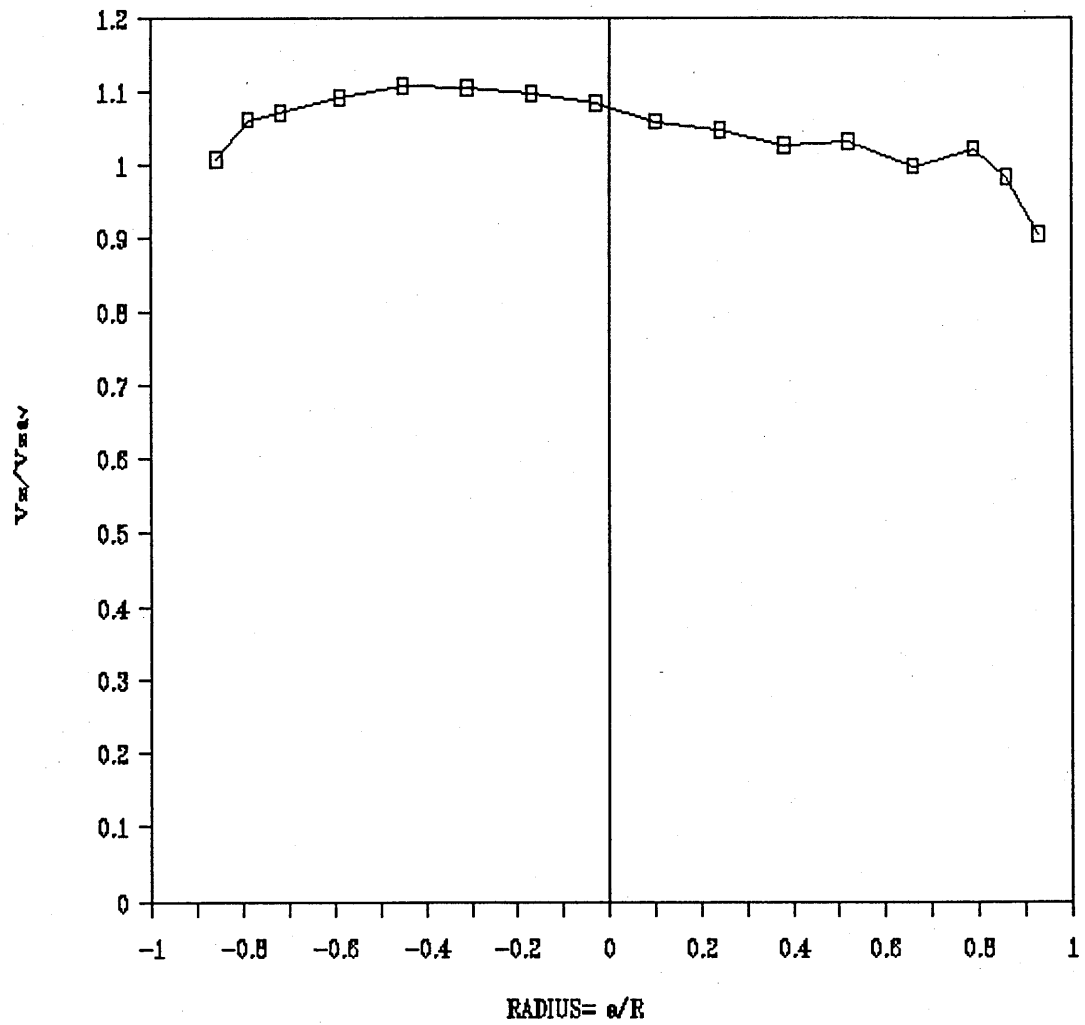


Figure 4.20 Axial velocity measurements on section 3 .

