

INLET BAFFLES AND THEIR EFFECT ON THE
VELOCITY DISTRIBUTION AND TURBULENCE
IN A PIPE

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the degree of Doctor of Philosophy

by

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Abstract

The term "baffle" has been used to describe two dimensional or three dimensional obstructions to the flow with certain definable physical properties. Two dimensional baffles such as meshes or screens have been used in wind tunnels to control turbulence, to reduce variations in the mean velocity and to generate desired velocity profiles. This last aspect has been investigated only in a few instances.

The experiments were conducted in air with the baffles fixed at the inlet to a pipe with only a smooth bellmouth entrance without any elaborate upstream arrangements. A three dimensional axisymmetric baffle was constructed with concentric cylinders of perforated sheet and the lengths of the cylinders adjusted such that the fully developed flow was obtained in a short distance. This has been called the "concave friction baffle".

The pressure and velocity variations within the baffle, the behaviour along the pipe of the velocity profiles generated and the pressure drop along the pipe have been investigated. The intensity of turbulence was measured immediately behind the baffle and further downstream and compared with those obtained by other investigators. An attempt has been made to derive a design formula for the concave friction baffle.

Velocity profiles generated by other arrangements of perforated cylinders were also investigated and those produced by perforated plates of uniform and varying resistance, meshes and honeycombs, and their behaviours compared.

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CHAPTER 1

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1.1.

HISTORICAL REVIEW OF THE PROBLEM OF PIPE FLOW

Since the classical experiments of Hagen (1839) flow in pipes has been the subject of numerous investigations covering most aspects of practical importance. For the hydraulic engineer the problem has been satisfactorily solved, by empirical or semi empirical methods gradually evolved over a period of more than a century, and it is now possible to predict reliably the energy losses in most classes of practical pipes. Nevertheless, what is yet lacking is a realistic theory based on a correct understanding of the process of turbulent energy decay. This however seems a long way off, though Schlichting (1950) may have been over pessimistic when he wrote "It is very likely that science will never achieve a complete understanding of the mechanism of turbulence because of its extremely complicated nature".

Hagen's (1794 - 1884) (Ref 1) paper of 1839 contains an important formula for the pressure drop along a pipe. It was of the form

$$h = \frac{1}{r^4} (A L Q + B Q^2)$$

h - drop of pressure head

r - radius

L - length

Q - quantity of flow

A & B - Empirical constants

About the same time Poiseuille (1799 - 1869) experimenting in Paris gave a formula for the pressure loss along a pipe.

Hagen, in a paper published in 1854, described the change in the state of the water as it flowed in a pipe, observed and recorded what is now termed laminar and turbulent flow. This is, however, often attributed to Reynolds, who, working years later, observed the same motion in the course of establishing his flow criterion. The parabolic velocity distribution for laminar flow, called Poiseuille flow, was derived independently by Neuman and Hagenbach between 1858 and 1860.

Weisbach (1806 - 1871) of Freiberg expressed the head loss formula as it is used in its present form.

$$h = f \frac{L \bar{u}^2}{d \cdot 2g}$$

h - head loss
 L - length
 d - diameter of pipe
 $\bar{u}$ - average velocity
 g - acceleration due to gravity
 f - friction factor.

Darcy (1803 - 1858) experimenting with various types of pipes observed a change in the resistance to the flow with different materials. Osborne Reynolds (1842 - 1912) used colour bands to demonstrate the existence of the two states of flow and formulated his dimensionless parameter to define the flow. This parameter is now called Reynolds number. He also made considerable theoretical progress when he inserted mean and fluctuating components in the Navier Stokes equation of motion and obtained in the equation additional stress components due to the turbulent fluctuations. These are sometimes referred to as the 'Reynolds Stress'.

Attempts to express the turbulent stress in terms of the mean velocity resulted in important developments in the study of turbulent pipe flow. In 1897 Boussinesq published a far reaching expression in the form

$$\tau = \rho \epsilon \frac{\partial u}{\partial y}$$

τ - shear stress
 ρ - density
 $\frac{\partial u}{\partial y}$ - the velocity gradient
 ϵ - Eddy Viscosity

Prandtl in 1925 made great progress in the study of pipe flow and fluid mechanics in general when he formulated his momentum transfer theory, which expressed the eddy viscosity in terms of the velocity gradient and a mixing length. This led to the logarithmic velocity distribution in turbulent boundary layers which agrees best with experimental data obtained in pipes. About the same time Taylor formulated his eddy viscosity theory and Von Karman the similarity hypothesis, both of which gave logarithmic forms for the velocity distribution. These are generally referred to as 'phenomenological theories' which depend on empirical constants for obtaining the final expressions. Prandtl made far reaching assumptions, but quoting Rouse (1958) "When the simplicity of the relationship is compared to the complexity of the phenomenon with which they deal it is the extent of their usefulness rather than of their inadequacy that is surprising". They have been very useful indeed. This was a meeting of the mathematician and the empirical investigator that has resulted in a more profitable approach to the study of fluid mechanics.

It is now possible to predict velocity distributions and pressure losses for fully developed conditions in both smooth and rough pipes. In 1911 Blasius analysed the existing data and expressed the friction factor as a power function of the Reynolds number. Nikuradse (1933) worked extensively on sand roughened pipes and obtained friction factors in terms of an average roughness parameter. The Colebrook-White formula for the friction factor in the transition region from smooth turbulence to rough turbulence effectively completed the study.

More recently it was suggested by Morris (1954) that the spacing of roughness is the parameter of primary importance and that it so happens with sand and commercial roughness that the spacing is equal to the roughness height. Data on flows with different forms and types of roughness is not sufficiently comprehensive to substantiate this.

The inadequacy of the phenomenological theories to explain the behaviour of different types of turbulence and turbulent flows made investigators search for a more realistic solution to the problem. The random nature of the flow suggested the possibility of a 'statistical' approach and in 1935 Taylor published four important papers which contained some of the fundamental ideas on which the present day study of turbulence is based. He derived a simplified expression for the decay of homogeneous isotropic turbulence.

$\bar{W}$ - rate of energy dissipation.

$$\frac{\bar{W}}{\mu} = 7.5 \left(\frac{\partial u'}{\partial y} \right)^2$$

μ - viscosity

u' - the turbulent fluctuation.

Where isotropy specifies a condition of turbulence where the rotation or change in the reference axes do not change the average value of any function of the velocity fluctuations. He also derived a relationship between the correlation of the turbulent intensities at two points and the scale of turbulence.

However, the statistical theory has not yet produced results to replace the older theories for pipe flow or other types of shear flow. Laufer in 1953 made a great contribution to the study of turbulence in fully developed pipe flow. His investigations of the different fluctuating components, the correlations, energy balance and energy spectra has greatly improved the understanding of the mechanism of energy decay in pipes.

Nevertheless the processes by which turbulence gains or transfers energy are not clear. Neither is the interaction between the mean and fluctuating motions nor how it is possible to impose a particular turbulence structure by imposing particular velocity distributions.

Studies in the inlet region are few or inconclusive. There is little agreement between the theoretical predictions for the inlet length required for the fully developed flow to be established and those obtained experimentally. Different investigators have given widely varying values with no definite variation with Reynolds number or inlet conditions. Extreme values of 25 to 100 diameters have been observed. Inlet conditions are known to influence the inlet length and it is generally assumed that a greater intensity of turbulence initially causes a reduction in the inlet length. Some aspects of this problem have been investigated in the present study.

Pipe flow has yet many problems which will keep investigators busy for years to come.

1.2.

SOME RELEVANT PAST WORK

1.2.1. Flow in the inlet region.

Theoretical investigations in this region were made by Latzko (Ref 2) and Ross (Ref 3). Latzko analysed the development of the flow for turbulent conditions by assuming that the velocity distribution was composed of a uniform central core and a $1/7^{\text{th}}$ power function at the boundary. The expression obtained for the total inlet length was

$$\frac{L}{d} = 0.69 Re^{0.25}$$

L - the inlet length

d - pipe diameter

 Re - Reynolds number

$$\frac{L}{d} = 6.9 \quad \text{at} \quad Re = 10^4$$

$$= 21.7 \quad \quad \quad = 10^6$$

These values are very much less than those obtained experimentally.

Ross gave an analytical solution for the development of the turbulent boundary layer in smooth pipes, by using the continuity, momentum and energy equations. A $1/7^{\text{th}}$ power function was not assumed as this restricts the validity of the theory for a particular range of Reynolds number. The expression obtained was

$$\frac{\Theta}{r} = \frac{0.043}{Re^{1/6}} \left(\frac{x - x_0}{2r} \right)^{5/6} \left[1 - \frac{(2+H)H \cdot \Theta}{r} \right]$$

 Θ - momentum thickness

r - pipe radius

H - shape parameter

 Re - Reynolds number x_0 - effective start of turbulent boundary layer.

The above expression is considered to be valid for a distance within 10 diameters from the inlet.

He also derived an expression for the pressure drop in the inlet region

$$\frac{p_0 - p}{\frac{1}{2} \rho \bar{u}^2} \approx \frac{0.17}{Re^{1/6}} \left(\frac{x - x_0}{2\tau} \right)^{5/6} \left[1 + \frac{1}{0.875 \log_{10} (Re \frac{x - x_0}{2\tau}) - 2.75} \right]$$

p - pressure

ρ - density

$\bar{u}$ - average velocity

This pressure drop expressed as a ratio of the fully developed pressure drop gives values of 2.5 at the inlet and 1.5, 10 diameters from the inlet for Reynolds numbers of 10^6 . Ross claims that there is good agreement with the very little available experimental data.

The experiments of Kirsten and Schiller (Ref 4) show that with smooth inlet conditions the velocity distribution at the wall can be expressed as a $\frac{1}{7}$ th power function, but not in the initial length of the inlet region, for values of Reynolds number between 10^4 and 5×10^4 . Other workers have, however, found that the $\frac{1}{7}$ th power function does apply for Reynolds numbers up to 20×10^4 and Schiller considered it safe to allow an inlet length of 100 diameters. Schiller also showed that where the boundary layer is wholly turbulent the inlet length increased with Reynolds number but with smooth inflow where a laminar boundary layer exists in the initial length the inlet length decreased with Reynolds number.

Kirsten (Ref 5) measured velocity profiles along a pipe and obtained values for the inlet length varying from 50 to 100 diameters for Reynolds numbers increasing from 20×10^3 to 80×10^3 . But according to Nikuradse the fully developed profile already exists between 25 and 40 diameters. Shapiro and Smith (Ref 6), experimenting with tubes of $\frac{3}{8}$ " to 4" diameter, found that the inlet length was nearly three times as long as that given by Latzko's formula.

Cox and Baugh (Ref 7),experimenting in a rectangular conduit,found that an "unstimulated" inlet length of 50 diameters was necessary and that roughening the inlet stimulated the growth of the boundary layer and reduced the inlet length to 40 diameters.

Shapiro and Smith (Ref 6) gave values for the apparent friction factor, which includes the pressure drop due to increase in the kinetic energy, that are within 10% of the value for fully developed flow, at about 20 - 30 diameters, and within 5% at about 40 - 50 diameters.

A summary of the above mentioned works would indicate that extreme values of 25 and 100 diameters have been observed for the inlet length. A value of about 40 diameters is often accepted as reasonable.

1.2.2. Fully developed flow.

Bakhmeteff in his lectures on the 'Mechanics of Turbulent flow', which were later published in 1936, made a comprehensive analysis of Prandtl's momentum transfer theory for the universal logarithmic velocity distribution in turbulent boundary layers. He explained its scope, limitations and assumptions and its agreement with the existing experimental data for pipe flow. These are now well known and experimental data for fully developed pipe flow is numerous. Von Karman's more rigid analysis agrees less than Prandtl's with experimental data. At the centre the actual profiles are found to deviate slightly from the theoretical curve.

Given below is the analysis following Prandtl.

τ_{xy} - Turbulent shear stress on the $\delta x \delta y$ face of a fluid element.

$$\tau_{xy} = -\rho \overline{u'v'}$$

u' - Turbulent fluctuation in x direction

v' - Turbulent fluctuation in y direction

Positive u' is on the average associated with negative v' hence the negative sign in the above expression.

$$|u'| = \ell' \left| \frac{du}{dy} \right| = \text{Constant} \times |v'| \quad \ell' - \text{a mixing length}$$

$$\overline{u'v'} = -k |u'| |v'|$$

u - mean velocity (local)

$$\overline{u'v'} = -\text{Constant} \times k \ell'^2 \left(\frac{du}{dy} \right)^2 \quad k - \text{correlation factor}$$

including k in the mixing length and writing $\text{Constant} \times k \ell'^2 = \ell^2$

and $\tau_{xy} = \tau$

$$\tau = \rho \ell^2 \left(\frac{du}{dy} \right)^2$$

If ℓ is proportional to y

y - distance from the wall

$$\tau = \rho C^2 y^2 \left(\frac{du}{dy} \right)^2$$

C - constant

Writing $\tau = \tau_0$ the wall shear stress

$$\text{and } \sqrt{\frac{\tau_0}{\rho}} = V^*$$

V^* - friction velocity

$$V^{*2} = C^2 y^2 \left(\frac{du}{dy} \right)^2$$

into grating this expression

$$u = \frac{V^*}{c} \ln y + B$$

When $u = 0$ say $y = y_0$

$$\therefore u = \frac{V^*}{c} [\ln y - \ln y_0]$$

N - Kinematic viscosity

B, β' - Constant

Writing $y_0 = \beta' \frac{N}{V^*}$

($\frac{N}{V^*}$ has the dimensions of length)

$$\therefore \frac{u}{V^*} = \frac{1}{c} \ln \frac{V^* y}{N} - \frac{1}{c} \ln \beta'$$

using experimentally obtained values and converting to $\log_{10}$

$$\frac{u}{V^*} = 5.75 \log \frac{V^* y}{N} + 5.5$$

1.2.3.

Turbulence in fully developed pipe flow.

Laufer (Ref 9) experimented in air in a 10" pipe at average velocities of 10 and 100'/sec allowing an inlet length of 30 diameters to obtain fully developed flow after screens, contractions and a 2½' length of sand paper roughening. 0.0001" diameter Wallaston platinum wire was used for the hot wire measurements. Turbulence intensities plotted in dimensionless form as a ratio of the friction velocity was 0.8 at the centre of the pipe for the fluctuations in all three directions. This ratio for the intensity in the direction of flow increased near the wall to a maximum value of 2.6 at $y/r = 0.015$

y - distance from wall

r - radius of pipe

The circumferential component increased to 1.6 further from the wall and the radial component showed a slight increase to a value of about 1.0 yet further from the wall.

He found that the direct viscous dissipation takes place in a very small region close to the wall $\gamma^* < 15$

$$\gamma^* = \frac{yV^*}{\nu}$$

V^* - Friction velocity

The maximum production of turbulent energy, viscous and turbulent dissipation and diffusion occurred in the region $\gamma^* = 11.5$

at the edge of the laminar sublayer. The central region received most of its turbulent energy by diffusion from the wall region. Spectrum studies confirmed the idea of energy being transferred from smaller to larger wave numbers.

1.2.4. Turbulence in artificially stimulated Boundary layers.

Laufer used sand paper to stimulate the growth of the boundary layer in the pipe. He, however, allowed 30 diameters of smooth pipe upstream of his test section and assumed that stimulation does not alter the turbulence properties of the fully developed flow and that they are the same as if the flow had developed naturally.

Klebanoff and Diehl (Ref 10) investigated artificially thickened boundary layers with zero pressure gradients. They investigated the mean velocity profiles and some turbulence properties. The experiments were done in a 4.5' wind tunnel, 0.04% low turbulence at 100 ft/sec. Various methods were used for thickening the boundary layer, rods of varying diameter, screens and sand paper.

When sand paper was used to stimulate the boundary layer a 3" layer was obtained at a distance of 10' from the leading edge for a speed of 55'/sec. This was equivalent to a 14' naturally developed layer. At a distance of 3' from the leading edge the velocity profiles showed similarity. The logarithmic velocity distribution obtained was

$$\frac{u}{V^*} = 4.2 \log y \frac{V^*}{N} + 5.75$$

The deviation from the above expression was found to increase with Reynolds number. A power function of the form

$$\frac{u}{V^*} = 8.16 \left(y \frac{V^*}{N} \right)^{1/7}$$

was then found to be in better agreement with experimental values.

Some turbulence properties were investigated in order to establish whether these were uniquely related to the velocity profile. Due to the lack of knowledge of the relationship between the mean velocity distribution and the turbulence properties this could not be assumed. The u' spectrum was found to be similar for artificially and naturally developed boundary layers but it was also found that the spectrum was insensitive to the distortions in the mean flow. The intensity of u' fluctuations showed similarity with similarity condition of the mean flow when expressed as a ratio $\frac{u'}{u}$ of the local mean velocity. $\frac{u'}{u}$ was also found to be sensitive to changes in the mean velocity profile

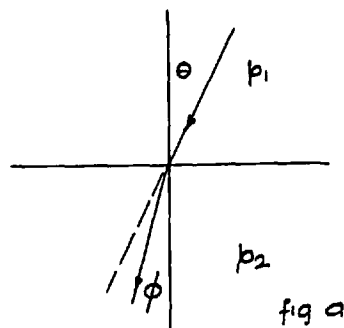
1.2.5. Effect of meshes on the flow in ducts.

Meshes are commonly used in wind tunnels to control turbulence and to reduce variations in the mean velocity, and have been studied mainly from this aspect. Taylor and Batchelor (Ref 11) obtained an expression for the deflexion of an airstream due to a gauze or mesh for flows at small angles of incidence.

$$\alpha = 1.1(1 - K)^{-1/2}$$

$$\alpha = \frac{\theta}{\phi}$$

$$K = \frac{p_1 - p_2}{\frac{1}{2} \rho u^2}$$



K - pressure coefficient

This expression for α is only true for meshes with one set of wires in the plane of the flow and the other perpendicular to it. They also derived an expression for the reduction in the disturbance to the mean velocity, across the mesh.

$$\frac{u_2}{u_1} = \frac{1 + \alpha - \alpha K}{1 + \alpha + K}$$

Where u_2 is the velocity disturbance downstream corresponding to u_1 , the disturbance upstream.

Prandtl gave a similar expression of the form

$$\frac{u_2}{u_1} = \frac{1}{1 + K}$$

and Collar (Ref 12)

$$\frac{u_2}{u_1} = \frac{2 - K}{2 + K}$$

These expressions, according to Taylor and Batchelor, are particular cases of the expression derived by them.

Prandtl's is true only when $\alpha=0$ which implies that K is infinitely large. This is not a practical case because even when the stream is perpendicular to the mesh $\alpha \neq 0$ due to its finite resistance. Collar's is correct only when $\alpha=1$ which according to Taylor and Batchelor is not true for meshes with finite resistance since this implies that lateral velocities are continuous across the meshes.

Taylor and Davis (Ref 13) gave an expression for the pressure coefficient across a perforated plate

$$K = \frac{1-\gamma}{\beta^2 C_c^2} - 1$$

γ - fraction of the pressure at contraction that is recovered downstream

C_c - Coefficient of contraction

β - Area Ratio $\frac{(\text{area of holes})}{(\text{Total Area})}$

No values are given in this paper for C_c or β . Echert and Pfluger (Ref 14) gave a simple formula for the pressure coefficient for perforated plates

$$K_1 = \left(\frac{1-\beta}{\beta} \right)^2$$

and Morgan (Ref 15)

$$K_2 = \frac{(1 - C_c \beta)^2}{C_c^2 \beta^2}$$

Morgan also gave the variation of C_c with β which he said was correct for $\beta < 0.5$

β	C_c
0.1.	0.612
0.2	0.612
0.3	0.622
0.4	0.631
0.5	0.644
0.6	0.662
0.7	0.687
0.8	0.722
0.9	0.781
1.0	1.000

Table 1.1

The last two expressions give widely differing values for the pressure coefficient

$$\begin{aligned} \text{Eq } \beta = 0.5 \quad K_1 &= 1.00 \\ K_2 &= 4.42 \end{aligned}$$

Wieghardt (Ref 16) obtained an expression for the pressure coefficient for meshes by analysing available data

$$K_m = 6(1-\beta)\beta^{-5/3} \left(\frac{u d_m}{N} \right)^{-1/3}$$

d_m - the wire diameter

The values for the area ratio given in (Ref 13) are as follows
For perforated plates with hexagonal arrangement of holes

$$\beta = 0.907 \frac{d_h^2}{l_h^2}$$

d_h - diameter of holes

l_h - distance between centres of holes

Perforated plate square arrangement of holes

$$\beta = 0.785 \frac{d_h^2}{l_h^2}$$

Square woven mesh

$$\beta = \left(1 - \frac{d_m}{l_m}\right)^2$$

d_m - wire diameter

l_m - distance between wires.

Dryden and Schubauer (Ref 17) gave a simplified theory for the reduction of turbulence by a mesh. U' is reduced in the ratio $\frac{1}{(1 + K_m)^{1/2}}$

where $U' = \sqrt{\frac{1}{3}(U'^2 + V'^2 + \omega'^2)}$

U', V', ω'
are the turbulent fluctuations, and K_m the pressure coefficient, by a single mesh, and in the ratio $\frac{1}{(1 + K)^{\eta/2}}$ by

η such meshes. Therefore for the reduction of turbulence it is more efficient to use η screens of coefficient K rather than one screen of coefficient ηK .

Grant and Nisbet (Ref 18) studied the turbulence intensities behind meshes and grids and found that, contrary to what is usually assumed, the turbulence was inhomogeneous and anisotropic. They found variations as large as 30% of the mean value of u' and also considerable differences in the different components. They expected this inhomogeneous pattern to persist a considerable distance downstream. An investigation behind a 0.02" thick perforated plate with $\frac{3}{8}$ " hole diameter also revealed the irregular nature of the turbulence intensity behind the plate.

1.2.6. Artificially produced mean velocity profiles.

Owen and Zienkiewicz (Ref 19) gave a design of a screen of horizontal rods of varying spacing as shown in fig b below, that produced a uniformly varying velocity profile downstream.

The method of design was as follows. The stream function was expressed in the form

$$\begin{aligned} \psi &= Uy + \psi' \quad \text{for } x < 0 \\ \psi &= Uy + \frac{1}{2}\lambda(y^2 - hy) + \psi' \quad \text{for } x > 0 \end{aligned}$$

where ψ' is the stream function for the disturbance. $\frac{\lambda h}{U}$ is

called the shear parameter. To satisfy the boundary conditions for all values of

$$\begin{aligned} \frac{\psi'}{Uh} &= \sum_{n=1}^{\infty} A_n e^{n\pi x/h} \sin(n\pi y/h) \quad x < 0 \\ \frac{\psi'}{Uh} &= \sum_{n=1}^{\infty} B_n e^{-n\pi x/h} \sin(n\pi y/h) \quad x > 0 \end{aligned}$$

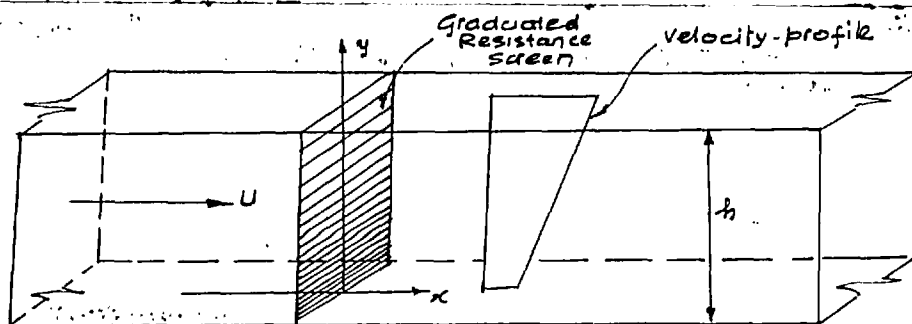


Fig b

Then using the following conditions

1. $\frac{\partial \psi}{\partial y}$ is continuous across the screen.

2. $\frac{\partial \psi}{\partial x}$ obeys the refraction condition given by $\alpha = 1.1(1 + K)^{-1/2}$

3. The difference between the total pressure on the upstream and downstream sides of the screen is equal to the resistance of the screen.

An expression was derived for the variation of the screen resistance

$$E(y) = \frac{-2\lambda h}{u} \left(\frac{1}{K_0} + \frac{1}{1+a} \right) \left(\frac{y}{h} - \frac{1}{2} \right)$$

$$a = 1.1(1 + K_0)^{-1/2}$$

K_0 - pressure coefficient at $y=0$

They experimented in a 20" square wind tunnel at 75'/sec with a screen designed for $\frac{\lambda h}{u} = 0.45$, K_0 was 1.15 and the diameter of the rods used was 0.125". The profile thus produced was found to persist, even the small irregularities, except for the gradual intrusion of the boundary layer. Increasing the value of K_0 and proportionately that of the entire screen would progressively transfer the velocity adjustment to the upstream side. However with increasing pressure drops across the screen the possibility of secondary flows occurring increases.

Elder (Ref 20) investigated the flow through arbitrarily shaped screens of non uniform resistance. He derived an expression involving the upstream and downstream velocity profiles, the shape of the screen and the variation of its resistance. Experiments were conducted with a straight screen of uniform resistance placed inclined to the stream, and a parabolic shaped screen of uniform resistance placed symmetrical with the screen.

The downstream velocity profiles agreed closely with those predicted by the expression. He also indicated a theoretical approach for the flow through a screen in a circular pipe.

An attempt was made by Zanker (Ref 21) to produce a 'flow straightener' that would give the fully developed profile in a pipe in a short distance. This was for use in conjunction with an orifice meter. A perforated plate was used together with a honeycomb. The diameter of the holes were varied but not made axisymmetric. The velocity profile measured one diameter away, was rounded but did not coincide with the fully developed logarithmic profile. The points on the profile are not sufficiently close to reveal any defects due to the ~~wake~~ from the 'flow straightener' it is not known whether the profile obtained was symmetrical or whether it did last a considerable distance. K the pressure coefficient was large, between 5.1 and 5.7.

CHAPTER 2

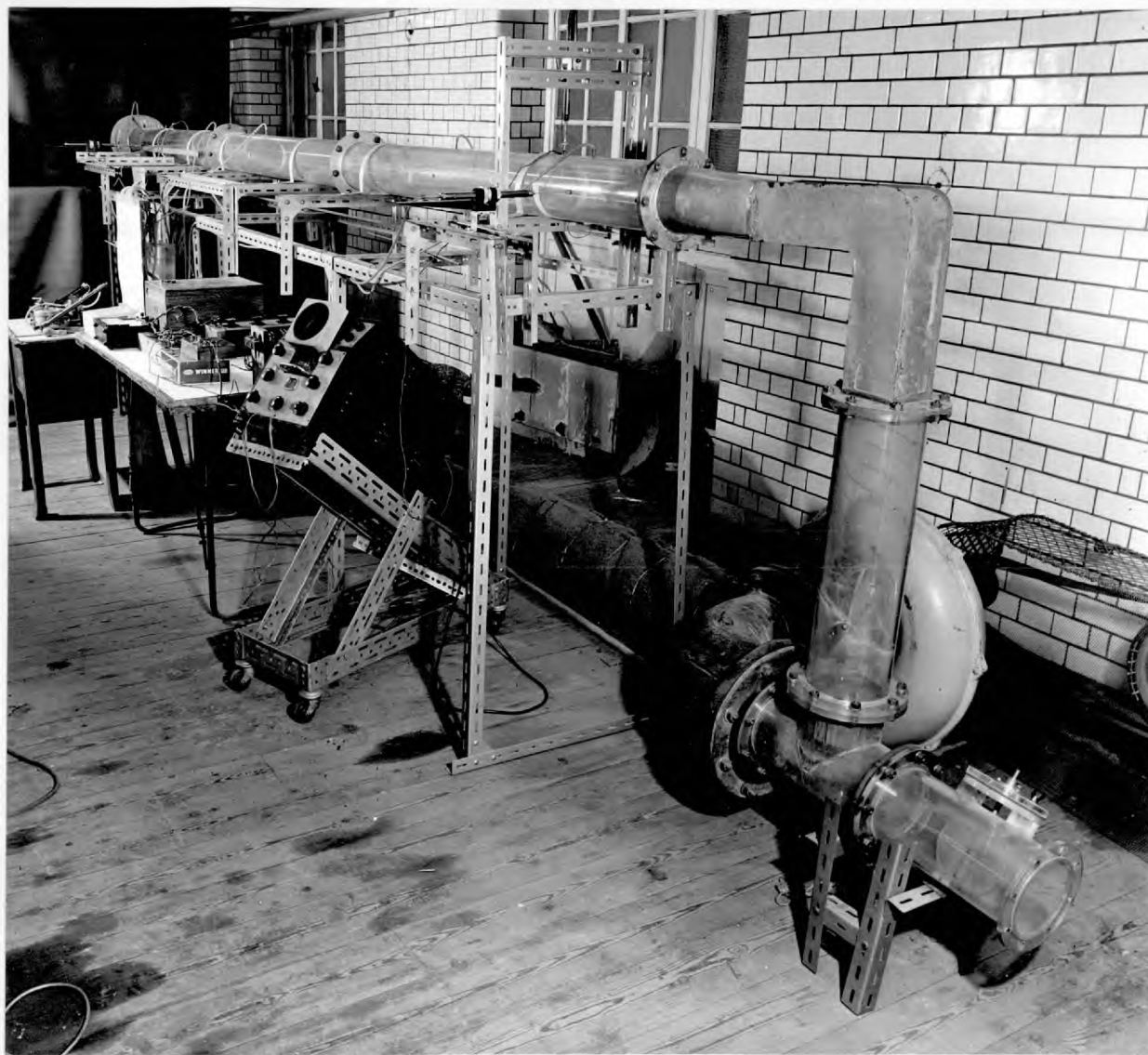
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2.1. DESCRIPTION OF APPARATUS

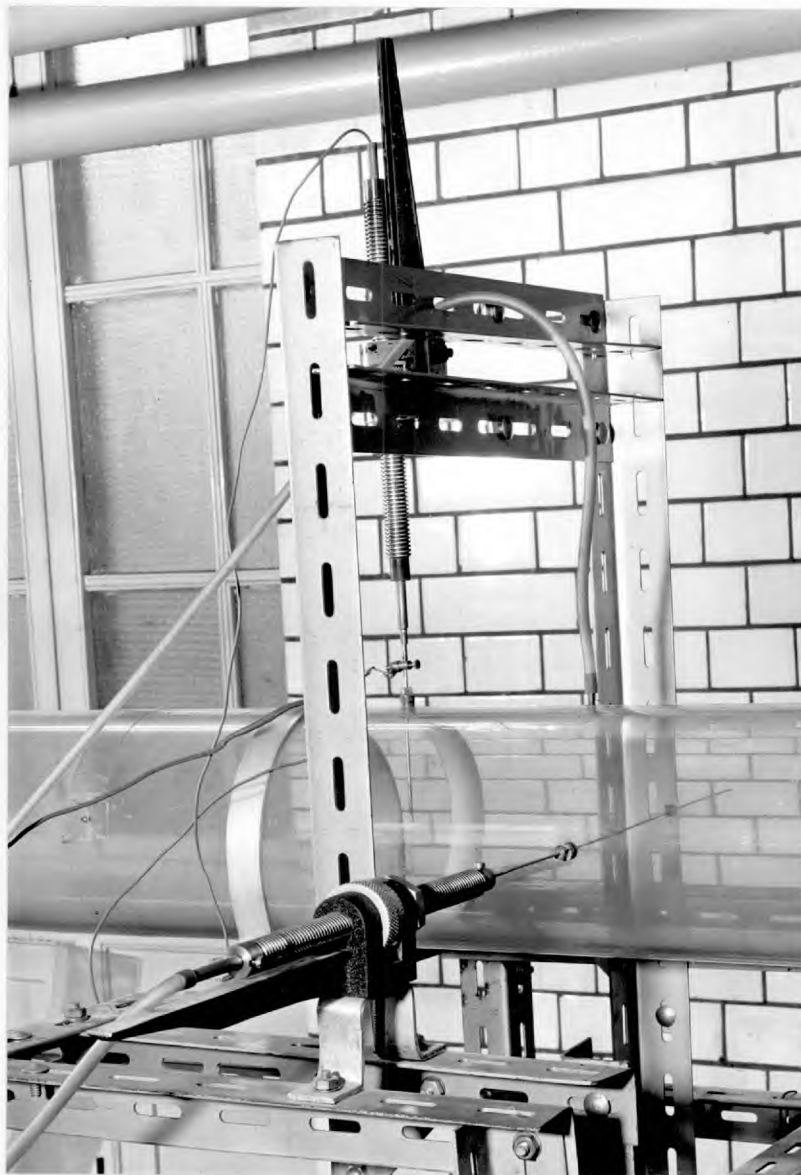
The pipe, which was of transparent perspex, had a total length of 15 ft, made up of three sections (fig 1). Of inside diameter $5\frac{1}{2}$ ". A smooth wax bellmouth with a 3" radius curve was fitted at the inlet to the pipe and the joint between the pipe and bellmouth made smooth with plasticine. Originally a perspex T junction connected the horizontal test pipe to the blower. However, when preliminary tests disclosed that disturbances originating at the junction were carried as far as 8 diameters upstream, a honeycomb was inserted at the end of the test pipe immediately upstream of the junction. But it was noticed that the honeycomb influenced the flow up to 4 or 5 diameters upstream. This was ultimately overcome by using a transition piece.

The transition piece joined the pipe to a square bend and was 12" in length. The bend itself was 3.88" square, 90° and slightly rounded off, constructed by soldering tin sheet. Cascades were fixed in the bend to prevent the development of secondary flows. A $5\frac{1}{2}$ " diameter vertical perspex pipe connected the bend to a T junction. The vertical pipe had a butterfly valve and further downstream a honeycomb to localise as far as possible the effects of the junction. One end of the horizontal member of the T junction was opened to the atmosphere through a butterfly valve and the other was connected to the centrifugal blower.

