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Certified that the attached dissertation on Experimental & Theoritical Determination of Characteristics of Dall Flow Tunes and a Comparative Study of them with the other Flow Measuring Devices was submitted by

Sri S.K. Acharaya

and accepted for the award of Degree of Master of Engineering in "Applied Thermodynamics"

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EXPERIMENTAL AND THEORETICAL DETERMINATION OF Characteristics of Dall Flow Tubes And A Comparative Study of Them with the other Flow Measuring Devices

A Dissertation Submitted in Partial Fulfilment of the Requirements for the Degree of

MASTER OF ENGINEERING

IN

APPLIED THERMODYNAMICS (Power Engineering Steam And I. C. E.)

By

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MECHANICAL ENGINEERING DEPARTMENT
UNIVERSITY OF ROORKEE,
ROORKEE (INDIA)

1963

C E R T I F I C A T E

CERTIFIED that the dissertation entitled "Experimental and Theoretical Determination of Characteristics of Dall Flow Tubes and A Comparative Study of them with the other Flow Measuring Devices" which is being submitted by Sri S.K.Acharya as a partial fulfilment of the requirement for the degree of Master of Engineering of the University of Roorkee is a record of bonafide work carried out by him under my supervision and guidance. The results embodied in this thesis have not been submitted for award of any other degree or diploma.

This is to certify further that he has worked for a period of ~~five~~ 6 months for the Master of Engineering Degree thesis at the University of Roorkee.

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C O N T E N T S

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CONTENTS

P R E F A C E

Invention is always dynamic. With the increase of scientific knowledge, the industrial products are getting new shapes to suit the increasing demands of greater accuracy and better working. There is also keen competition amongst the industrial enterprises to put in the market their products having better usage to mankind. To that end they use the latest techniques of design and production to win the market. The same is also true in case of fluid flow measuring devices. This thesis work deals with the latest type of differential pressure flow measuring device know as 'Dall Flow Tube'. The work embodies the characteristic study of this new device and a comparative study of it with other types of differential pressure producing flow metering devices . The author hopes that this work may give some informations to those who are interested in fluid flow metering problems.

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LIST OF SYMBOLS.

A_1	Area of Cross-section of pipe.
A_2	Area of the throat section of the meter.
dA	Elimentary Cross-sectional area.
C, L	Length of the cylindrical section.
C_D	Coefficient of discharge.
D	Diameter of the pipe.
d	Diameter of the throat section.
f	Coefficient of friction from Darcy's formula.
G	Specific gravity of fluid.
g	acceleration due to gravity.
H	Rectangular weir reading.
h	Differential head in meters.
h_f	Frictional head loss.
J	Roughness Coefficient.
K_L	Pressure loss Coefficient in the Converging portion.
L	Characteristic length.
m	Area ratio $= (d/D)^2$
n	Bazin's Coefficient.
O	Refers to stagnation condition in case of compressible fluids.
p	Pressure of fluid at any point.
	Suffixes 1 and 2 refer to the sections 1 and 2 res.
Q	Volume rate of flow.

$(R_e)_d$ Renolds number referred to throat conditions.

$(R_e)_D$ Reynolds number referred to pipe diameter D.

T Absolute temperature.

U Velocity of fluid at any point.

V Average velocity of fluid at any section.

W Weight rate of flow of fluid.

w Specific weight of fluid.

(x,y) Co-ordinates of any point along x- axis and y-axis respectively.

Z Datum head.

ϕ Angle of opening of diffuser.

α_1 Correction coefficient for non uniform velocity distribution in pipe section.

α_2 Correction coefficient for non-uniform velocity distribution in the throat section.

ϵ Compressibility coefficient.

ρ Density of fluid.

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INTRODUCTION

In successful conduct of modern, large scale continuous processes, accurate and reliable measurements of fluid flow is of prime importance. It provides basis for accurate cost and yield data on the operation and is a guarantee of quantity sales to consumers. It is equally important in plant test work undertaken to increase production, to obtain design data or to eliminate operating difficulties.

There are various types of devices for flow measurements and their characteristics differ according to their design and use. A list of them is given below :

CLASSIFICATIONS:

The devices for flow measurement may be classified in many ways. But they may be conveniently grouped under two headings :

A. Meters for measuring the flow of fluids through closed conduits and

B Meters for measuring the flow through open conduits.

Under group A all fluids are included -

that is to say, liquids (water, petrol, oil etc.) and gases (air, coal gas, steam etc.) Under group B. Since they are open to atmosphere, the fluids are limited to liquids, the vast majority of which consists of water or aqueous solutions, such as sewage effluent.

The meters under group A can be subdivided as follows:

1. Mechanical flow meters.
2. Differential pressure flow meters.
3. Shunt type flow meters (combining certain characteristics of 1 and 2).
4. Variable aperture flow meters.

Group B are subdivided into:

1. Weir flow meters.
2. Venturi flume or standing wave flow meters.
3. Meters for steam gauging.

The reader will appreciate the difficulty that it is not possible to describe all the types of meters described above, since each type will consist a treatise of its own. For this reason only the differential pressure flow meters under heading A- 2 will be discussed in detail. In order to have some idea about the different types of meters available in the market a short note is given below:

Under the heading of Mechanical Flow meters the following meters are available.

a. Reciprocating piston types.

b. Semi rotary circular piston types.

c. Nutating disc type.

d. Helix and other types.

e. Single and multijet fan types.

f. Turbine type.

g. Propeller types.

h. Popeller type with electronic counter.

i. Twin rotær propeller, type.

The meters under the heading 'Shunt Flow Meters' are :

a. Meters for liquids.

b. Meters for gases.

c. Water-damped meters for gases.

d. Meters for low pressure gases.

Variable-aperture flow meters are :

a. Cone and float type.

b. Cone and disc type.

c. Piston type.

d. Hinged - gate type.

For complete informations of all these meters (5) may be referred.

DIFFERENTIAL PRESSURE FLOW METERS :

Since the present study is mainly on these types of meters, a complete analysis is given in subsequent chapters. Before going into detailed study, a general classification of these types of meters is given below. They constitute two parts:

1. Primary elements.
2. Secondary Elements.

The primary elements are the detecting devices and the secondary elements are measuring devices. The primary elements are :

a. The Venturi tube.

b. The Dall tube.

c. The Orifice plate.

d. The nozzle.

e. The pilot static tubes.

f. The double throat venturi tube.

g. The Dall Orifice.

h. The Pitot-venturi and Dall Pitot.

Of the various forms of flow meters available, those of the differential pressure type have the widest application in industry. The reasons for this are as follows :-

a. The meter can be built to suit any shape of conduit, which need not be horizontal or vertical.

b. There are no moving parts immersed in the metered fluid; consequently there is no mechanical wear in this portion of the meter.

c. With very few exceptions they can be used to meter all fluids.

d. An indication of the instantaneous rate of flow, a record of variations in flow rate and an integration of the rate of flow can easily be obtained.

e. The limits of the dimensions of the conduit are determined only by constructional considerations.

f. The principle of operation facilitates the addition of various forms of mechanisms for the transmission of flow records over a distance.