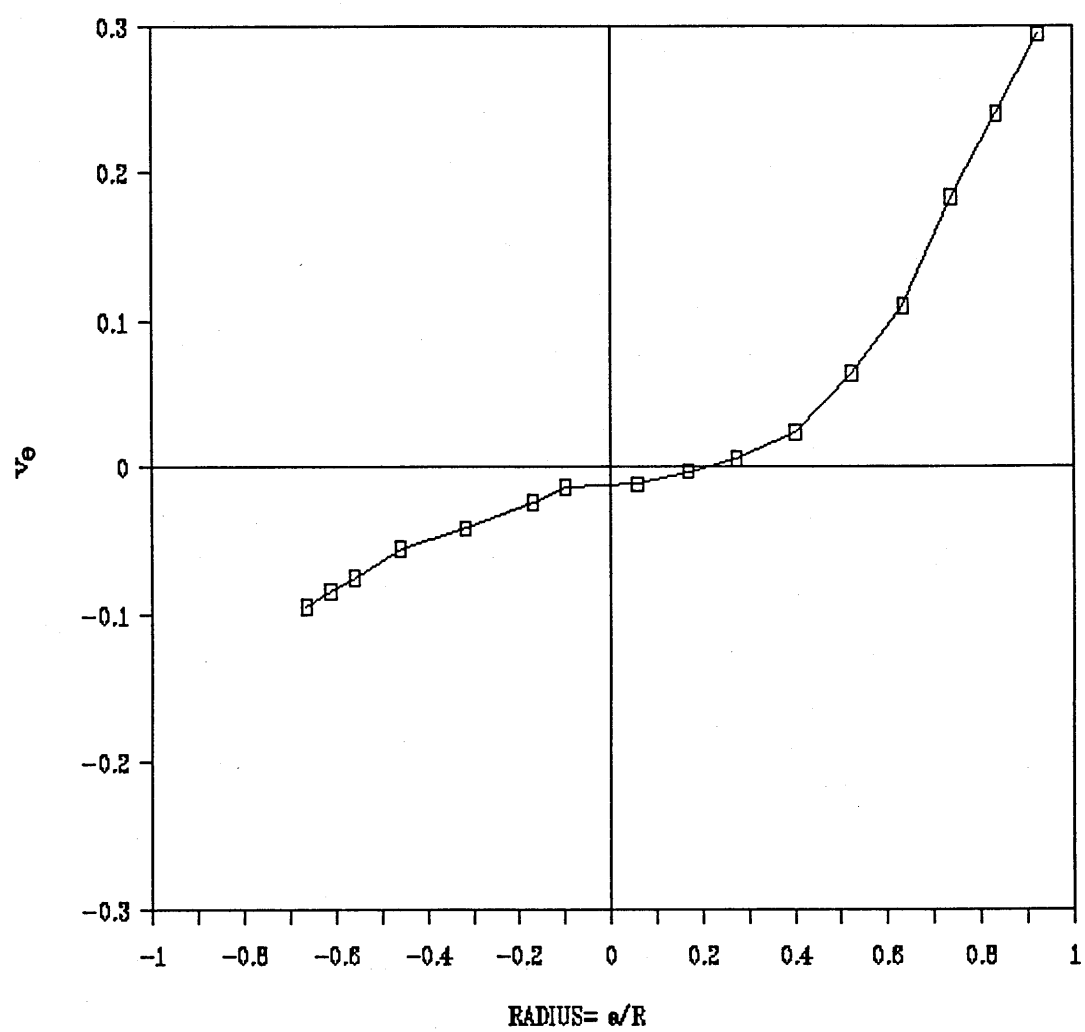


Figure 4.21 Tangential velocity measurements on section 1 .
S 0.077

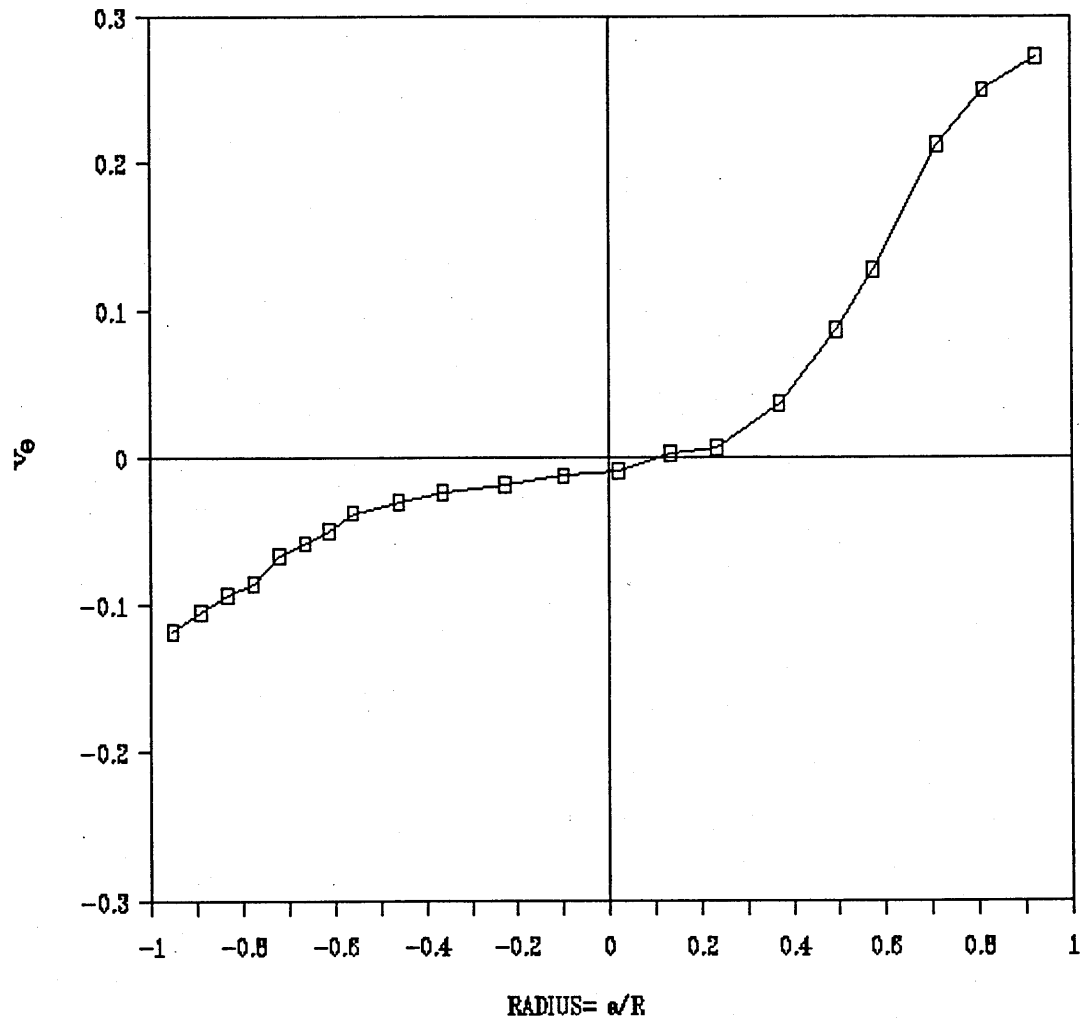


Figure 4.22 Tangential velocity measurements on section 2.