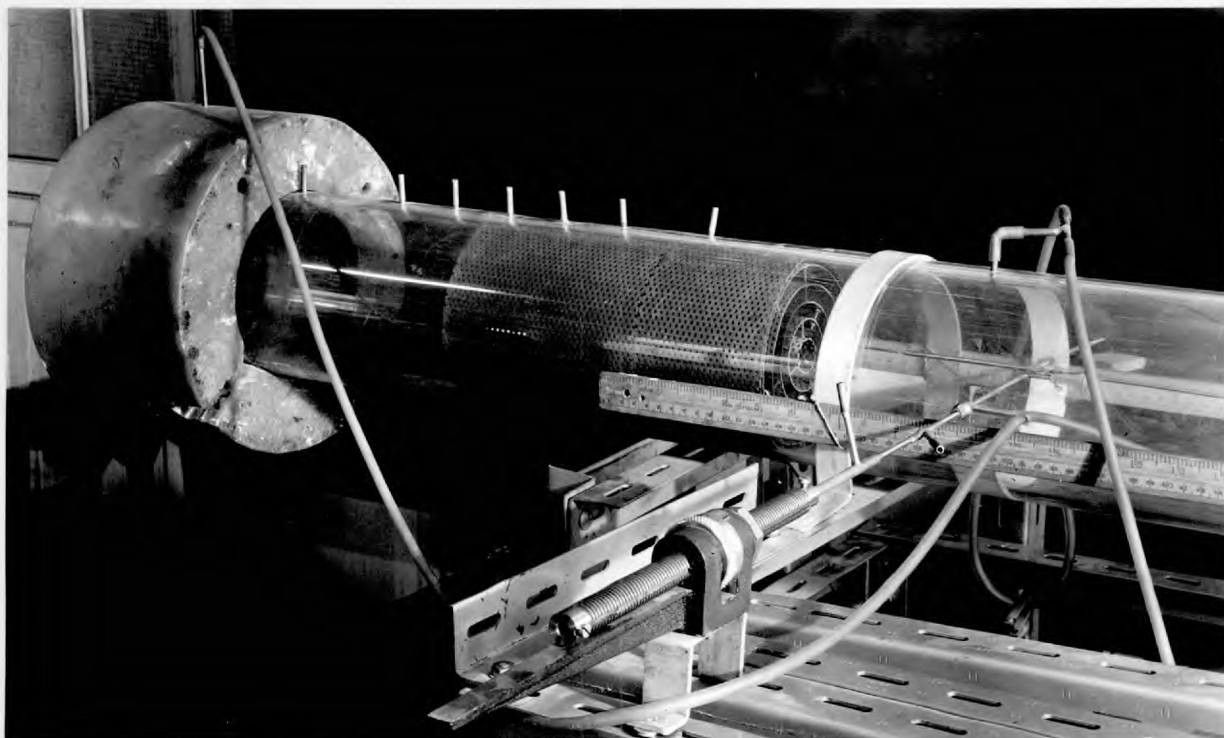
This centrifugal air blower was belt driven from a motor at a speed of around 1800 r/m and could deliver up to 600 cu ft/min S.T.P. The motor was a 8.5 HP shunt wound constant speed one with a speed around 900 r/m. A shunt field resistance permitted small adjustments to the speed. The air was discharged through a padded cylindrical mesh. The flow through the pipe was regulated mainly by the butterfly valves and partly by the shunt field resistance. Conditions were maintained constant throughout individual tests by adjusting to a reference pressure. Usually this was from a pressure tapping at the end of the test pipe. The test pipe was supported on and strapped firmly to a slotted angle iron frame. There was a considerable amount of vibration transmitted from the blower to the rest of the apparatus, probably through the timber floor.



General View of Apparatus



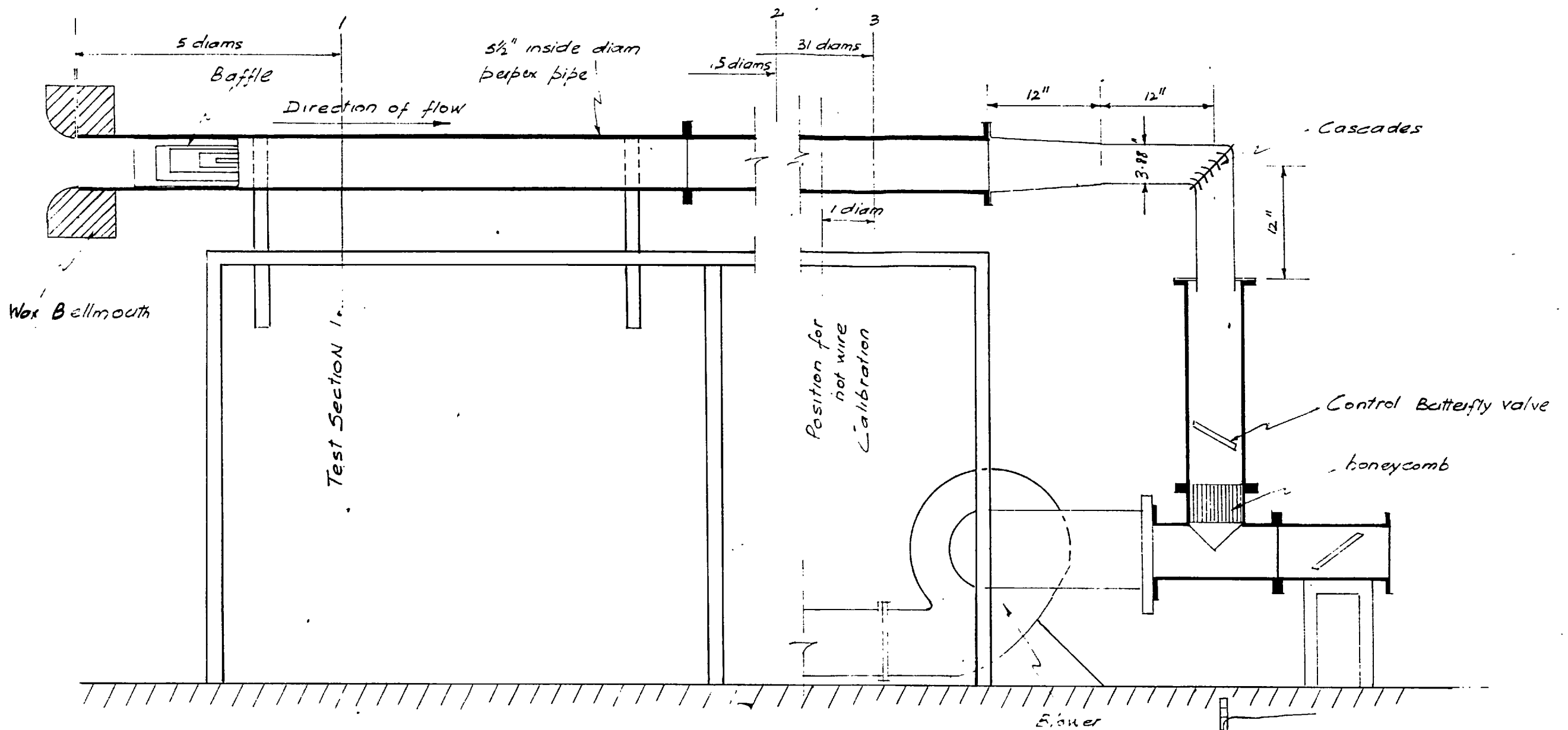
Pitot Tube and Hot Wire probe



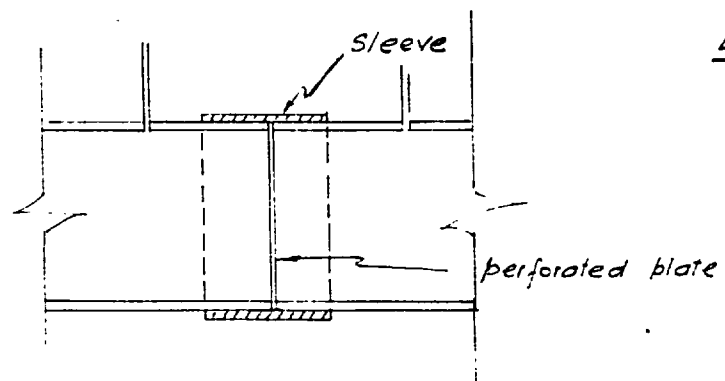
Concave Friction Baffle



Graduated Resistance Perforated Plate



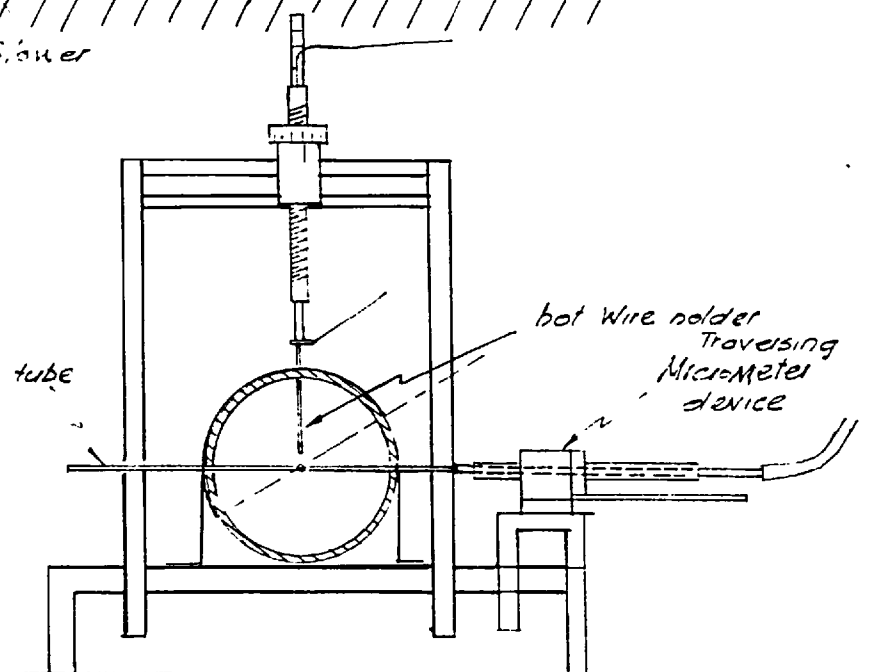
Layout of Apparatus
Scale 1/2"

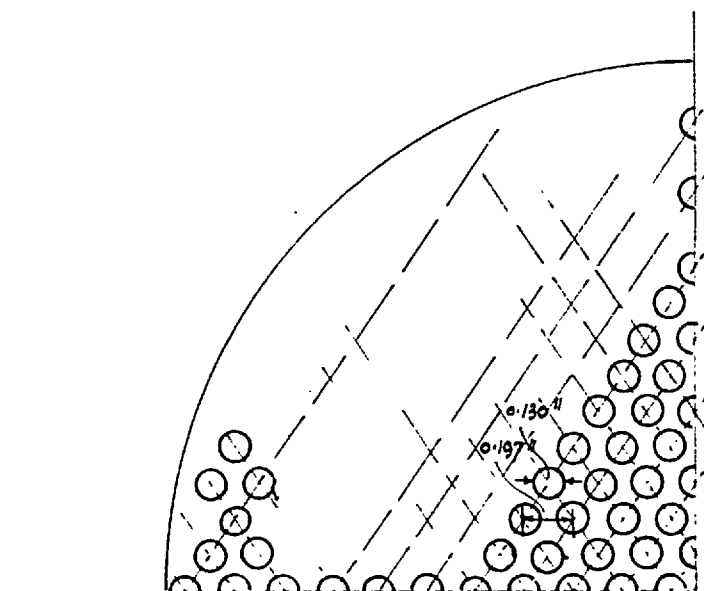


Fixing of Perforated Plate
in pipe Scale 1/2" 1"

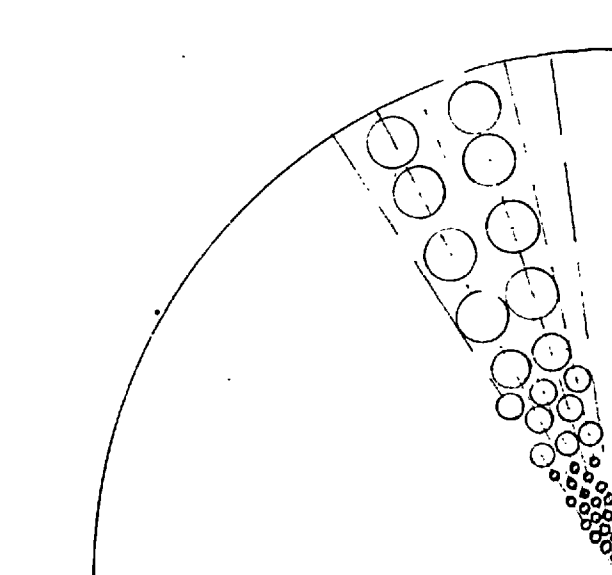
Fig 1

Arrangement for
hot Wire Calibration
Scale 1/2" 1"

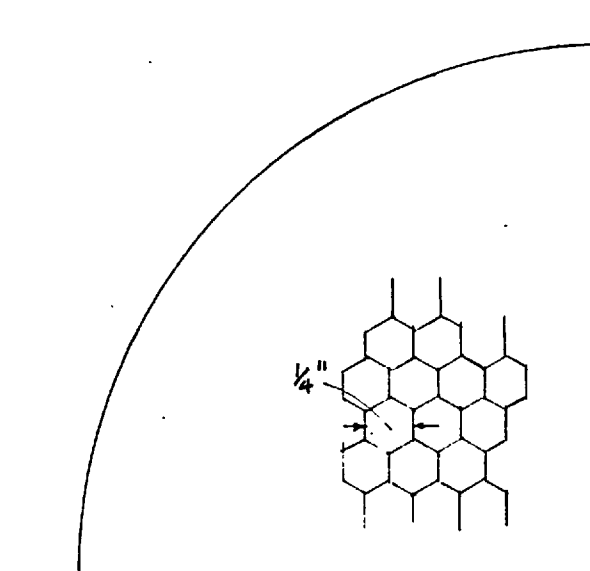




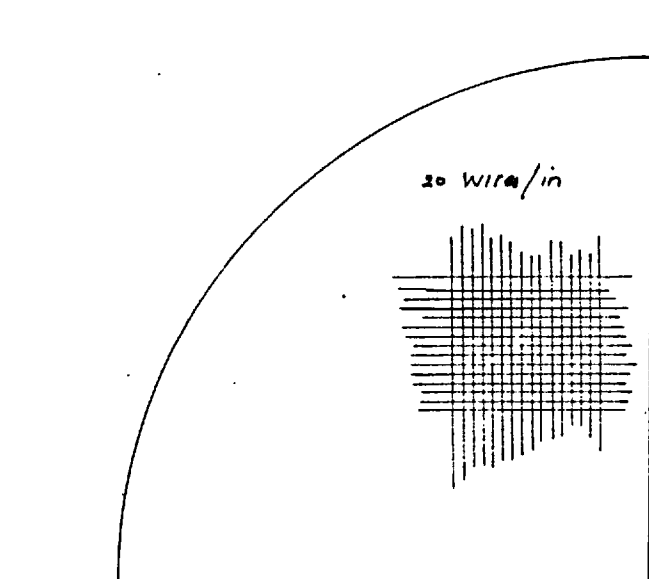
PERFORATED Plate (Hexagonal Pattern)
UNIFORM RESISTANCE



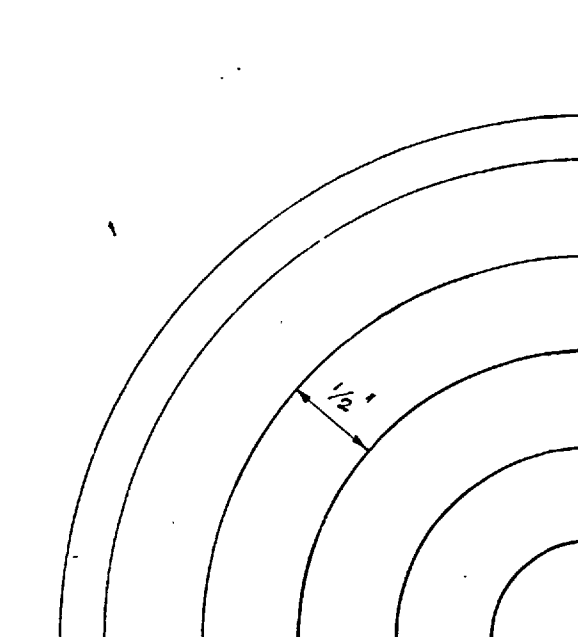
PERFORATED Plate (RADIAL Pattern)
GRADUATED RESISTANCE



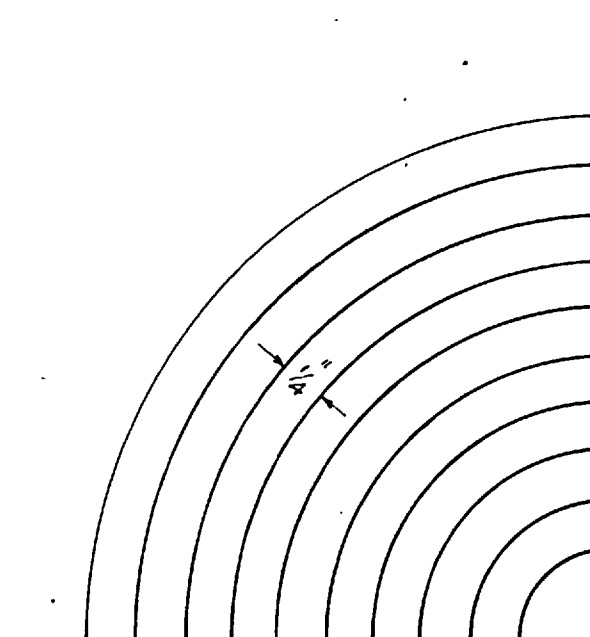
HONEYCOMB



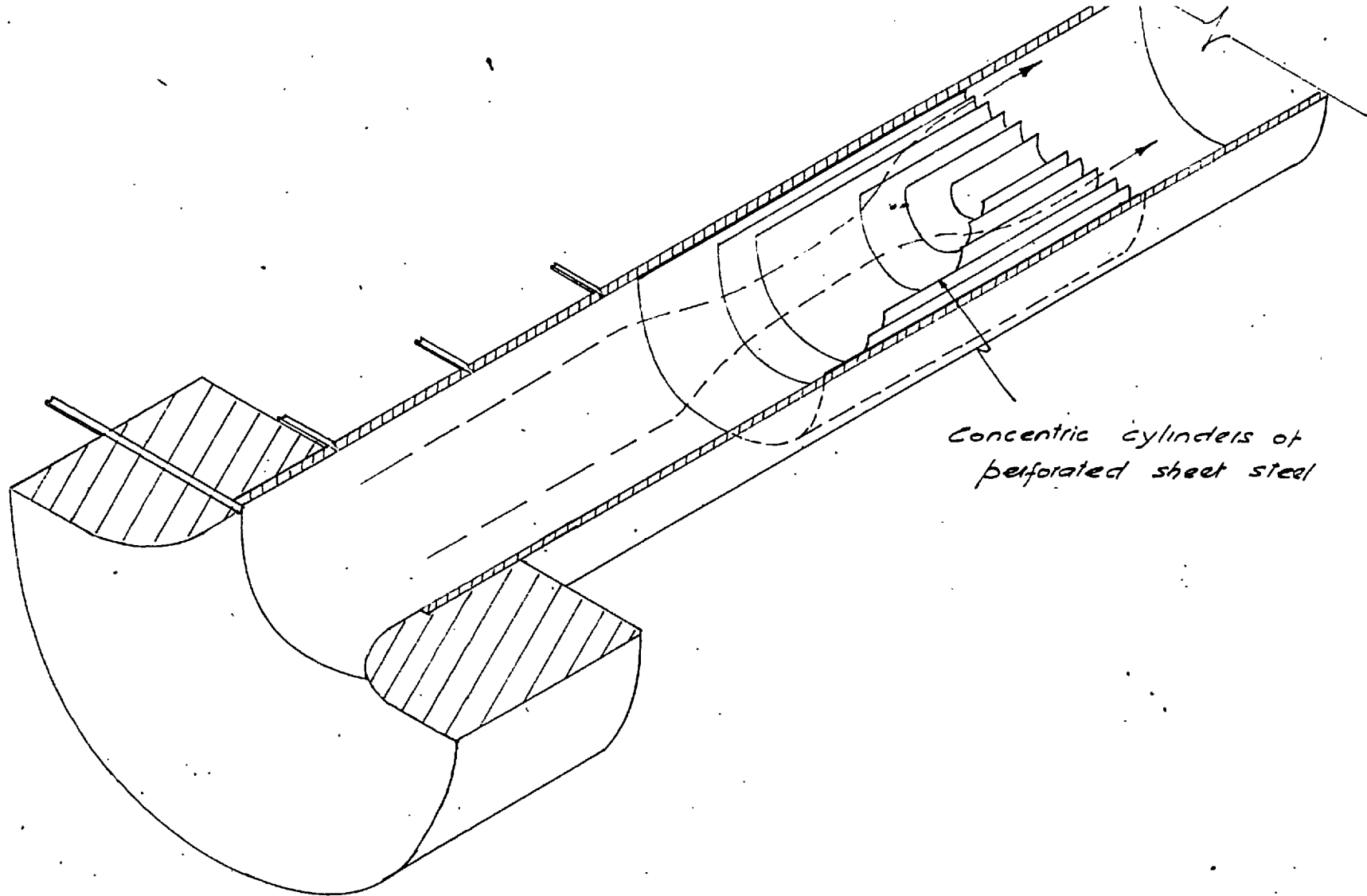
SQUARE WIRE MESH



CONCENTRIC CYLINDERS
(Large spacing)

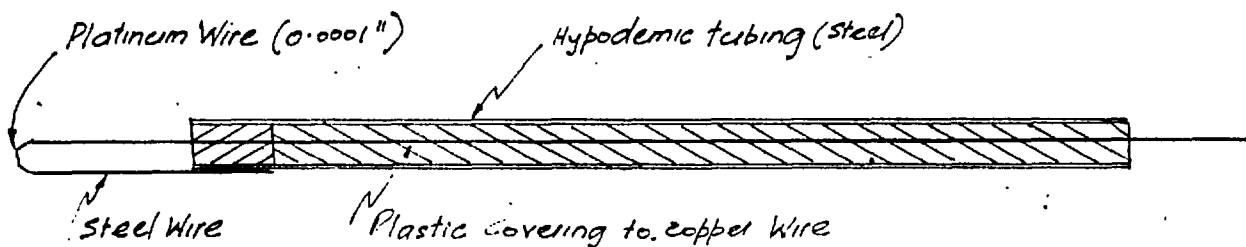


CONCENTRIC CYLINDERS
(Small spacing)



Concentric cylinders of
perforated sheet steel

Fig 3 ISOMETRIC. SKETCH OF SECTIONED PIPE AND
CONCAVE FRICTION BAFFLE



HOT WIRE HOLDER

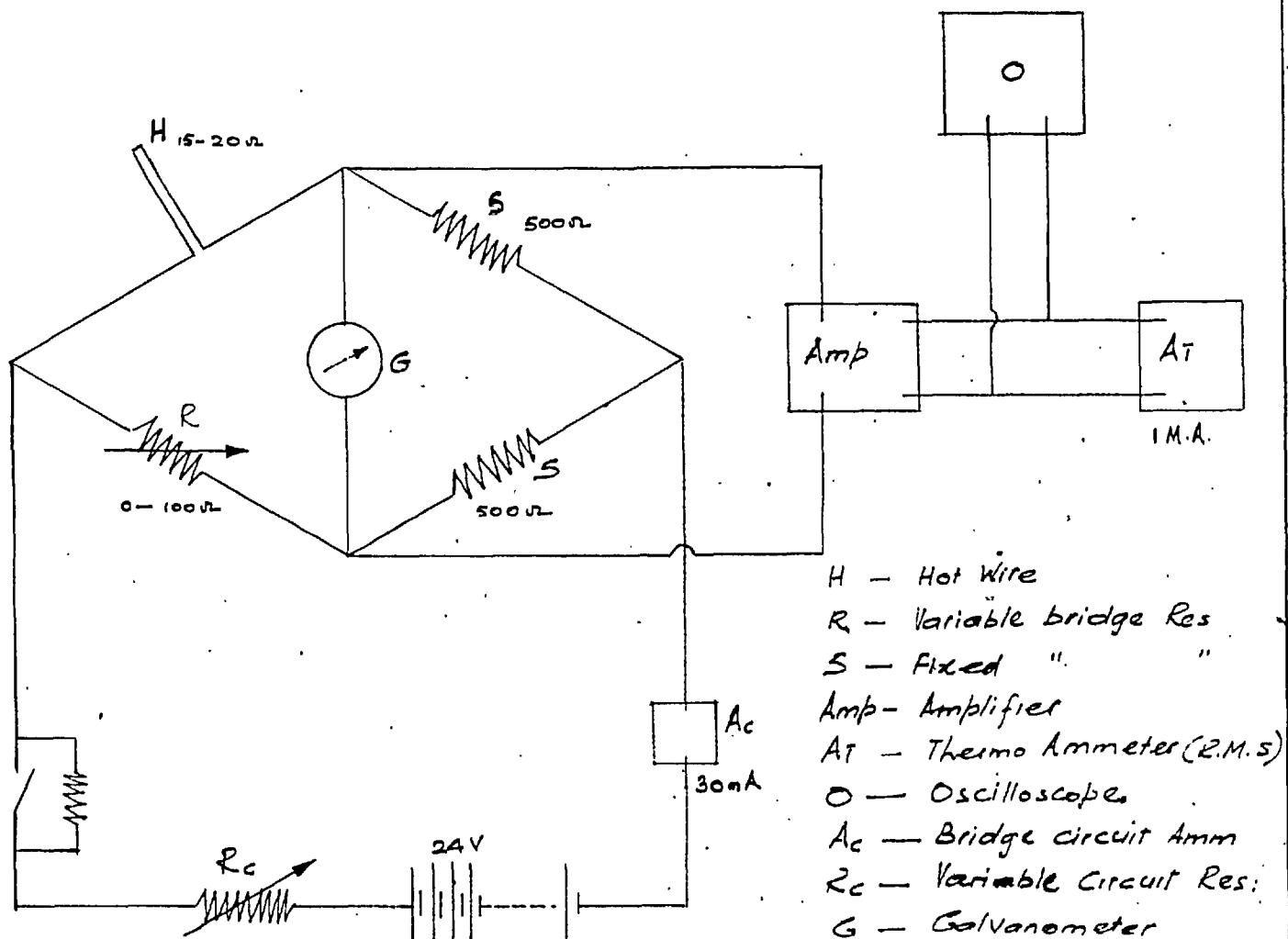


Fig 4 HOT WIRE CIRCUIT

2.2.

MEASURING EQUIPMENT

At test Section 1, 5 diameters from the inlet, a standard pilot static tube was used to obtain the velocity and pressure heads. Its dimensions were

Outside tube	Inner diameter	= 0.100"
	Outside "	= 0.138"
Inner tube	Inner diameter	= 0.043"
	Outside "	= 0.068"

The supporting tube extended across the pipe, thus keeping conditions similar while the tube was traversed across the pipe. The tube was inserted into the pipe through a slot cut horizontally in the wall immediately downstream of the tube. Traversing was done by means of a micrometer traversing device fixed to the frame and attached to the tube beyond the tapping leads. With the scale reading set correctly with the tube against the wall, the distance from the wall could be read direct correct to 0.002". Two pressure tappings on the wall connected together and disposed at 120° to each other were also used at this section as a check on the static pressure reading.

At test sections 2 and 3, 15 and 31 diameters from the inlet 0.043 inside diameter total head probes were used. These were constructed of hypodermic tubing with the supporting tubes extending right across the pipe and with the same traversing device as before. The tube facing the stream projected ¼" from the tube across the pipe. Wall pressure tappings, 0.04" diameter, were used to obtain the pressure along the pipe and were all connected by ¼" P.V.C. tubing to the manometers.

The manometer board had 9 vertical tubes and 2 inclined tubes 1 in 5, connected to a large water container. Two sensitive alcohol manometers with variable inclinations from 1.2 to 1.100 were also used and when fluctuations were small these gave very accurate readings.

The pressure tappings at the test sections were directly above the edge of the pilot tubes and no interference from the pilot tubes to these readings was observed.

Reading taken by total head and pressure tubes can be defective due to various reasons some of which are given below with the corresponding errors as found or estimated by various investigators. In the present work one or more of these effects would tend to predominate depending on the conditions under which the measurements were made.

(a) Effect on the total head reading of turbulent fluctuations. Goldstein (Ref 22) has shown that when the total head tube is placed in a turbulent stress the head recorded is

$$p + \frac{1}{2}\rho u^2 + \frac{1}{2}\rho \bar{u}'^2$$

u - mean velocity

u' - resultant turbulent fluctuation.

(b) Effect on the pressure of turbulent fluctuations. Fage (Ref 23) expressed this in the form

$$p_m = p + G\rho[\bar{v}'^2 + \bar{\omega}'^2]$$

p_m - measured value

p - static pressure

$\bar{v}'^2, \bar{\omega}'^2$ - mean value of the fluctuations perpendicular to axis of tube.

G - constant.

For fully developed pipe flow was found to be 0.25, and for isotropic turbulence $p_m = p + 0.5\rho\bar{v}'^2$

G varies with different conditions of flow.

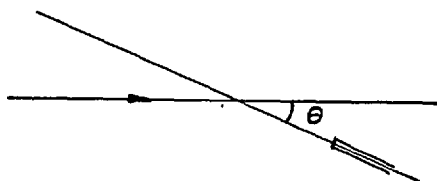
(c) Effect of the angle the pilot tube makes with the stream. Fage (Ref 23) gave corrections for this

EG

Velocity 40'/sec

$$\theta = 30^\circ \quad \frac{H_\theta - H_0}{\rho u^2} = -0.0461$$

$$\theta = 10^\circ \quad = -0.0006$$



H - Total head

(d) Effect of the inclination of the stream on the pressure reading. Fage (Ref23) gave an expression for the correction for a tube 0.125" outside diameter, with 12 holes.

$$p_{\theta} = p_0 - 0.31 \rho u^2 \sin^2 \theta \quad \text{for } \theta < 20^\circ$$

(e) Displacement effect due to shear. Macmillan (Ref24) gave

$$\frac{\delta}{D} = 0.15 \quad \text{for } \frac{y}{D} > 2$$

δ - the displacement towards the higher velocities

D - tube diameter

y - Distance from the wall

Near the wall an additional correction has to be applied

$\frac{y}{D} = 0$	$\frac{u_c}{u} = 0.015$
$= 1$	$= 0.003$
$= 2$	$= 0.0$

u_c - the velocity correction

u - local velocity

(f) Viscosity effects.

The tube sizes used were small, ~~the~~ ^{but} viscosity effects would be small.

2.3. BAFFLES TESTED: CONSTRUCTION AND DATA

2.3.1. The Uniform resistance perforated plate was of standard hexagonal pattern. The perforations were not symmetrical about the axis.

Thickness of plate = 0.027"

Hole diameter = 0.130"

Distance between hole centres = 0.197"

Area Ratio - $\frac{\text{Area of holes}}{\text{Total area}} = \beta = 0.380$

$$K_1 = \left(\frac{1 - \beta}{\beta} \right)^2 = 2.66$$

$$K_2 = \left(\frac{1 - C_c \beta}{C_c \beta} \right)^2 \quad (\text{Ref 15}) = 10.60$$

K_1 and K_2 are pressure coefficients.

2.3.2. The graduated resistance perforated plate was constructed by drilling holes of varying diameter along radii. The pattern of holes was axisymmetric. The plate was filed and emery papered to remove burrs produced by drilling. The variation of pressure coefficient was calculated on the assumption that the pressure was uniform across the pipe both upstream and downstream of the plate.

Then $\frac{K_r}{K_m} = \left(\frac{U_{max}}{U_r} \right)^2$ U_{max} - the central velocity

$K_m = 4.0$ (assumed) K_m - Pressure coefficient at centre.

U_r, K_r - Corresponding values at any radius.

The tables shown below give the variation of k_2 with β for values of C_c obtained from Ref (15) and the variation of area ratio with radius for values of $\frac{U_{max}}{U_T}$ obtained from the

logarithmic velocity distribution for Reynolds number around 140×10^3

β	k_2
0.1	149.00
0.2	51.00
0.3	18.90
0.4	8.80
0.5	4.42
0.6	2.30
0.7	1.17
0.8	0.55
0.9	0.18
1.0	0.00

Table 2.1

Radius r''	$\frac{U_{max}}{U_T}$	K_T	β
0.25	1.01	4.08	0.515
0.50	1.03	4.24	0.510
0.75	1.05	4.40	0.500
1.00	1.06	4.44	0.499
1.25	1.08	4.68	0.490
1.50	1.11	4.92	0.485
1.75	1.14	5.20	0.476
2.00	1.19	5.68	0.462
2.25	1.25	6.24	0.450
2.50	1.40	7.84	0.415
2.75	1.86	13.85	0.345

Table 2.2

The area ratio increases towards the centre of the perforated plate and the diameter of the drilled holes decreased towards the centre of pipe.

- 2.3.3. The mesh was of cylindrical brass wire woven in a square pattern, fig 2

$$\text{Number of meshes/sq inch} = 4.00$$

$$\text{Wire diameter} = 0.015$$

$$\text{Area Ratio} = 0.400$$

Pressure coefficient

$$K_m = 6(1-\beta)\beta^{-5/2} R_d^{-1/2} = 4.57$$

for $\bar{u} = 40'/\text{sec}$

$$R_d = \frac{\bar{u} d_m}{N}$$

When testing with the baffles at the entrance the plates and mesh were held firmly between the wax bellmouth and the perpex pipe. When the plates were tested inside the pipe they were held between two pipe sections with an outside coupling sleeve fig 1. The plates and mesh were always in the same position 5 diameters upstream of test section 1. The extension piece used when testing with baffles inside the pipe was 12" long.

- 2.3.4.

The friction baffles were fabricated from perforated sheet steel, with hexagonal pattern perforations. Cylinders were formed by butt welding the edges of the correct length of sheet and rolling into a circular form. fig. 3. A spacer piece in the form of a slotted cross was first used to hold the cylinders in place, but this caused interference to the flow and was abandoned. Instead thin threaded rods $\frac{1}{16}$ " diameter and small nuts were used to space the cylinders, a rather tedious method but the only one available for fixing that would allow dismantling and refixing when necessary. The data for the baffles are given overleaf.

Plate Thickness	= 0.039"
Hole diameter	= 0.097"
Area Ratio	= 0.36
K_1	= 3.16
K_2	= 12.50

CONVEX BAFFLE (SHORT AT WALL)		CONCAVE BAFFLE (SHORT AT CENTRE)	
Diameter	Length	Diameter	Length
1.5"	15.0"	1.0"	2.75"
2.5	13.5	2.0	3.50
3.5	12.0	3.0	4.25
4.5	10.5	4.0	7.00
		5.0	7.75
		5.5	10.50

2.4

HOT WIRE EQUIPMENT

2.4.1. Principle of hot wire measurements.

The hot wire technique is now adopted as a reliable method for measuring turbulent fluctuations in an air stream. The method is based on the fact that when a thin wire of suitable material is heated to a temperature above that of the air stream the wire resistance changes due to the cooling effects of the air stream. The fluctuating resistance is therefore a measure of the turbulent fluctuations. The materials generally used for the hot wire are platinum, platinum irridium and Tungsten. The disadvantage of platinum is its low mechanical strength and tungsten oxidises over 300° C and is difficult to solder. Diameters of wires used vary between 0.00005" and 0.0005". (Ref25).

Two different methods are available for measuring turbulence,— the constant current method and the constant resistance method. In the former the current in the wire is kept constant. This is the commonly used method and the one used for the present measurements. The constant resistance method requires a feedback system which until recently had proved difficult to construct. Hinze (Ref26) describes the different methods for measuring turbulence.

2.4.2. Construction of hot wire probe.

The hot wire probe consisted of a diameter hypodermic tubing with a single insulated copper wire threaded through. fig 4 . The wire was exposed beyond the tubing, a length of about $\frac{1}{2}$ ". A steel wire soldered to the hypodermic tubing which projected to the same extent as the copper wire formed the other prong of the hot wire. The distance between the prongs was about 0.1". The wire used was Wollaston platinum of 0.001" unetched diameter, and 0.0001" etched diameter.

To facilitate etching, the wire was soldered slightly looped. The tip of the hot wire was then dipped in a bubble of dilute nitric acid contained in a small cavity in a copper plate. A small current from a 1.5 volt battery was then passed through the wire. The resistances of the wires used were between 12 and 20 ohms with an etched length of about 0.03".

2.4.3. The hot wire Circuit.

The circuit used is shown in fig 4 . The wire was connected to one arm of a Wheatstone bridge with an adjustable resistance box with a minimum adjustment of 0.1 ohm on the opposite arm. Two 500 Ω fixed resistance formed the other arms of the bridge. A sensitive galvanometer was connected as shown. The current to the bridge was supplied from a battery of 12, 2 volt cells connected in series with a variable resistance and a fixed high resistance with a short circuiting switch. A sensitive milliammeter was also connected to the external bridge circuit and under balanced conditions the ammeter read twice the wire current.