A complete analysis of the theory pertaining to these types of flow meters is given in Chapter I. In Chapter II, the experimental setups and the results obtained from them are given . The discussion of various types of errors that come in the metering problems is given in Chapter III. In Chapter IV, the comparative study of the different differential pressure flow - meters is outlined and finally the installation techniques, the testing and repairing of these meters are given in the concluding Chapter V.

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CHAPTER I

CHAPTER I

DESCRIPTION OF DIFFERENTIAL PRESSURE-TYPE METERS AND THE THEORETICAL ANALYSIS OF FLUID METERING PROBLEMS:

A SHORT HISTORY:

Venturi in 1797 postulated the operating principle of the differential pressure flow metering device. This device is named after him as 'Venturi meter'[4]. His interest was mainly academic and it was left to Clemens Herschel (1887) to develop an instrument for measuring water flow, the same form of which is used to this day. One of the first attempts to find the factors governing the coefficient of discharge of these meters and therefore to predict its value was by Gibson (1915) who showed that among other things, the coefficient of discharge depends upon friction losses in the cone and type of velocity distribution at the pressure measuring section.

Another interesting paper by Pardoe (1919) also attempted to predict the value of the coefficient of discharge and in so doing calculated the losses in the conical contraction section between the main and the throat section. (He did not take effect of Reynold's number) . Smith (1923) showed the dependence of the coefficient of discharge upon viscosity for venturi - meters used for oil flow measurements (in laminar region).

The various factors involved, have been more recently analysed, and set out in A.S.M.E. 'Flow Measurement' publications. An excellent summary of this work and present position was given by Jorrisen in 1951 (2) Pioneering work in Eureope was done by Camichel in Toulouse, France, by Schlag in Liege, Belgium, In 1932 at a meeting held in Milan, Italy, the Committee on Fluid Measurement of the International Federation of the National standardizing Association (I.S.A. 30) pointed out the interest of systematic experimental research in venturis. In the year preceding the second World War numerous studies were undertaken, particularly in Belgium, France, Germany, Great Bretain, and Italy, and results were published susceptible of being used for a completed standardization. In 1938, the Italian Ente nazionale per l' unificazione nell' Industria (U.N.I.) submitted a project to a public enquiry and this project served as a basis for discussion at Helsinki, 1939 meeting I.S.A. 30 Committee. It was at this meeting that, for the first time, international rules were established for flow measurements with venturi tubes. In May, 1948, the subject of venturi tubes was one of the most important topics discussed at the Paris meeting of the Committee on Flow Measurement of the International Organisation for Standardization (I.S.O. 30)).

Twenty years ago experiments[3] were being made in the United Kingdom on venturitubes without a parallel length between the upstream and downstream cones, in an attempt at simplification, but the flow coefficient was found to be unstable. These experiments originated from the observation that a freely discharging cone has an expanding rather than a parallel jet. Throatless tubes were experimented with, and a suitable design developed for which the coefficient remained virtually constant at normal flows and for normal pipe sizes. Thus the Dall tube, designed by Mr. H. E. Dall of the Kent Hydraulic research team, developed. This differential pressure producer was first manufactured in 1946. In the year 1956 A. L. Torrisen studied the characteristics of the Dall Flow tube and verified the claims of the manufacturer [1].

THEORETICAL ANALYSIS OF METERING PROBLEMS - BASIC PRINCIPLE

In its simplest form a differential pressure type flow meter consists of a detecting element operating in conjunction with a measuring unit. The detecting element is constituted by the differential pressure producing device, which is inserted into the pipe line through which the fluid to be metered is passing. This is essentially a flow restricting device through which the Kinetic energy of the flowing fluid changes. The measuring or secondary element is responsive to the differential pressure produced. In all cases the basic principle underlying operation of these meters (except pitot tubes) is that an increase in the velocity of flow is accompanied by a decrease in the pressure of the fluid under consideration, hence the derivation of the term 'differential pressure meter'. The required increase in the velocity is obtained by decreasing the net Cross-sectional area of the flow.

FLOW CHARACTERISTICS:

Theoretically, there will be three conditions of flow possible when fluid flows through a closed conduit.

1. Stream-line or viscous flow.
2. Combined viscous and turbulent flow.
3. Turbulent flow.

The condition 1 can be obtained at very low velocities and the only forces causing the reduction in pressure in the direction of flow are those due to the shear stresses within the fluid itself. The stresses are set up by fluid in one plane sliding over the fluid in the adjacent plane, and they disappear immediately the fluid comes to rest. This condition of flow is only possible in a pipe of uniform cross-sectional area.

In the turbulent flow region, however, discrete particles of the fluid behave as separate entities which superimpose, across the velocity gradient, a multitude of vortices and cross currents. The velocity gradient is now no longer entirely dependent on the viscous shear between successive layers of fluid, but is influenced by the movement of slower particles travelling away from the walls of the conduit and in so ding they accelerate the slower moving particles in their path. Under these conditions the pressure drop in the direction of flow is directly proportional to the change in Kinetic energy of the fluid viscous forces are still present, but their effect is negligible.

DISCUSSION:

Under the viscous flow conditions, the pressure drop is directly proportional to the velocity of flow. In the differential pressure flow devices, this condition of flow cannot exist. Because due to change of cross-sectional area of the flow, there is a change in the direction of stream lines and hence there is a change in momentum. Strictly speaking, even at low velocities through a differential producing device, in practice the flow characteristics is a combination of viscous and turbulent, but the viscous forces may be so great that they predominate.

Generally differential pressure flow meters are designed to operate entirely within the turbulent flow region. Hence in this Chapter, the equation for turbulent flow is given more importance. However, the other factors that enter into theproblem have been also duly considered.

FLOW EQUATIONS:

The following theoretical formula may be derived from Bernoulli equation and the equation of continuity for the case of incompressible fluids[2]:

$$V_2 = \frac{1}{\sqrt{\alpha_2 - \alpha_1 m^2 + K_L}} \sqrt{2g\left[\left(z_1 + \frac{p_1}{w}\right) - \left(z_2 + \frac{p_2}{w}\right)\right]} \quad (1)$$

This may be written as

$$V_2 = \frac{C_D}{\sqrt{1+m^2}} \sqrt{2g\left[\left(z_1 + \frac{p_1}{w}\right) - \left(z_2 + \frac{p_2}{w}\right)\right]} \quad (2)$$

and the rate of flow is expressed by

$$Q = \frac{C_D A_2}{\sqrt{1-m^2}} \sqrt{2g\left[\left(z_1 + \frac{p_1}{w}\right) - \left(z_2 + \frac{p_2}{w}\right)\right]} \quad (3)$$

The relation between C_D and K_L is easily found to be

$$C_D = \sqrt{\frac{1-m^2}{d_2\left(1-\frac{d_1}{d_2}\right)m^2 + K_L}} \tag{4}$$

Alldwance must be made for possible roughness-coefficient . Finally the volumetric rate of flow is given by

$$Q = \frac{C_D J A_2}{\sqrt{1-m^2}} \sqrt{2g\left[\left(z_1+\frac{p_1}{w}\right)-\left(z_2+\frac{p_2}{w}\right)\right]} \tag{5}$$

when the fluid is considered as compressible, an addition correction takes for compressible effects into consideration. The weight rate of flow for compressible fluids is expressed by

$$W = \frac{C_D J \epsilon A_2}{\sqrt{1-m^2}} \sqrt{2g(p_1-p_2)w_0} \tag{6}$$

DISCUSSION:

The Bernoulli theorem is usually the foundation for calculations in the field of fluid flow measurement. In its convential form it represents the law of conservation of energy and does not take into account[(6)]:

1 Irreversibility due to friction, impact etc.
2. Varying velocity distribution across the conduit due to the state of turbulence, which can be fundamentally related to the Reynolds number and roughness of the wall.