S 0.099

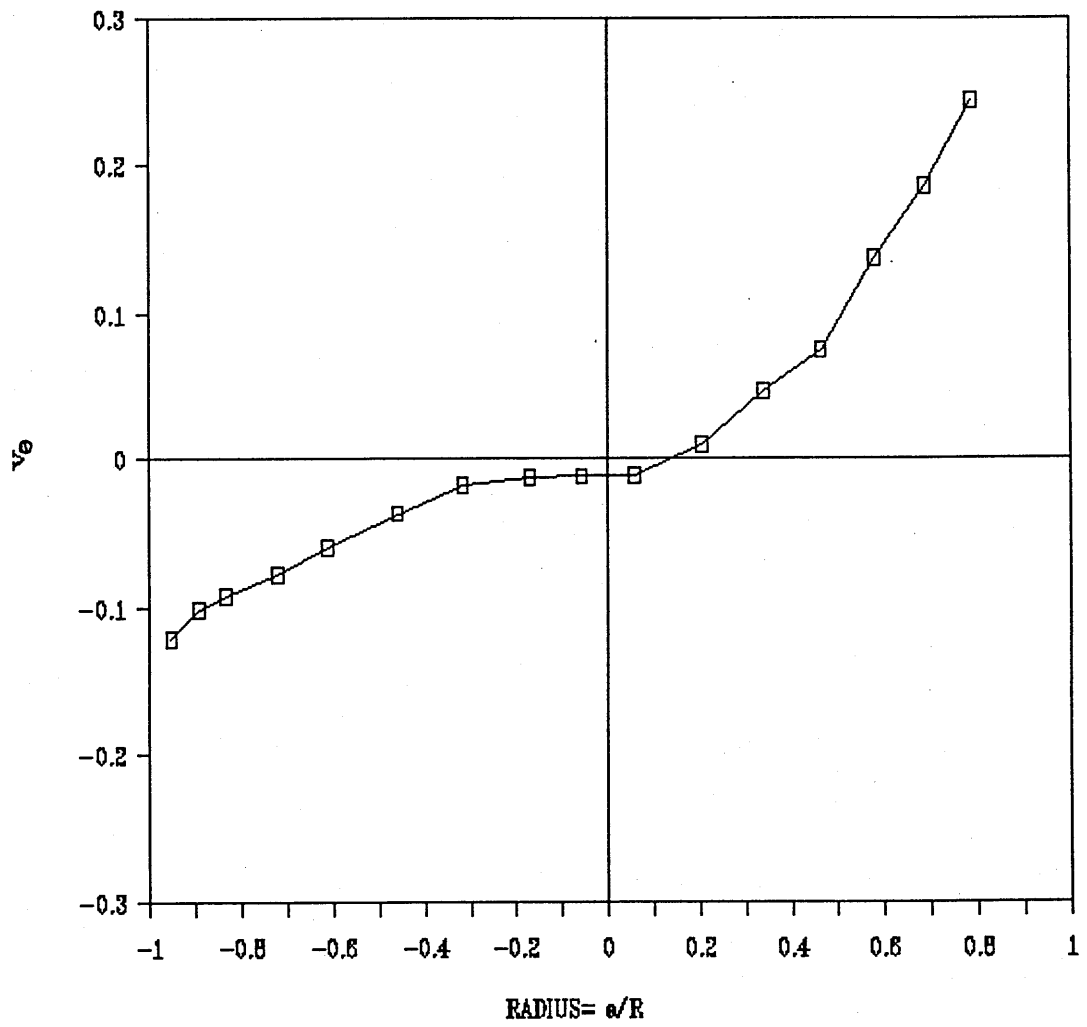


Figure 4.23 Tangential velocity measurements on section 3.