The galvanometer terminals were connected to the input terminals of a transistor amplifier. The amplifier had two stages of amplification. But there was no compensation circuit for the thermal lag of the hot wire. This is usually included in an amplifier used for hot wire measurements. The power supply to the amplifier was from a 120 volt dry battery.

The output from the amplifier was connected to a thermomilliammeter. The R.M.S. value of the signal could be read on the calibration scale attached to the ammeter. The amplified signal could also be connected to an oscilloscope on which the fluctuations could be observed and oscillogram taken.

2.4.4. Calibration of Amplifier and hot wire and the measurement of intensity of turbulence.

The amplifier was calibrated by feeding in a Sine signal from a signal generator at varying voltage and frequencies. The input was measured on a millivoltmeter and the R.M.S. value of the output current on the thermoammeter. Thus the amplification for different frequencies could be obtained. A mean amplification was used in the calculations.

The wire was calibrated at the centre of the test pipe, 30 diameters from the inlet with the concave friction baffle at the inlet. The hot wire probe was attached to a micrometer traversing device and held inside the pipe through a copper plug, tightened flush with the surface. The wire was centered in the pipe by aligning with the edges of a strip of paper pasted on the outside of the pipe of length equal to half the circumference. The probe was made radial as accurately as possible. The velocity at the centre of the pipe was measured one diameter behind with a total head tube and the wall pressure tapping. It was observed that the hot wire probe did not interfere with these readings. The bridge was balanced and the wire current noted for speeds varying from 9'/su to 60'/su. Total head and pressure readings were obtained on an alcohol manometer with inclinations varying from 1.5 to 1.50. A typical set of hot wire calibration curves is shown in fig K_1-K_3 .

For measuring the intensity of the turbulent fluctuation in the direction of flow the hot wire was placed perpendicular to the stream at the required position. The bridge was balanced and the ammeter readings noted. A typical hot wire calculation for the turbulent intensity is given in Appendix 1, Page 33.

2.5.5. Factors affecting hot wire measurements (Ref 24).

- a. Thermal lag of wire.
- b. Cooling effects of the supports.
- c. Non uniformity of velocity distribution along the wire.
- d. Noise and mechanical vibrations.
- e. The intensity of turbulence in the calibrating stream may vary the calibration.

EXPERIMENTS PERFORMED

With Concave friction baffle Concave face upstream the following experiments were performed.

Velocity profiles and pressure drops in pipe for different Reynolds numbers using pilot tube and hot wire.

Pressure drop across baffle for varying Reynolds numbers.

Measurement of turbulent intensities behind the baffle and at the downstream end of the pipe.

Pressure and velocity variations within the baffle.

Velocity profiles at different sections of the pipe were obtained with the concave friction baffle flat face upstream, with convex friction baffle, uniform and graduated resistance perforated plates, wire mesh and honeycomb. With the uniform resistance perforated plate, pressure drops across the plate were measured and also turbulence intensities at the downstream end of the pipe.

CHAPTER 3

	Pages.
3.1. Preliminary investigations of the concave friction baffle.	44 — 50
3.2. Derivation of the formula for the baffle.	51 — 55
3.3. Discussion of the different terms of the formula.	56 — 62
3.4. Performance of the baffle.	63 — 76

The physical characteristics of a two dimensional baffle as defined on page 1 can usually be described by one parameter. This for perforated plates is the area ratio, ie. the ratio of hole area to total area. The parameter used to describe a square woven mesh is the ratio of the wire diameter to the distance between the wires. The relationships between the pressure drop across the baffles and these parameters as suggested by other investigators are given on pages 18 and 19

For a complete description of a varied resistance plate or mesh an expression of the variation of the above mentioned parameters is necessary and this is normally adjusted so as to produce a specific velocity profile behind the baffle. A baffle of this type designed to produce a uniformly varying velocity profile was described in Ref 19. A varying resistance perforated plate simply designed to produce a fully developed velocity profile in a pipe was tested in these experiments but did not give any useful results.

Three dimensional baffles can be constructed with uniform or varied resistance plates or meshes. Elder investigated three dimensional meshes (ref 20), both theoretically and experimentally. Three dimensional

baffles can also be constructed of varying lengths of perforated or unperforated sheet with uniform or varied spacing placed in the direction of flows. Arrangements of perforated concentric cylinders were investigated in these experiments, particularly an arrangement with the lengths of cylinder increasing towards the wall, which showed interesting characteristics as described in the following pages.

3.1.

Preliminary experiments showed that the velocity profiles in a pipe downstream of a baffle made of concentric cylinders of perforated metal were affected by the slightest non-alignment or eccentricity of the cylinders. This is a serious defect if symmetrical profiles are desired downstream, however the profiles can be observed and errors of this kind rectified by adjusting the alignment of one or more of the cylinders. The cylinders were formed by hand and differed from the perfect cylinder due to local irregularities, but these did not cause the profiles to become unsymmetrical. The difference is illustrated in fig. c

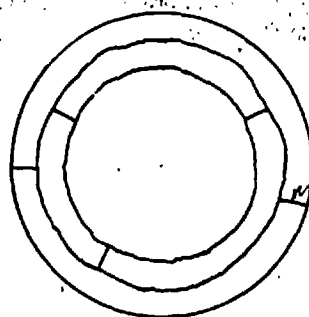
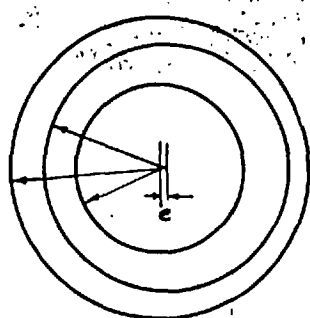


Fig C

Major defect causing
error in Velocity profile
(eccentricity of cylinders)

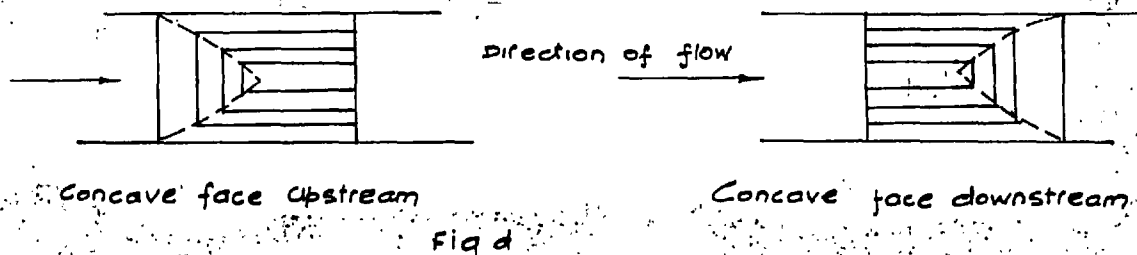
Minor error which does not
effect the Velocity profile
(irregularities in cylinders)

The arrangement of the spacing rods was irregular but this did not cause any eccentricity, although near the baffle it did cause local defects in the velocity profile. These rods were of small diameter and the wakes they caused would only have been of the same order as the local turbulence fluctuations so that lasting distortions of the velocity profile were hardly to be expected.

The lengths of the cylinders were adjusted by trial and error to give an approximation to the required profile in a short distance for a Reynolds number $\frac{ud}{N} \approx 140 \times 10^3$. The necessary adjustments to obtain the fully developed profile were affected by small changes in the relative

lengths of the cylinders and the final adjustment involving nearly all the cylinders was a very difficult task. As observed in fig. 4.1.1 and discussed later, the adjustments were not continued to the ultimate limit. For convenience of adjustment a larger spacing was used.

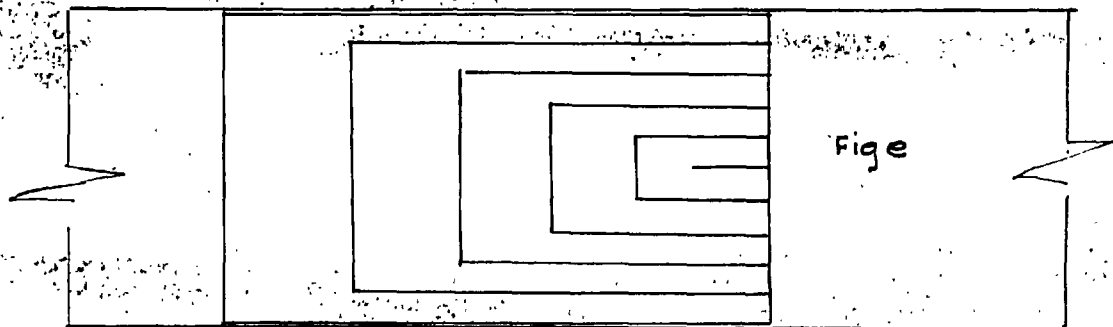
There are two possible ways of using the baffle, either with the flat face upstream or with the concave face upstream. Symmetrical profiles could not be obtained with the flat face upstream and most of the experiments were performed with the concave face upstream.



The edge of the baffle was at least diameter from the inlet. During earlier experiments with the baffle edge flush with the inlet to the pipe it was difficult to obtain symmetrical profiles. Also, when the baffle was flush with the inlet, the noise became objectionably great.

Observations with Titanic chloride fumes disclosed an outward movement of the fluid within the baffle. These observations were, however, only possible for very low flows, flows much smaller than those for which velocity traverses were made. With the direction of the baffle reversed the fumes showed an inward movement of the air within the baffle. The ideas on which the following theory is based were formed after preliminary observations of the flow of the air through the baffle.

3.2. When cylindrical tubes of varying lengths are arranged concentrically in a pipe as shown in fig.e the several flow paths,



being different in length, are liable to experience pressure gradients of different magnitudes. At the downstream end these gradients all terminate in a plane over which the pressure tends to be substantially constant. The unequal pressure gradients in adjacent paths produce radial pressure gradients within the cylinders.

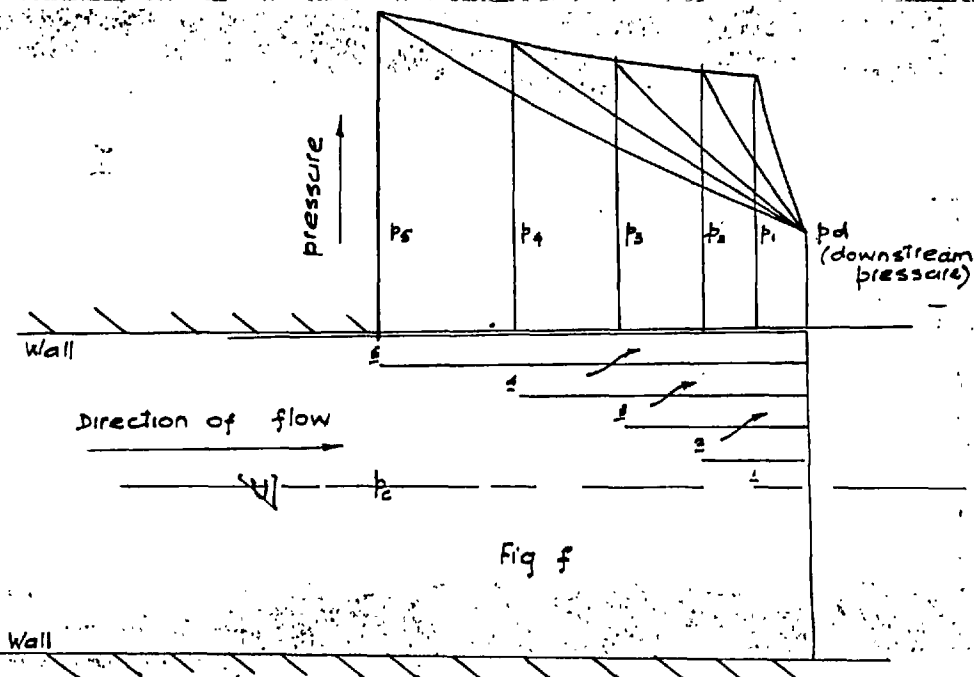
Also, due to the non uniform distribution of resistance across the pipe, the pressure and velocity fields just upstream of the cylinders are influenced by its presence, the flow tending to be displaced towards the centre.

Assuming the cylinders are made of impervious material, the fluid entering the space between any two cylinders leaves by the downstream end of the same space, so that except for certain minor readjustments the entire change in the velocity pattern takes place at the upstream end and none occurs within the cylinders themselves.

However, when the cylinders are made of perforated sheet so that they are porous, considerable radial currents can develop within the cylinders and

the distribution of velocity at the exit is no longer that at inlet. The change which occurs within the cylinders is then determined by the porosity and the radial differences of pressure.

The pressure variations likely to occur are shown in fig. f



where p_5, p_4 etc. are the pressures at the numbered points shown at the inlet to the spaces of the baffle. There is likely to be a pressure drop towards the centre of the pipe immediately upstream and a corresponding movement of fluid towards the centre. In fig. f the downstream pressure p_d has been considered constant across the pipe and accordingly the pressure in the different spaces falls from p_5 to p_d, p_4 to p_d etc. causing a pressure gradient across the cylinders and consequently causing a flow perpendicular to the axis of the pipe within the baffle.

For simplicity assume

- 1) That the pressure drops along the spaces
is much larger than the pressure variations
at the inlet to the different spaces
i.e. $p_s - p_d \gg p_s - p_c$
- 2) Between p_s and p_d , p_4 and p_d etc. the
pressure drops linearly
- 3) The flow is axisymmetric.

$$p_{3x} = p_3 - \left[\frac{p_3 - p_d}{l_3} (l_3 + x) \right] \quad \text{-----} \quad 3.2$$

$$p_{1x} = p_1 - \left[\frac{p_1 - p_d}{l_1} (l_1 + x) \right] \quad \text{-----} \quad 3.3$$

If $p_2 \approx p_3 \approx p_1 = p$

and $p_1 - p_d = p'$

Then

$$p_{2x} = p - \frac{p'}{l_2} (l_2 + x) \quad \text{-----} \quad 3.1a$$

$$p_{3x} = p - \frac{p'}{l_3} (l_3 + x) \quad \text{-----} \quad 3.2a$$

$$p_{1x} = p - \frac{p'}{l_1} (l_1 + x) \quad \text{-----} \quad 3.3a$$

Since l_1 , l_2 and l_3 are not equal the pressures at a particular value of x are not equal. Therefore defining a pressure drop coefficient K' across the cylinder

$$K' = \frac{p_{1x} - p_{2x}}{\frac{1}{2} \rho V^2} \quad \text{-----} \quad 3.4$$

ρ - fluid density

V - velocity through the perforations

The flow through the perforated cylinder per unit area is βV

○

$$\beta - \text{Area ratio} = \frac{\text{Area of holes}}{\text{Total Area}}$$

substituting for V from 3.4

$$\beta V = \frac{\beta}{(\frac{1}{2} \rho K')^{1/2}} (p_{1x} - p_{2x})^{1/2} = C (p_{1x} - p_{2x})^{1/2} \quad \text{-----} \quad 3.5$$

$$C \equiv \frac{\beta}{(\frac{1}{2} \rho K')^{1/2}}$$

Considering the flow into and out of space 2 through the perforations of an elementary circumferential area of the inner and outer cylinders

$$\begin{aligned} \text{from } 1 \text{ to } 2 & C 2\pi (\tau_2 - s_2) dx (p_{1x} - p_{2x})^{1/2} \\ \underline{2} \text{ to } 3 & C 2\pi (\tau_2 + s_2) dx (p_{2x} - p_{3x})^{1/2} \end{aligned}$$

s - spacing (distance between two cylinders)

Nett flow out of space 2 through the elementary circumferential area

$$\begin{aligned} &= 2\pi C \left[(p_{2x} - p_{3x})^{1/2} (\tau_2 + s_2) - (p_{1x} - p_{2x})^{1/2} (\tau_2 - s_2) \right] dx \\ &= 2\pi C \left[\{ (p_{1x} - p_{2x})^{1/2} + (p_{2x} - p_{3x})^{1/2} \} s_2 dx \right. \\ &\quad \left. - \{ (p_{1x} - p_{2x})^{1/2} - (p_{2x} - p_{3x})^{1/2} \} \tau_2 dx \right] \\ &\equiv f(p) \times 2\pi C \end{aligned}$$

Therefore the nett flow through the perforations into the whole length of the space is $2\pi C \int_{-l_2}^0 f(p) = 2\pi C \int_0^{l_2} -f(p) \quad \text{--- 3.6}$

The integral $\int_0^{-l} \{ (p_{1x} - p_{2x})^{1/2} - (p_{2x} - p_{3x})^{1/2} \} dx$

after substituting 3.1a, 3.2a and 3.3a is

$$2/3 p'^{1/2} l_2 [f_1(l) - f_2(l)] \quad \text{--- 3.7}$$

$$f_1(l) \equiv (1 - l_2/l_3)^{1/2}$$

$$f_2(l) \equiv (l_2/l_1 - 1)^{1/2}$$

Similarly the integral $\int_0^{l_2} \{ (p_{1x} - p_{2x})^{1/2} + (p_{2x} - p_{3x})^{1/2} \} dx$

$$= 2/3 p'^{1/2} l_2 [f_1(l) + f_2(l)] \quad \text{--- 3.8}$$

Inserting 3.7 and 3.8 in 3.6 the nett flow from space

$$\underline{z} \text{ is } \frac{4}{3} \pi C p'^{1/2} l_2 \left[\gamma_2 \{f_1(l) - f_2(l)\} + \frac{S}{2} \{f_1(l) + f_2(l)\} \right] \text{ ————— } 3.9$$

The decrease in the mean flow between inlet and exit

$$\text{from space } \underline{z} \text{ is } = U_{u2} \times 2 \pi r_2 S - U_{d2} \times 2 \pi r_2 S$$

$$\therefore 2 \pi r_2 S (U_{u2} - U_{d2}) = \frac{4}{3} \pi C p'^{1/2} l_2 \left[\gamma_2 \{f_1(l) - f_2(l)\} + \frac{S}{2} \{f_1(l) + f_2(l)\} \right]$$

$$U_{u2} - U_{d2} = \frac{2}{3} \frac{C}{S} p'^{1/2} l_2 \left[\gamma_2 \{f_1(l) - f_2(l)\} + \frac{S}{2 \gamma_2} \{f_1(l) + f_2(l)\} \right] \text{ ————— } 3.10$$

If γ is the pressure drop coefficient for the baffle

$$\text{then } \gamma = \frac{p'}{\frac{1}{2} \rho \bar{u}^2} \text{ ————— } 3.11$$

$\bar{u}$ - mean velocity of flow

Substituting for p' and C in 3.10

$$\frac{U_{u2} - U_{d2}}{\bar{u}} = \frac{2 \beta (\gamma/K')^{1/2}}{3 S} l_2 \left[\gamma_2 \{f_1(l) - f_2(l)\} + \frac{S}{2 \gamma_2} \{f_1(l) + f_2(l)\} \right] \text{ ————— } 3.12$$

An approximate solution to equation 3.12 can be obtained

by expressing ly the length of a cylinder at distance

y from the wall in a non dimensional logarithmic

$$\text{form and using generalized suffixes } \frac{ly}{d} = \alpha_n \log \left(\frac{d}{n_s} \right) \text{ ————— } 3.13$$

$$\therefore f_1(l) = \left(1 - \frac{\alpha_n \log d/n_s}{\alpha_{n-1} \log d/(n-1)s} \right)^{1/2}$$

d - diameter of pipe, $y_2 = ns$
 $y_1 = (n+1)s$
 $y_3 = (n-1)s$

If it is assumed that $\frac{\alpha_n}{\alpha_{n-1}} = 1$ (an estimate of the error

caused by this assumption has been made in Appendix 1

page 139)

then

$$f_1(l) = \left(1 - \frac{\log d/n_s}{\log d/(n-1)s} \right)^{1/2} = f_2 \eta \text{ ————— } 3.14$$

Similarly

$$f_2(l) = \left(\frac{\log d/n_s}{\log d/(n+1)s} - 1 \right)^{1/2} = f_{1,n} \text{ ————— } 3.15$$

Substituting for $f_1(l)$ and $f_2(l)$ in 3.12 and generalizing the suffixes

$$\frac{u_{ay} - u_{dy}}{\bar{a}} = \frac{2\beta}{3s} \left(\frac{\gamma}{K'} \right)^{1/2} l_y \left[(f_{2n} - f_{1n}) + \frac{s}{dy} (f_{2n} + f_{1n}) \right]$$

$$\therefore l_y = \frac{\frac{3s}{2\beta} \left(\frac{K'}{\gamma} \right)^{1/2} \left[\frac{u_{ay} - u_{dy}}{\bar{a}} \right]}{\left[(f_{2n} - f_{1n}) + \frac{s}{dy} (f_{2n} + f_{1n}) \right]} \text{ ————— } 3.16$$

Dimensionally

k'/γ is a ratio of pressure drop coefficients across and along the baffle

β the area ratio

The velocity difference is in a dimensionless form

$\frac{u_{ay}}{\bar{a}}$ has to be expressed as a function of l_y

$\frac{u_{dy}}{\bar{a}}$ is the required velocity

The denominator function is nondimensional

$$\frac{s}{dy} = \frac{1}{d/s - 2(n - 1/2)}$$

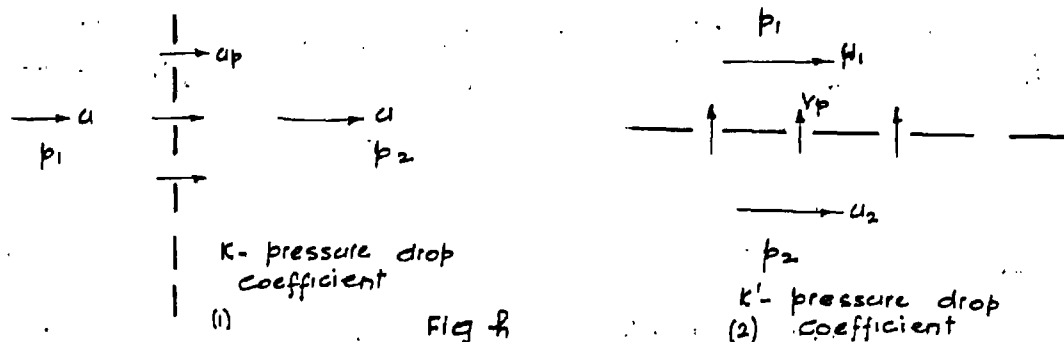
dy is the diameter to the centre of any space considered

s The spacing in the numerator has the dimension of length

In the following pages the various terms of equation will be further examined.

3.3.1.

The pressure drop coefficient K' across the perforated cylinder and the area ratio β



K' and β both depend on the material of the baffle. As mentioned earlier the pressure drop coefficient for a perforated plate for the conditions shown in fig. h(1) has been expressed in two ways in previous work

$$K_1 = \left(\frac{1-\beta}{\beta} \right)^2 \text{ --- Ray 14}$$

$$K_2 = \left(\frac{1-C_c\beta}{C_c\beta} \right)^2 \text{ --- Ray 15}$$

C_c - coefficient of contraction.

It is possible to have the same area ratio β with a few holes of large diameter or with many holes of small diameter. Therefore it is reasonable to expect what could be called a "pattern factor" to affect the value of K . This "pattern factor" could be implied in the coefficient of contraction and in that sense the latter expression is more appropriate. But how exactly the hole diameter or arrangement of holes affects the coefficient of contraction or the pressure drop coefficient is not known and has not been sufficiently studied.

There is, however, a basic difference between the conditions for which K has been defined in the above expressions and those existing in this baffle as seen on comparing fig. h(1) and fig. h(2). In fig. h(2) $V_p \ll u_1$ or u_2 whereas in fig. h(1) $u_p > u$. If two comparable

Reynolds numbers were written based on the velocity of flow through the perforations the value of Re will be much smaller for the latter conditions. K is known to be influenced by Re though this has not been considered in the expressions given above. If the same perforated plate was investigated under the two conditions K' can be expected to be smaller than K . The velocity difference $u_1 - u_2$ will also affect the value of K' .

In increase in β will cause a decrease in K' and an effective decrease in the overall length of the baffle assuming that all other factors remain unaltered. There will however be a limit to the possible increase in the area ratio.

3.3.2.

The pressure drop coefficient γ along the pipe (across the baffle) and the spacing s .

If a baffle composed of perforated cylinders with equal spacing and of equal length is considered, fig. f(1), the pressure drop coefficient γ will be a function of the spacings and the length l .

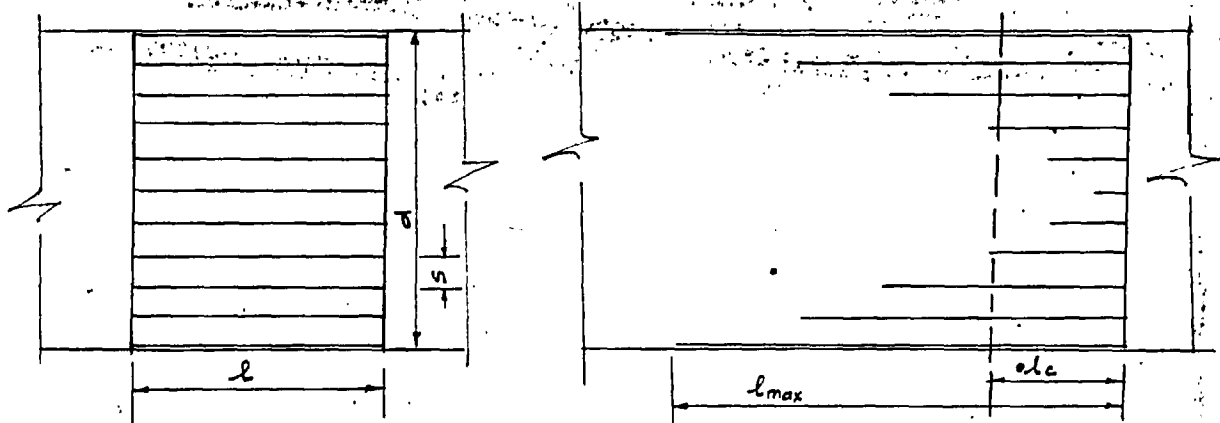


Fig. 1

in a non dimensional form.

$$\gamma = \phi \left(\frac{s}{d} \right)^{\alpha} \left(\frac{l}{d} \right)^{\lambda} \quad \text{--- 3.17}$$

α, λ - experimental constants

ϕ - constant for a particular material

d - pipe diameter

ϕ will be a function of the friction properties and the thickness of the material. Considering a baffle with equal spacing but with the lengths adjusted to give a desired profile fig. f(2) and γ determined for such a baffle, then from expression 3.17, the equivalent length l_c of a flat baffle could be established.

$$\text{If } l_{max} = \eta l_c \quad \text{--- 3.18}$$

η can be expected to be a constant for all baffles designed to give a particular conversion of velocity profiles. Therefore, for a required spacing s and pressure drop coefficient γ , the maximum length of the baffle could be obtained. It is evident that there must be a lower limit of γ for a particular spacing below which the baffle could not function.

The choice of spacing will be influenced by a number of factors. There are two extreme cases—in one a very large spacing between the cylinders and in the other a very narrow spacing.

In the first case there is a possibility of the upstream disturbances and the large scale eddies being carried through and affecting the downstream flow. This could cause the flow to become unsymmetrical.

In the second case, the overall length of the baffle will be small and pressure gradients may develop across the pipe downstream of the baffle.

There will be an optimum spacing for which the above conditions are best adjusted such that the fully developed flow is obtained in the shortest possible length.

3.3.3. The inlet and exit velocities.

It is assumed that the approach velocity to a particular space is not affected by radial flows inside the baffle and that it depends only on the resistance to the several flows. Since the spacing is the same this will primarily be a function of the length. For simplicity, assuming a power function as follows

$$\frac{u_{ay}}{\bar{u}} = \left(\frac{\bar{l}}{l_y} \right)^\delta \quad \bar{l} - \text{the cylinder length where } u_{ay} = \bar{u}$$

$$\frac{u_{ay}}{\bar{u}} = \left(\frac{\gamma l_{max}}{l_y} \right)^\delta \quad \gamma - \text{a constant} \quad 3.19$$

$$\bar{l} = \gamma l_{max} \quad l_{max} - \text{the maximum length of the baffle}$$

γ and δ must be experimentally determined.

The second term in the expression

$$\frac{u_{ay} - u_{dy}}{\bar{u}} \quad \text{is the required velocity, and for the fully developed logarithmic profile}$$

$$\frac{u_{dy}}{V^*} = 5.75 \log \frac{V^* y}{\eta} + 5.5$$

V^* - friction velocity

and $u = \bar{u}$ when $y = 0.225 \tau$

$\frac{u_{dy}}{\bar{u}}$ can easily be plotted for the required Reynolds number and the velocity corresponding to the midpoint of each space taken as the mean velocity for that space.

The logarithmic profile could also be drawn for a rough pipe and the baffle designed to give the required profile

3.3.4.

In equation 3.16 the denominator function

$$F_n = \{f_{2n} - f_{1n}\} + \frac{s}{d_y} \{f_{2n} + f_{1n}\}$$

gives the following values for $\frac{d}{s} = 12, 14, 16$ and 22 .

n	Fn			
	$\frac{d}{s} = 12$	14	16	22
(1.125)				(0.155)
2	0.1060	0.0870	0.0600	0.0748
3	0.1040	0.0820	0.0636	0.0504
4	0.1460	0.0960	0.0717	0.0504
5	0.2520	0.1300	0.0960	0.0450
6	0.9120	0.2490	0.1420	0.0639
7	-	0.8480	0.4160	0.0632
8	-	-	0.8020	0.0841
9	-	-	-	0.1224
10	-	-	-	0.2010

Table 3.1

The calculations for $\frac{d}{s} = 22$ is shown in App.1 page 136
 For $n = 1, \frac{d}{(n-1)s}$ becomes infinitely large, and the values obtained for F_n are too large. If however n is given a value slightly greater than 1 a more realistic value of F_n is obtained. In the above table the value of F_n for $n = 1.125$ has been included for $\frac{d}{s} = 22$.

3.3.5. The effect of Reynolds number.

The Reynolds number does not appear directly in the expression 3.16 and is not expected to affect the expression to any considerable degree. $\frac{C_d \Delta p}{\rho u}$ in this non-dimensional form changes only slightly with Re. δ and K' will change with Re but since they appear as a ratio the expression 3.16 may not be greatly affected by it.

3.3.6. Summary and limitations of the theory.

If the various experimental constants are determined, the variation in the length l_y for a concave friction baffle can be obtained from expression 3.16 for any pipe diameter, spacing and pressure drop.

A more rigid analysis of the problem would be desirable but is difficult due to the complex nature of the phenomenon. The discontinuity in the flow cannot be reduced to a mathematical plane as can be done with a simple screen. The variation of the resistance along and across the baffle has to be considered. The medium of flow within the baffle is not homogeneous, unless the spacing is reduced to the same magnitude as the perforations, in which case the baffle will act differently. The double deflexion of the stream lines make the flow pattern complex.

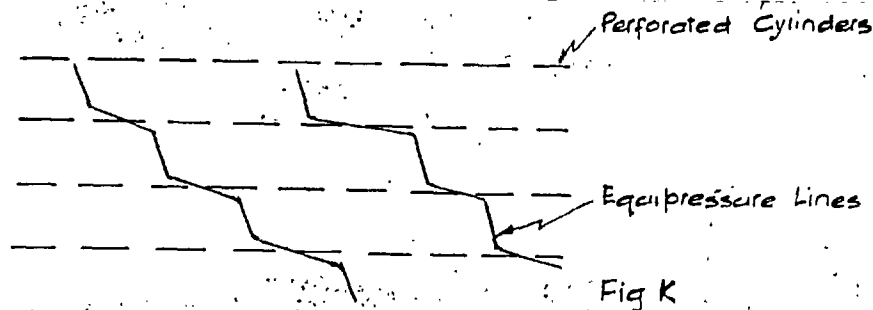
Certain inconsistencies in the theory have been mentioned earlier. At the wall either the assumptions for the velocity variation or the pressure variation is incorrect, and the denominator function F_n needs adjustment there.

A baffle designed on the above theory and with experimentally determined constants was constructed and tested and found to be about 8 percent less effective than intended.

3.4. PERFORMANCE OF THE BAFFLE

The experimental pressure contours within the baffle were obtained with a 0.1" pressure probe of hypodermic tubing with 4, 0.025" diameter holes round the circumference drilled radially 2" from the end of the tube. Fig.A.10 and fig.B.1 show the pressure contours for the two directions of the baffle and this confirms the general pattern of the flow observed visually.

In fig.A.10 with the concave face upstream the equipressure lines follow the extremities of the baffle at both ends except near the wall on the upstream side. Actually the equipressure lines can hardly be smooth curves as shown in fig.A.10, they would rather be stepped across the cylinders as shown in fig.K



The pressure readings of such probes are liable to error when the stream is oblique to the probe. The reading is then higher than the undisturbed pressure reading of that region (lower pressures) though this error can be reduced somewhat by using 4 holes equispaced round the circumference. An estimate of this error is made below using the correction for inclination given by Page (page 35).

Observations on an alcohol narometer inclined 1:2 at the inlet to space 3 fig.A10 were as follows:

Total head	= 8.0 mm.
Suction pressure	= 58.5 mm.
Velocity head	= 50.5 mm.
Correction to pressure head	= $0.31 \sin^2 \theta \times 2 \times \text{Velocity head}$
" Taking θ	= 15°
$\sin^2 \theta$	= 0.067

the correction in mm. inclined 1:2 to the pressure reading
 $= 0.31 \times 2 \times 50.5 \times 0.067$
 $= 2.05 \text{ mm.}$

Actual suction pressure = 60.55 mm.

An error of about 3% in the pressure readings. Therefore the equipressure line shown in fig.A10 will actually be inclined more. Since the probe was 0.1" diameter it occupied a large part of width, 0.5", of each space. The resulting contraction tends to increase the velocity locally and so to reduce the pressure (large - ive pressure). This may compensate the error due to inclination to a certain extent.

Upstream of the baffle the pressure lines, fig.A10, the pressure lines are inclined towards the centre of the pipe disclosing a region of low pressure there, so that the fluid converges towards the centre where correspondingly the velocity is greater within the baffle itself. There is a double deflexion of the stream, towards the centre upstream, and towards the wall within and this occurs when the concave face is upstream.

The assumptions regarding the pressure variation, linear with distance and equal at the inlet to each space, made in the theory are only partially justified by fig.A10 and fig.A11. Near the wall at the upstream end

the inclination of the pressure lines actually indicate flow towards the centre even with the baffle. The pressure lines also show a distortion at the centre. Downstream of the baffle the pressure is constant across the pipe and this was correctly assumed in the theory.

Fig.A11 shows the pressure drop along the different spaces in the baffle. The maximum difference of pressure across the spaces occurs at the inlet to the innermost cylinder and between the centre and the wall this is nearly 25% of the total pressure drop along the axis of the baffle. However, this is considerably less than predicted according to the theoretical assumption of equal pressure at the inlets and linear pressure drop with the baffle. The theoretical pressure for the same conditions as for fig.A10 and fig.A11 would be as shown in fig.A12 and there is a considerable difference between this and the actual pressure conditions. A more correct assumption that will be in better agreement with actual conditions is shown in fig.A12

The mean velocity given in fig.A10 was calculated from the pressure reading 4" from the inlet and is expected to be about 2% higher than the actual value as observed from other tests where symmetrical profiles were obtained and the mean velocity from the profile and pressure reading compared. The downstream profiles in this case were not symmetrical due to the interference caused by the attachments and tubing for the pressure probe, placed inside the pipe downstream of the baffle. The higher values of the mean velocity calculated from the pressure tapping are probably due to the large curvature of the bellmouth.