3. Curvature of the flow filament which may lead to separation of main fluid stream of the wall surface of the measuring device, resulting is the contraction of main stream and eddy formation.

4. A displacement of boundary layer.

These conditions which are mainly due to viscosity influence, differ from those of irrotational flow, the latter being a basis of mathematical approaches to fluid dynamics. Inspite of these limitations, important conclusions can be drawn from potential theory, particularly for fully developed turbulence, when viscosity influences may become irrelevant. This statement may be considered a paradox in the case of a fully developed turbulence with its high verticity. Neverthless, the experimental results closely approach calculations based on irrotational flow.

CORRECTION COEFFICIENT FOR NON UNIFORM VELOCITY VARIATION IN PIPE SECTION - .

The value of correction coefficient used in previous equations is due to non-uniform velocity distribution. The total energy of a stream made up of stream tubes, at any cross-section, is equal to the total energy at some other section plus the losses occuring between the two sections. Since the quantity flowing past each section is steady flow during the same time, interval is same, the average energy per pound of fluid at any section is equal to the average at the second section plus the losses between the two sections. This average energy at any section may be easily found to be (7)

$$H_{av.} = \frac{p}{w} + Z + \frac{1}{2g} \quad \frac{\int U^3 \, dA}{\int U \, dA} \qquad (7)$$

The last term in this expression can be evaluated exactly only when the velocity variati n in terms of dA is known. If we use the average velocity in c mputing the kinetic energy of the fluid per pound of fluid flowing, we introduce an error equal to the difference between $1/2g \left(\frac{\int U^3 dA}{\int U \, dA} \right)$ and $\frac{q^2}{2g}$

in which U represents the average velocity. The true average Kinetic energy head $1/2g \left(\frac{U^3 dA}{VA} \right)$ is always larger than the average value $V^2/2g$ so that the former may always be expressed as $\alpha V^2/2g$. varies between 1 and 2, when $\alpha = 2$ the velocity varies parabollically.

DISCUSSION:

It will be clear from the above considerations that we are using the formula for flow calculations with the assumption of irrotational flow. More over the flow at the walls of the pipe line is not irrotational and a little more consideration of this effect will be given in Chapter 3.

In the present problem, the meters are installed almost horizontally, hence the variation of datum head is not there. Again the metering fluid is water and it is tacitly assumed that there is no variation of density under the normal operating conditions. Under this circumstance, the flow equation may be written as

$$Q = \frac{C_D \, J \, A_2}{(1 - m^2)^{\frac{1}{2}}} \, (2gh)^{\frac{1}{2}} \qquad (8)$$

PRIMARY ELEMENTS - CENTRIFUGAL TYPE METER.

If a fluid is flowing through a right angled bend in a pipe and the bend is in the form of a smooth arc of a circle, then, owing to the tendency of the fluid to continue to move in a straight line, the pressure of the fluid on the pipe on the outside of the bend will be greater than the pressure inside. The difference of pressure will depend upon the density of the fluid and upon its velocity. It has been found by experiment that the pressure difference bears a reasonably constant relationship to the mass rate of flow of the fluid. The relationship may be expressed by the relation.

$$W = C_D \; A \; \left(63.34 \; (p_1 - p_2) \right)^{\frac{1}{2}} \qquad (9)$$

p in psi.

DISCUSSION:

There are two right angled bends on the pipe line of 13 ¾" diameter installed in the senior Hydraulic Laboratory. Hence it was thought necessary to establish whether the relationship between the differential pressure and the discharge in the pipe line is constant or not. So two manometers were used on suction and discharge sides of the centrifugal pump to study the effect. More over the liquid being water the relationship between discharge and differential head may be related as

$$Q = 12\, C_D\, A\, (63.34 \times h)^{\frac{1}{2}} \qquad (10)$$

SIMPLE U- TUBE MANOMETER :

With a common glass U-tube manometer the difference in liquid levels in the two legs of the manometer may be measured directly with the linear scale attached to the instrument. This measurement is related to the pressure differential across the meter [8].

During the experiments, simple inverted U-tube manometers were used and the direct reading of differential pressure in inches were obtained. From this the actual difference of pressure is obtained by taking into account the weight of air andthe expression for it may be easily found to be $h = h_m\ (1 - w'/w)$ (11)