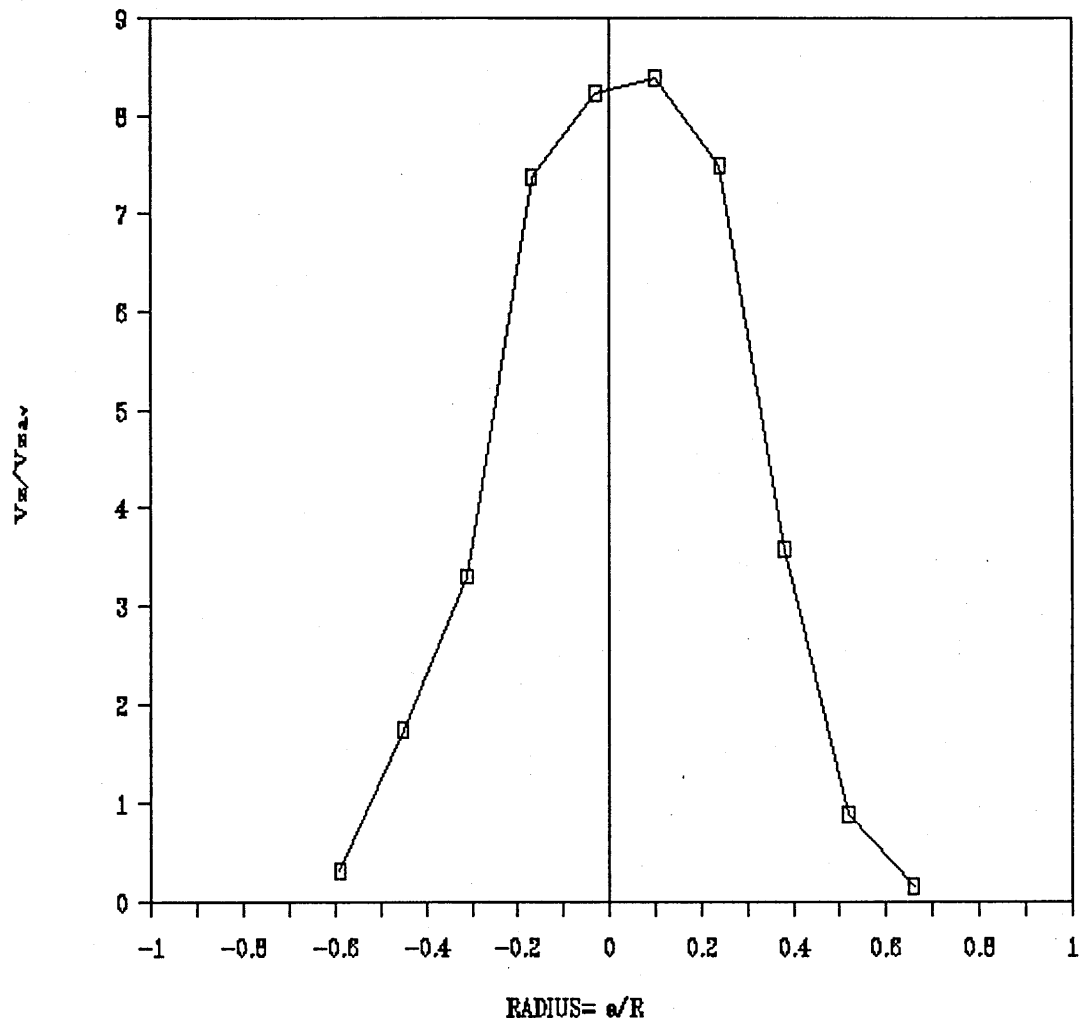


Figure 4.24 Axial velocity measurements on section 4.

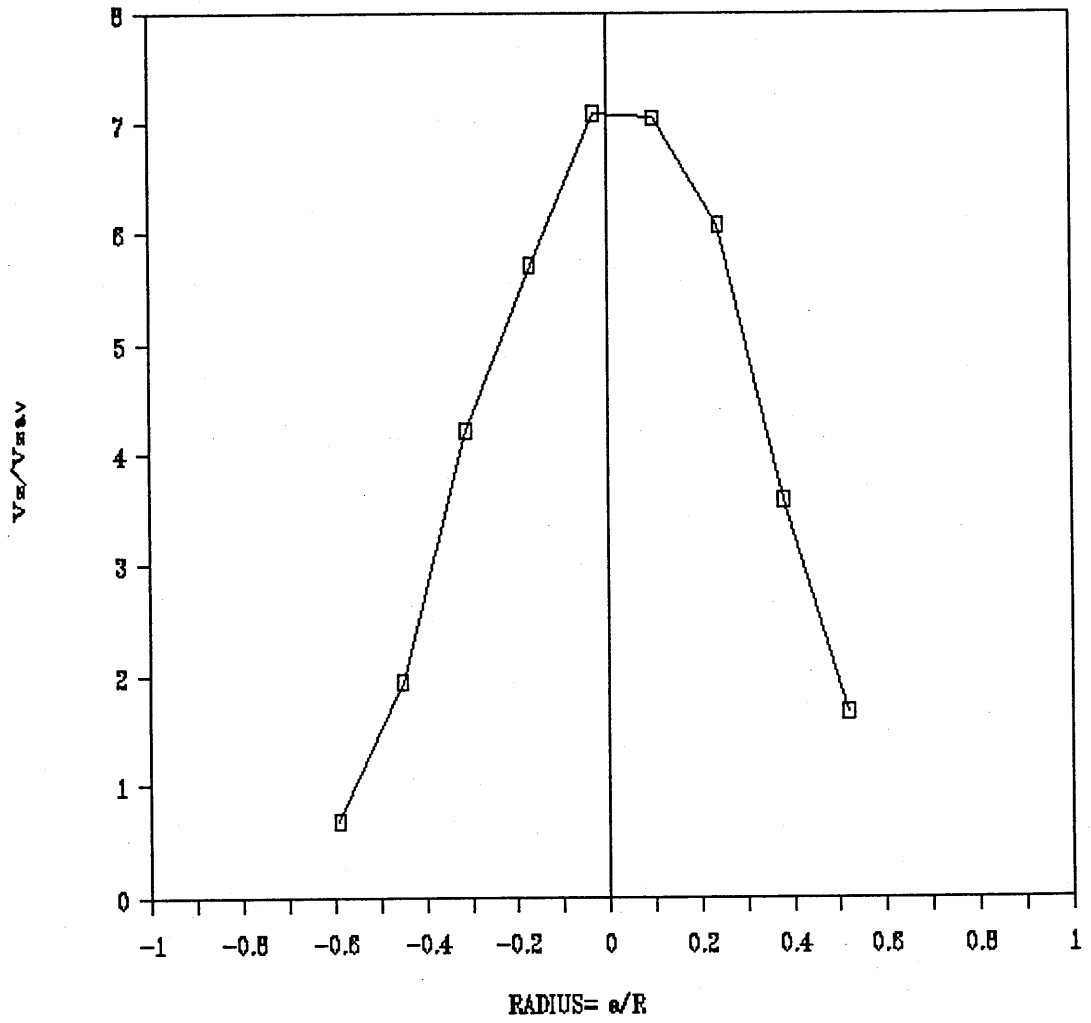


Figure 4.25 Axial velocity measurements on section 5.