The flow for the pressure investigation with the baffle is the reverse position, with the flat face

upstream fig.B1 was adjusted so that the same pressure drop was obtained across the axis of the baffle as it was thought that this will give the most suitable basis for comparing the two pressure investigations. The mean velocity in this case was lower for the same pressure drop due to the difference in the inlet conditions. As seen from fig.B2 nearly a third of the total pressure drop occurs at the inlet within the first half inch of the baffle.

Both upstream and within the baffle the flow was towards the axis, there being no double deflexion of the stream lines as in the former position. Pockets of low pressure exist on the downstream side, which accounts for the nonsymmetry of the profiles produced by the baffle in this position. Even after repeated attempts a perfectly symmetrical profile could not be obtained. It was generally noticed that if there were pressure variations across the pipe downstream of any baffle it was almost impossible to obtain symmetrical profiles.

However, the profiles in fig.B3 are not very different from the desired profile. Since the directions of flow are quite opposite in the two positions the upstream velocity adjustment must be different. Therefore, the inlet velocities to the spaces of the baffle not only depend on the lengths of the cylinder but also on their relative position, and the relationship between the inlet velocities and the lengths shown in fig.A15 will be correct only for a concave arrangement of cylinders.

The maximum pressure difference between the centre and the wall is again about 25% of the total pressure drop along the axis fig.B2 and occurs at the exit to the innermost cylinder.

Total head readings were obtained at the inlet, exit and 1.5" from the inlet inside the concave friction baffle with the concave face upstream using a shielded total head tube of small diameter. This tube was first tested in an airstream and gave correct total head readings with inclinations of the tube with the air stream up to 45° . The flow for the total head investigation in the baffle was the same as that for which pressure contours were obtained in fig. A10 and the velocities obtained are shown in fig. A13. This shows an increase in the average velocity of every space between inlet and outlet, which is inconsistent with the pressure contours of fig. A10.

The pressure gradients at the inlet and exit are steep and positioning the total head tube accurately inside the baffle was difficult. The slightest error in positioning the total head tube could cause a considerable error in the reading when placed in a steep pressure gradient. No quantitative results could reliably be obtained from these readings. It was, however, noted that the velocity profiles are flat at the inlet and symmetrically rounded about the space centre at the exit. Within the baffle the profiles are eccentric towards the outer side, obviously due to the outward movement of the air.

Hot wire measurements of the inlet velocities to the baffle are shown in fig. A14. The flow was the same as for the profiles of fig. A.2. The average outlet velocities obtained from fig. A7 are also shown. These are consistent with the pressure contours and the variation of the inlet velocities with the cylinders is shown in fig. A15, together with the expression derived from it. Values are shown in Appendix 1 page 134.

Referring again to fig.A14 the outlet velocity in the space adjacent to the wall is less than the inlet velocity which at first sight appears to be inconsistent with the general outward flow. However, this is possible since at the upstream end the flow is actually towards the centre in the space adjacent to the wall, the pressure contours here being different from what was expected as discussed earlier.

Fig.A.2.3 shows a hot wire velocity traverse at the upstream edge of the baffle. The profile is flat except for the boundary layer. Along the axis the velocity increases from 48.1 ft/sec at the upstream edge of the baffle to 64.0 ft/sec at the inlet to the innermost cylinder. Fig A.14

An attempt was made to obtain the value of K' , the pressure drop coefficient across the perforated cylinders, but this was not successful. A single perforated cylinder was fixed axially inside the pipe with its upstream end closed with a perforated disc fig.L

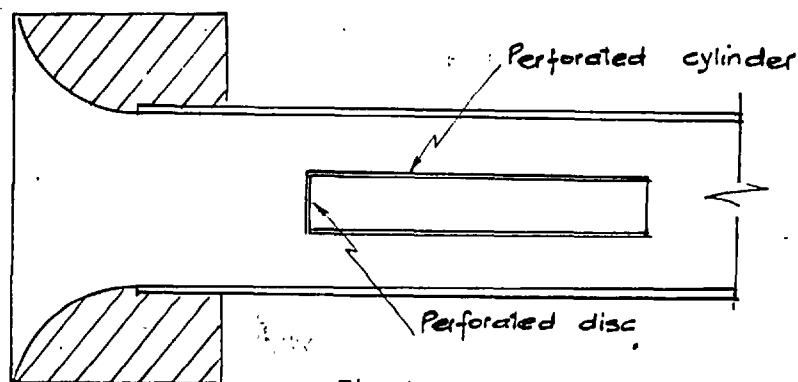


Fig L

It was expected that the pressure drop produced across the disc would persist some way downstream so that a pressure gradient would be produced between the outside and the inside of the cylinder, and by making velocity and pressure traverse at two sections some idea of K' could be obtained. But the pressure 1" downstream of the

disc was found to be uniform throughout the cross section of the pipe. If such an investigation is to be successfully carried out in a pipe a more elaborate arrangement with two blowers will be necessary. An approximate value of K' was obtained using the derived design expression and experimental data from the tested concave friction baffle.

Fig.E1 shows the values of γ the pressure drop coefficient across the baffle for flat friction baffles (i.e. cylinders of equal length). With smaller spacings γ appears to be affected by Re more than with larger spacings. The expression obtained for the variation of γ with length of cylinder and spacing on the assumption that γ is independent of Re is also shown in fig.E1

A concave friction baffle was designed using expression 3.16 (pages 135-138). The calculations are shown in Appendix 1 pages 135-138. The baffle was constructed and tested and the profiles obtained shown in fig.C1. The profile obtained was flatter than the desired logarithmic profile. The pressure coefficient γ was equal to unity the designed value. Another unexpected feature of fig.C1 is that the profile, though different from the fully developed profile, remains unchanged along the length of the test pipe. With a smaller spacing of $\frac{1}{4}$ " as used for this baffle it was more difficult to obtain symmetrical profiles than with the baffle with $\frac{1}{2}$ " spacing.

The discrepancy between the expected profiles and those actually obtained may be due to incorrect pressure assumptions in the theory and errors in the experimental coefficients. By suitably adjusting the theory and with better experimental values of the required coefficients it will be possible to obtain a correctly

designed concave friction that will give the fully developed logarithmic profile in a short distance. The low pressure drop coefficients and the elimination of elaborate upstream arrangements makes the concave friction baffle a useful means of achieving this.

The total drag coefficient for the perforated plate used in the construction of the friction baffle can be calculated from the values of pressure drop coefficient for a flat baffle given in fig E.I. This includes both the form drag and the surface drag and due to the arrangement of plates in the baffle the form drag would be a considerable part of the total drag. It is however difficult to separate accurately these two components of the total drag.

The average velocity through the baffle is greater than the average velocity in the pipe, due to the cross sectional area occupied by the plates as shown in the table below.

Spacing	plate thickness	Area occupied by baffle	$\frac{\bar{u}_b}{\bar{u}}$	ϕ
$\frac{1}{2}$ "	0.039"	2.14 sq"	1.10	1.21
$\frac{1}{4}$ "	0.039"	3.95	1.20	1.44

Table 3.2

$\bar{u}$ —Average velocity through pipe

$\bar{u}_b$ — " " " baffle

$$C_d = D$$

$$= \frac{\sum \text{Surface area} \times \rho \bar{u}_b^2}{2}$$

C_d —Total drag Coefficient

D — Total drag

$$\phi = \left(\frac{\bar{u}_b}{\bar{u}}\right)^2$$

Assuming that there is no change of momentum between the upstream and downstream sides

$$D = (p_1 - p_2) \frac{\pi d^2}{4}$$

p_1 — upstream pressure
 p_2 — downstream pressure
 d — pipe diameter

$$D = \gamma \frac{\rho \bar{u}^2}{2} \cdot \frac{\pi d^2}{4}$$

γ — pressure drop coefficient

$$\therefore C_d = \gamma \frac{d^2}{4(d + 2 \sum d_y)} \phi$$

$\sum d_y$ — summation of cylinder diameters (excluding one adjacent to the pipe)

The drag coefficient calculated from the values of from Fig. E. 1 are given in table below:

Spacing	Length of Baffle	Area Ratio	Hole Diam. Plate Thickness	Cd
1/2"	4"	0.36	2.48	0.0318
1/2"	2"	0.36	2.48	0.0364
1/4"	4"	0.36	2.48	0.0488

Table 3.3

Francis and Minton (Ref 32) investigated the drag coefficient of perforated plates at zero incidence. They concluded that perforated surfaces, particularly

those with holes, are surprisingly rough in the aerodynamic sense, the roughness depending on the perforated area and on the diameter to depth ratio of the perforations. Some relevant data from Ref 32 is shown in Table 3.4 , and fig.m gives the plotted values from Tables 3.3 and 3.4 . fig m - page 74

Plate Mark	Area Ratio	Hole Diam. Plate Thickness	Cd
A	0.355	2.3	0.071
B	0.355	6.8	0.049
C	0.355	21.6	0.038
D	0.355	84.0	0.028
E	0.089	2.4	0.028
F	0.355	5.0	0.038
G	0.089	5.0	0.022

Table 3.4

Cd for plate A is considerably higher than that for the perforated plate used in the present experiments even though both have similar values for the characteristic parameters of perforated plates. The form drag for a series of plates as used in the baffle was expected to be higher than for a single plate. However, the thickness of Plate A was 0.048" as compared with 0.039" for the thickness of the baffle plates and the tests on Plate A were in a negligible pressure gradient at an average speed of 100 ft/sec whereas the baffle had steep pressure gradients across it and the velocities were around 50 ft/sec.

The minimum surface area of perforated plate required to convert a flat velocity profile in a pipe to the fully developed logarithmic profile can be obtained using these total drag coefficients. Assuming that the pressure upstream and downstream of the baffle are uniform across the section of the pipe the minimum pressure drop to produce this conversion is given by Bernouilli's equation

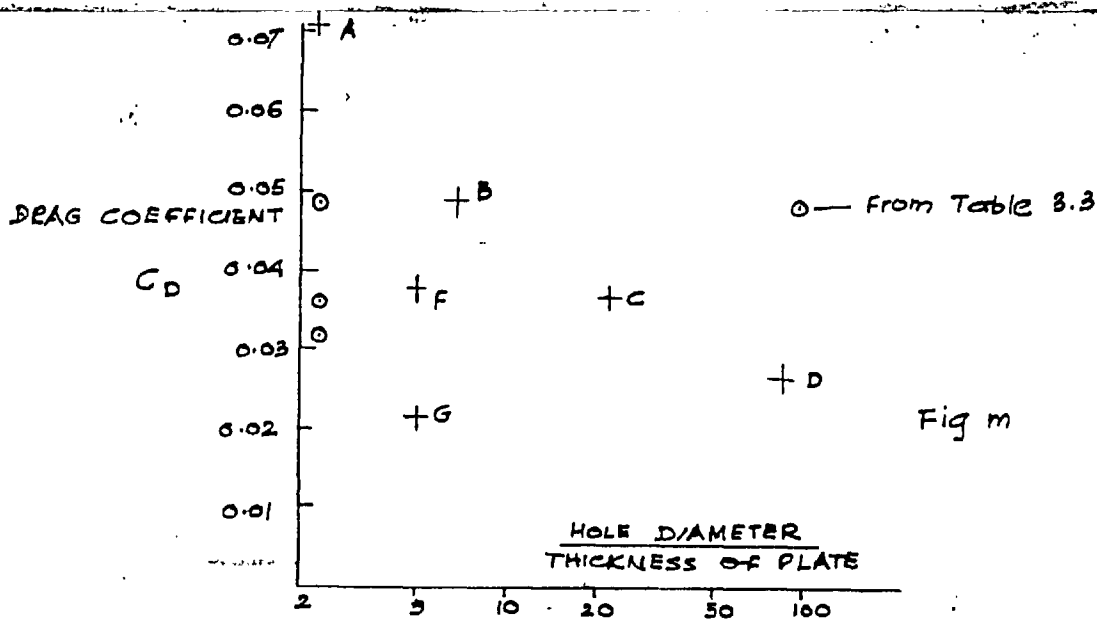
Eq Ref Fig A.1.1

$$\begin{aligned} Re &= 169.0 \times 10^3 \\ V_c^* &= 2.70 \\ N &= 162.7 \times 10^{-6} \\ \bar{u} &= 60.0 \text{ ft/sec} \\ u_d &= 70.3 \text{ ft/sec} \\ \left(\frac{u_d}{\bar{u}}\right)^2 &= 1.38 \end{aligned}$$

Then by Bernouilli's equation

$$p_1 - p_2 = 1.38 \frac{\rho \bar{u}^2}{2} - \frac{\rho \bar{u}^2}{2} = 0.38 \frac{\rho \bar{u}^2}{2}$$

$$\text{i.e. } \gamma = 0.38 = \frac{p_1 - p_2}{\frac{\rho \bar{u}^2}{2}} \quad (\text{For tested Concave friction baffle } \gamma \approx 1.0)$$



If it is assumed that the difference in the momentum of the two profiles upstream and downstream of the baffle is small, then the baffle must provide a minimum drag of

$$A_p \times 0.38 \frac{\rho \bar{u}^2}{2}$$

A_p — crosssectional Area of pipe

Considering the baffle as a series of plates with a constant drag coefficient C_d , the total drag of the baffle is

$$C_d \times \frac{\rho \bar{u}^2}{2} A_b \phi$$

$$\therefore A_b = 0.38 \frac{A_p}{C_d \phi}$$

A_b — Total surface area of baffle

Spacing	C_d	A_b	$\frac{l'_c}{l_c} = \gamma = 0.38$	$\frac{l'_c}{l_c} = \gamma = 1.0$
$\frac{1}{4}"$	0.0488	$5.40A_p$	0.69"	1.345"
$\frac{1}{2}"$	0.0364	$8.64A_p$	2.2	5.410

Table 3.5

The value of l'_c , the characteristic length (ie the equivalent length of a flat baffle ref page 58) of the ideal concave friction baffle are given in table 3.5, together with the corresponding values for the actual baffle tested. The baffle designed with a spacing of $\frac{1}{4}"$, and $\gamma = 1$ did not give the desired velocity conversion. One reason for this may be that with a spacing of $\frac{1}{4}"$ a pressure drop of $\gamma = 1$ is insufficient to produce this conversion

Taylor (Ref 33) derived an expression for the drag coefficient of a perforated plate perpendicular to the fluid stream

$$C_d = \frac{K}{(1 + \frac{1}{4}K)^2}$$

Taylor and Davis (Ref 13) obtained experimental data for perforated plates with various area ratios. Table 3.6 shows the theoretical and experimental values. There is little agreement between the two.

β	K Ref. 13	C_d Ref. 13	$C_d = \frac{K}{(1 + \frac{1}{4}K)^2}$ Ref 33	C_c from $K = \frac{(1 - C_c\beta)^2}{C_c\beta^2}$
0.530	2.50	0.790	0.950	0.725
0.530	2.49	0.835	0.990	0.764
0.493	3.29	0.840	0.945	0.722
0.420	6.00	0.950	0.960	0.691
0.188	40.1	1.35	0.330	.
0.38	*4.44			0.850
0.41	*4.48			0.785
0.55	*1.65			0.796

Table 3.6

* Values from present experiments

The value of the coefficient of contraction calculated from the experimental values of K both from data given by Taylor and Davis and the present experiments are larger than those given by Morgan (page 19)

CHAPTER 4.

PAGES

4.1.

BEHAVIOUR AND CHARACTERISTICS

OF ARTIFICIALLY PRODUCED

VELOCITY PROFILES.

78 — 93

If a sufficient length of similar boundary conditions is available, the velocity profile of the fluid flowing within the enclosed boundaries acquires an unchanging shape, the shape and the length required for its development depending on the boundary conditions. This is true for straight lengths of duct with cross sections that do not produce secondary flows. However, by suitable changes in the cross section or the resistance properties of the boundaries the fully developed conditions can be stimulated in a shorter length than is otherwise required. Also, any desired velocity profile can be stimulated by suitable arrangements of resistance across the path of flow.

The subsequent behavior of these artificially produced velocity profiles cannot, however, be easily predicted and would depend on the method by which they were produced and the associated turbulence structures. Some aspects of this problem have been investigated and described in the following pages with reference to the velocity profiles generated by the tested baffles.

4.1.

The flow immediately downstream of the friction baffle, figs. A.5 and A.7, can be described as a number of concentric wakes rather than a series of jets. At the exit from a nozzle jets are characterized by a potential core, although in the fully developed region jets and wakes have comparable properties. No such potential core exists in this case, the turbulent boundary layer appears, from fig. A.13, to have developed across the space between the baffle cylinders.

Wakes and jets are often classified as 'plane' or 'circular' according to the shape of their cross section. The flow immediately behind the baffle is 'circular' at the centre and tends to a two dimensional 'plane' cross section near the wall. Shear flows are usually classified for their turbulence properties as 'free' turbulence or 'wall' turbulence. There is a transition from 'free' to 'wall' turbulence within 10 diameters from the baffle. This is, however, not a very correct description of the flow since the wall effects exist even immediately behind the baffle, and 'free' turbulence is not associated with pressure gradients and confined spaces, and these do exist in the flow behind the baffle.

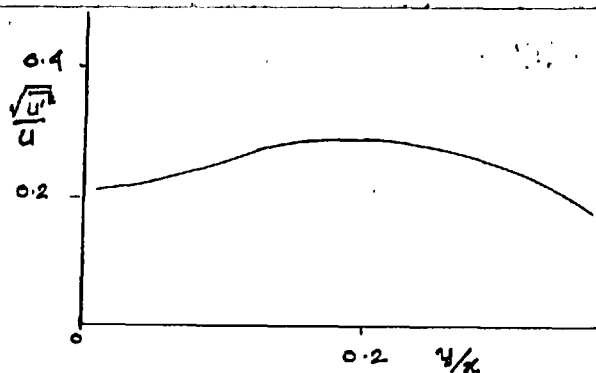
In fig. A.5 the velocity profile $\frac{1}{2}$ " behind the baffle is well defined, though the peaks on one side do not correspond to the centres of the spaces, due probably to a slight non alignment of the baffle cylinders. The asymmetry of the velocity profile should be attributed to the line of supporting rods which was near the axis of the traverse on the left side of the profile.

A hot wire traverse taken 2" downstream of the baffle is shown in fig. A.7.1 including readings close to the wall which were not possible in the pilot traverse of fig. A.5. The flow conditions were the same for the profiles in figs. A.2 and A.7.1. The average velocity calculated for fig. A.7.1 is 4% greater than that of fig. A.2 which may be regarded as the true value. Considering a set of figures from Appendix 1 page 132, Given below, this difference is not excessive

Distance from Wall	1 Hot Wire Current M.A.	I ²	Velocity ft/sec
2.00"	25.00	625.0	50.5
1.75"	24.95	621.0	49.5

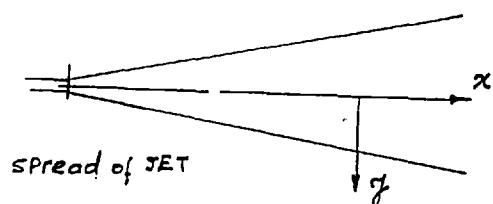
A difference of 0.05 mA in the ammeter reading gave a velocity difference of 1 ft/sec, and the smallest division of the milliammeter was 0.5 mA.

The intensity of the turbulence in the direction of flow was measured 2" downstream of the baffle and given in fig. A.7.2. and fig. A.8. Fig. A.7.2 gives the intensity corresponding to the profile of fig. A.7.1 while figs. A.8.2 and A.8.3 give a more detailed investigation behind two spaces of the baffle. If these values are compared with the intensities for fully developed regions of free jets and of wakes as obtained by Townsend (Refs 27 and 28), even though the conditions of flow are different, it will be observed that the intensities obtained in this case are very low.



Turbulence intensities in jet
(Ref 25)

(1)



SPREAD of JET

u' - Fluctuation in x direction
 u - Local mean velocity
 u_{1max} - Maximum Velocity defect
 d_c - Diameter of Cylinder

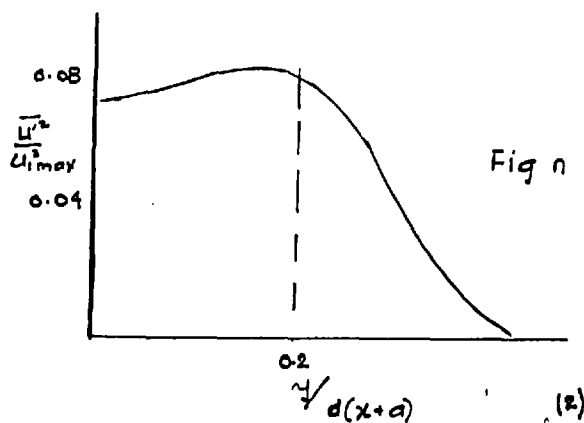
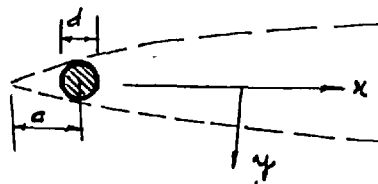


Fig n

(2)

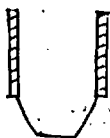
Turbulence intensities in a
wake of a cylinder (Ref 26)



Wake behind cylinder

The maximum intensity in the fully developed region of the free jet is nearly 30% of the local velocity, fig n ; whereas fig. A.8.3 shows an intensity of only 3 to 5% of the local velocity.

In a search for an explanation of this difference the hot wire equipment was investigated against a manufactured unit based on the constant temperature method. The two hot wires were fixed downstream of a large meshed grid about 12" apart and at similar positions relative to the mesh wires. The manufactured unit gave readings that were 30% higher than those obtained with the equipment used for these measurements. This may be partly due to the absence of compensation for thermal lag and if there are higher frequency fluctuations behind the baffle in the pipe the difference due to this cause will be even greater. The construction of the wire was not perfect. The etched part of the wire was not rigidly fixed and was liable to twist in the air stream. The wire was looped across the supports so as to facilitate etching off a small part (0.5 - 1.0 mm) of the silver coating.



Hot Wire as constructed



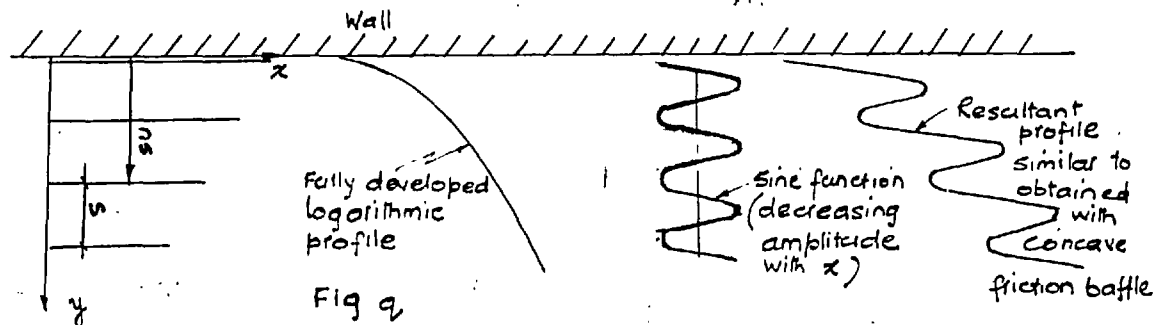
Correct Construction for hot Wire

The wire should have been held taut between the supports, since looping permits the wire to vibrate and the vibration affect the hot wire response. To measure the intensity in the direction of flow the wire must be held perpendicular to the stream. The wire response changes when it is not perpendicular to the stream. Making the looped wire perfectly perpendicular to the air stream was very difficult.

Referring again to figs. A.8.2 and A.8.3 the fluctuations decrease at points corresponding to the centres of the spaces between the baffle cylinders and increase at the quarter points of the spaces corresponding with regions of maximum velocity gradients. This is an interesting feature and if compared with fig 7(1). and 7(2). the flow is similar to a combination of jets and wakes with the intensities increasing in the region where the two merge. As far as known no information is available of turbulence measurements in a flow of this type and comparison with work is not possible.

An attempt was made to obtain theoretically a continuous expression for the velocity profile along the pipe by considering it to be a superimposition of a decaying periodic function and a constant log function of the form.

$$u = \epsilon \bar{u} \left(\frac{x}{d} \right)^p \sin 2\pi \left[\gamma - \left(ns + \frac{s}{4} \right) \right] + A \log By$$



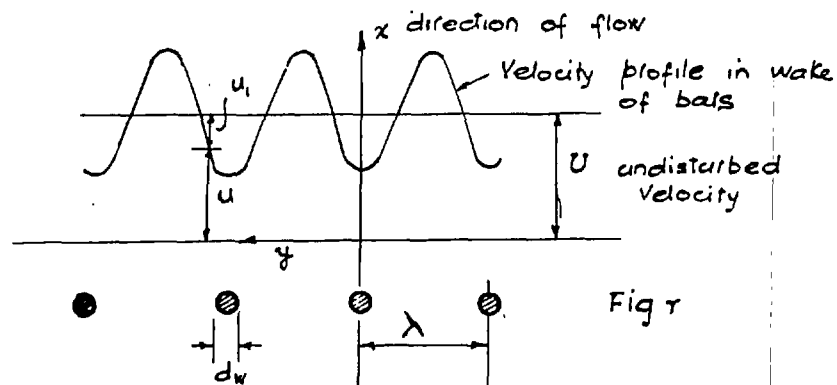
u - velocity at any point

d - pipe diameter

ϵ, p - constants to be derived

$\bar{u}$ - average velocity

and $A \log By$ by the fully developed logarithmic profile. Schlichting (Ref 29a) refers to a somewhat similar but much simpler case that was successfully investigated both theoretically and experimentally by Olsson for the flow behind a row of bars in a free stream, with no pressure gradients.



He assumed a velocity distribution of the form

$$u = A_1 \left(\frac{x}{\lambda}\right)^{-1} \cos\left(\frac{2\pi y}{\lambda}\right)$$

where A_1 is a constant and the power -1 was obtained by a dimensional analysis of the equation of motion in its simplified form

$$-U \frac{\partial u_1}{\partial x} = \frac{1}{\rho} \frac{\partial \tau}{\partial y}$$

The shear stress τ was expressed as

$$\tau = \rho l^2 \frac{\partial u}{\partial y} \sqrt{\left(\frac{\partial u}{\partial y}\right)^2 + l'^2 \left(\frac{\partial^2 u}{\partial y^2}\right)^2}$$

l was assumed to be a constant, and $l' = \frac{\lambda}{2\pi}$

and the expression for u_1 obtained was

$$u_1 = \frac{U}{8\pi^2} \left(\frac{\lambda}{l}\right)^2 \frac{\lambda}{x} \cos(2\pi y/\lambda)$$

Experimental values were found to be in agreement with this expression for $\frac{x}{\lambda} > 4$. Also $\frac{l}{\lambda}$ was evaluated at 0.103

for $\frac{\lambda}{d} = 8$

With the flow conditions in the pipe the pressure gradients cannot be ignored and the rate of decay of the wakes cannot be obtained from a simple dimensional analysis and suitable assumptions regarding the mixing lengths was not possible. However, the rate of decay of the velocity differences at the centre of the pipe was obtained from figs. A5 and A6. Figs. A.6.1 and A.6.2 show velocity measurements taken 1 and 2 diameters downstream of the baffle. No systematic variation can be identified in the velocity measurements 1 diameter downstream except at the centre where the velocity gradients are only half those behind the other spaces. There is only a scatter of points 2 diameter downstream.

Considering the flow at the centre of the pipe the maximum difference in the mean velocity decreases as $x^{-0.7}$ up to 1 diameter. x been the distance from the baffle and $x^{-0.9}$ between 1 and 2 diameters.

The values given by Schlichting (Ref29b) for the decrease in the centre line velocity difference in free jets and wakes are

Two dimensional	jet	$x^{-0.5}$
Circular	jet	$x^{-1.0}$
Two dimensional	wake	$x^{-0.5}$
Circular	wake	$x^{-0.66}$

The values obtained in the pipe are closest to the circular free jet.

From these values obtained and from pressure measurements downstream of the baffle that are discussed later, it appears that the steeper the pressure rise the slower the rate of decay of the velocity differences. With a rising pressure gradient the rate of decay is slower than with a zero or a falling pressure gradient.

Velocity profiles obtained 10 and 11 diameters downstream are given in figs.A1,A2,A5 and A6, for different values of Reynolds number. Figs.A5 and A6 were obtained using a vertical water manometer and are considered to be less accurate than figA1 for which an inclined alcohol manometer was used. Fig.A2.2 is a hot wire traverse 10 diameters downstream. In all these the points close to the wall lie below the profile obtained further downstream and above at the centre. This is noticeable even for the measurements at low values of Reynolds number in fig.A1.2 even though the differences are negligible. Had the baffle cylinders been adjusted perfectly, an unchanging profile could have been obtained. The effects of the baffle wakes are negligible at 10 diameters. The scatter of points was small and smooth curves were obtained.

Figs.A1, A2 and A5 give velocity profiles obtained 26 and 27 diameters from the baffle. The velocity profile calculated from

$$\frac{u}{V^*} = 5.75 \log \frac{V^* y}{\eta} + 5.5$$

is given in fig.A2 and it is seen that the curves do not coincide at the centre of the pipe. It has been found by investigators that measured fully developed profiles do diverge at the centre from the logarithmic profile given above. This is seen in Schlichting's (Ref29c) logarithmic plot of Nikuradse's data. But in the present measurements there appears to be a larger deviation than is usually obtained for naturally developed flows.

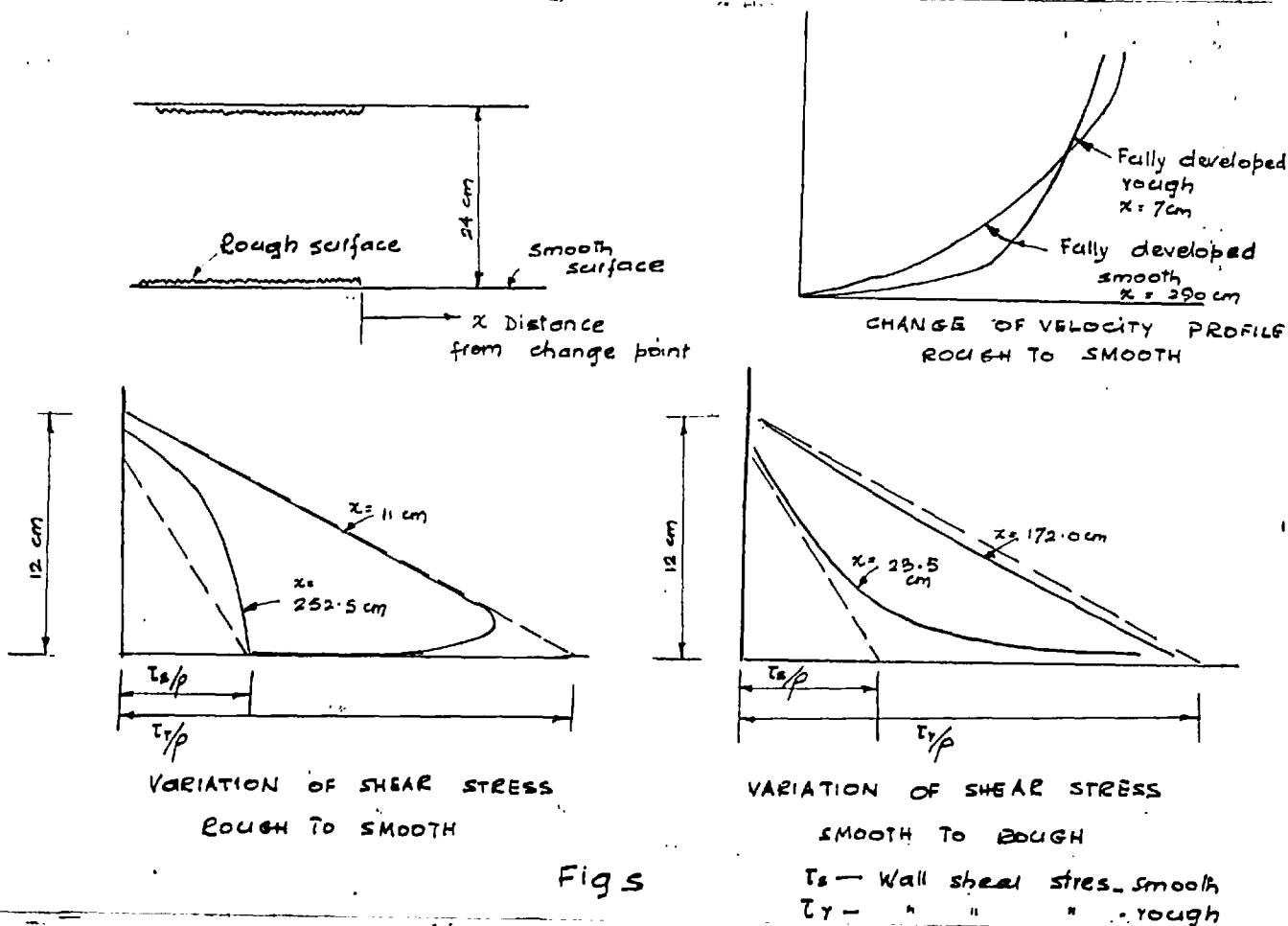
It seems quite possible that this is also due to the inaccurate adjustment of the baffle cylinders and with further refinement of the baffle closer agreement could be obtained at the centre. The logarithmic plot of three of these profiles is given in fig.A4 and all the points lie below the expected line. The friction velocities were obtained from the pressure drop in 10 diameters near the end of the pipe and these are larger than the corresponding calculated values (Appendix 1 Page 30). The values from fig.A.2.1 have been plotted again using the calculated values of the friction velocity and these points are in better agreement with the given line.

The pressures along the pipe are shown in fig.A3 together with those calculated from the Prandtl Nikuradse resistance formula. In all three flow conditions the points do not lie well on the calculated line but show a systematic variation from it. Immediately behind the baffle, as seen in fig.A11, the pressure rises at a diminishing rate for a distance of about 1 diameter. The pressure drop appears to be smaller than that given by the calculated line up to about 15 diameters fig.A3 and beyond that greater. The mean kinetic energy changes slightly along the pipe and this can account for the deviation of the pressure drop from that expected in fully developed flow.

With no baffle in position at the inlet the flow was very unsteady so that recording values of total head seemed pointless. At the end of the test pipe 31 diameters from the inlet the fluctuations of the vertical manometer were as much as $\frac{1}{4}$ " whereas with the baffle in position the fluctuations were less than 0.1" and about 3 or 4 mm on an alcohol manometer inclined 1.5. Also at the wall the pressure varied but these variations were somewhat less than the variation in the total head.

Some important questions arise from the above discussion of the velocity profiles. Is similarity of velocity profile a sufficient criterion for the flow to be fully developed? [The term "similarity and self preserving condition" could also be used to describe this state of flow and as used by Townsend has a wider relevance in other kinds of shear flows. However, the term "fully developed" used here refers to the unchanging state of the mean flow and turbulence properties.] If the rate of strain corresponding to the fully developed flow is stimulated artificially in a short length in such a manner that it does persist, would the structure of turbulence be the same as if the flow had developed naturally, assuming of course that the boundary conditions are the same?

Klebanoff and Diehl (Ref10) investigated the properties of an artificially thickened boundary layer for a smooth plate. A brief review of this work is given on Page 15. One of their important conclusions was that "The mean velocity distribution appears to afford a simple means of identifying a fully developed turbulent boundary layer". Their study did not include measurements of the shear stress. However Jacobs (Ref30) made an interesting study of the velocity distributions and shear stresses in a duct in the transition from a smooth wall to a rough wall and from a rough wall to a smooth wall, with abrupt changes of the surface. He found that the wall shear stress assumed the fully developed value corresponding to either a rough or a smooth surface almost immediately after the change although the stress distribution and the velocity profiles kept changing for a length of about 10 times the height of the duct.