Fig 1

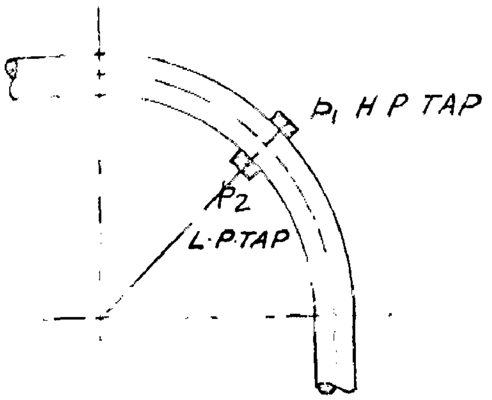


Fig 2

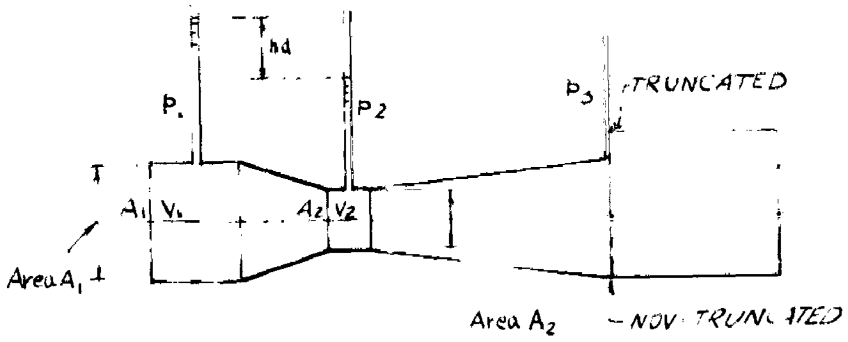


FIG 3

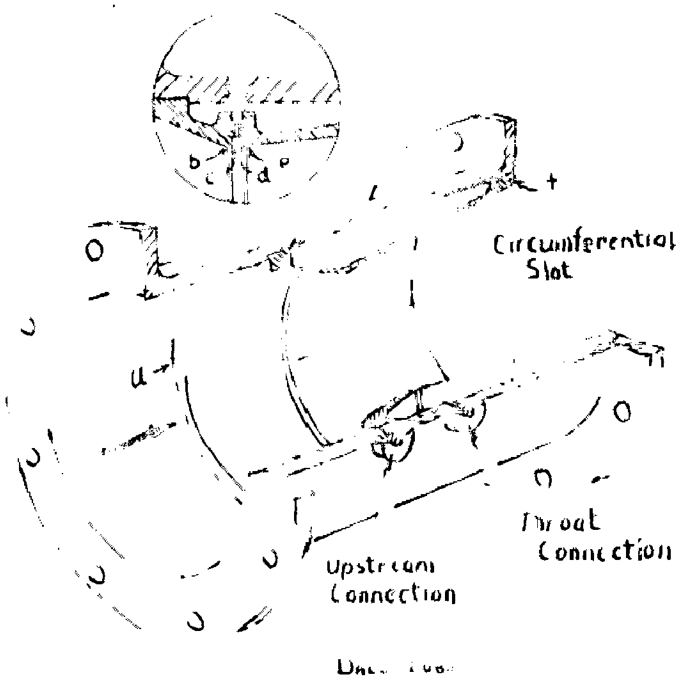
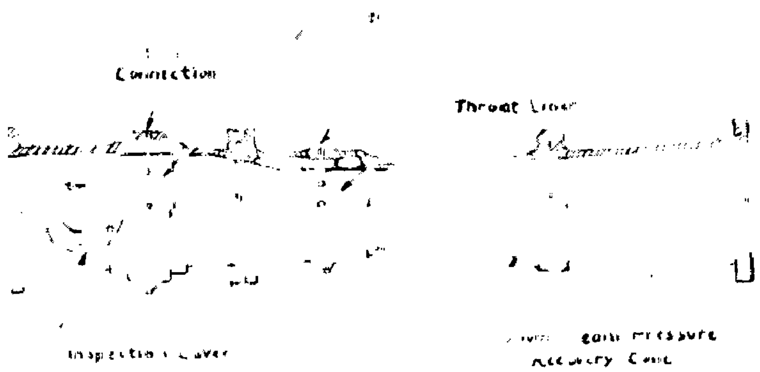


FIG 4



CLASSIC VENTURI TUBE

Where h_m measure differential pressure in the manometer.

h is the actual differential pressure of the meter.

w' is the weight of air at main pressure lbs/cft.

w is the weight of water lbs/cft. (10).

THE METHOD OF DIMENSIONS:

Geometrical Similarity: The method of dimensions (8) may be used to determine the form of relationship between

1. The pressure difference p_{12} between any two sections in a chosen system where liquid flows steadily between the solid boundaries (pressure difference due to difference in level having been deducted).

2. The mass M per second crossing any transverse section (M not varying with time)

3. the density of the fluid.

4. the viscosity of the fluid. and

5. 'l' , a length that specifies the linear dimensions of the walls of the system.

Since only length l is specified, it follows that we are assuming any change in the length to apply to all lengths in the system. If the size of the tube be changed, it must be reduced or magnified throughout being kept geometrically similar.

If no other variables are concerned we may write the relationship

$$F_1 \; (p_{12} \; , M , l , \rho \; , \mu) \quad = 0 \; .$$

The function F_1 involves the variables and constants depending on the shape of the walls and the position of the two chosen points.

Any change in units used to measure mass, length and time must have the function unchanged in value. Hence whenever the variables occur in the function, they must occur in products that have zero dimensions when expressed in terms of mass, length and time.

let $p_{12}^{a} \quad M^{b} \, l^{c} \quad \rho^{d} \, \mu^{e}$ be such a product

having zero dimensions.

By dimension analysis we get

$(p_{12} \; l^{2} \, \rho \, \mu^{-2} \;)^{a} \quad (M \, l^{-1} \, \mu^{-1} \;)^{b}$ as the two products . According to the van driest method of analysis we find that there will be two independent dimensionless groups.

Hence:

$$F_2 \quad (p_{12} \, l^{4} \, \rho \; M^{-2} , \quad M \; l^{-1} \mu^{-1} \quad) = 0 \qquad (12)$$

Solving for first product and taking the square root and transposing we have .

$$M = l^2 (p_{12} \rho)^{\frac{1}{2}} \quad f_1 \; (M/\rho \mu) \qquad (13)$$

This also can be written as

$$M = \frac{l^3 \; p_{12} \rho}{\mu} \quad f_2 \left(\frac{M}{l \; \mu} \right) \qquad (14).$$

Equation 13 and 14 show that $M/l^2 \; (p_{12} \quad)^{\frac{1}{2}}$ and $M \; / \, l^3 \, p_{12} \; \rho \;)$ do not change in value if $M/l\mu$ be kept constant, even if M , l and change Thus either of these products has a single value for one value of $M/ \, l\mu$ and a curve may be plotted (from experiment or theory) to relate either of these and $M/l \mu$. This curve is general for all liquids, rates of flow, and sizes of the system, provided that the system and two chosen points be kept geometrically similar.

In almost every case, when the value of M/l is large (a large rate of flow) the kinetic energy terms predominate, and M^2 varies p_{12} approximately. Thus f_1 in equation 13 tends to a constant value. Similarly for small values of $M/l \; \mu$ (small rate of flow) viscous forces predominate and $M/ \rho \; \propto \; p_{12} \, / \mu$, then f in equation 14 generates into a constant and no turbulence is produced.

If the sizes of the system can be specified by a single variable l, but requires more (such as the case of a circular orifice of radius r in a plate thickness Δ , placed in a tube of radius R) the application of above theory would give in place of equation 13.

$$M = l^2 (p_{12} \rho) \; f_1 \left(\frac{M}{l \mu} \quad \frac{\Delta}{r} \quad \frac{R}{r} \right) \quad -(15)$$

If f_1 is to remain unchanged in value we must keep (in general) $M/l\mu$ and Δ/r and R/r unchanged in value and hence all system must be made geometrically, similar unless the effect of Δ/r and R/r on the function is about to be investigated.

Consequently, if the boundary is kept geometrically similar and $M/l\mu$ is unchanged, the stream lines and eddies remain geometrically similar (unless, the eddies be moving and the functions vary periodically with time.) In any of the equations, the mass passing per second may be written in terms of Q, the volume passing per second or V, defined as the mean velocity over the transverce section or as the velocity at any chosen point in any chosen direction, may be used. The product M/l then is replaced by $Q\rho/l\mu$ or $Vl\rho/\mu$ which is nothing but Reynold's number.

CONCLUSIONS:

From the above discussion, it is clear that if two meters are geometrically similar, their characteristics will be identical under similar conditions. Themass rate of flow or in the present case of incompressible liquid the volume rate of flow will be proportional if the geometrically characteristics of the meters will be proportional provided that Reynolds number is same in both cases. This is perfectly true provided the other factors do not come into picture. But in actual practice this is not the case, because the factor like roughness cannot be precisely controlled and hence there will be some variations. Again the boundary layer theory conforms to this statement. After a certain degree of smoothness has been attained even polishing to the equivalent of a glass finish does not reduce frictional losses. However, this roughness effect is very small, but it does call for the introduction of a small correction, which is known as the 'pipe size factor'. More over the considera-tion of the dimensional analysis helps to a great extent in thefield of model testing.