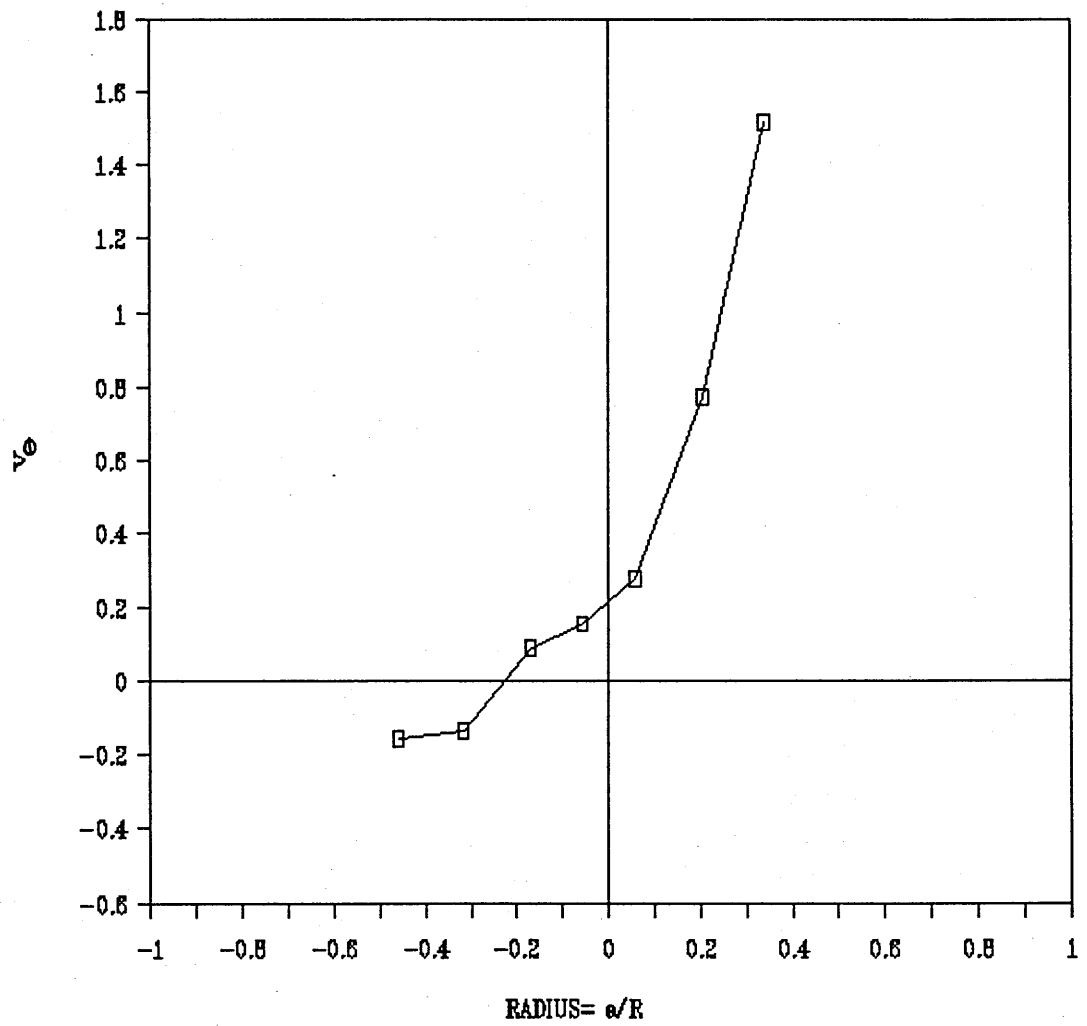


Figure 4.26 Tangential velocity measurements on section 4.