These are very interesting observations. No turbulence measurements were made in this investigation. It would have been useful if turbulent intensities were measured for these conditions.

Laufer's use of sand paper lining to roughen the initial length of the smooth pipe, in his experiments, would cause the fluid at the rough wall to be retarded faster than if it was smooth, and although a certain length was allowed for readjustment to the smooth profile, an overall shortening of the total length was possible. The properties of artificially thickened boundary layers mentioned in the last paragraph could be expected to apply in pipes as well. Laufer's method can be described as essentially a gradual 'turbulence' process. But the use of a baffle, such as a concave friction baffle in a pipe to create the fully developed flow in a shorter length, is a different process where the necessary adjustment in the total energy of the fluid is affected right across the entire section of the pipe and involving large radical components of velocity. With the baffle, too, a certain length is required to allow the disturbances caused by it to decay. An observation of Klebanoff and Diehl which is relevant to the differences in the methods that has been mentioned above is that once similarity conditions were established the turbulence properties, those that were measured, were independent of the method of thickening though the different methods required varying lengths for similarity conditions to be established.

From the discussion above it was expected that the turbulent intensities measured in the pipe would be the same as that in naturally fully developed flow. The intensity in the direction of flow measured at 10 and 26 diameters downstream from the baffle are shown in fig. A9, plotted nondimensionally as a ratio of the friction velocity. The values measured are very low when compared with those of Laufer (Ref 9) shown in figs. t

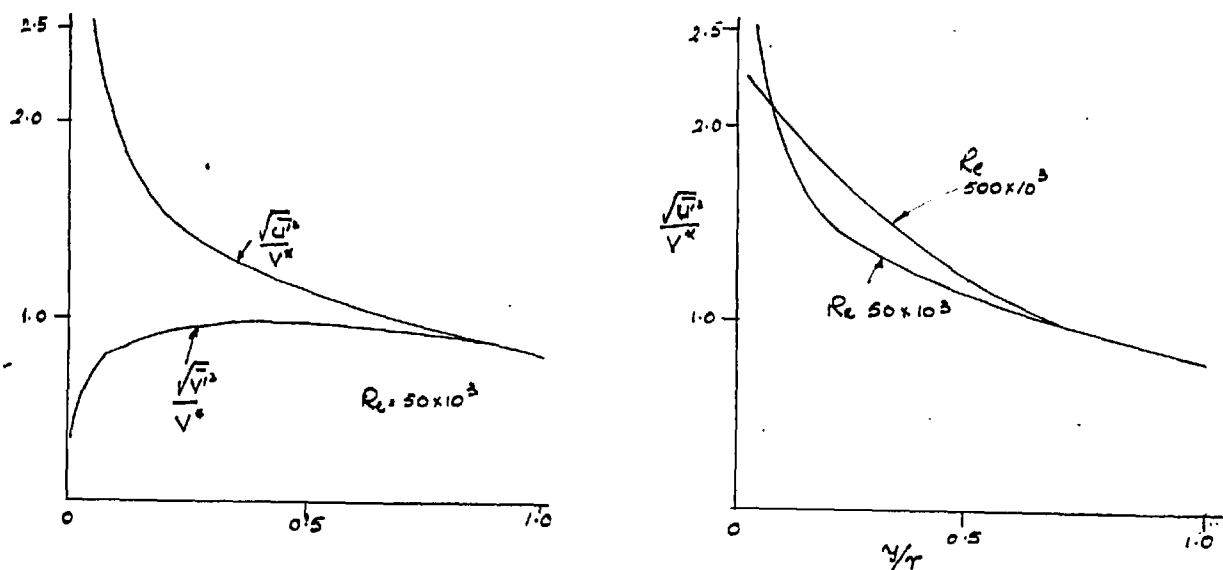
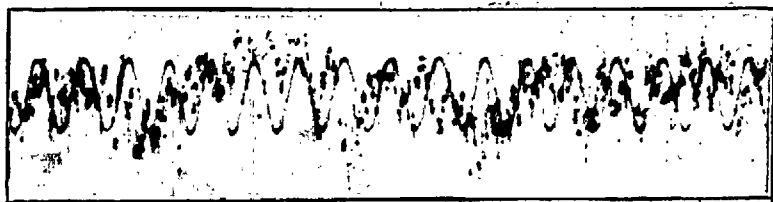


Fig t

TURBULENCE INTENSITIES - FULLY DEVELOPED PIPE FLOW (Laufer)

Since the pressure drop in the pipe, and therefore the friction velocity, was somewhat different from the expected values for fully developed flow, small differences in the intensity of turbulence could have been expected. But these differences are large, due to other causes. From the pressure and velocity measurements, the shear stress distribution could have been expected not to differ greatly from the fully developed stress distribution. Unfortunately, this was not measured, nor was the intensity perpendicular to the flow. Without these, it is impossible to comment on the correctness of the ^{turbulence} measurements made. It may be that these measurements are too low, due to defects in the measuring equipment, some of which were discussed earlier. Oscillograms taken 26 diameters downstream are shown on Page 89



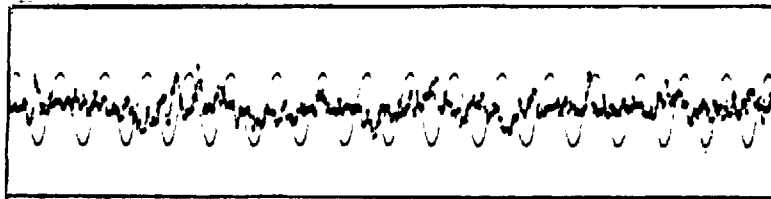
0.1" from Wall, $\frac{y}{\tau} = 0.036$

$$\frac{\sqrt{u'^2}}{u} = 0.0805$$



0.6" from Wall, $\frac{y}{\tau} = 0.218$

$$\frac{\sqrt{u'^2}}{u} = 0.0390$$



2.75 from Wall, $\frac{y}{\tau} = 1.00$

$$\frac{\sqrt{u'^2}}{u} = 0.015$$

Oscillograms taken 26.3 diams downstream of Concave friction baffle

$Re = 129.1 \times 10^3$ (As for A.2).

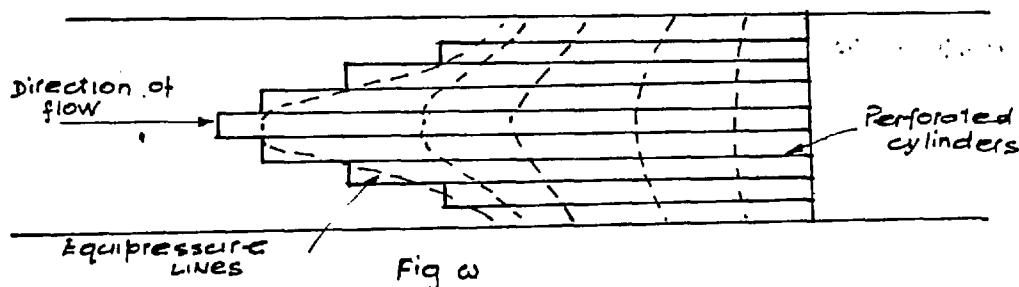
Film speed = 10"/sec

Calibration Curve = 50 c/s

If similar velocity profiles could be produced with different turbulence fields associated with them, their subsequent behaviour would provide useful information regarding the interaction of the mean velocity and the turbulence field. The graduated resistance perforated plate was designed to produce the fully developed profile with a turbulence structure different from the naturally developed turbulence field. The area ratio was reduced towards the wall (i.e. less open space) and the hole diameter was increased in the same direction (page 37). Although it is hardly possible to compare the turbulence generated by this baffle with the complex naturally developed field without an extensive investigation of the turbulence behind the baffle, it was expected that this baffle would produce a close approximation to the fully developed profile with a different turbulence structure, the large eddies being near the wall and the smaller eddies at the centre.

Unfortunately the baffle did not function as expected. Downstream conditions were very uncertain and the profiles varied at the same section for the same flow conditions. A set of profiles obtained are shown in fig. G1. The average velocities at the different *radii* do not agree due to the whirling flow in the pipe produced by the pressure variation across it on the downstream side of the baffle. Traverses were also made with a honeycomb at the inlet but the profiles did not show any noticeable difference.

The behaviour of velocity profiles produced by a convex arrangement of perforated cylinders was also investigated. The cylinder lengths of the convex friction baffle were uniformly increased towards the centre (page 39) and the baffle was fixed with the convex face upstream. There was no cylinder lining the wall as for the concave baffle. No pressure readings were taken inside the baffle but the equipressure lines expected are as shown in fig.ω



PROBABLE EQUIPRESSURE LINES WITH CONVEX FRICTION BAFFLE

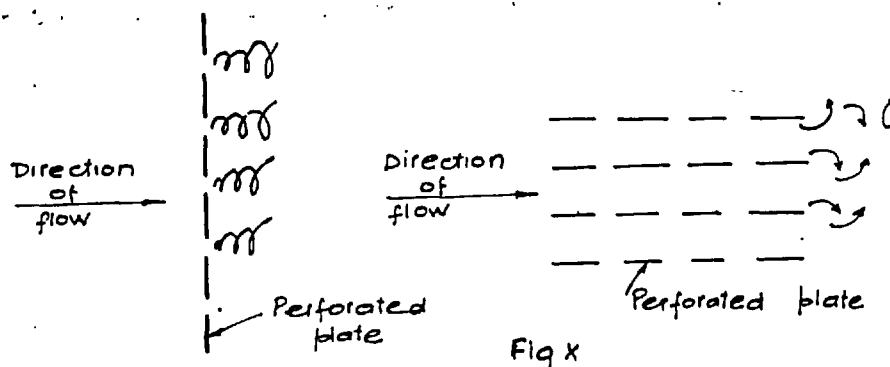
The wakes immediately behind the baffle decay in a short distance, at 3 diameters the profile is peaked at the centre and at the wall fig.D1 and fig.D2. At 10 diameters the profile is flat for a large part of the cross section of pipe. The interesting feature of these profiles is the rapid changes they undergo in the short length of pipe, to give a profile that is not very different from the fully developed one at 26 diameters. The behaviour of these profiles is a contrast to those produced by the uniform resistance perforated plate as discussed below. The pressure drops corresponding to the above mentioned profiles are much higher than for the fully developed flow at the same Reynolds number, fig.D1

Velocity profiles in the pipe with the uniform resistance perforated plate are shown in fig.F1 with the plate at the inlet immediately behind the bellmouth and with an extension pipe upstream of the plate. The central flat core extends for a large part

of the pipe cross section, the profiles being flatter for the lower Reynolds numbers. In the central region the profiles change very little along the test length though the boundary layer has developed unsymmetrically.

The difference in the subsequent behaviour of similar flat profiles is seen on comparing fig.D1 and fig.F1. The flat profile at 10 diameters in fig.D1 changes to the round profile at 26 diameters, whereas the flat profile at 5 diameters in fig.F1 remains essentially the same. This difference is obviously due to the different turbulence fields produced by the baffles and associated with the profiles, and any explanation must be based on a detailed investigation of the turbulence fields. However some visual observations that were made in the two cases may be worth noting.

A smoke jet held close to the perforated plate was found to whirl through the individual perforations fig.X, forming eddies with their axes in the direction of flow



Whereas with the perforated no such whirling was observed and the eddies shed by the cylinder appeared to be with their axis perpendicular to the direction of flow, though this was not easily observed. Also there

was less spreading of the smoke jet with the cylinders than with the plate in the short distance that the smoke could be observed. These visual observations were with very low flows. The intensity of turbulence measured 31 diameters downstream of the perforated plate is shown in fig.F2

All the profiles with the perforated plate have unsymmetrical boundary layers. This cannot be attributed to unsteady atmospheric conditions upstream because the flow downstream was steady and there was no evidence of large scale whirl. The perforation of the plate of a hexagonal pattern was not axisymmetric and velocity traverses were made with the plate rotated through various angles, but this did not reveal any connection between the nonsymmetry of the velocity profile and the pattern of perforations.

A velocity profile was obtained with the perforated plate inclined at a small angle of 2° to the axis investigate the possibility of the nonsymmetry being caused by very small inclinations of the plate with the axis of the pipe. The profile obtained is shown in fig.F3 , and this confirms the tendency of the stream to deflect towards the normal as found by Taylor and Batchelor (Ref.11) page 17 , but the profile is uniformly inclined across the pipe and is very different from the nonsymmetrical profiles of fig.F1 , which is due to some other cause.

Fig.H1 shows the profiles obtained with the fine wire mesh and fig.J1 gives a set of profiles obtained with the honeycomb. The behaviour of the profiles obtained with these baffles is very uncertain.

Some conclusions derived from this discussion of the behaviour of artificially produced velocity profiles ~~is given in the following pages.~~

CONCLUSIONS

1. The concave friction baffle is a simple means of producing the fully developed flow in a pipe within a short distance without any elaborate upstream arrangements.
 2. With suitably adjusted lengths and spacing the fully developed profile can be obtained within 10 diameters from the baffle with a low pressure drop coefficient of about unity.
 3. The formula derived for the design of such baffles gives an approximation to that required and further investigation of the formula for the concave friction baffle is necessary.
 4. The velocity profiles and pressure drops along the pipe behind the concave friction baffle are in close though not exact agreement with those expected for fully developed flow. However, perfect agreement seems attainable with baffles of this type.
 5. The turbulence intensities measured are different from those obtained by investigators of fully developed flow in pipes. The reasons for this were investigated but an explanation has not been conclusively established.
 6. Every two-dimensional baffle, such as perforated plates and meshes which were tested, gave an unsymmetrical profile which remained unsymmetrical along the length of the test pipe.
 7. Velocity profiles produced by the convex friction baffle changed rapidly to nearly the fully developed profile at the end of the test pipe, whereas the flat profiles produced by the uniform resistance perforated plate showed little change.
 8. Changes in the velocity profiles are influenced by the nature of the turbulence produced by the baffles and associated with the generated velocity profiles.
 9. Similar profiles produced with different turbulence fields subsequently behave differently.
 10. The inlet length required for producing a lasting fully-developed profile depends not only on the intensity of turbulence but also on the nature of the turbulence produced at the inlet.
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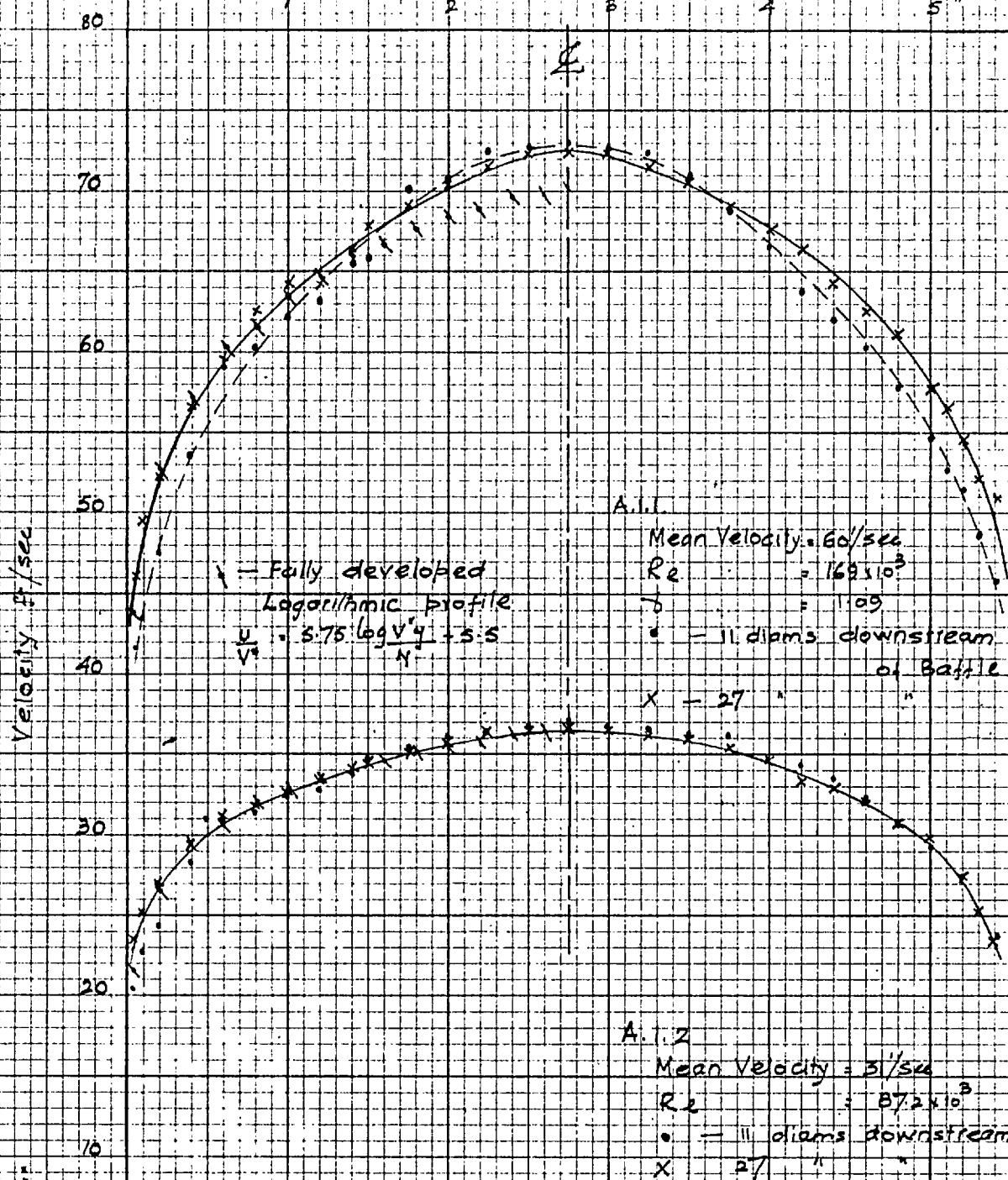


Fig A.1 Velocity Profiles in pipe
with CONCAVE FRICTION BAFFLE

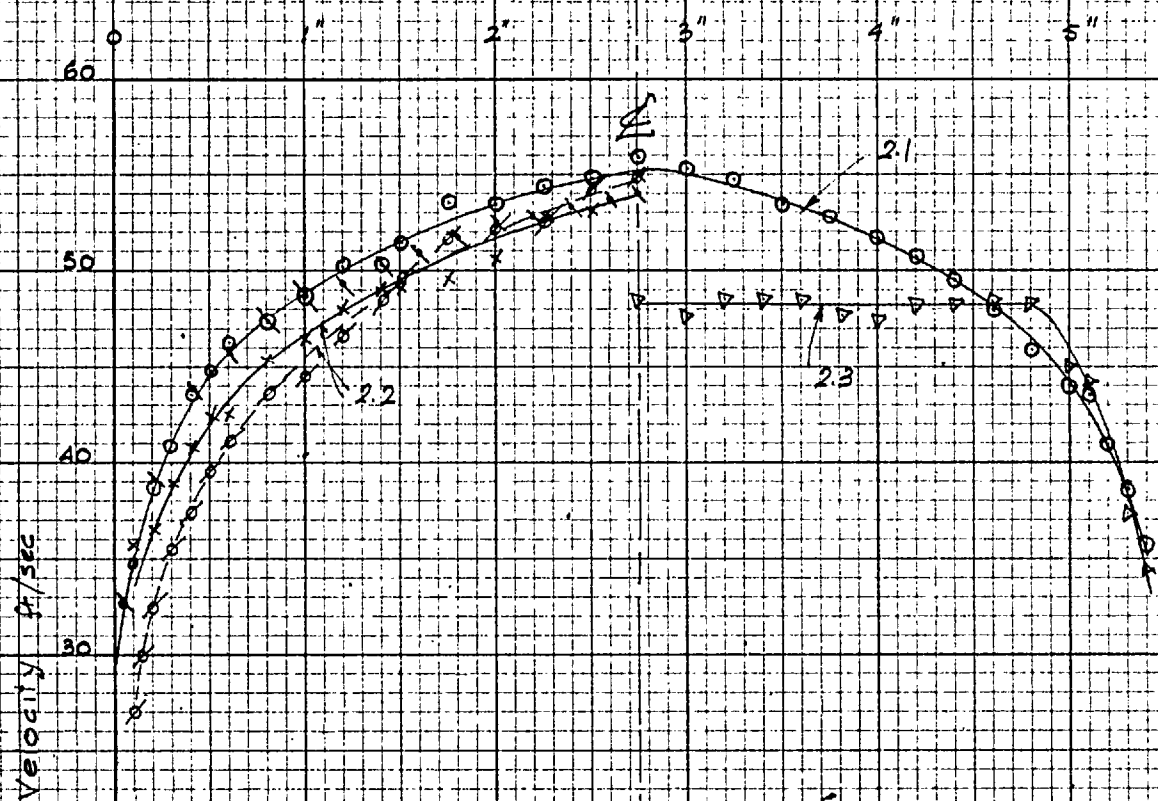


Fig A.2 Velocity Profiles in pipe
with CONCAVE FRICTION BAFFLE

$$\text{Mean Velocity} = 45.8 \text{ ft/sec}$$

$$Re = 129.1 \times 10^3$$

A.2.1 ○ 26 diams downstream of Baffle

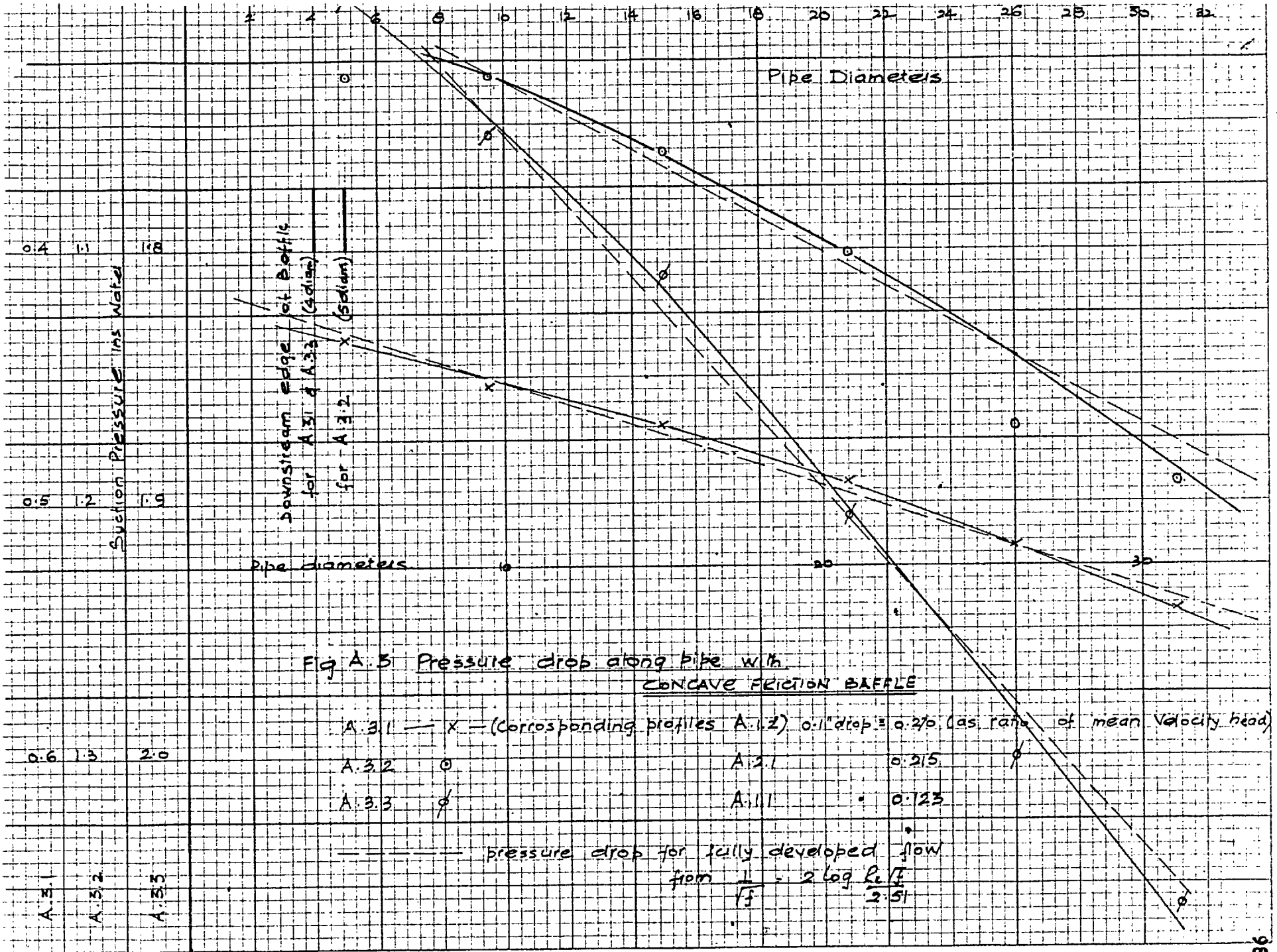
A.2.2 Hot Wire traverse X — 26 diams
φ — 10 "

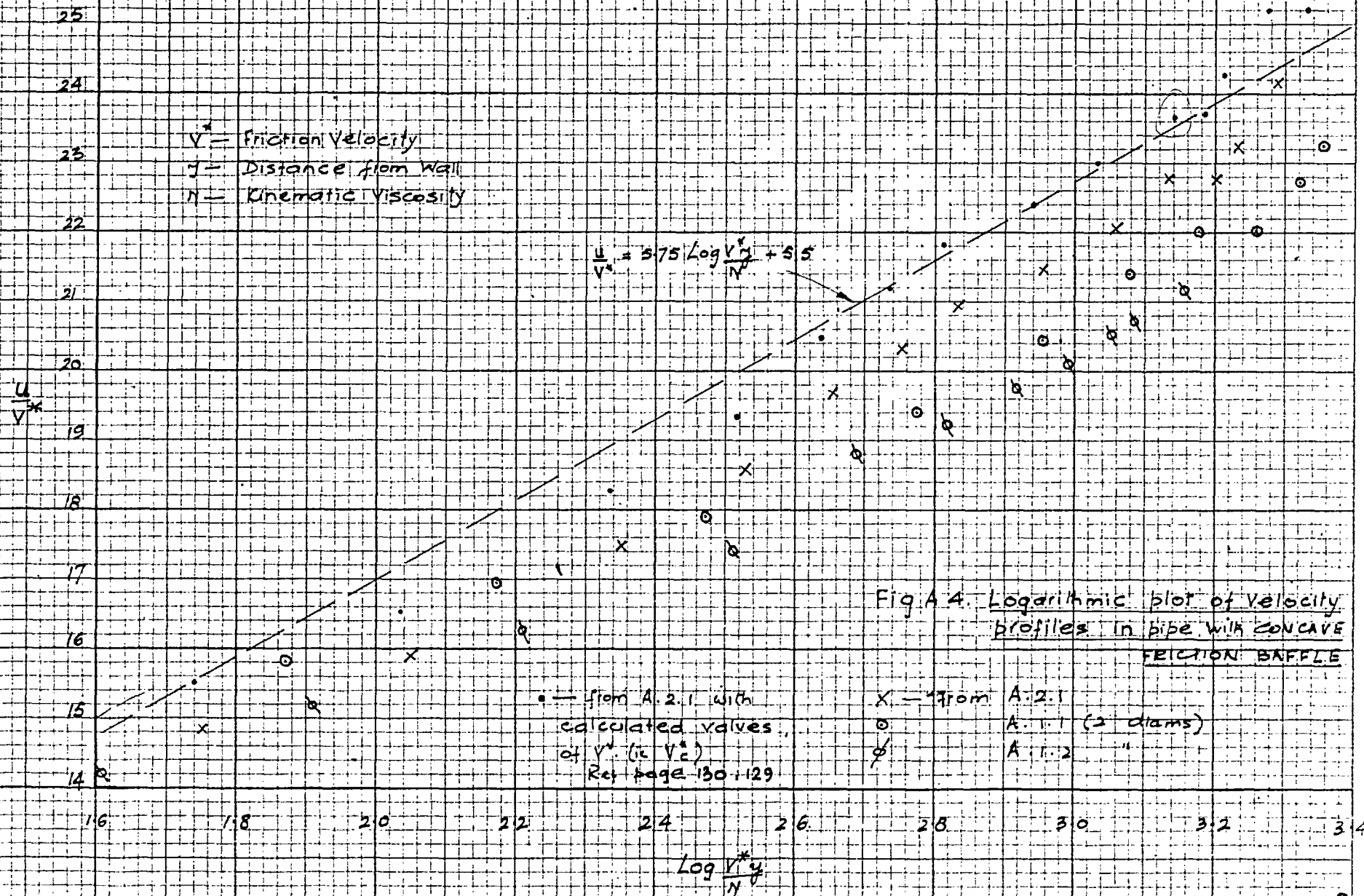
A.2.3 "

Δ — Upstream end of Baffle

† — Fully developed logarithmic profile

$$\frac{u}{V_*} = 5.75 \log \frac{V_*}{V} + 5.5$$





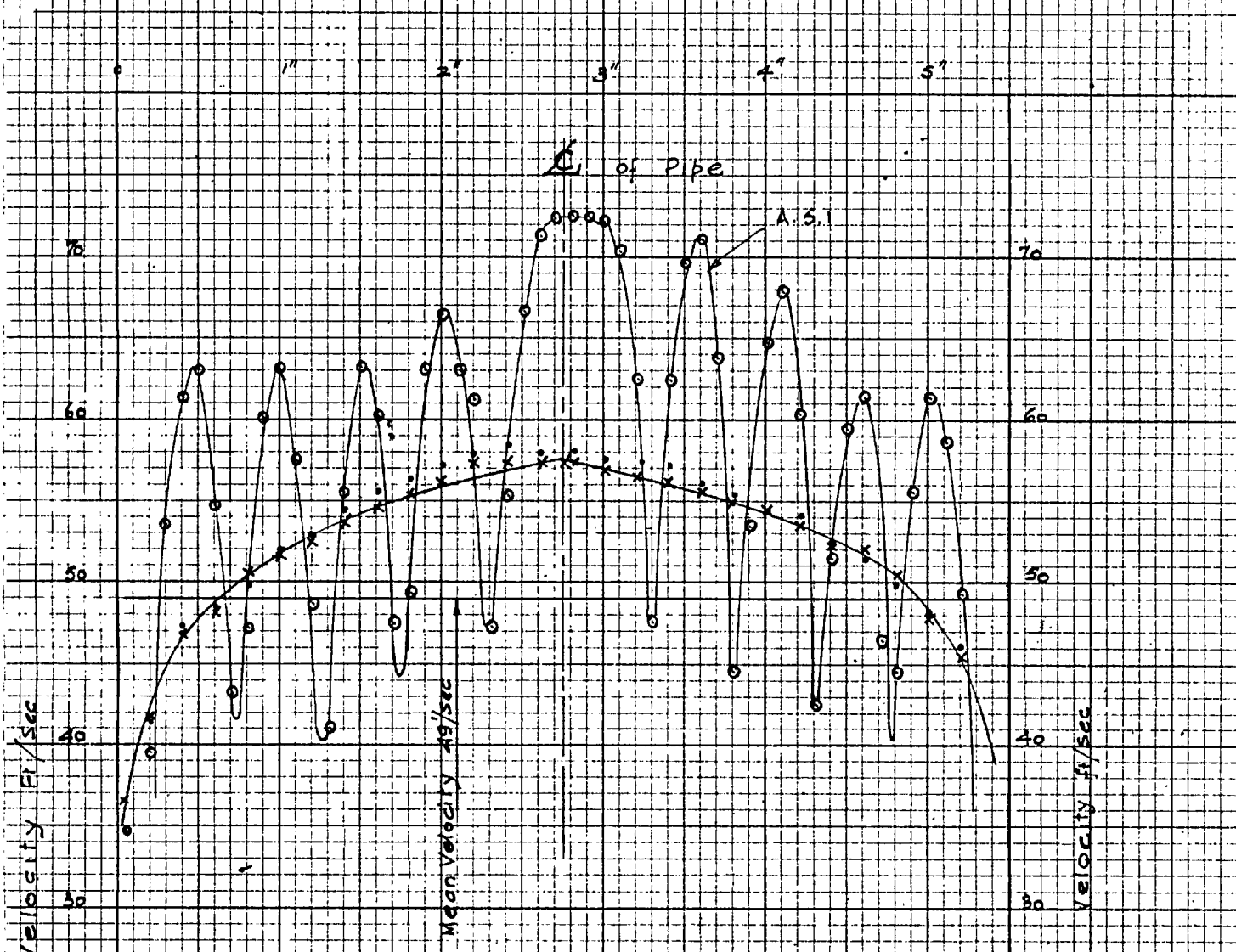


Fig. A.5 Velocity profiles in pipe
with Concave FRICTION BAEFLE

$$U = 49/\text{sec}$$

$$Re = 148 \times 10^3$$

A.S.1 — $\circ$ 2" behind baffle

A.S.2 — $\cdot$ 10.3 diams downstream

A.S.3 — $\times$ 26.3 " "

Velocity ft/sec

Velocity ft/sec

0 1" 2" 3" 4" 5"

70

60

50

40

30

60

50

40

30

A.6.1

o - 1 diam downstream

• " "

x 27 " "

A.6.2

o - 2 diams downstream

Fig A. 6 Velocity profiles in pipe with concave FRICTION BAFFLES

Mean Vel. $\bar{u} = 48.5 \text{ ft/sec}$

$Re = 136.8 \times 10^3$

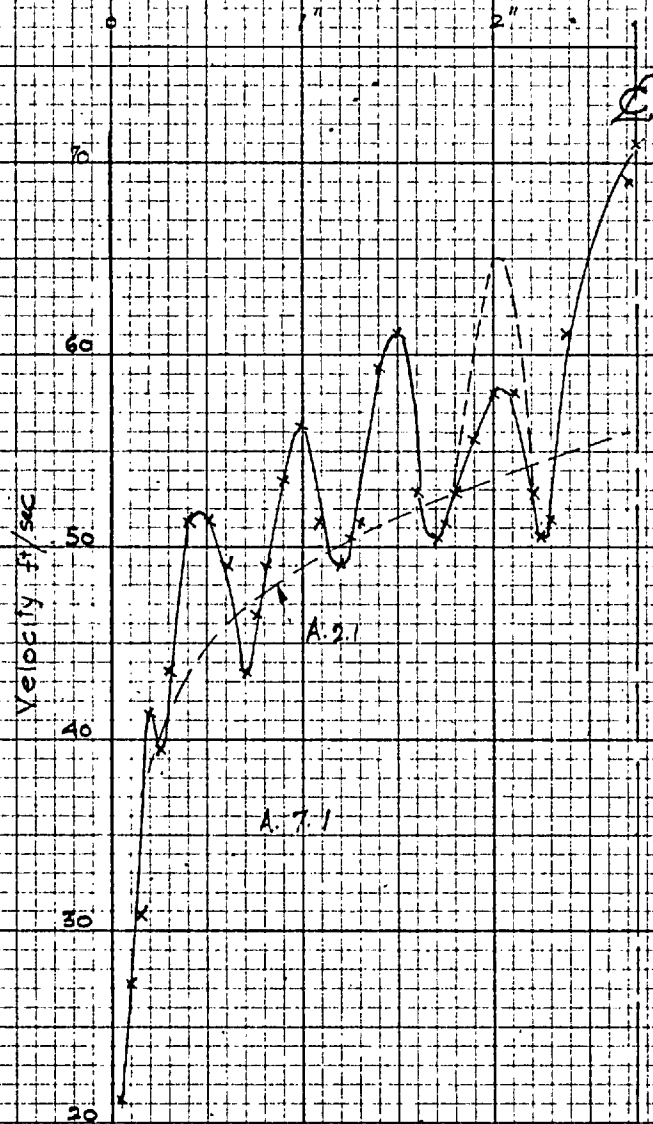


Fig A.7

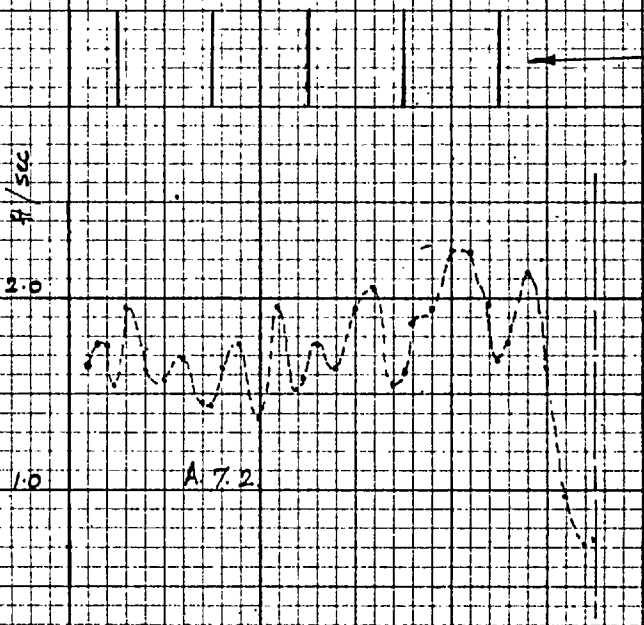
Hot Wire traverse in
pipe 2" downstream
of CONCAVE FRICTION
BAFFLE