A SHORT DESCRIPTION OF THE DIFFERENT TYPES OF METERS.

In order to have a clear picture of the different types of differential pressure flow meters as regards their geometrical characteristics are concerned, it seems reasonable to give a brief description of each type of meter.

DALL FLOW TUBE : Fig. 3, The Dall tube consists of a short length of 'lead in ' parallel pipe, followed by converging and diverging sections [11]. A small gap made between these sections. The flow first strikes the dam at a , which would be expected to increase the head loss. There is a narrow cylindrical section on either side of the throat slot. Therefore after the flow passes through the inlet cone it encounters a sharp edge at b, and then another sharp edge at c. The flow next has to traverse the open throat slot, after which it strikes two more sharp edges d and e. At point f the flow undergoes sudden enlargement to the pipe diameter. The cones are steep in addition to being truncated, and the whole device is only about two diameters long. The recovery cone has an included angle of about 15^{o}.

The vonventional Bernoulli theorem applies when the stream lines are parallel with the walls of the tube through which the fluid is flowing at the point of pressure measurement and by utilising a parallel throat length, the conventional form of venturi is designed to ensure this condition. The slot ~~is~~ formed between the smaller diameters of two cones, has a substantial included angle and the lower of two pressures is measured in this slot. The effect of abrupt change in contour of the glowresults in a substantial curvature of the stream lines at the slot and this adds a 'stream line' head to the differential head produced in accordance with the Bernoulli equation. That is to say, the pressure at the throat is increased by a significant amount and the discharge coefficient is, therefore, much lowerthan unit. In addition, the abrupt reduction in diameter at the point where the upstream pressure is measured results in a local increase of pressure which again, increases the differential pressure produced by this device.

Inspite of the abrupt change in direction of the stream line of the slot, there is no break away of the jet from the walls and consequently no eddies or disturbances. In fact the slot is analogous to the anti-stalling device on an air craft, which prevents the break away of air stream at high angles of incidents, The elimination of eddy losses and the reduction in friction, resulting from the short length of the cones, results in a high recovery.

Due to the uncomplicated nature of the D ll tube contour, in the larger sizes these detecting elements can easily be fabricated in mild steel. Dall tube inserts of fibre glass f r mounting between adjacent flanges in a pipe line, have also been used for metering corrosive fluids.

Usually Dall tubes are used for measuring the flow of water, compressed air, sewage, low pressure air, relatively clean and non-corrosive gases, and low pressure steam eg. exhaust steam.

VENTURI METER (Fig. 4).

Essentially, a venturi tube consists of the following:

1. A converging portion, where the transformation of pressure into kinetic energy takes place. This is the essential part of the pressure difference device. According to the shape of this portion venturi tubes are classified into:

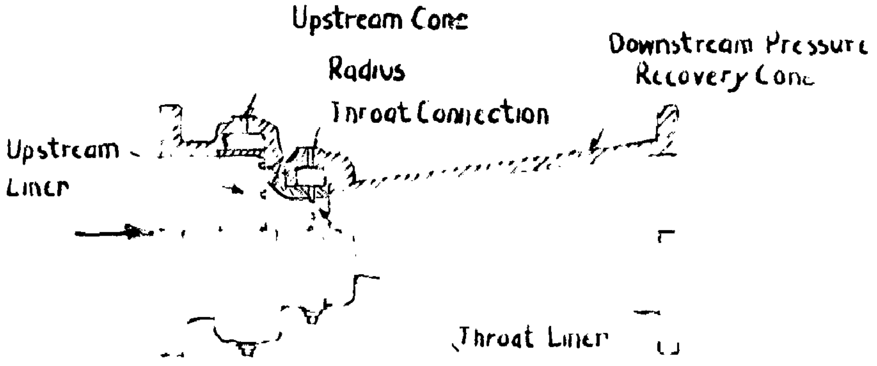
a. Herschel type, also called conical entrance type, in which the entrance portion is converging-entrance cone.

b. I.S.A. nozzle type in which the entrance portion consists of a standard I.S.A. 32 nozzle.

2. A cylindrical throat portion.

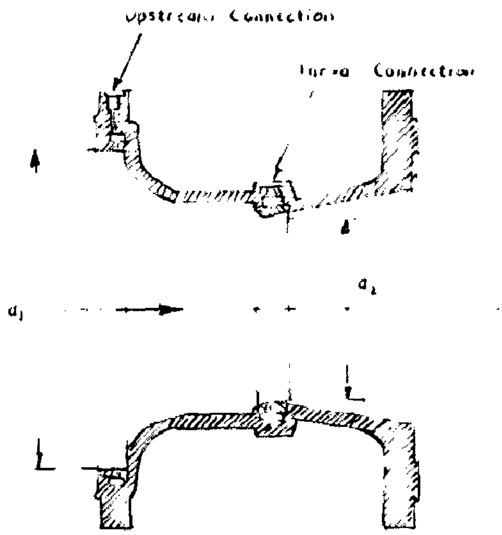
3. A divergent diffuser. This is the characteristics part of the venturi tube. Its adjunction of the converging portion allows a partial recuperation into pressure energy of the kinetic energy existing in the throat

FIG 5



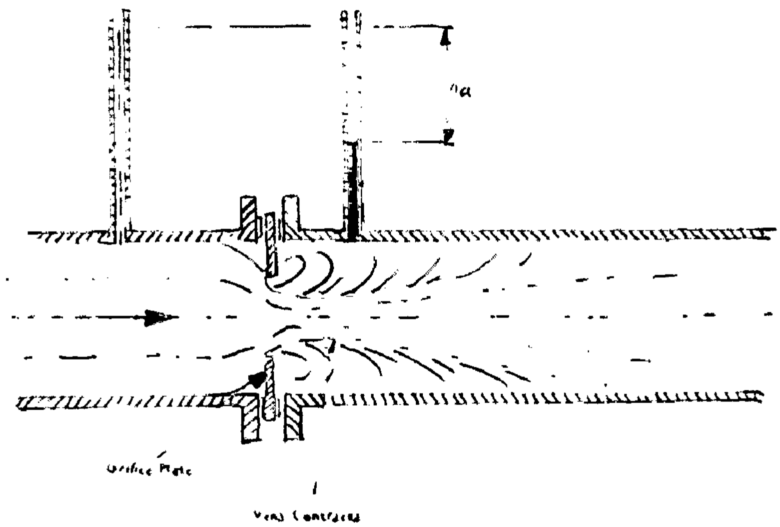
VENTURI TUBE

FIG. 6



DOUBLE THROAT VENTURI TUBE

FIG. 7



PLAIN-TYPE ORIFICE PLATE

section. This recuperation is progressive and is best eaffected when the eddying zone is small in the decelerating portion. The optimum value of the divergent angle is from 5 to 7^{o}.

When the diameter of the downstream section of the diffuser is smaller than the diameter of the pipe, the venturi is said to be truncated. Distinction is thus made between the following.