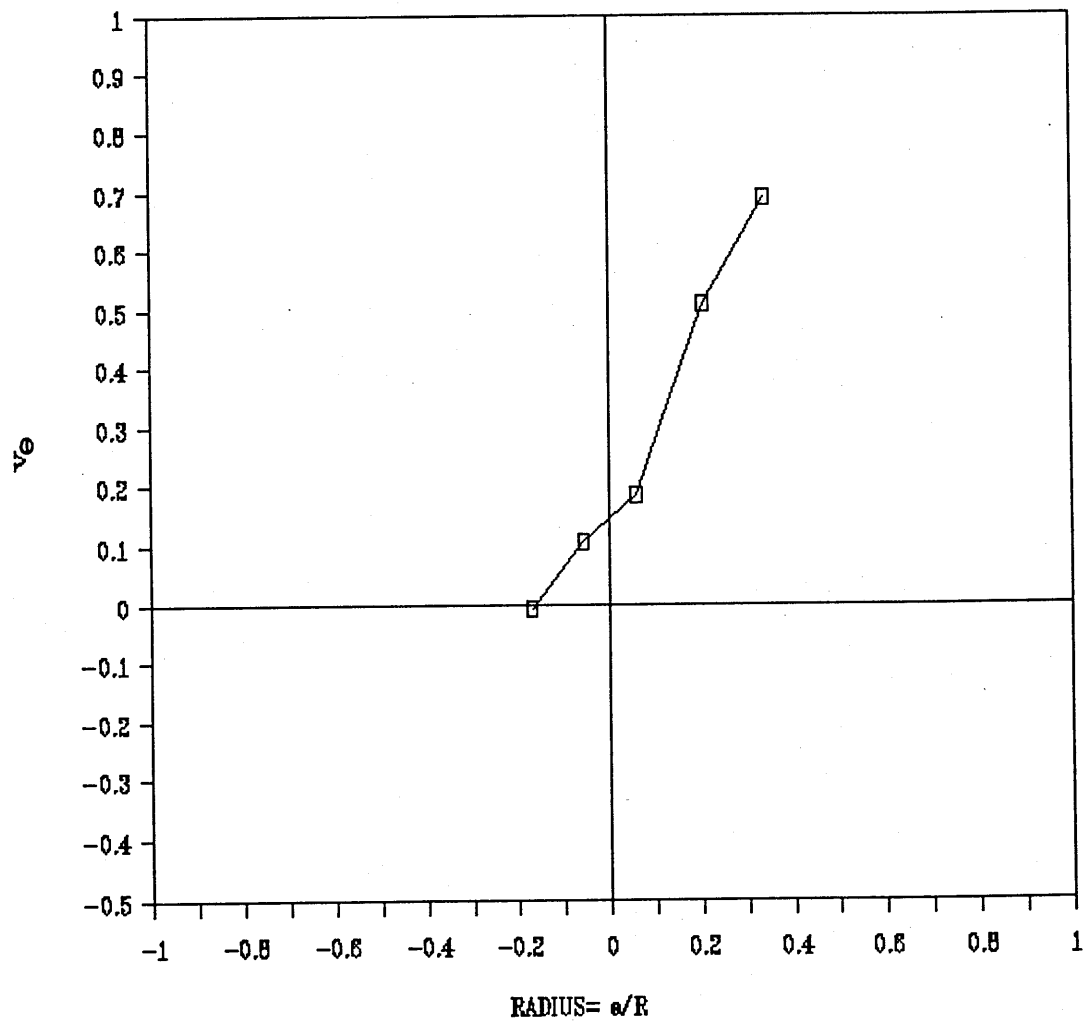


Figure 4.27 Tangential velocity measurements on section 5.