Flow adjusted to
Mean Velocity = $35.8/sec$
 $Re = 129.1 \times 10^3$
(same as Fig. A.2.1)

Mean Velocity
from A.7.1 = $47.9/sec$.

u' — turbulent fluctuation
in direction of
flow

Turbulence intensity
in direction of flow
 $\sqrt{u'^2}$



Position of
baffle cylinders

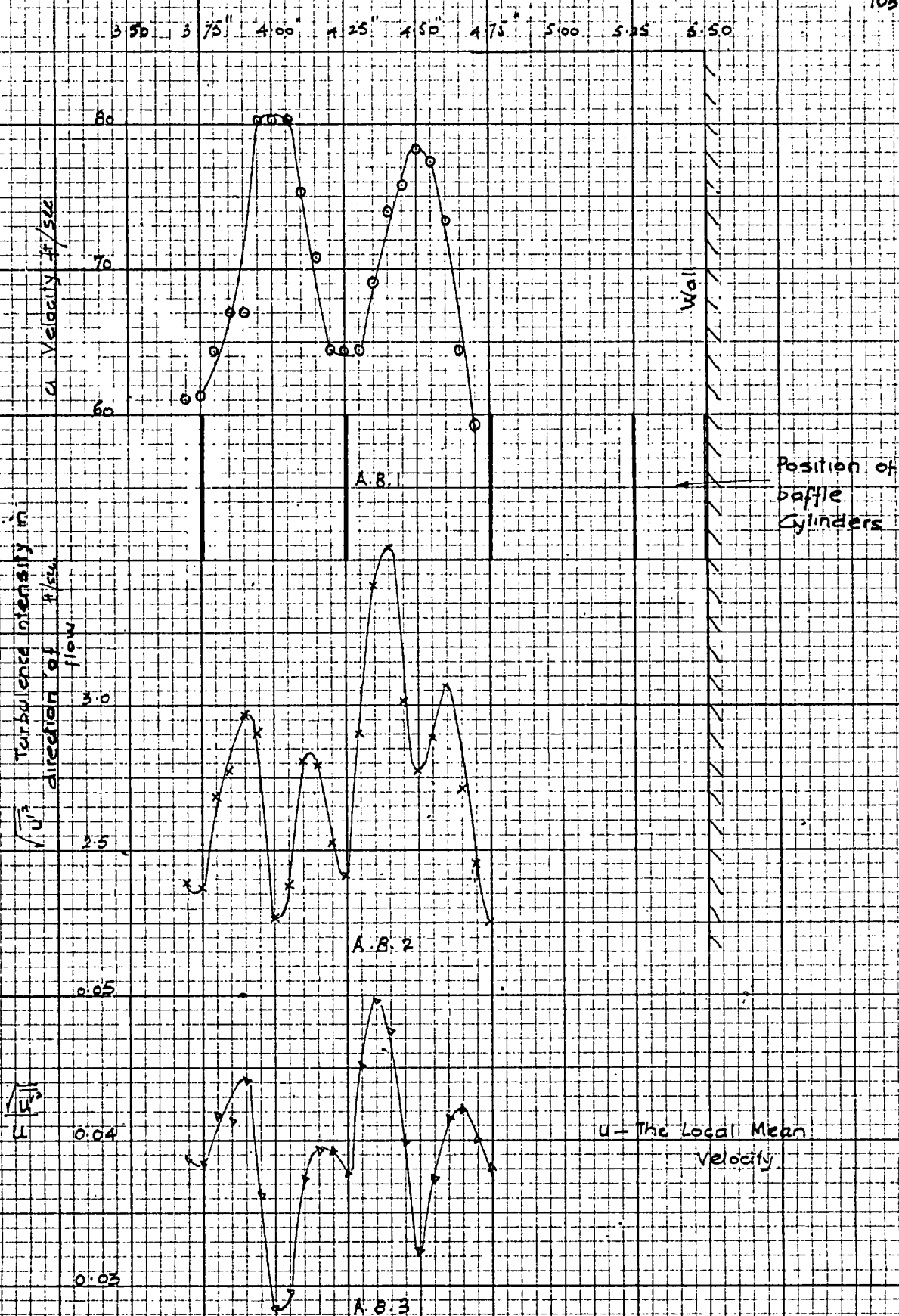


Fig. A.8 HOT WIRE TRAVERSE 2 BEHIND CONCAVE FRICION (CORRESPONDING TO TWO SPACES OF BAFFLE

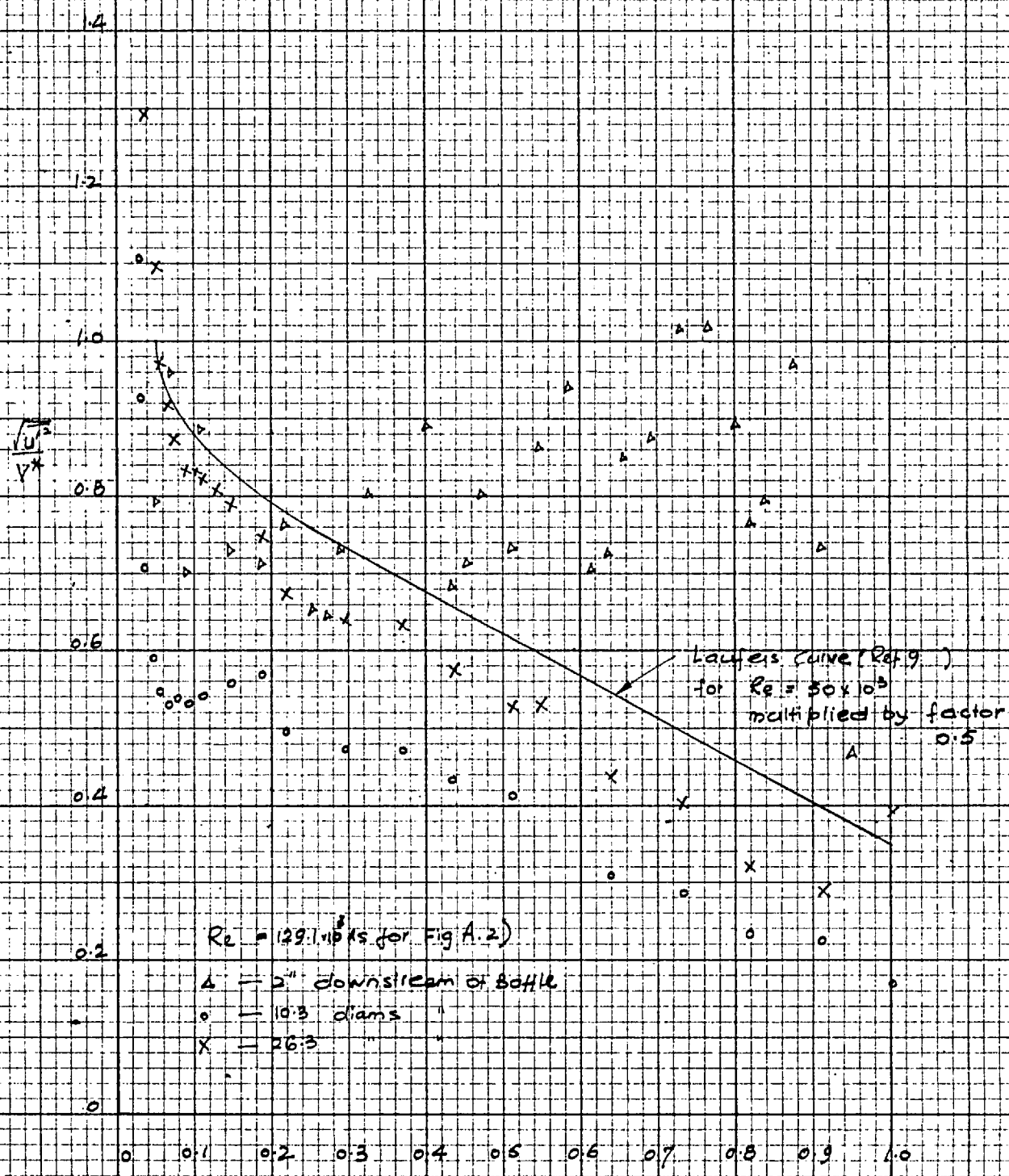
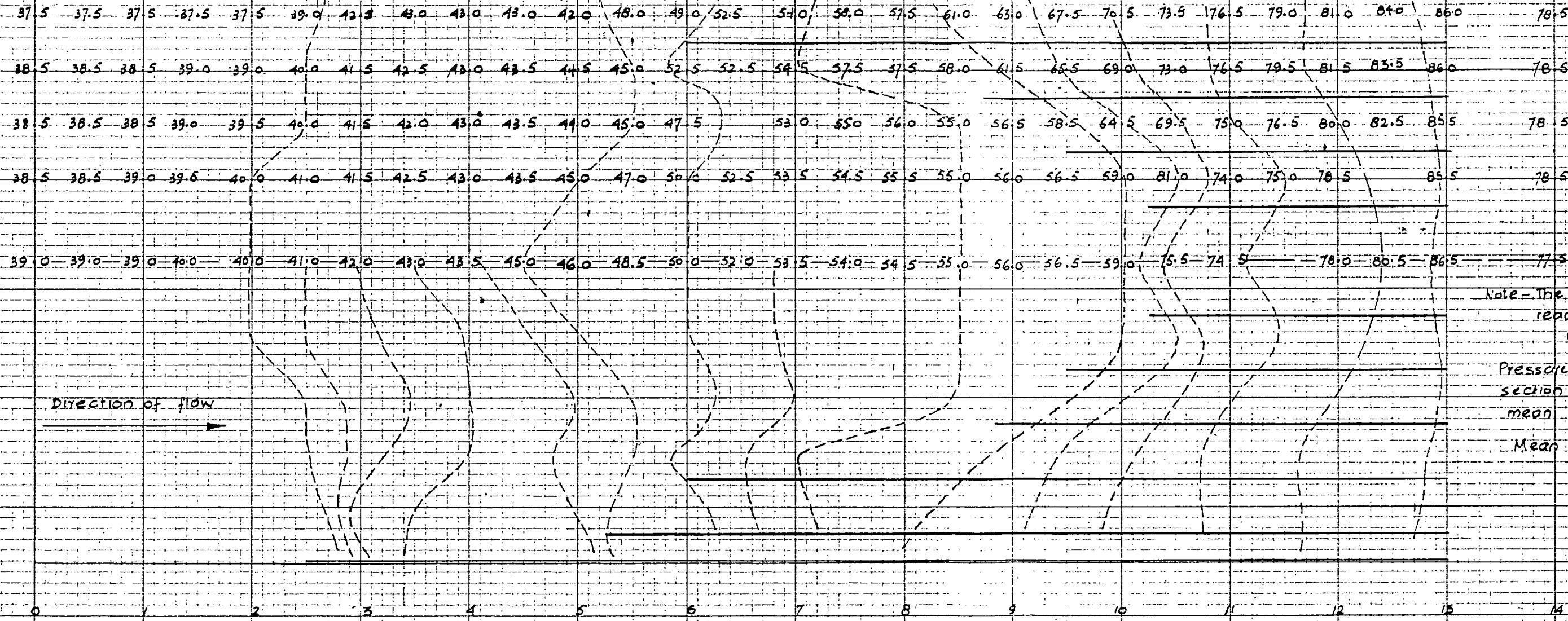


Fig. A.9 Intensity of Turbulent fluctuation in direction of flow in pipe with concave friction baffle

$\sqrt{u'^2}$ — Turbulence Intensity in direction of flow.
 v^* — Friction Velocity
 y — Distance from Wall
 r — Radius of Pipe



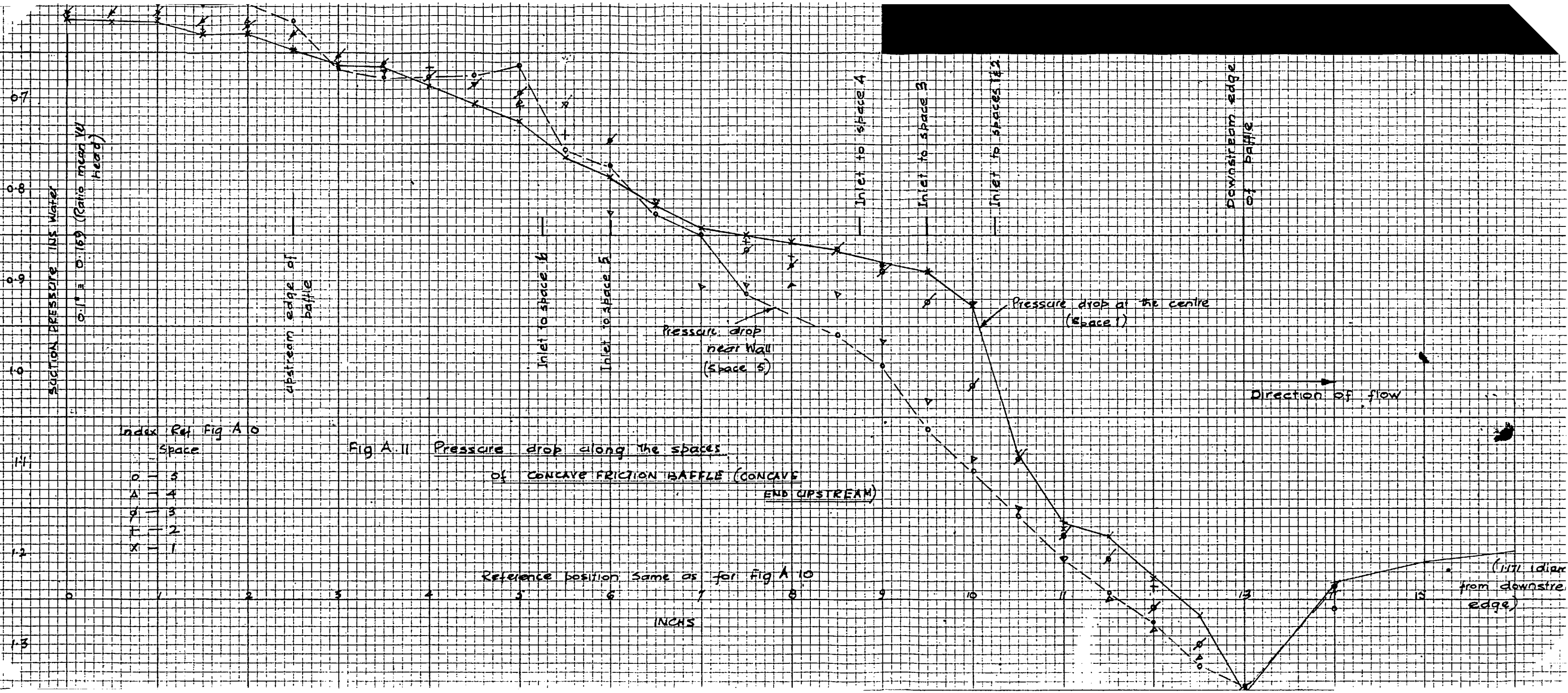
Index for Fig A.10
Space 5

Note - The numbers shown are the reading on an alcohol manometer in mm inclination 1:2

Pressure Drop between Contours (top section = 0.1375' water = 0.265 (ratio to mean velocity head)

Mean Velocity = 51.1'/sec
 $Re = 145.6 \times 10^3$
 ϕ (with maximum pressure difference) = 1.145

Fig A.10 Pressure Contours for CONCAVE FRICTION BAFFLE (CONCAVE END UPSTREAM)



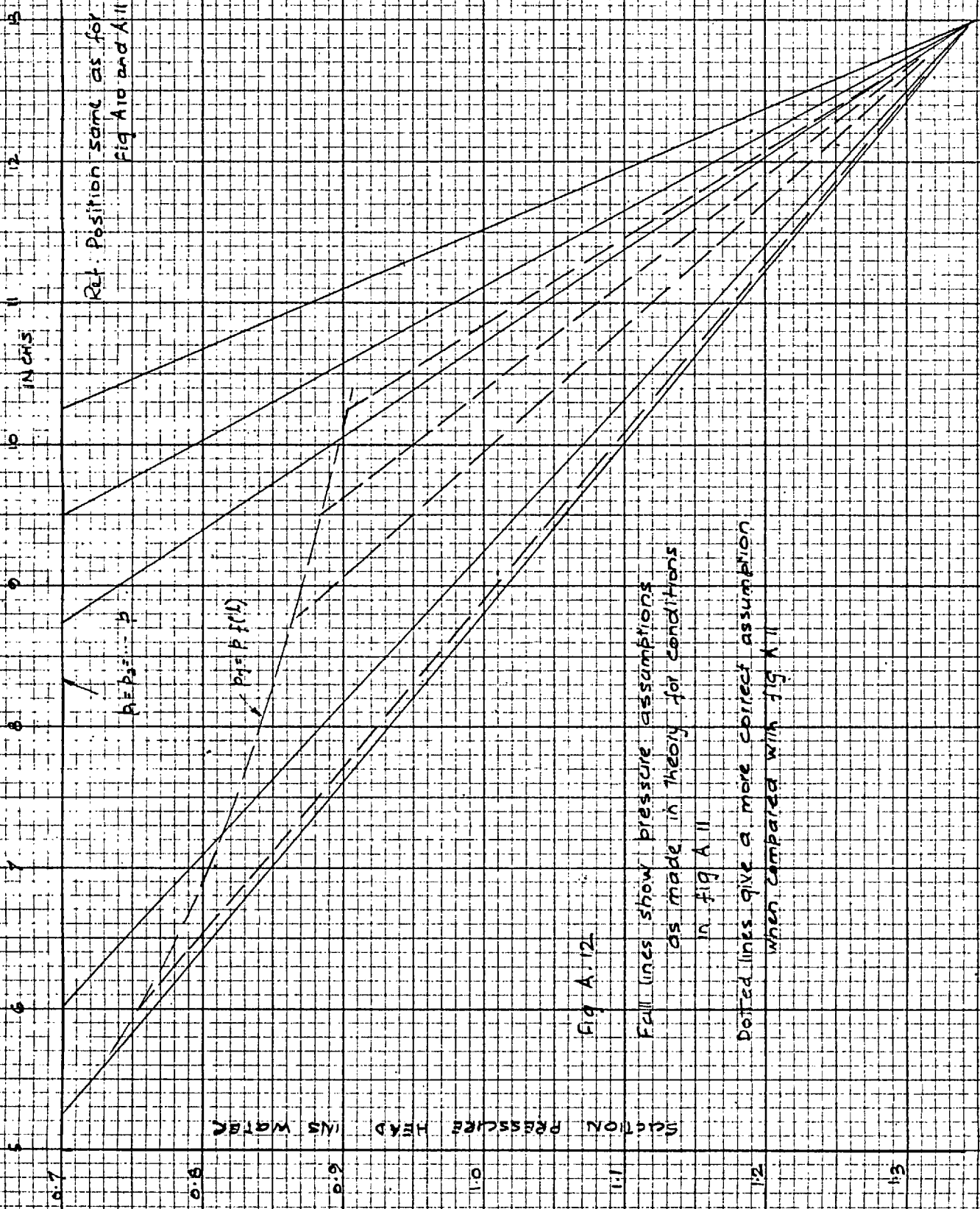
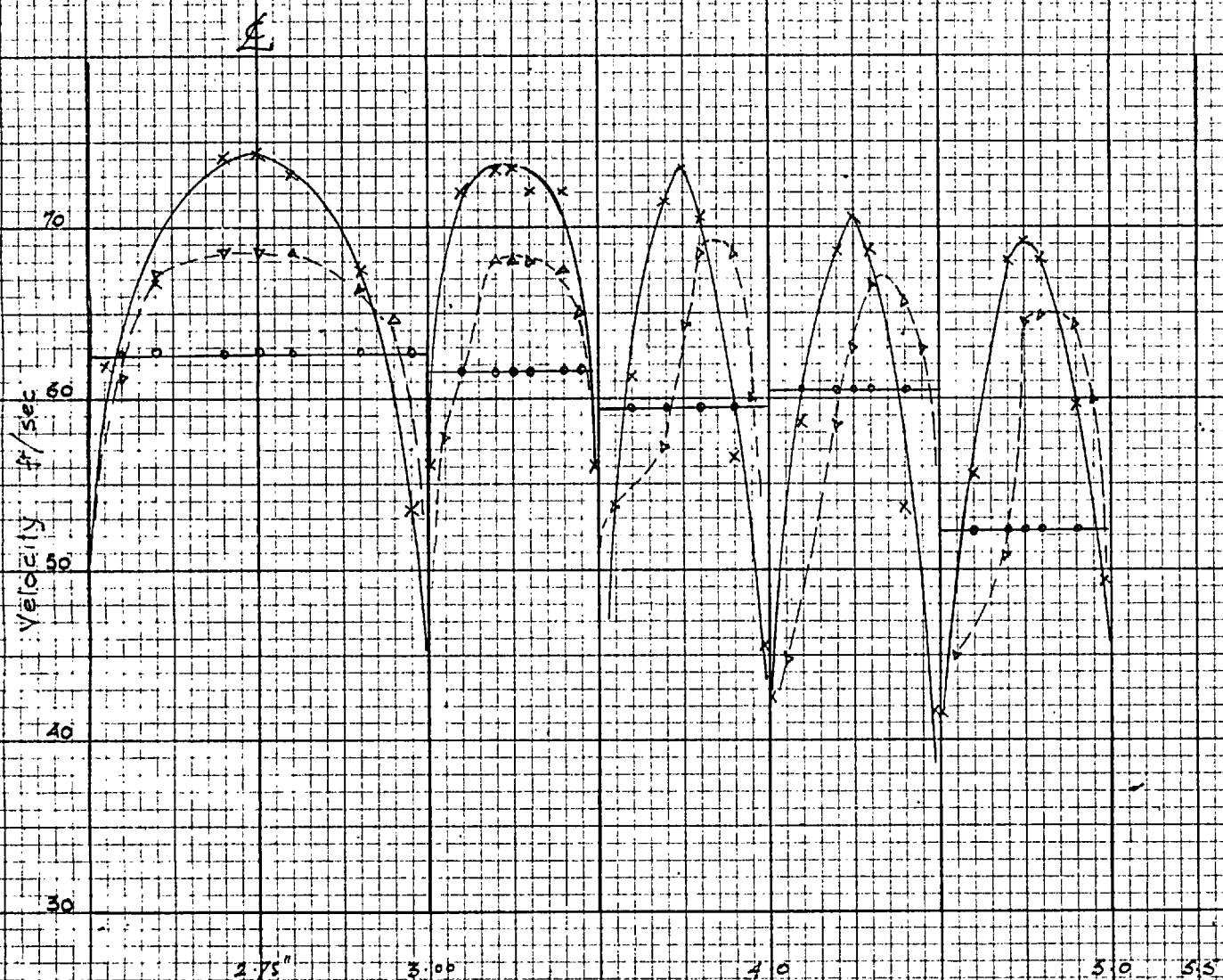


Fig A.12

Fall lines show pressure assumptions as made in theory for conditions in fig A.11

Dotted lines give a more correct assumption when compared with fig A.11



X Mean Velocity of Inlet

	2.75"	3.00"	4.0"	5.0"	5.5"
A " 1.5" from exit inside baffle	62.5/sec	61.5/sec	59.4/sec	60.4/sec	57.2/sec
O " at Exit	64.3	63.0	63.5	58.5	55.0
	66.8	69.6	64.6	60.7	57.0

Fig A.13 Velocity profiles in CONCAVE FRICTION BAFFLE

$$Re = 145.0 \times 10^3 \quad (0.1 \text{ for } A:10)$$

Direction of flow



Exit Velocities
from Fig A.7

48.1

49.0

50.5

52.6

64.0

62.5'/sec

63.9

48.0

63.9

63.9

54.3

51.0

53.6

50.3

48.1

54.10

50.10

46.0

47.7

37.4

10 9 8 7 6 5 4 3 2 1 0

INCHS

Fig. A.14 Inlet and Exit Velocities (hot wire measurements)
for CONCAVE FRICTION BAFFLE
 $Re = 129 \times 10^3$ (as in A.2)

0.14

0.12

0.10

0.08

0.06

0.04

0.02

$\log \frac{U_{in}}{\bar{u}}$

(Ref. page 184.)

U_{in} - Inlet Velocity
 $\bar{u}$ - Mean Velocity
 L_y - Length of Cylinder,
 baffle
 L_{max} - Length of
 longest cylinder

$$\frac{U_{in}}{\bar{u}} = \left(\frac{K L_{max}}{L_y} \right)^{\delta}$$

$$\log \frac{U_{in}}{\bar{u}} = \delta \log \left(\frac{L_{max}}{L_y} \right) + \delta \log K$$

$$\delta = 1/3.45$$

$$K = 0.8706$$

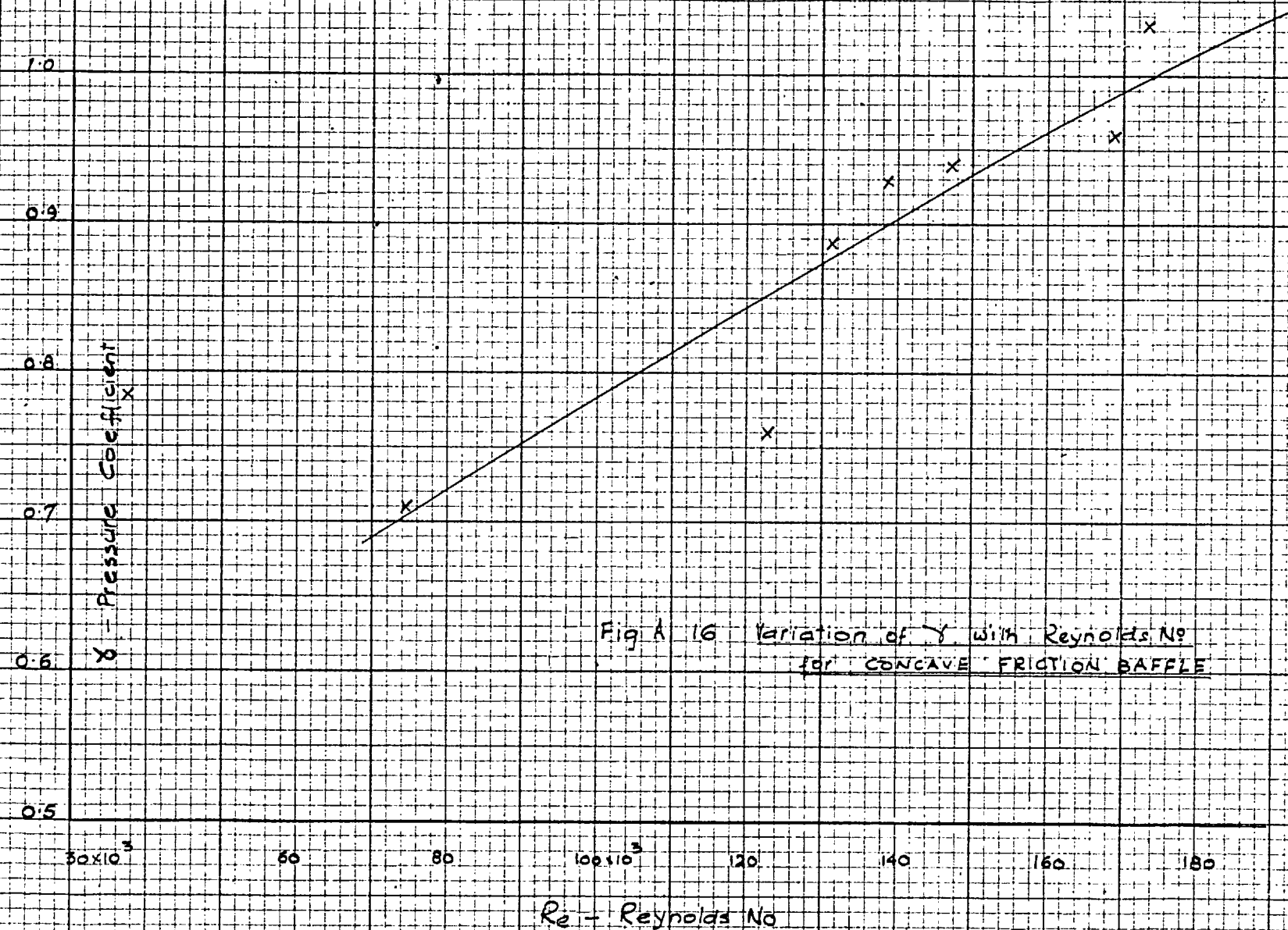
$$\frac{U_{in}}{\bar{u}} = \left(0.87 \frac{L_{max}}{L_y} \right)^{1/3.45}$$

Fig A.15 Variation of Upstream Velocity with
 Cylinder Length for CONCAVE
 FRICTION BAFFLE

(Values from Fig A.14.)