Venturi truncated with a divergent angle of 5 to 7^{o}.
Venturi truncated with a divergent angle exceeding 7^{o}
Venturi non-truncated with divergent angle exceeding 7^{o}
Venturi nontruncated with divergent angle 5 to 7^{o}.

<u>NOZZLE TYPE VENTURI TUBES:</u> (Fig. 5)

The nozzle type venturi consists of a standard I.S.A. 32 nozzle, followed by a cylindrical throat and a diffuser. Fig. 4 shows the characteristic of this device and makes apparent the essential difference between this type and the venturi tube and the standard nozzle. It will be noticed that where as in thelater, the downstream pressure hole is located in the down stream face of the nozzle, in the venturi, it is situated in the throat, immediately following the nozzle profile and preceding the cylindrical section. The first study of this type of venturi was made by Schlag. Schlag and

Jorrisen give results of nozzle venturi calibrations, effected between the years 1934 and 1947. A full discussion of all these will be given in Chapter 4.

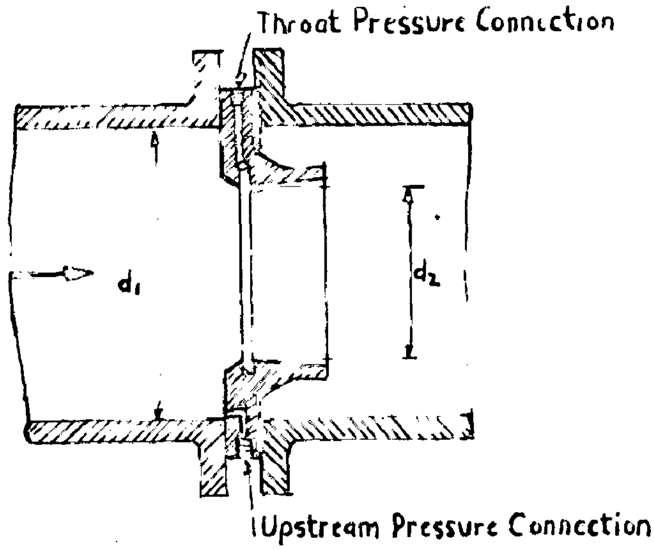
THE DOUBLE THROAT VENUTER TUBE (Fig. 6)

A s in the case of the Dall tube, the function of this device is to give a low pressure loss chaaracteristics. From Fig. 6 it is seen that in general the design of this detecting element is very similar to the standard nozzle, venturi tube, but at the end of the normal parallel throat section a second contraction is formed, by a buttress, and the pressure tappings are contained in this .

THE ORIFICE PLATE (Fig. 7)

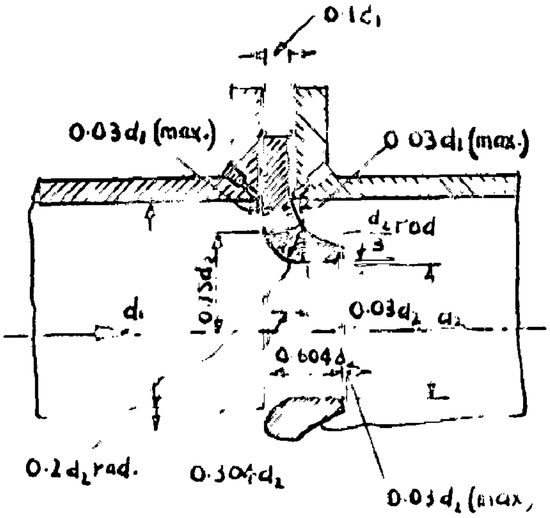
Figure seven illustrates schematically details of an orifice meter installation. The installation consists principally of a flat plate which has a hole in its centre, concentric with the pipe and in which is clamped between two pipe flanges so that its plane is perpendicular to the axis of the pipe. The pressure tap holes are drilled in the flanges and are connected to a U-tube manometer by means of small pipes. The change in pressure which occurs as the fluid flows through the orifice is indicated by the difference in liquid levels in the two legs of the manometer.

FIG 8



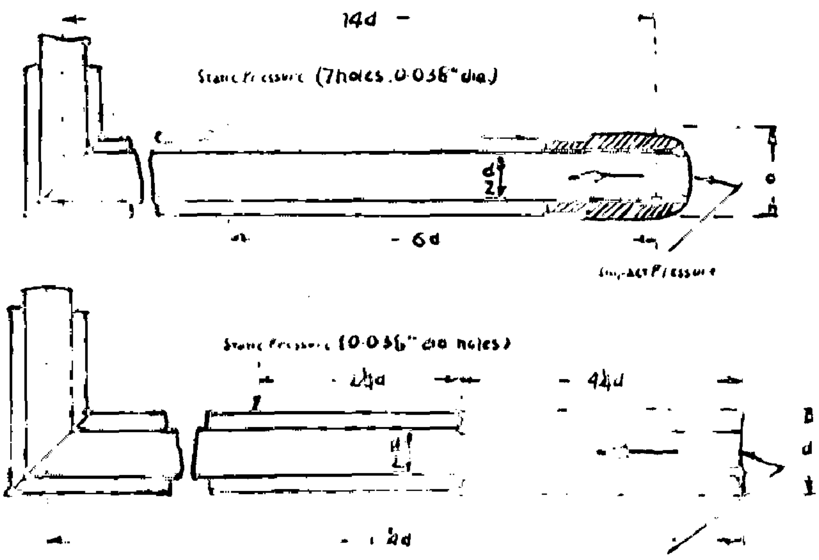
Dall Orifice

FIG. 9



Nozzle

FIG. 10



N.P.L. type Pitot Tube

The type of plate now universally adopted is of the square edge form . That is to say, the bore of the whole forms a right angle with the upstream face, while rounding the upstream edge would have the effect of reducing the contraction of the jet and thus increase the pressure loss, this operation will greatly increase the manufacturing costs. Thus by using the square edge, no special machinery or checking gages are required in the construction and duplication of orifice plates Since the plate has only to withstand the differential pressure it produces and not the static pressure in the main , it is made quite thin. The usual thickness is 1/16" for mains upto 6" diameter and 1/8" for mains of larger diameters. There are various arrangements for pressure tappings.

THE DALL ORIFICE: (Fig. 8).

The Dall orifice is a development of the Dall tube. At the expense of increased head loss, the included angle of both the upstream contraction and down stream expansion are made somewhat larger than in the Dall tube and also reduced in length. It may be regarded as shorten version of the Dall tube designed for insertion between the adjacent flanges in a pipe line; the shortened expansion cone projects into the downstream pipe. The gap required between the adjacent flanges is of the order of 1½ inches, and both the upstream and throat tappings are contained in the distance piece carrying the upstream and downstream cones.

THE NOZZLE (Fig. 9)

The nozzle type of differential pressure producting device is vertually a venturi tube with the curved form of approach but without downstream expansion cone. Various standard designs of nozzle are available of which the I.S.A. nozzle is the one most widely known. This is of the short form, the inlet consisting of two circular arcs of different radius. The design also is adopted by the British Standard Institution and the proportions for this are as illustrated in the Fig. 9.

Single tappings or a piezometer ring may be used, and a complete assembly can be provided in corporating both upstream and throat tappings.