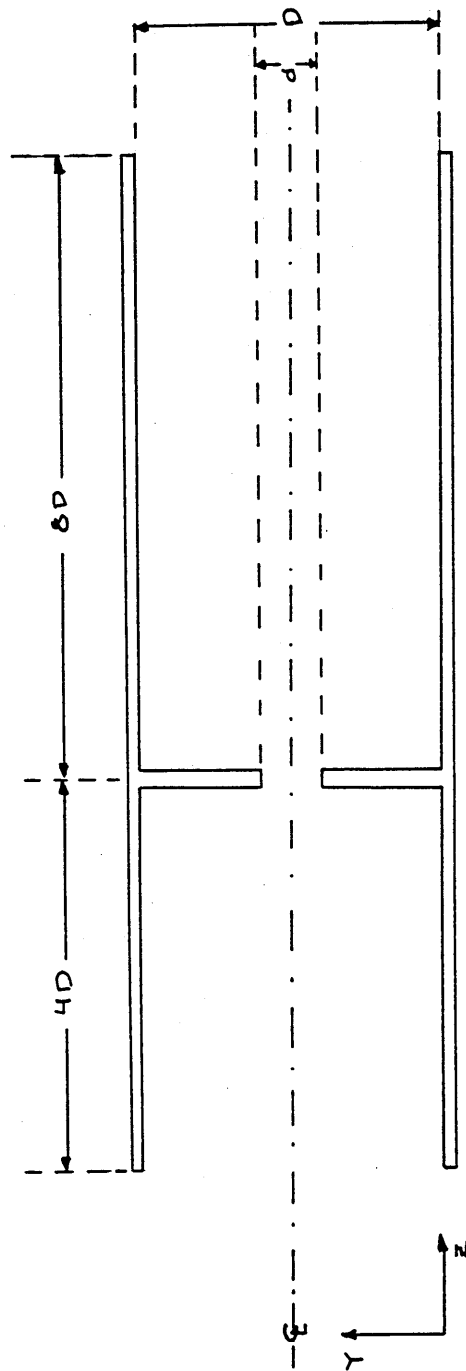


Figure 5.1 The computational test section

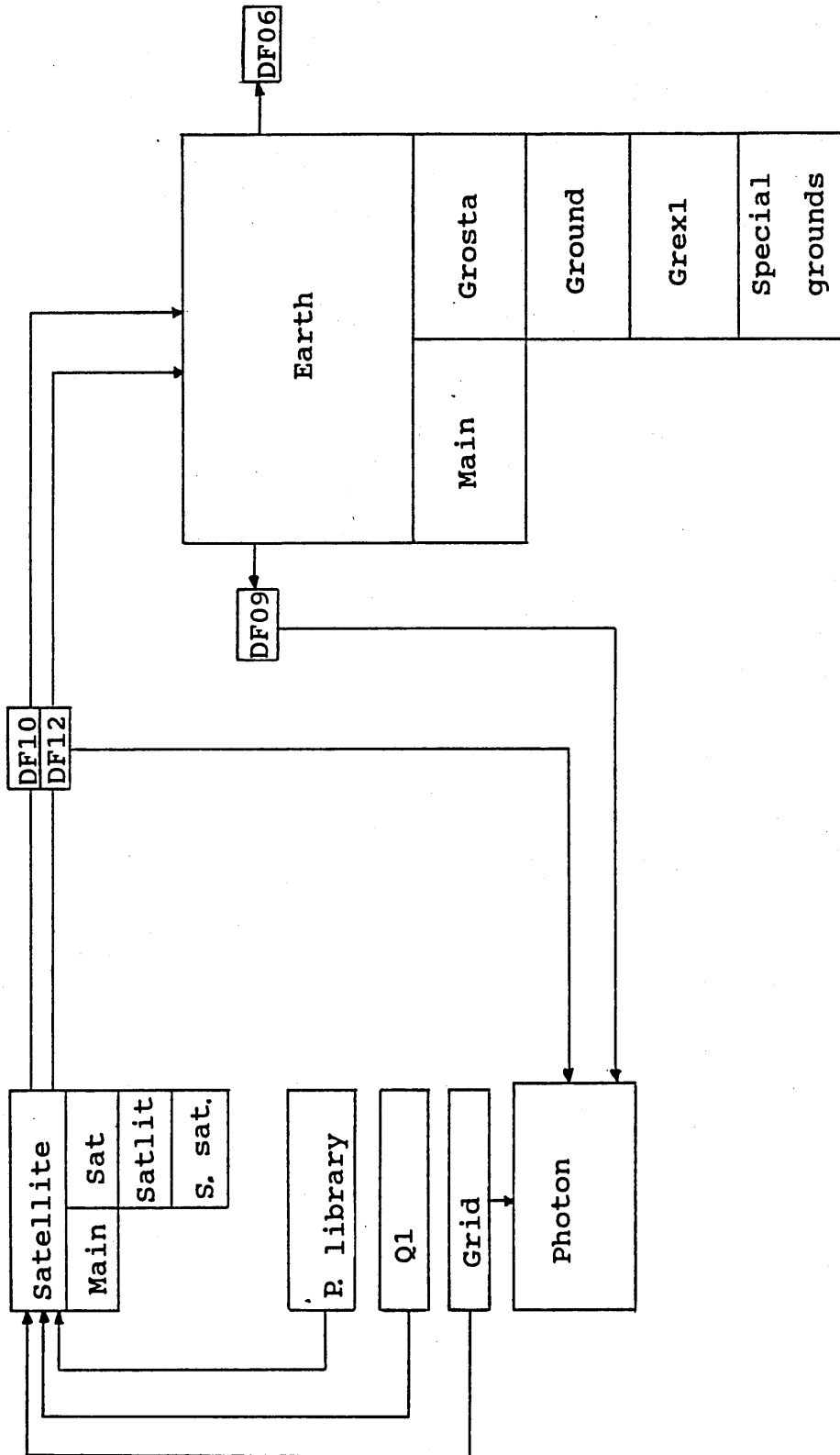
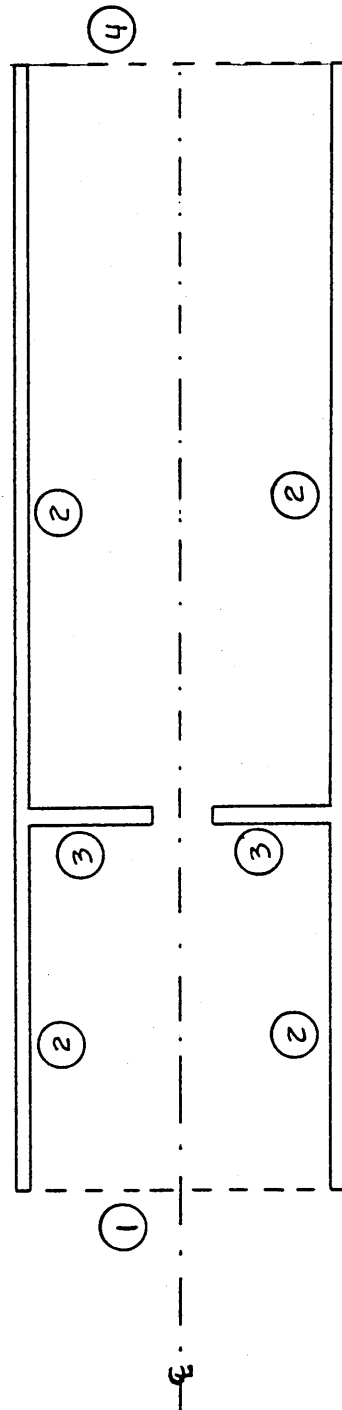