$\log \frac{L_{max}}{L_y}$

0 0.1 0.2 0.3 0.4 0.5 0.6



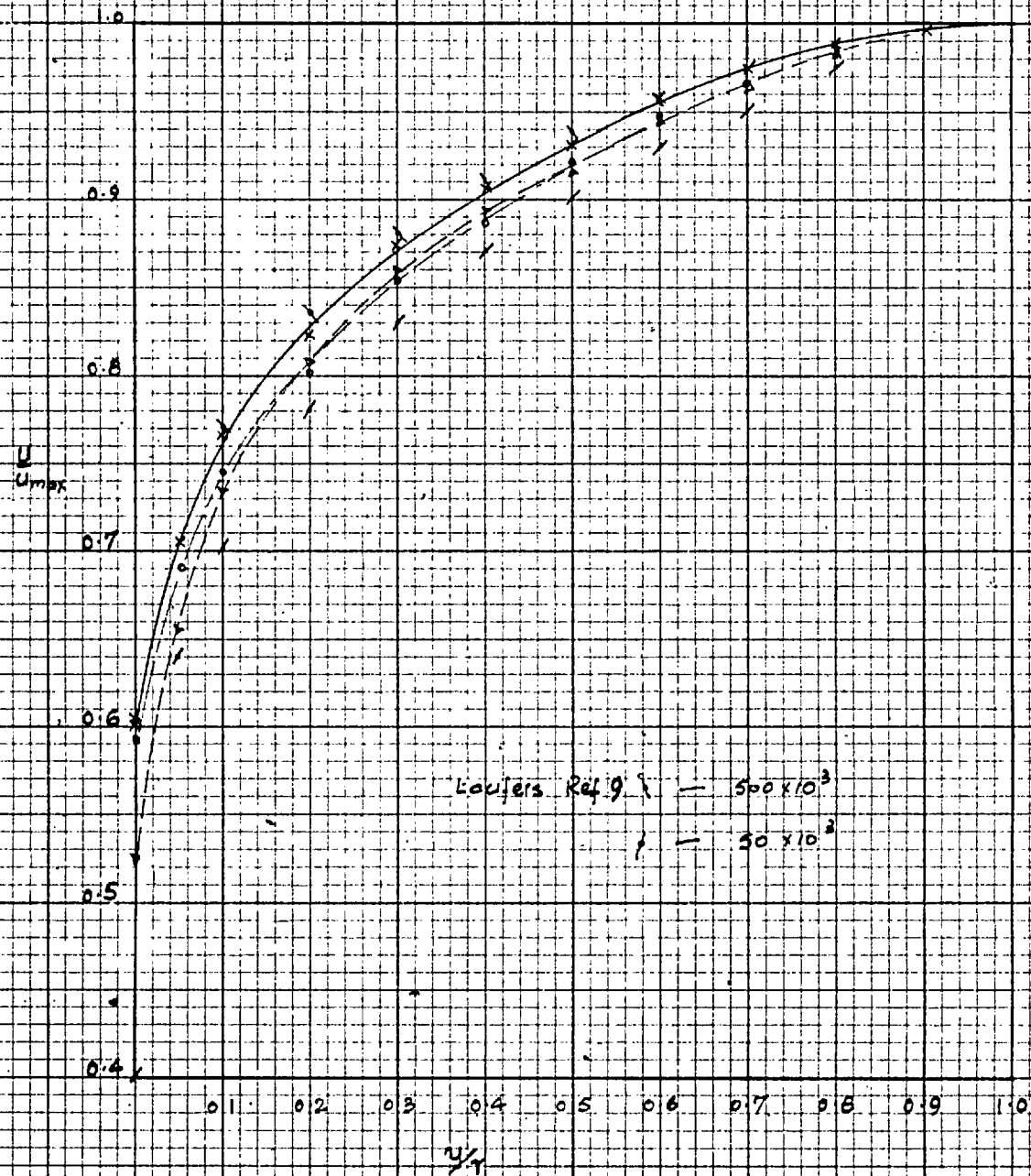


Fig A.17 Nondimensional Plot of Velocity Profiles

x Fig A.1.2
 o Fig A.1.1
 Δ Fig A.2.1

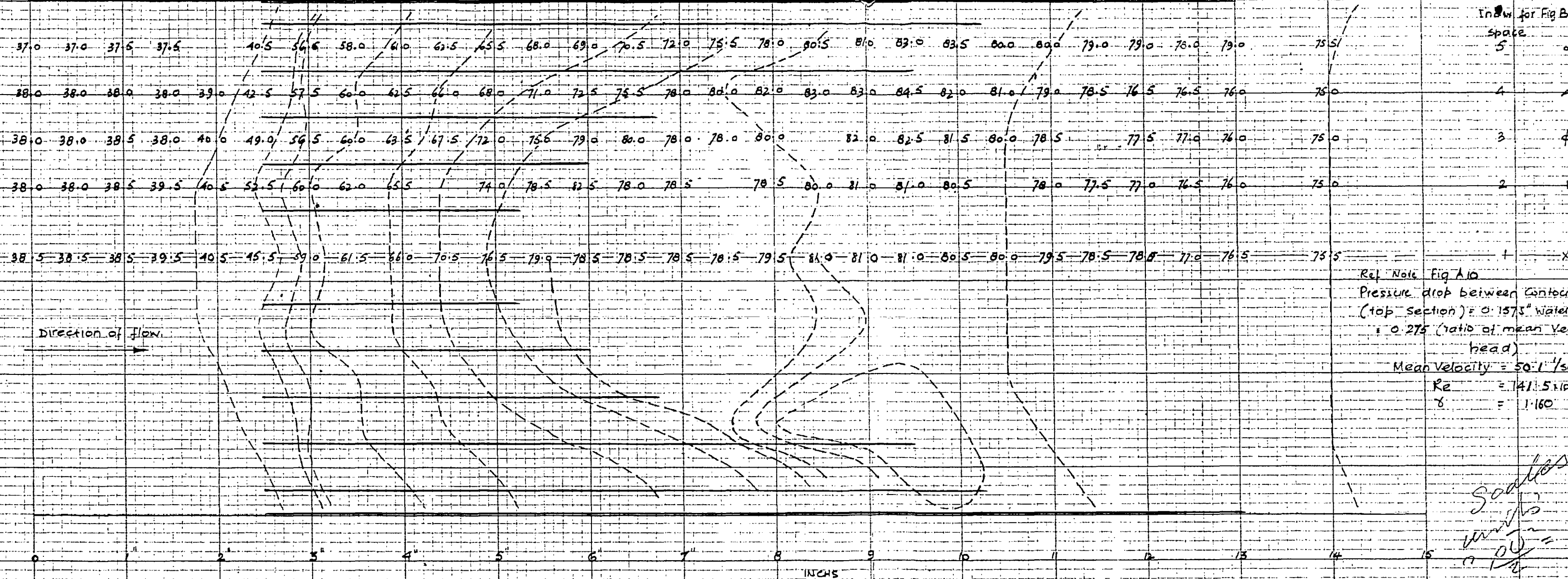


Fig B1 Pressure Contours for Concave Friction Baffle (Concave End Downstream)

Ref. Note Fig A10
 Pressure drop between Contours
 (top section) = 0.1575" water
 = 0.275 (ratio of mean Vel
 head)
 Mean Velocity = 50.1 ft/sec
 $Re = 141.5 \times 10$
 $\gamma = 1.60$

Soak
 units
 $\frac{1}{100} =$
 1.1601

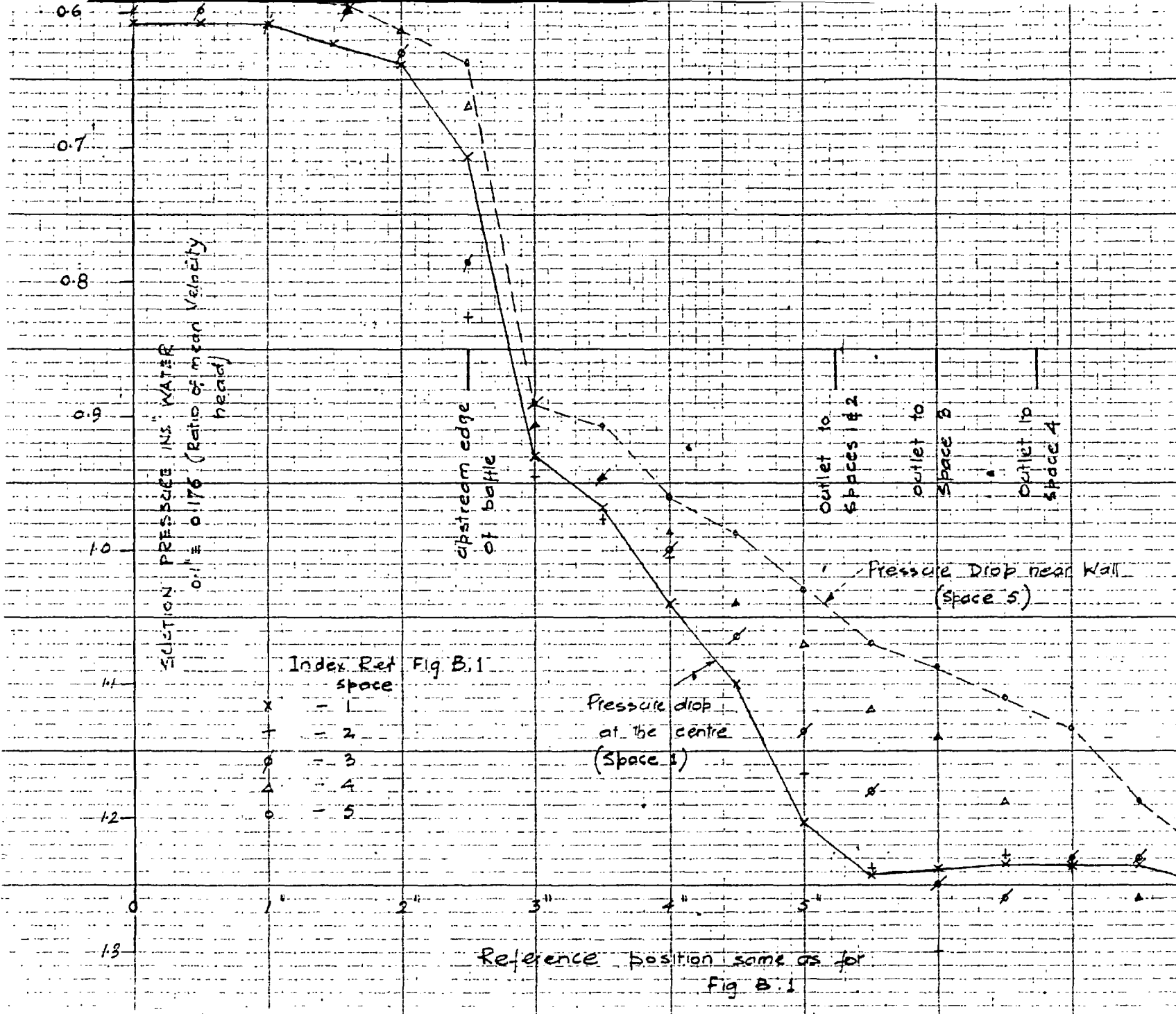
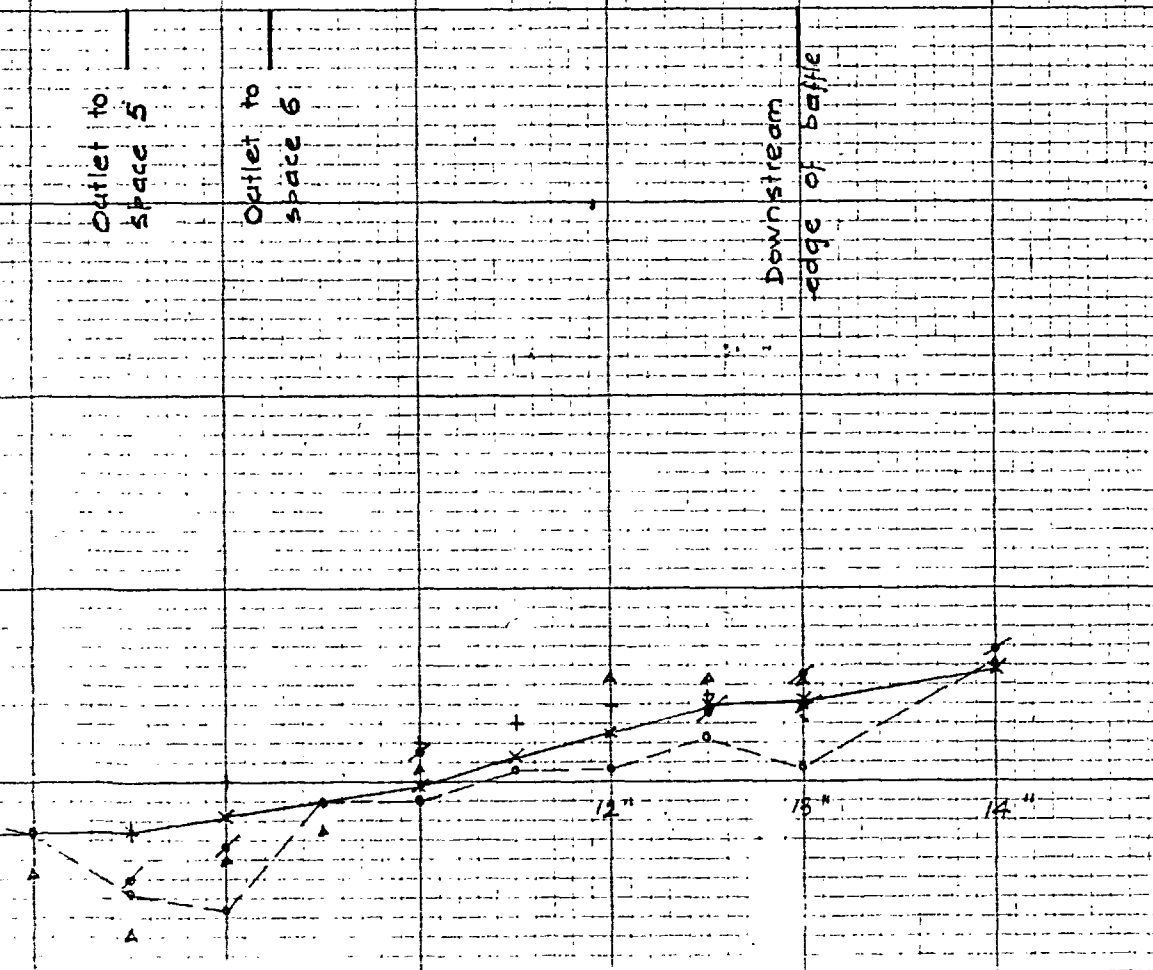


Fig. B. 2. PRESSURE DROP ALONG THE SPACES OF CONCAVE FRICTION BAFFLE (CONCAVE END DOWNSTREAM)



Velocity ft/sec

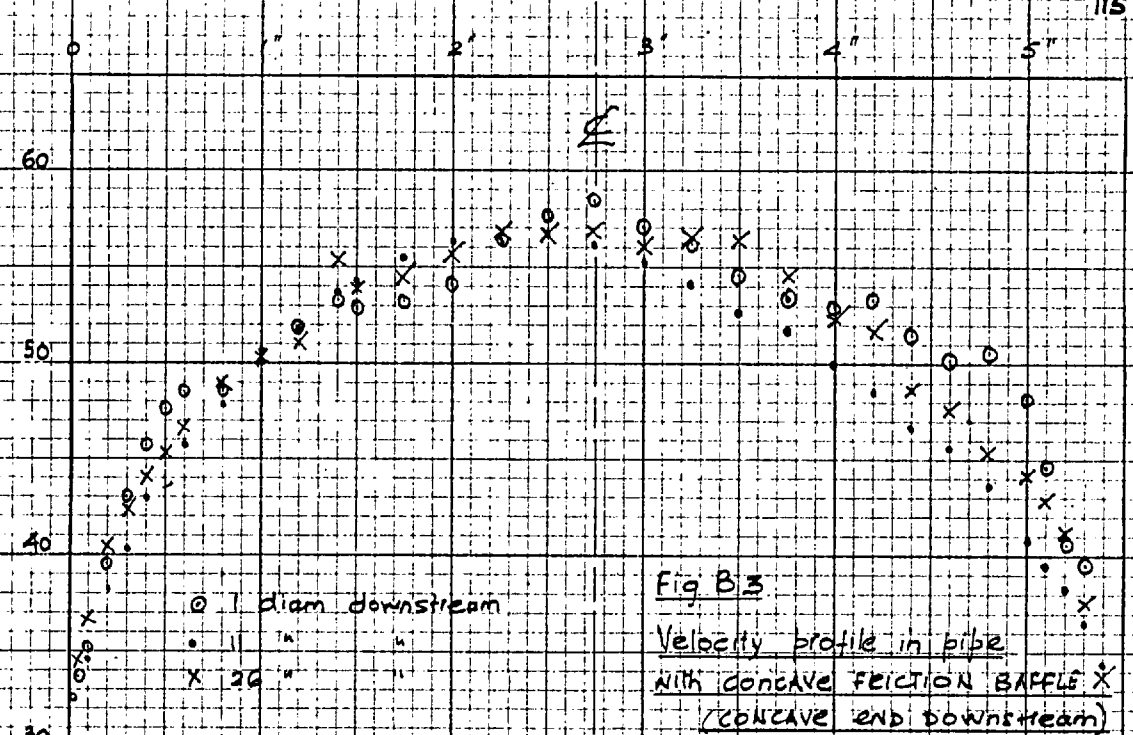


Fig B.3

Velocity profile in pipe
with concave friction baffle
(concave end downstream)

Velocity ft/sec

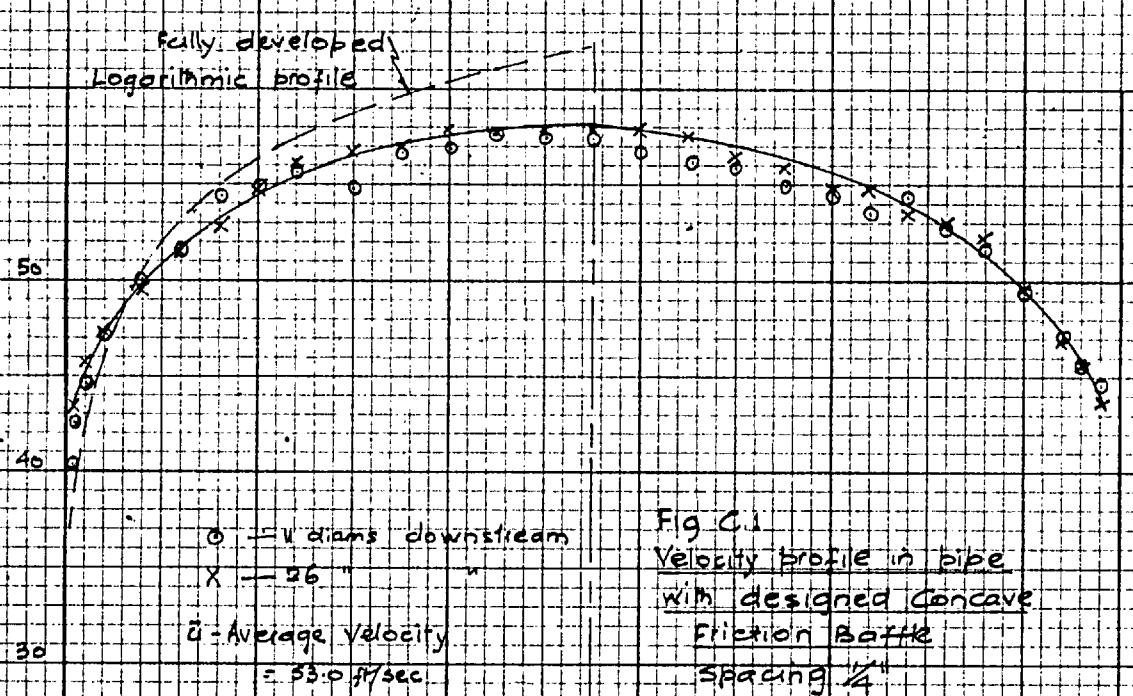


Fig C.1

Velocity profile in pipe
with designed concave
friction baffle
spacing 1/4"

$\bar{u}$ - Average Velocity
= 33.0 ft/sec

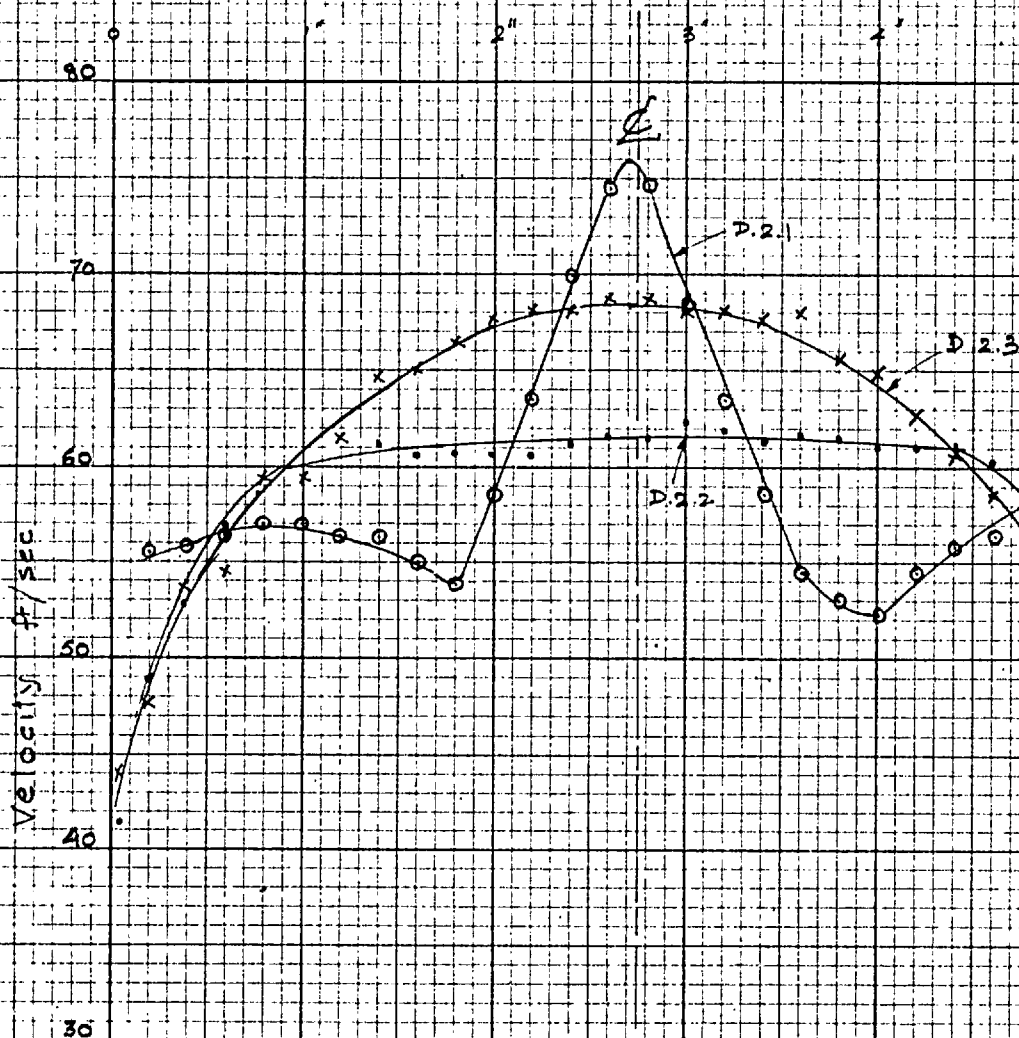


Fig D.2 Velocity profiles in pipe
with CONVEX FRICTION BAFFLE
(CONVEX END UPSTREAM)

$$U = 54.3/34$$

$$Re = 153,103$$

D.2.1	0	3	diam's	downstream of Baffle
D.2.2	.	13	"	"
D.2.3	X	29	"	"

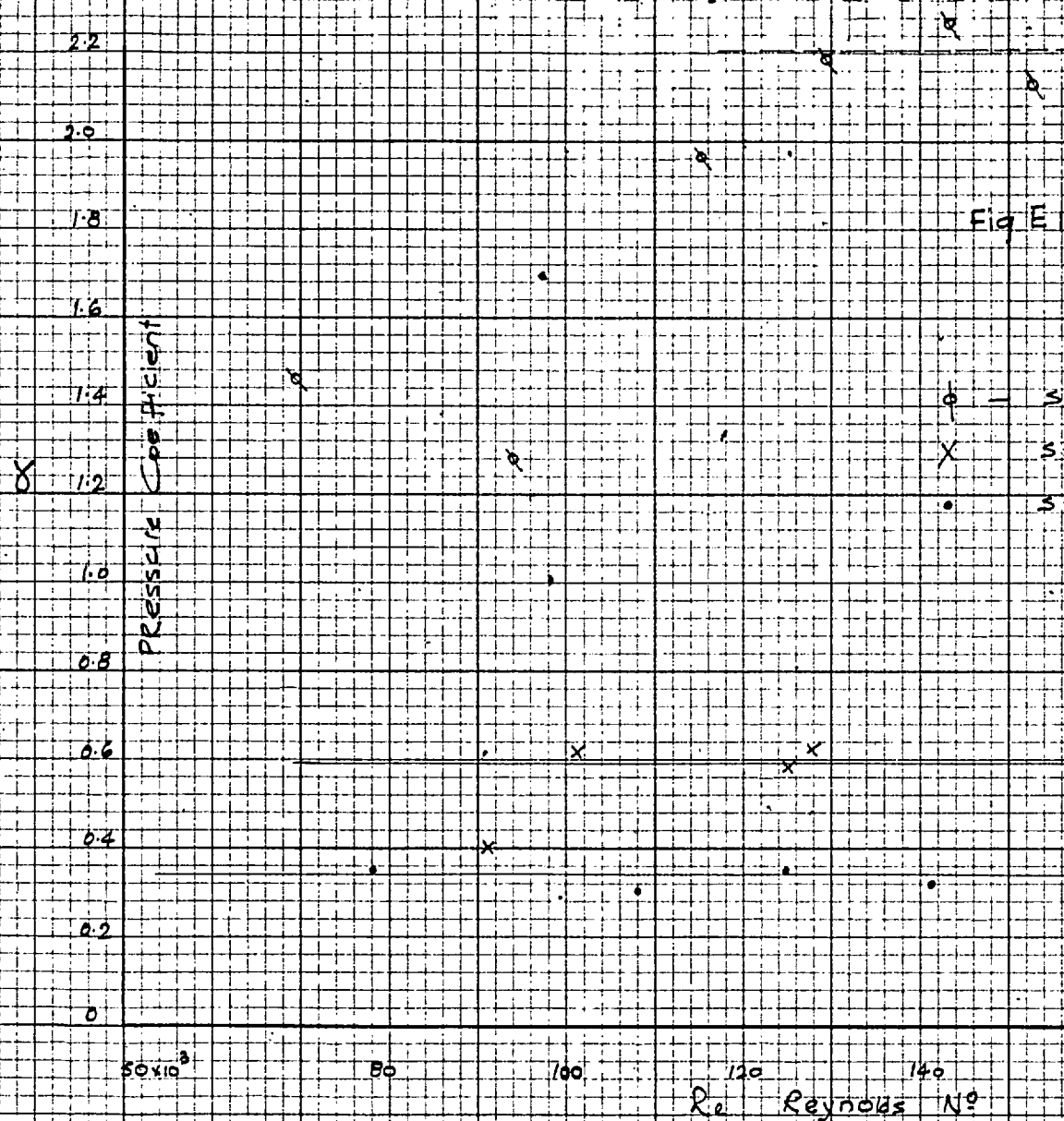


Fig E Variation of Pressure Coefficient γ With Reynold No for Flat Friction

BAFFLE

$$\gamma = \phi \left(\frac{S}{d} \right)^{0.189} \left(\frac{L}{d} \right)^{0.860}$$

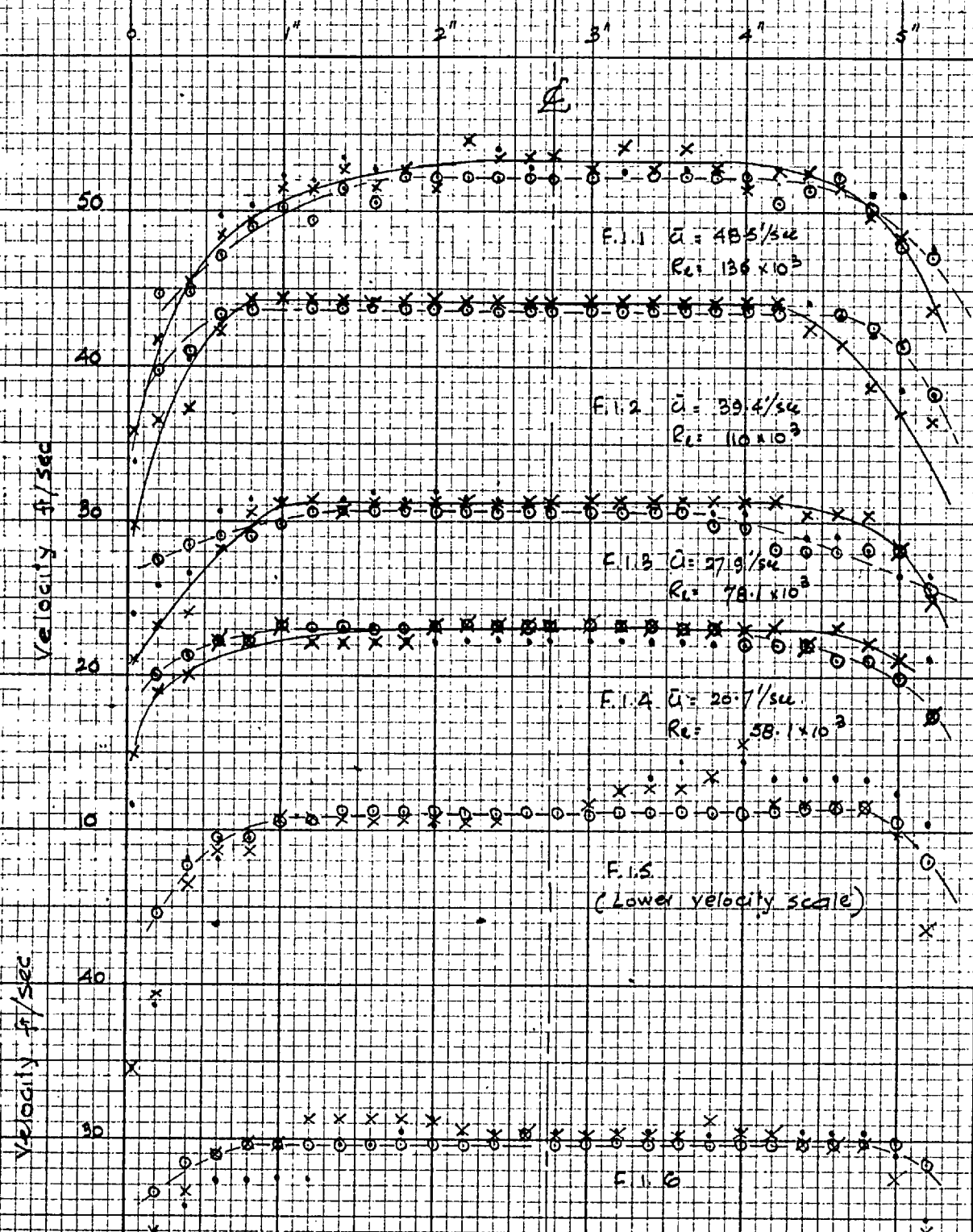
S - spacing
L - Length

ϕ - $S = \frac{1}{4}"$ $L = 4"$ $\gamma = 2.2$ (for higher Values of Re)

X $S = \frac{1}{2}"$ $L = 4"$ $\gamma = 0.535$

$\bullet$ $S = \frac{1}{2}"$ $L = 2"$ $\gamma = 0.342$

$$\gamma = 0.0084 \left(\frac{d}{S} \right)^{1.89} \left(\frac{L}{d} \right)^{0.860}$$



Velocity Profiles in pipe with
Fig F Perforated Plate
(Uniform Resistance)

o - 0.5 diams behind plate
• - 15 " " "
X - 31 " " "

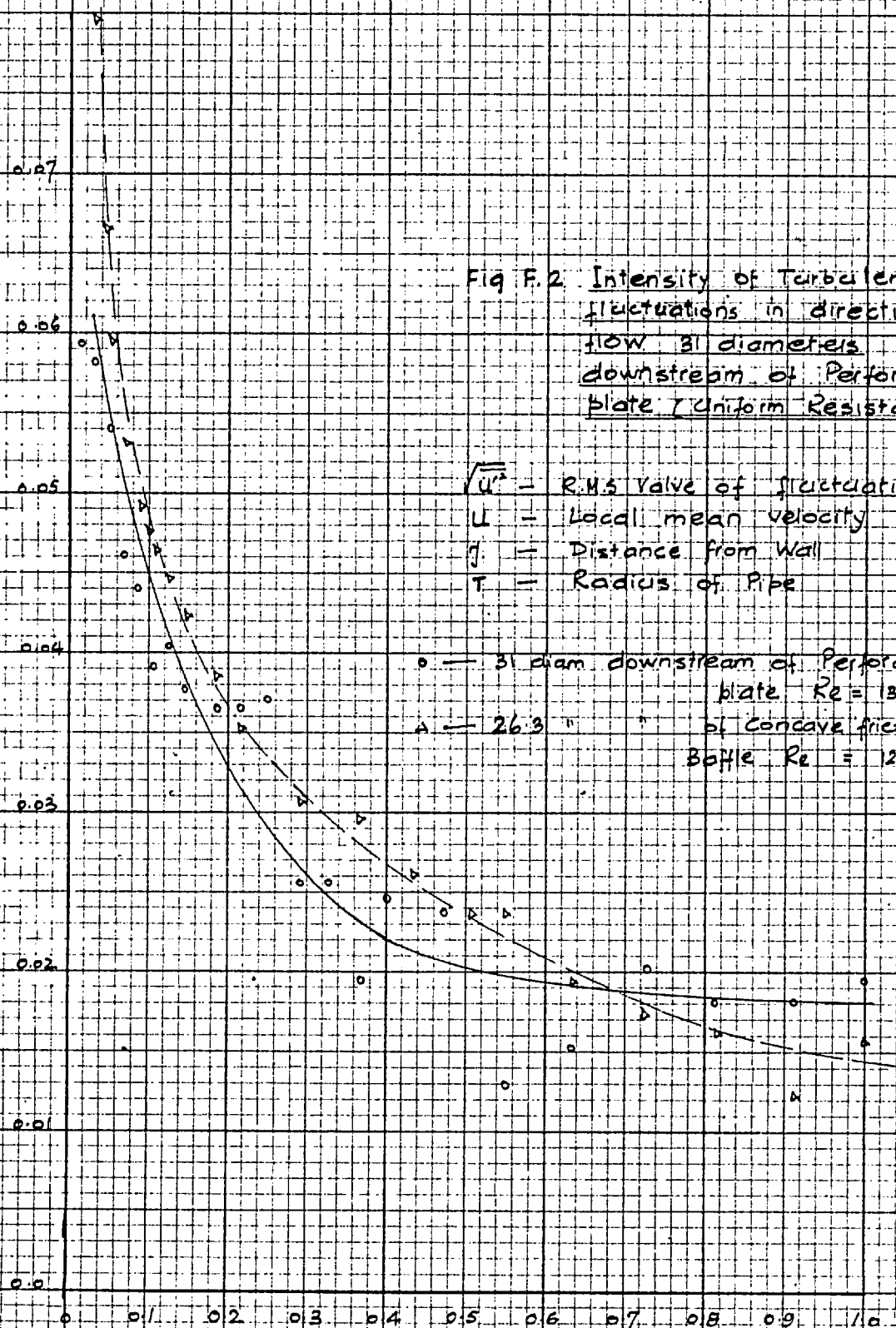
F.1.1 - F.1.4 With plate at Inlet
F.1.4 and F.1.5 " " " inside pipe

$$\frac{\sqrt{u'^2}}{U}$$

Fig F.2. Intensity of Turbulence fluctuations in direction of flow 31 diameters downstream of Perforated plate (Uniform Resistance)

$\sqrt{u'^2}$ = R.M.S. Value of fluctuation
 U = Local mean velocity
 y = Distance from Wall
 T = Radius of Pipe

○ — 31 diam downstream of Perforated plate $Re = 136 \times 10^3$
 △ — 26.3 " " of Concave friction Baffle $Re = 129.1 \times 10^3$



$\frac{y}{T}$

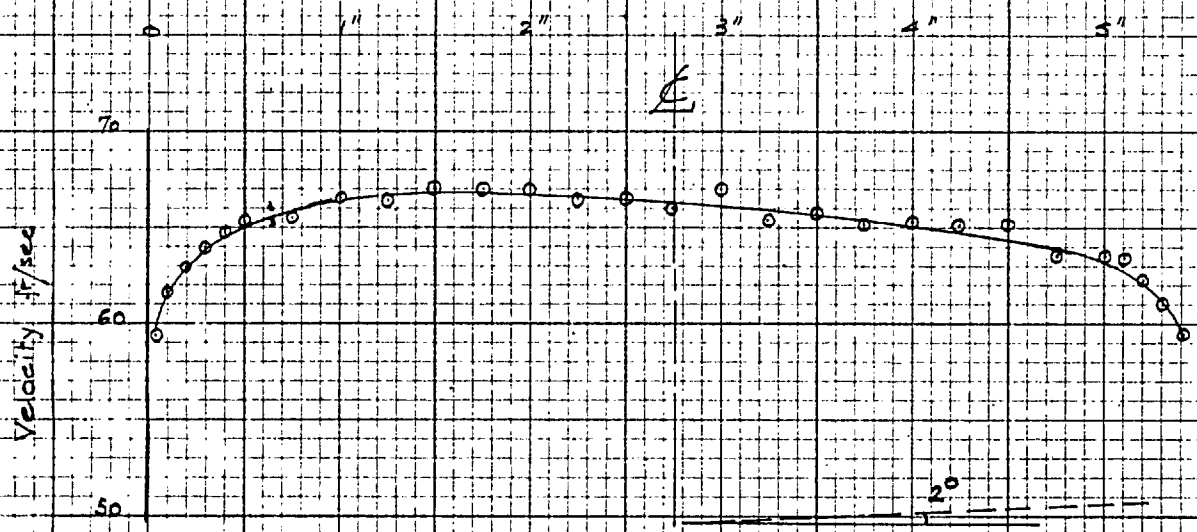


Fig F.3. Velocity Profile in pipe 5 diams downstream
of uniform resistance perforated
plate inclined at 2° to the
stream

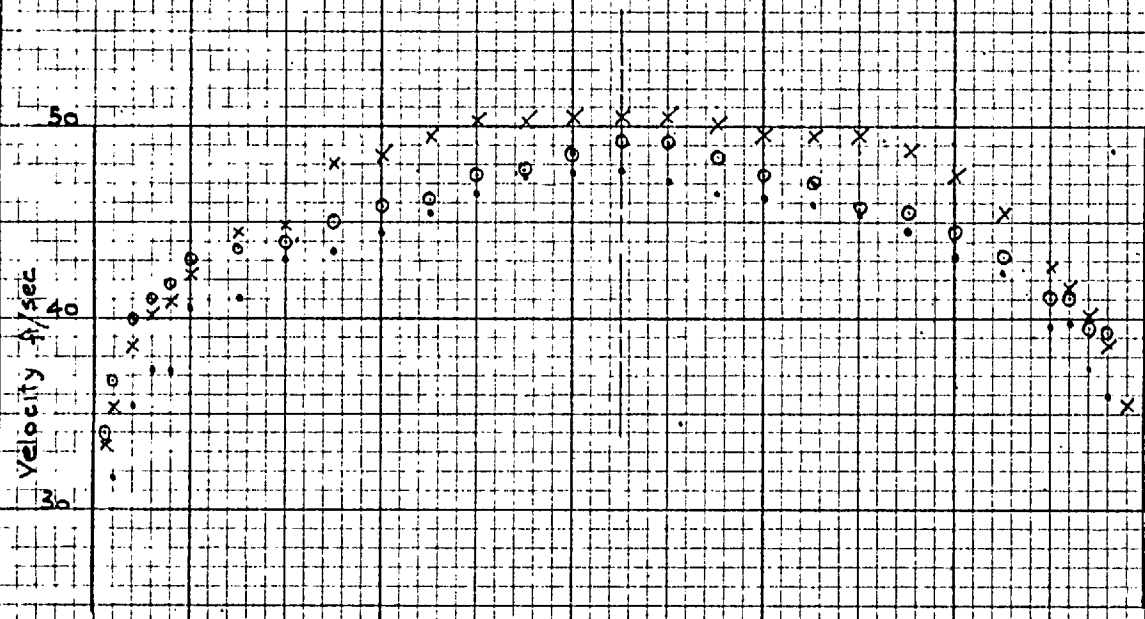


Fig G.1. Velocity Profiles in pipe
with graduated resistance
perforated plate

○ — 5 diams
• — 15 " "
x — 31 " "