THE PITOT TUBE [12] (Fig. 10).

The first description of a tube used to measure pressures for velocity distributions was given by Henry Pitot in 1732. Darey mentioned improvements to the instruments in 1854. Airey and Guy give a reasonable summary of the history and status of informations about the tubes upto 1910.

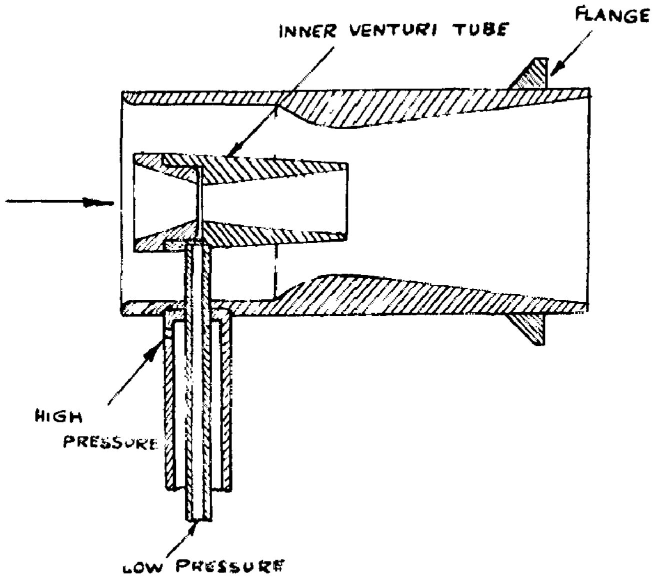


FIG. 11 - DOUBLE VENTURI PITOT HEAD

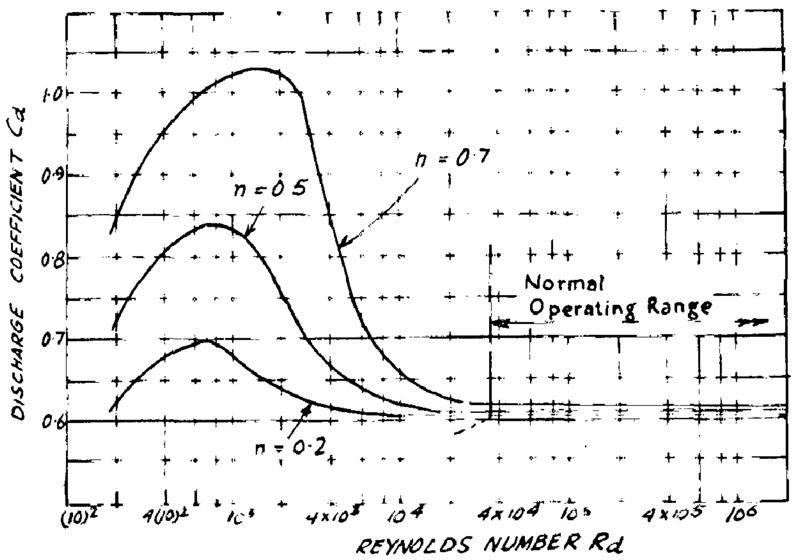


FIG. 41 - CURVES RELATING DISCHARGE COEFFICIENT WITH REYNOLDS NUMBER FOR ORIFICE PLATES

A pitot tube or similar velocity measuring device consists essentially of three parts, namely 1. The head or the instrument section.

2. The pressure connecting lines between the head and the pressure indicating device. and

3. The indicating device.

a. Pitot tube: A cylindrical tube with an open end pointed upstream, used in measuring upstream pressures.

b. Pitot static tube: A parallel or coaxial combination of a pitot and a static tube. The difference between the impact pressure and the static pressure is a function of the velocity flow past the tube.

When the velocity distribution is symmetrical with respect to the pipe axis

$$\alpha \frac{V^2}{2g} = \frac{\int_0^2 2\pi \gamma \frac{w}{g} \frac{u^3}{2} d\gamma}{\int_0^R 2\pi \alpha w u d\gamma} \quad (16)$$

Where $\alpha = \dfrac{\int_0^A u^3 dA}{A V^3}$

<u>THE PITOT VENTURI AND DALL PITOT</u> (Fig. 11).

For certain applications a higher differential pressure is required than that produced by a single or double tip pitot tube. The differential head may be magnified by mounting a small venturi tube on the end of a pitot tube. The lower pressure is obtained from the venturi throat, measuring point, and the differential head obtained is about four times greater than that produced by a single pitot tube.

A greater magnification can be obtained by using a Double venturi tube head of the form illustrated in fig. 11. The lower pressure is now obtained from the throat of the inner venturi, the downstream end of which coincides with the throat of the outer venturi. A flange may be mounted near the end of the outer venturi, The function of the flange being to increase the suction at the outlet end of the outer venturi, thus increasing to a limited extent, the amplification factor. The overall diamensions of this device are of the order of 1¾" dia. by 3½" long. An alternative design utilises a Dall-tube head. Depending upon the size of the duct or pipe, the differential head produced is from 6 to 10 times that produced by a single tip pitot tube. The flange of the end of the Dall tube has the function of increasing the amplification factor in the manner described above.

SUMMARY OF CHAPTER 1.

1. In the beginning of the chapter a short history is given in order that some idea can be gathered about the development of the differential pressure flow measuring device. Some historical note is also given

inside the Chapter whereever it is felt necessary.

2. Next a theoretical study of the fluid metering problem is analysed. From the analysis it is found that the true viscous or true turbulent flow in the closed conduit is not possible. There always exists some influences of both . But in the normal working range the flow is turbulent and the effect of viscosity is not predominant. Hence the turbulent flow characteristic is dealt with more detail.

3. The Bernoulli theorem for turbulent flow region is fully discussion and the flow equations are derived. In the derivation, the various effects that come in the picture have been incorporated. It has been more seen that the irrotational flow is assumed in the derivation of these equations. At high Reynold's number these conditions are in good agreement with the actual problems.

4. A simple theory of the bend meter is also given, because this also comes under the family of differential producer flow meters. Here also the irrotational flow is assumed. This assumption is valid because the effect of viscosity on the flow is negligible in the normal workin range of the liquid which is water in the present case.

5. A dimensional analysis for such types of meters is given. It is found from this analysis that the mass rate of flow can be predicted very easily if the geometrical similarity is maintained between the model and the prototype. More over the non dimensional factor i.e. the Reynolds number also should be same in both cases. In practice, there will be some difference from the theoretical value due to the pipe size factor.

6. Then a short description of different types of differential producer flow meters is given in order to know the geometrical characteristics of them. It is essential while investigating the difference in their working characteristics. Of course it is not possible to deal with all themeters in detail in this present work, as it will make a treatise of its own.

7. Endeavour is made to explain the working of these meters in order that while doing the comprative study of their characteristics, it may be of immense help.

8. While the Pitot tube is essentially a velocity measuring instrument, it has been included in the Chapter because of the fact that it is included in the same category of the differential pressure producing devices.

9. Since the experiments that are conducted in the laboratory are with water, the theoretical investigations are based on incompressible fluid flow. Some passing

remarks have been also given to take into account the compressibility factor in case of gases and vapours.