Figure 5.2 Phoenix components



- ① Inlet plane
- ② Pipe wall
- ③ Orifice plate
- ④ Outlet plane

Figure 5.3 Location of the boundary conditions

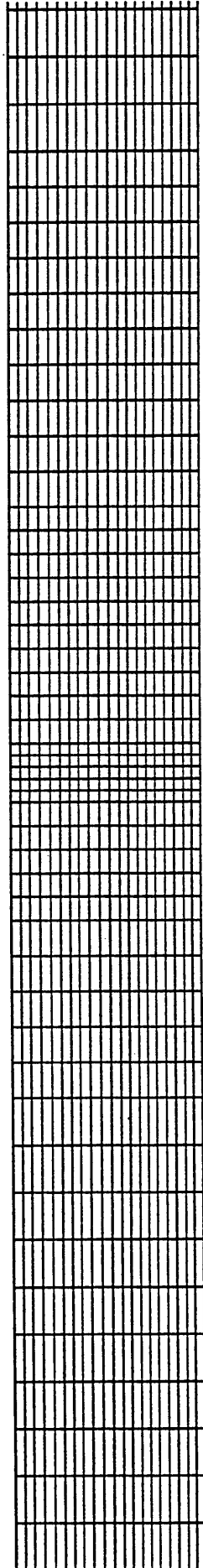


Figure 5.4 The computational grid

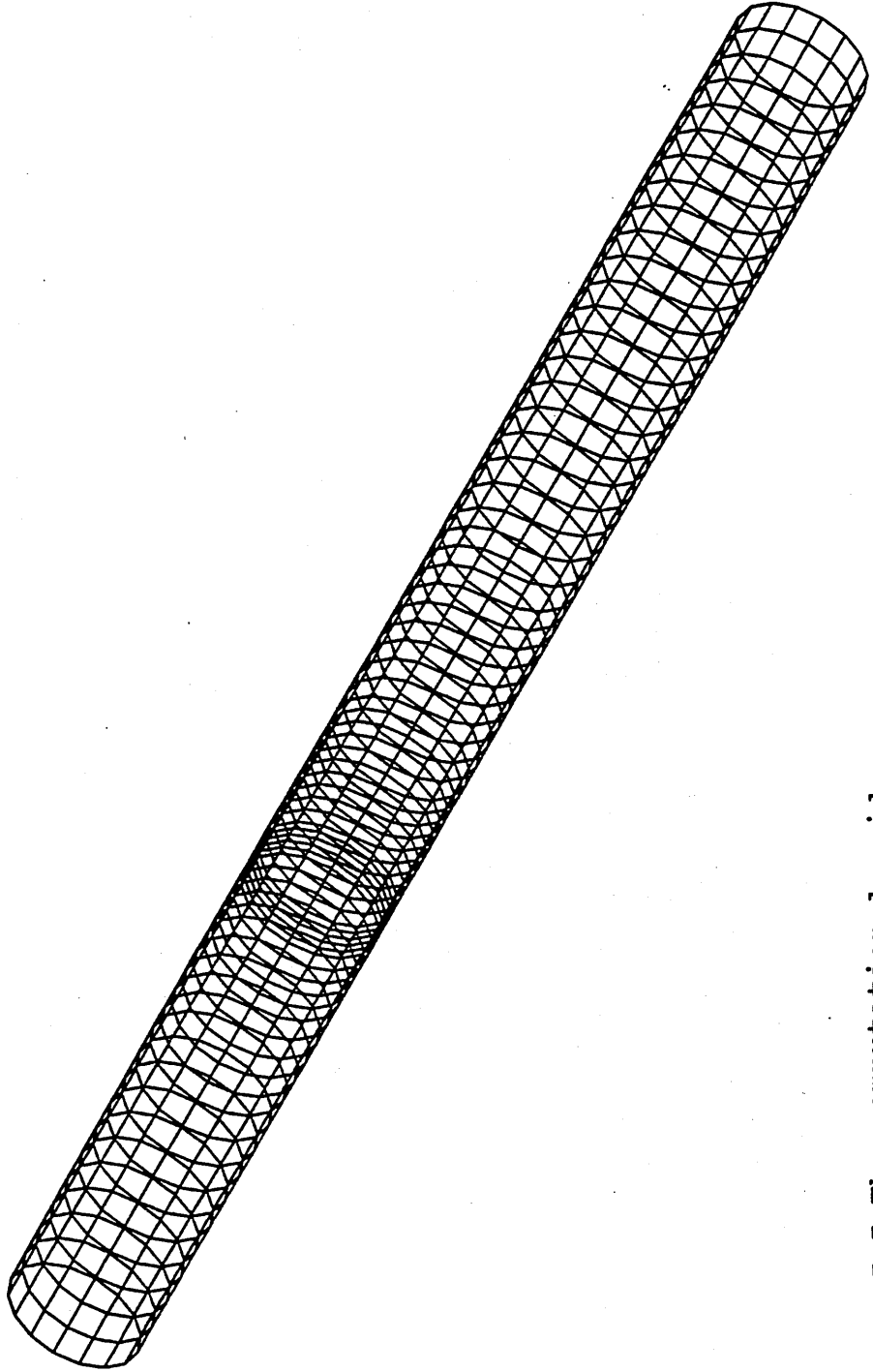
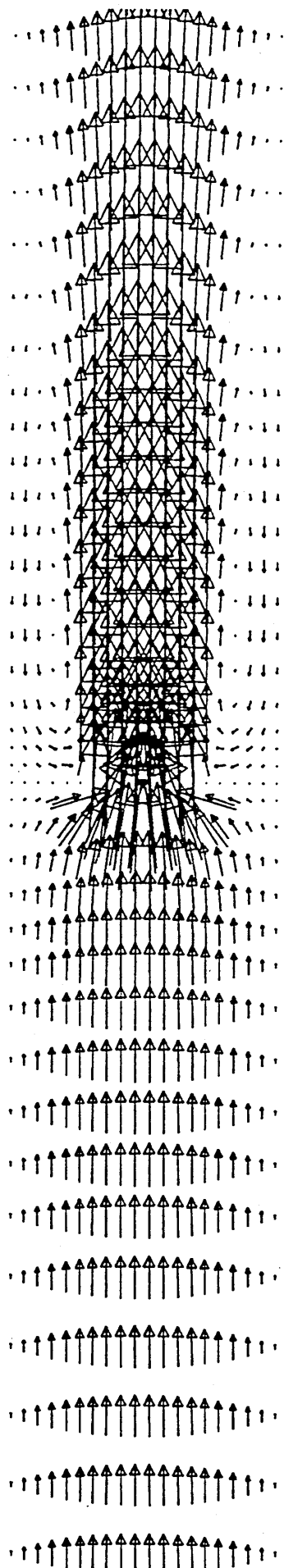


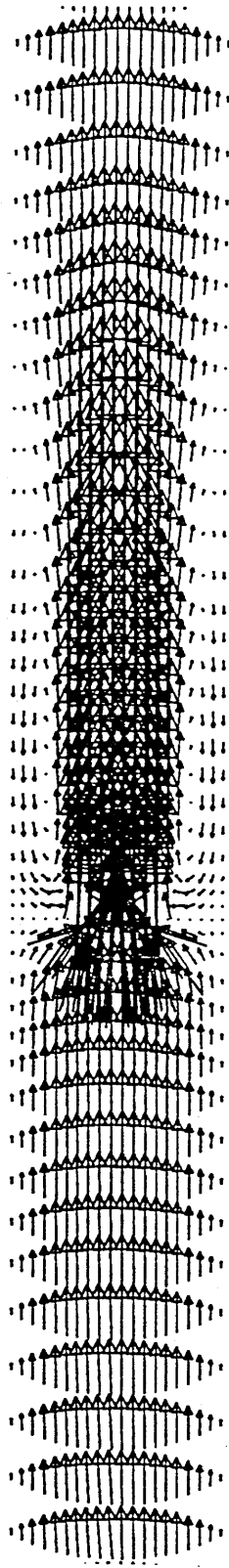
Figure 5.5 The computational grid



Vector scale: 5.42 ms⁻¹



Figure 5.6 Velocity vector diagram
(with swirl)



Vector scale: 4.87 ms/
→

Figure 5.7 Velocity vector diagram
(without swirl)