Velocity ft/sec

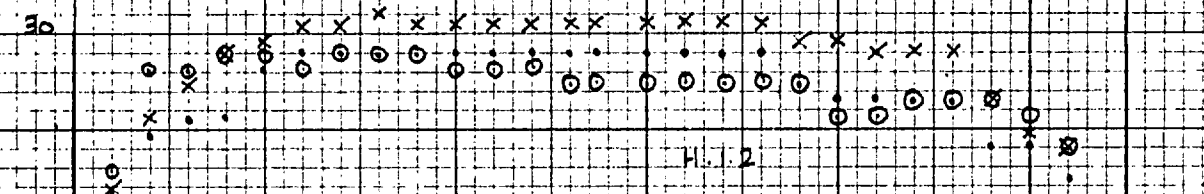
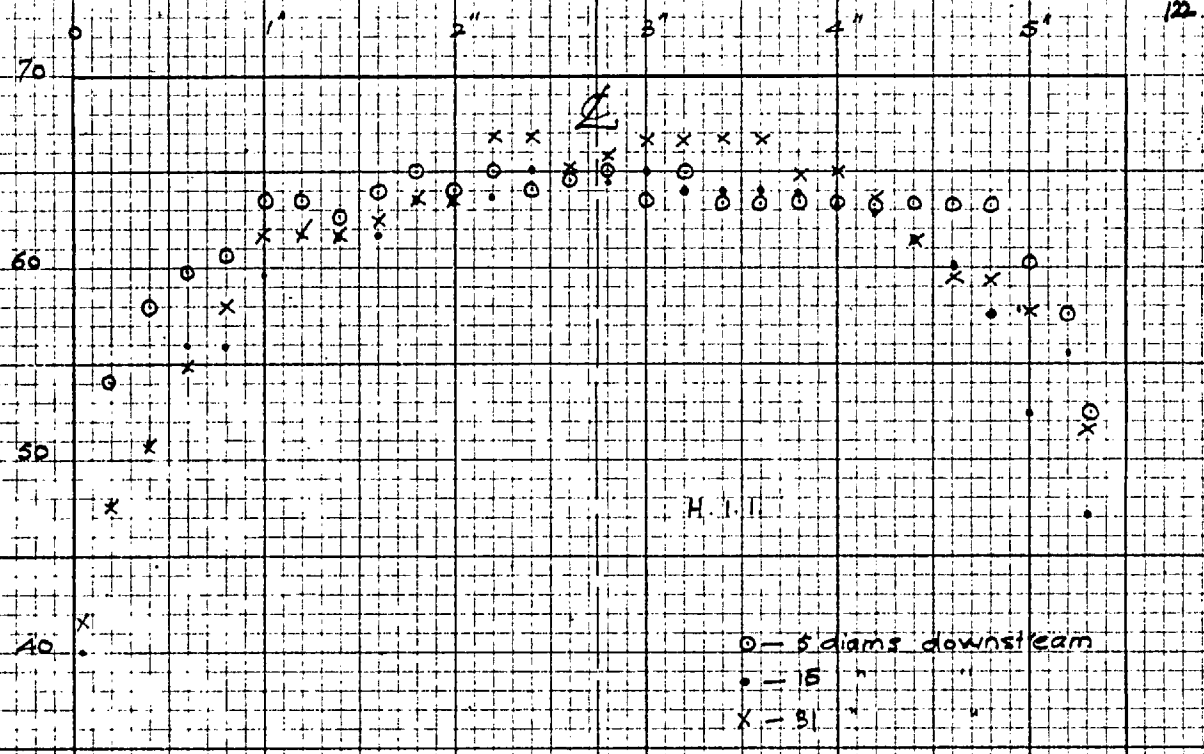


Fig H.1.3 Velocity Profiles in Pipe
 WITH WIRE MESH

H.1.1 Mesh at inlet
 H.1.2 " inside the Pipe

Velocity ft/sec

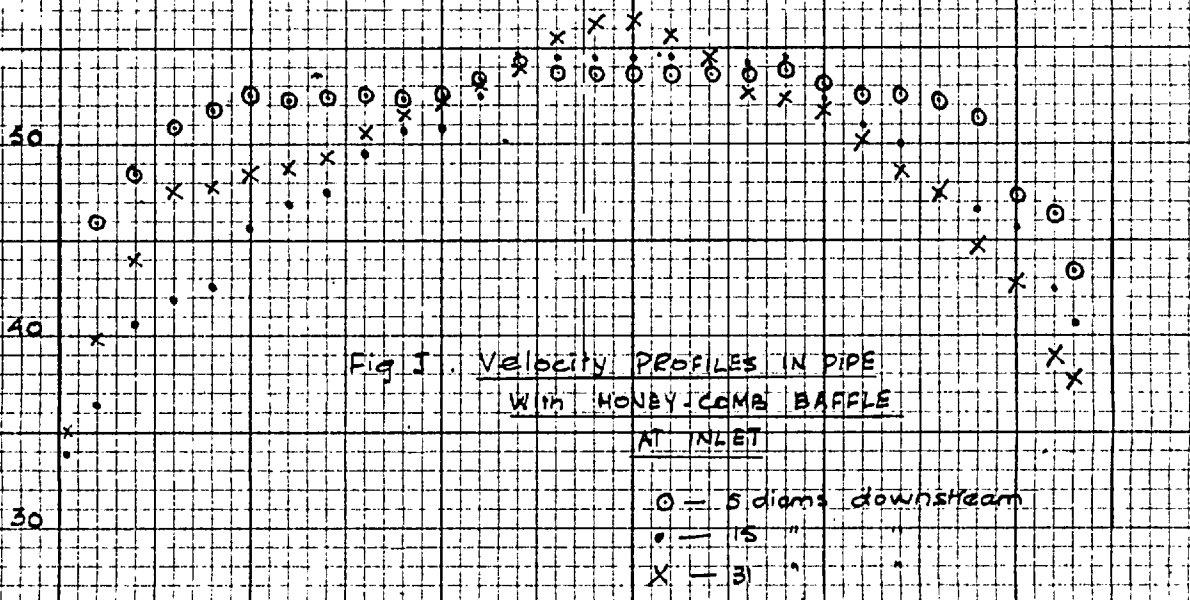


Fig I Velocity Profiles in Pipe
 WITH HONEY-COMB BAFFLE

AT INLET

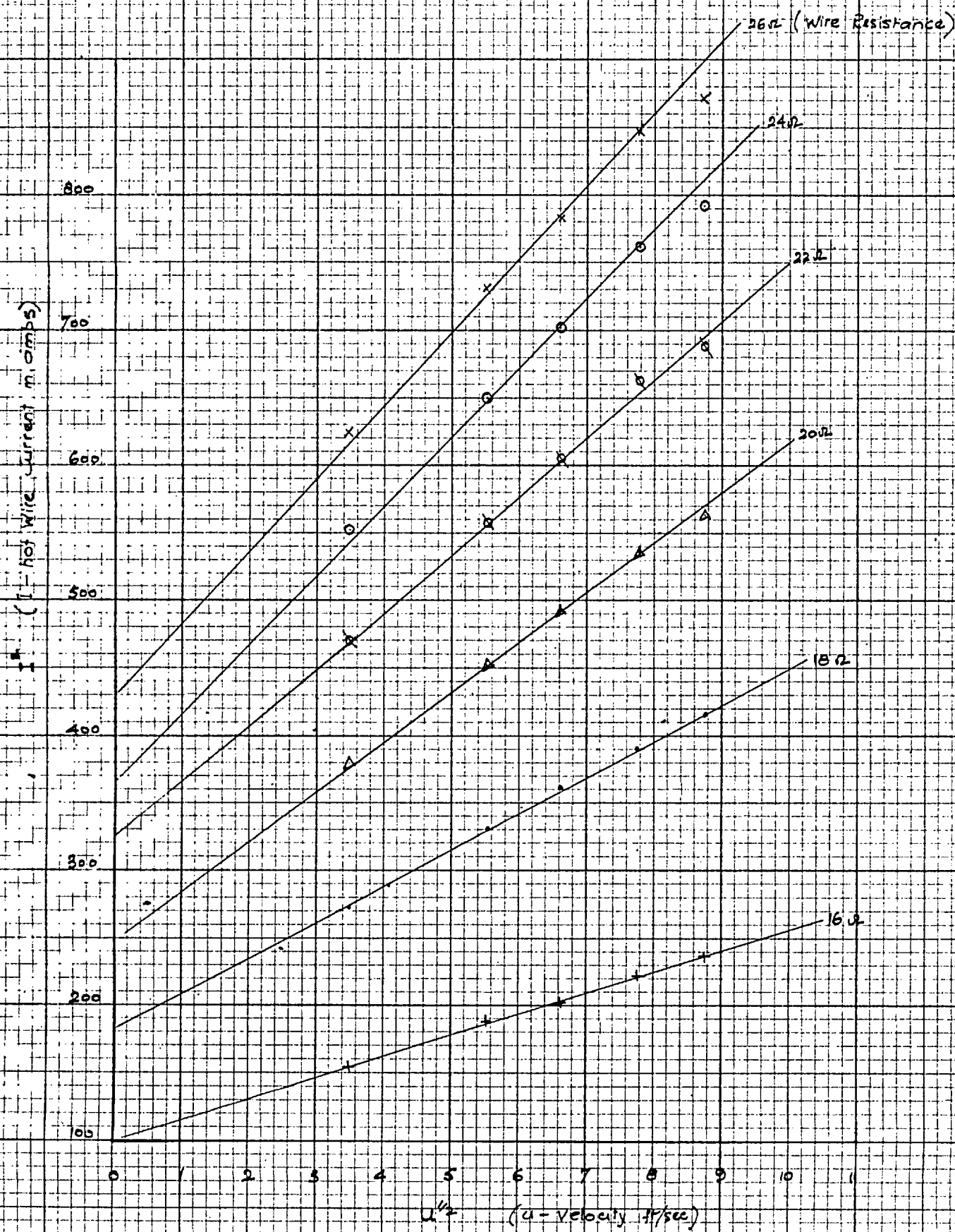
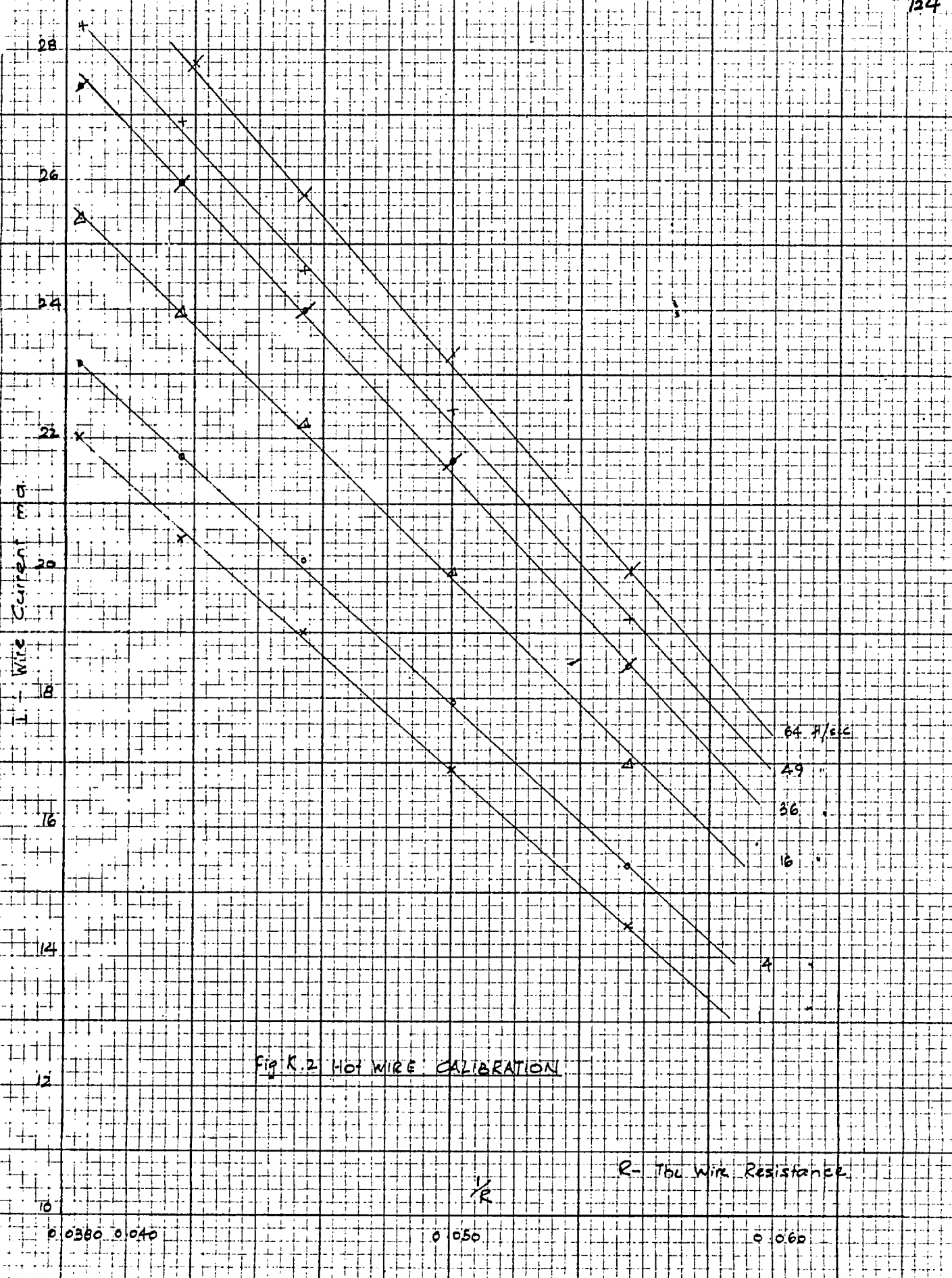


Fig K.1 Hot Wire Calibration



GRANENT OF LINES FROM CIR K 2 (0.10%)

0.60

0.50

0.40

0.30

0.20

0

10

20

30

40

50

60

70

80

Velocity ft/sec.

Fig K 3 Hot Wire CALIBRATION

APPENDIX I

127 ————— 139

MEAN AIR VELOCITIES FROM WATER MANOMETER

$$\begin{aligned} \text{Total head reading on vertical manometer} &= x_1'' \\ \text{Pressure reading} &= x_2'' \\ \text{Velocity head} &= x'' = x_2 - x_1 \end{aligned}$$

$$\frac{1}{2} \rho_a u^2 = \frac{x}{12} \rho_w g$$

$$u = 66.6 \sqrt{x} \text{ at } 20^\circ\text{C}$$

$$u = A \sqrt{x}$$

TEMP °C	A
15	65.9
16	66.0
17	66.1
18	66.2
19	66.4
20	66.6
21	66.7
22	66.8
23	66.9

$$\text{Specific gravity of absolute alcohol} = 0.8$$

$$\rho_a \text{ — Density of air at } 20^\circ\text{C} = 0.00234 \text{ lbs sec}^2/\text{ft}^4$$

$$\rho_w \text{ — " " Water " } = 1.935 \text{ lbs sec}^2/\text{ft}^4$$

$$g = 32.17 \text{ ft/sec}^2$$

$$u \text{ — Velocity of air ft/sec}$$

TYPICAL VELOCITY TRAVERSE

Values for fig. A.2.1:--

Horizontal traverse 26 diameters from downstream face
of concave friction baffle, using alcohol manometers inclined
1:5

Zero reading of manometer = 11.0 mm

Atmospheric temperature = 20 °C

Distance From Baffle (Pipe Diameter) y	Pressure Reading mm
0	175.0
4.5	175.0
10.0	
15.8	179.5
21.0	196.5
26.0	200.0

Distance From Wall y"	Total Head mm	Mean Velocity u ft/sec	Distance From Wall y"	Total Head mm	Mean Velocity u ft/sec
0.05	161.5	52.75	2.75	88.0	55.90
0.10	156.0	55.00	3.00	91.0	55.10
0.20	146.5	58.60	3.25	92.5	54.70
0.30	140.0	40.90	3.50	97.0	55.20
0.40	132.5	45.40	3.75	101.0	52.60
0.50	128.0	47.80	4.00	105.0	51.40
0.60	124.0	46.10	4.20	108.0	50.60
0.80	120.0	47.30	4.40	113.0	49.25
1.00	115.0	48.70	4.60	118.0	47.80
1.20	110.0	50.10	4.80	125.0	45.70
1.40	110.0	50.10	5.00	131.0	43.90
1.50	105.0	51.40	5.10	132.5	43.40
1.75	98.0	53.40	5.20	140.0	40.90
2.00	96.0	53.40	5.30	146.5	38.60
2.25	95.0	54.20	5.40	156.5	34.80
2.50	92.5	54.70			

Logarithmic Plot of Values for Profile A.2.1

Sown in Fig. A.4

y^* — friction velocity from experimental data
(page 130) = 2.205

V_c^* — friction velocity calculated = 2.115

N — kinematic viscosity.

y^* Distance From Wall	$\frac{u}{y^*}$	$\log \frac{y y^*}{N}$	$\frac{u}{V_c^*}$	$\log \frac{y V_c^*}{N}$
0.05	14.81	1.753	15.50	1.734
0.10	15.86	2.054	16.55	2.035
0.20	17.50	2.356	18.25	2.336
0.30	18.55	2.531	19.35	2.512
0.40	19.65	2.656	20.50	2.637
0.50	20.30	2.753	21.20	2.734
0.60	20.90	2.852	21.80	2.813
0.80	21.45	2.957	22.40	2.938
1.00	22.05	3.054	23.00	3.035
1.20	22.75	3.133	23.70	3.114
1.40	22.75	3.200	23.70	3.181
1.50	23.25	3.251	24.50	3.211
1.75	24.15	3.297	25.20	3.278
2.00	24.15	3.356	25.20	3.336
2.25	24.60	3.405	25.60	3.386
2.50	24.80	3.451	25.90	3.433
2.75	25.30	3.494	26.40	3.474

Analysis of Data Corresponding to Figs. A.1.1.,
A.1.2 and A.2.1.

Fig	$\bar{u}$ ft/sec	Re	f	f_c	V^*	V_c^*
A.1.2:	31.0	87.3×10^3	0.0230	0.0185	1.660	1.490
A.2.1.	45.8	128.9×10^3	0.0136	0.0170	2.205	2.115
A. 1.1.	60.0	169.0×10^3	0.0139	0.0162	2.920	2.700

was obtained from the pressure drop in the pipe between 20.8 and 31 diameters from the inlet.

VALUES FOR NONDIMENSIONAL VELOCITY PROFILES, FIG. A.17

y/r	y''	Fig A. 1.2		Fig A. 2.1		Fig A. 1.1	
		$\bar{u}$ ft/sec	u/u_{max}	$\bar{u}$ ft/sec	u/u_{max}	$\bar{u}$ ft/sec	u/u_{max}
0.0	0.000	22.0	0.600	29.0	0.525	43.0	0.594
0.05	0.138	25.8	0.705	36.2	0.655	50.0	0.690
0.1	0.275	28.1	0.767	40.6	0.735	54.1	0.746
0.2	0.50	30.1	0.822	44.6	0.808	58.1	0.802
0.3	0.825	32.0	0.874	47.4	0.856	62.0	0.855
0.4	1.100	33.1	0.905	49.4	0.894	64.4	0.888
0.5	1.380	34.0	0.930	50.7	0.919	66.6	0.920
0.6	1.650	35.0	0.956	52.2	0.945	68.4	0.944
0.7	1.925	35.6	0.973	53.2	0.963	69.9	0.964
0.8	2.200	36.1	0.986	54.0	0.976	71.2	0.983
0.9	2.480	36.4	0.995	54.9	0.991	72.2	0.996
1.0	2.750	36.6	1.000	55.3	1.000	72.5	1.000

$$\frac{u}{V^*} = 5.75 \log \frac{V^* y}{\nu} + 5.5$$

$$\frac{1}{\sqrt{f}} = 2 \log \frac{Re \sqrt{f}}{2.51}$$

$$V^* = \bar{u} \sqrt{f/8}$$

$$h = f \frac{\bar{u}^2 l}{2g d}$$

f — Experimental friction Coefficient

f_c — Calculated " "

$\bar{u}$ — Average Velocity

V^* — Friction Velocity.

V_c^* — Calculated friction Velocity

Calculated logarithmic profile for conditions
corresponding to Figs. A.1.1 and A.2.1

$$\frac{u}{V^*} = 5.75 \log \frac{V^* y}{\nu} + 5.5$$

For Fig A.1.1 $u = 68.5 + 15.5 \log y$

Fig A.2.1 $u = 48.5 + 12.15 \log y$

y''	$\log y$	A.1.1 u ft/sec	A.2.1 u ft/sec
0.05	-1.301	43.30	32.79
0.02	-0.699	52.69	39.06
0.4	-0.398	57.33	43.70
0.6	-0.222	60.06	45.82
0.8	-0.097	61.99	47.33
1.0	0.0	63.50	48.50
1.2	0.079	64.72	49.45
1.4	0.146	65.76	50.26
1.6	0.204	66.66	50.96
1.8	0.255	67.45	51.58
2.0	0.301	68.16	52.14
2.2	0.342	68.80	52.63
2.4	0.380	69.40	53.10
2.6	0.415	69.94	53.51
2.75	0.439	70.30	53.80

Variation of the pressure drop coefficient

Fig. A. 16 for concave friction bafile

Re	γ
37.8×10^3	0.784
74.5	0.710
122.2	0.760
151.0	0.807
158.5	0.923
147.0	0.936
158.5	0.960
173.0	1.036

HOT WIRE TRAVERSE—Fig. A.9

26.5 diameters from downstream face of concave
friction baffle

Calibration curves for hot wire figs K1, K2 and K3

Typical hot wire calculation page:- 133

Distance from Wall y"	y/r r-radius	Hot Wire Current I m.A	I ²	U ft/sec Velocity	Thermo Amm Reading m.A	$\frac{\sqrt{U^2}}{U}$	$\frac{\sqrt{U^2}}{ft/sec}$	$\frac{\sqrt{U^2}}{V^*}$
2.750	1.00	25.20	636.0	54.9	0.131	0.0159	0.875	0.396
2.50	0.909	25.15	631.0	52.8	0.100	0.0123	0.650	0.294
2.25	0.818	25.10	630.0	52.8	0.130	0.0161	0.685	0.310
2.00	0.728	25.00	625.0	50.5	0.140	0.0175	0.884	0.400
1.75	0.637	24.95	621.0	49.5	0.155	0.0195	0.965	0.437
1.50	0.546	24.90	620.0	49.0	0.188	0.0238	1.165	0.527
1.40	0.509	24.90	620.0	49.0	0.189	0.0238	1.165	0.527
1.20	0.436	24.80	616.0	48.0	0.205	0.0262	1.260	0.571
1.00	0.364	24.75	611.0	46.4	0.230	0.0298	1.391	0.631
0.80	0.291	24.65	611.0	45.2	0.240	0.0312	1.410	0.639
0.60	0.218	24.50	600.0	42.3	0.265	0.0352	1.489	0.674
0.50	0.182	24.50	600.0	42.3	0.292	0.0388	1.641	0.745
0.40	0.145	24.40	600.0	40.8	0.315	0.0425	1.735	0.786
0.35	0.127	24.35	595.0	39.5	0.328	0.0448	1.770	0.803
0.30	0.109	24.25	591.0	39.0	0.340	0.0464	1.810	0.820
0.275	0.100	24.20	590.0	38.5	0.345	0.0477	1.835	0.831
0.250	0.091	24.10	586.0	36.6	0.355	0.0498	1.825	0.826
0.200	0.073	24.10	581.0	36.6	0.372	0.0522	1.911	0.866
0.175	0.064	24.05	580.0	36.0	0.400	0.0562	2.025	0.918
0.150	0.055	24.00	576.0	35.8	0.420	0.0594	2.215	0.965
0.125	0.046	24.00	576.0	35.8	0.471	0.0666	2.421	1.098
0.100	0.035	24.00	576.0	35.8	0.562	0.0796	2.850	1.290

The expression used for calculating the intensity of turbulence was

$$\frac{\sqrt{u'^2}}{u} = \frac{4 \left\{ \frac{1}{R_0 R} \left(\frac{\partial I}{\partial \sqrt{R}} \right) u = \text{constant} + \frac{I}{2R} \right\} \left(\frac{R+s}{s} \right) i_0}{(I^2 - I_0^2) \times A_f} \quad \text{Ref 31}$$

$\sqrt{u'^2}$ — Root mean square of turbulence fluctuation in direction of flow

u — Mean velocity.

R_0 — Cold resistance of wire

R — Box resistance

I — Wire current

i_0 — Thermo ammeter reading

A_f — Amplified constant

s — Constant resistance of bridge

$\left(\frac{\partial I}{\partial \sqrt{R}} \right)$ — From fig. K3

I_0 — From fig. K1

Typical calculation for distance 1" from wall

fig. A. 9. (page 132)

cold resistance $R_0 = 13.8 \Omega$

$I = 24.75 \text{ m.A}$

$u = 46.4 \text{ ft/sec.} \quad \text{— fig. K1}$

$R = 22 \Omega$

$A_f = 0.265 \frac{1}{2}$

$i_0 = 0.230 \text{ mA}$

$s = 500 \Omega$

$I_0^2 = 326 (\text{mA})^2$

$\frac{\partial I}{\partial \sqrt{R}} = 0.545$

$$\begin{aligned} \frac{\sqrt{u'^2}}{u} &= \frac{4 \left\{ \frac{1}{22 \times 13.8} (0.545) + \frac{24.75 \times \frac{1}{1000}}{2 \times 22} \right\} \left(\frac{500+22}{500} \right) \frac{0.230}{1000}}{(1000)^2 \{ 611 - 326 \} \times 0.265} \\ &= 0.0298 \end{aligned}$$

Variation of Length and Inlet Velocity for Concave Friction

Baffle.

$$l_{max} = 10.5''$$

u_{ay} — upstream Velocity

u_{dy} — Downstream Velocity

Values from A.7 plotted in fig. A.15

l_y''	$\frac{l_{max}}{l_y}$	$\log \frac{l_{max}}{l_y}$	u_{ay}	$\frac{u_{ay}}{\bar{a}}$	$\log \frac{u_{ay}}{\bar{a}}$
7.75	1.355	0.1319	47.70	1.040	0.0170
7.00	1.500	0.1761	50.10	1.095	0.0395
4.25	2.470	0.3927	53.60	1.170	0.0682
3.50	3.000	0.5441	55.90	1.395	0.1446
2.75	3.820	0.5821	63.90	1.395	0.1446

Flat Friction Baffles (Equal Length)

Variation of γ the pressure coefficient with Reynolds No.

Re	γ	Re	γ	Re	γ
91×10^3	0.400	69.9×10^3	1.469	47×10^3	0.375
101.0	0.622	104.0	1.280	78.0	0.348
125.0	0.571	115.5	1.960	108.0	0.286
127.8	0.625	116.5	1.985	124.8	0.356
157.5	0.610	129.0	2.180	141.0	0.306
162.0	0.526	145.0	2.270	159.5	0.402
		152.5	2.120	184.0	0.382
Length	4"	4"		2"	
Spacing	$\frac{1}{2}"$	$\frac{1}{4}"$		$\frac{1}{2}"$	

DESIGNED OF CONCAVE FRICTION BAFFLE

$$l_y = \frac{\frac{3}{2} \left(\frac{K'}{\delta} \right)^{\frac{1}{2}} S}{F_n} \left[\frac{u_{ay} - u_{dy}}{\bar{u}} \right] \quad \text{-----} \quad 3.16$$

$$\gamma = \phi \left(\frac{S}{a} \right)^{\alpha} \left(\frac{l}{a} \right)^{\lambda} \quad \text{-----} \quad 3.17$$

$$\frac{u_{ay}}{\bar{u}} = \left(\frac{\gamma l_{max}}{l_y} \right)^{\delta} \quad \text{-----} \quad 3.19$$

$$l_{max} = \eta l_c \quad \text{-----} \quad 3.18$$

l_y — the cylinder length

l_{max} — the maximum length of baffle

K' — pressure coefficient across perforated sheet

γ — pressure coefficient across baffle

u_{ay} — Upstream velocity at the inlet to each space

u_{dy} — downstream velocity from logarithmic profile

$\bar{u}$ — average velocity

S — spacing

ϕ — area ratio

F_n — denominator function

l_c — characteristic length (equivalent length of flat baffle causing same pressure drop across

$\phi, \alpha, \lambda, \gamma, \delta, \eta$ — constants

n	$\frac{d}{ns}$	$\frac{d}{(n-1)s}$	$\frac{d}{(n+1)s}$	A $\log \frac{d}{ns}$	B $\log \frac{d}{(n-1)s}$	C $\log \frac{d}{(n+1)s}$	$\frac{A}{B}$	$\frac{A}{C}$	$1 - \frac{A}{B}$	$\frac{A}{C} - 1$	f_{2n}	f_{1n}	$f_{2n} - f_{1n}$	$f_{2n} + f_{1n}$	S/d_j	$\frac{s/d_j \times}{(f_{2n} + f_{1n})}$	F_n
1	22.00	∞	11.00	1.3424	∞	1.0414	0.0	1.290	1.000	0.290	1.00	0.538	0.462	1.538	0.0480	0.0738	0.5358
2	11.00	22.00	7.34	1.0414	1.3424	0.8657	0.776	1.200	0.224	0.200	0.473	0.447	0.026	0.920	0.530	0.0488	0.0748
3	7.34	11.00	5.50	0.865	1.0414	0.7404	0.830	1.170	0.170	0.170	0.412	0.412	0.00	0.824	0.0612	0.0504	0.0504
4	5.50	7.34	4.40	0.7404	0.8657	0.6435	0.856	1.150	0.144	0.150	0.380	0.387	-0.007	0.767	0.0666	0.0511	0.0504
5	4.40	5.50	3.66	0.6435	0.7404	0.6635	0.868	1.140	0.122	0.140	0.363	0.374	-0.011	0.737	0.0769	0.0566	0.0450
6	3.66	4.40	3.14	0.5635	0.6435	0.4969	0.876	1.120	0.124	0.120	0.352	0.360	-0.011	0.715	0.0909	0.0650	0.0639
7	3.14	3.66	2.75	0.4969	0.5635	0.4393	0.881	1.130	0.119	0.130	0.345	0.360	-0.015	0.705	0.1110	0.0782	0.0632
8	2.75	3.14	2.44	0.4393	0.4969	0.3874	0.882	1.130	0.118	0.130	0.344	0.360	-0.016	0.704	0.1429	0.1001	0.0841
9	2.44	2.75	2.20	0.3874	0.4393	0.3424	0.883	1.130	0.117	0.130	0.342	0.360	-0.018	0.702	0.2000	0.1404	0.1224
10	2.20	2.44	2.00	0.3424	0.3874	0.3010	0.885	1.140	0.115	0.140	0.339	0.374	-0.035	0.713	0.3300	0.2360	0.2000

$$F_n = \{f_{2n} - f_{1n}\} + \frac{s}{d_j} \{f_{2n} + f_{1n}\} \text{ --- from expression 3.16 (page 55)}$$

$$\text{for } d_s = 22$$

The baffle designed is for a spacing $s = 14''$, $\gamma = 1$ and $\frac{d}{s} = 22$. The function F_n has been calculated, shown on page 136. The variation of the upstream velocity with length is given by fig. A.15

$$\frac{U_{dy}}{\bar{u}} = \left(\frac{0.87 l_{max}}{l_y} \right)^{1/3.45} \quad \text{-----} \quad 3.19a$$

From fig. E1

$$\gamma = 0.0084 \left(\frac{d}{s} \right)^{1.89} \left(\frac{l}{d} \right)^{0.860} \quad \text{-----} \quad 3.17a$$

Using a simplified form of this expression

$$\gamma = 0.0084 \left(\frac{d}{s} \right)^2 \frac{l}{d} \quad \text{-----} \quad 3.17b$$

the characteristic length of the baffle already tested was obtained, equal to 5.41". Since its maximum length was 10.5" $\eta = 1.94$ from 3.18

$$\text{For } \frac{d}{s} = 22 \quad \gamma = 1 \quad l = 1.345 = l_c \quad \text{from 3.17b}$$

$$\therefore l_{max} = 1.94 \times 1.345 = 2.6''$$

the approximate value of K^1 obtained by applying the design formula for the data obtained from the tested baffle was 1.62. The area ratio for the material was 0.36. Substituting these values in expression 3.16

$$0.755 F_n l_y = \frac{1.325}{l_y^{1/3.45}} - \frac{U_{dy}}{\bar{u}}$$

U_{dy} was obtained from the fully developed logarithmic profile and the required variation of length is given in the table below.

n	F_n	$\frac{Cdy}{a}$	ly''
	-	-	2.60
1	0.1050	0.755	2.40
2	0.0748	0.925	2.22
3	0.0504	1.000	2.02
4	0.0504	1.045	1.82
5	0.0450	1.080	1.71
6	0.0638	1.110	1.47
7	0.0632	1.133	1.40
8	0.0841	1.155	1.28
9	0.1224	1.174	1.13
10	0.2010	1.193	0.97

Estimate of the error caused by the assumption made in the theory of page 54 that $\frac{\alpha_n}{\alpha_{n+1}} = 1$ $\frac{\alpha_n}{\alpha_{n-1}} = 1$

From Table	From Table.	% Error	$\frac{\alpha_n}{\alpha_{n+1}}$	$\frac{\alpha_n}{\alpha_{n-1}}$
$l_4/l_5 = 1.061$	$\log \frac{d}{ns} / \log \frac{d}{(n+1)s} = 1.150$	8.37	0.925	
$l_4/l_3 = 0.893$	$\log \frac{d}{ns} / \log \frac{d}{(n-1)s} = 0.856$	5.26		1.04
$l_5/l_6 = 1.161$	$\log \frac{d}{ns} / \log \frac{d}{(n-1)s} = 1.140$	1.84	1.02	
$l_5/l_4 = 0.939$	$\log \frac{d}{ns} / \log \frac{d}{(n+1)s} = 0.868$	7.59		1.08

$$\frac{dy}{d} = \alpha_n \log \left(\frac{d}{ns} \right) \quad \text{--- 3.13}$$

Pressure drop coefficient for perforated plate K

Test	Average Velocity Head ins Water	Pressure Drop Across Water ins Water	K
F. 1.5	0.530	2.350	4.440
F. 1.6	0.175	0.770	4.400

$$\text{Area Ratio } \beta = 0.38$$

$$K = \left(\frac{1-\beta}{\beta} \right)^2 = 2.66$$

$$= \frac{(1-c_c\beta)^2}{c_c^2\beta^2} = 10.60$$

$$= \frac{(1-0.85 \cdot 0.38)^2}{0.85^2 \cdot 0.38^2} = 4.440$$

$$c_c = 1.000$$

$$c_c = 0.629 \quad (\text{page 19})$$

$$c_c = 0.85 \quad (\text{From Test})$$

c_c — Coefficient of Contraction

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