-:0:-

CHAPTER II

CHAPTER II.

EXPERIMENTAL SETUPS FOR SHORT VENTURIS AND DALL FLOW TUBES.

INTRODUCTION:

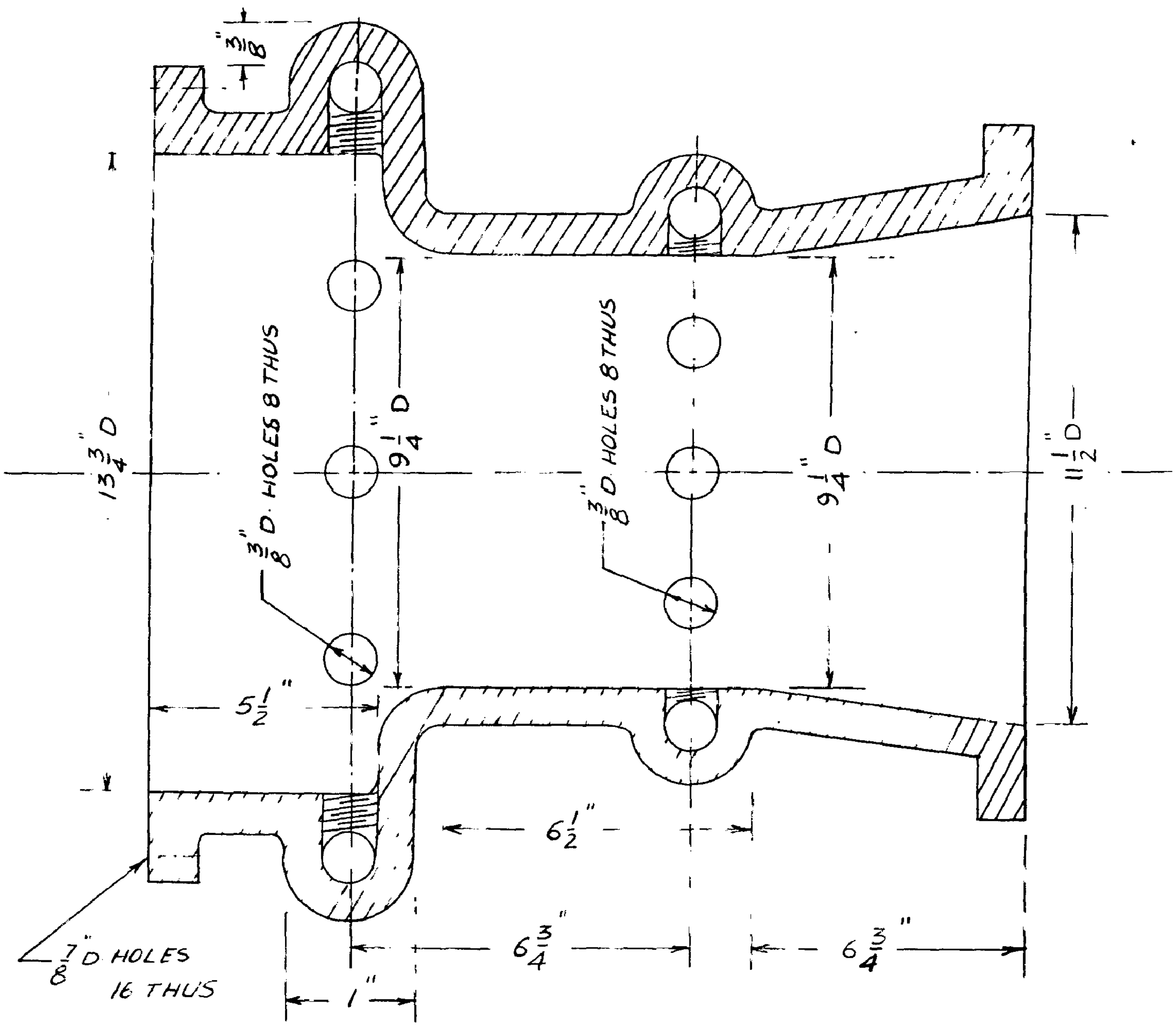
It was proposed to calibrate the short venturi tubes installed in the senior Hydraulic Laboratory and also to calibrate the Dall tubes. The calibrations of the short venturis having rounded entrance have been done first in experiment set ups No. 1 and No. 2. Then from this the calibration of the following meters are also done :

1. Bend meters, one on suction side and one on discharge side of the pump.
2. Calibration of rectangular weir of the Senior Hydraulic Laboratory.
3. Calibration of the shunt water meter used in experiment set up no. 1.
4. Calibration of water meter on filled in series with the short venturi in experiment no. 2.
5. Calibration of the recording meter used in experiment No. 2.

Thercalibration of a Dall tube which was installed in Junior Hydraulic Laboratory is also done. A model

FIG 13

SHORT VENTURI DIMENSIONS ON MAIN PIPE $13\frac{3}{4}$" DIA



was prepared for the Dall tube of 6 inches diameter filled in the Junior Hydraulic Laboratory and the calibration curve for this also is dealt with here.

Finally the discussion of all the results obtained is given at the end. The experimental observations are attached at the end in Appendix.

EXPERIMENTAL SET UP NO. 1. (Fig. 12 and 13)

Centrifugal Pump: - Escherwyss, Zurich

H = 18 m. n = 97.0 T/min.

Q = 230 l/sec. N = 71.5 ps.

No. 7922.

The centrifugal pump as mentioned above is installed for the 13¼" pipe line in the Senior Hydraulic Laboratory. The crossectional view of the short venturi used for flow measurement is as shown in Fig. 13.

PLAN OF PROCEDURE:

The first and foremost problem in this experiment is to measure the correct discharge. Since the quantity of discharge is large enough to be handled by any measuring tank, other means are resorted to.

It has been found by experiments that if the flow to any converging cane is smooth andthe stream lines take the form of entrance cone, the head loss is very small and is equivalent to loss of head due to friction through a pipe of 1' long and uniform diameter [15]. This statement is given with the assumption that thereis no further contraction beyond the contracted section. For a pipe of 13¼" inside diameter the equivalent length is 3'-6". This assumption is correct in this experiment, because the entrance section is smooth as shown in Fig. 13. The theory to be dealt with in the Chapter III supports this statement also. To find the correct values of discharge coefficient, the method described in Chapter III is adopted and the values of discharge coefficients, discharge and other factors are calculated as shown in Table 1 of Appendix.

In order to get the frictional head loss in the meter with the help of equivalent length as found above, observations were taken for the head loss in straight portions of the uniform pipe. The distance between two points in the same horizontal plane, is 20.5'.

CALIBRATION OF THE METER

The physical properties of water are determined at operating temperature of 74^{o}F. At this temperature the value of absolute viscosity μ is 1.92×10^{-5} lbsec/ft^2. The density of water = 62.25 lbs/cu. ft. A calibration curve is drawn between Reynold's number $(R_e)_D$ and

FIG 15

CALIBRATION OF SHORT VENTURI ($13\frac{3}{4}''$ DIA)

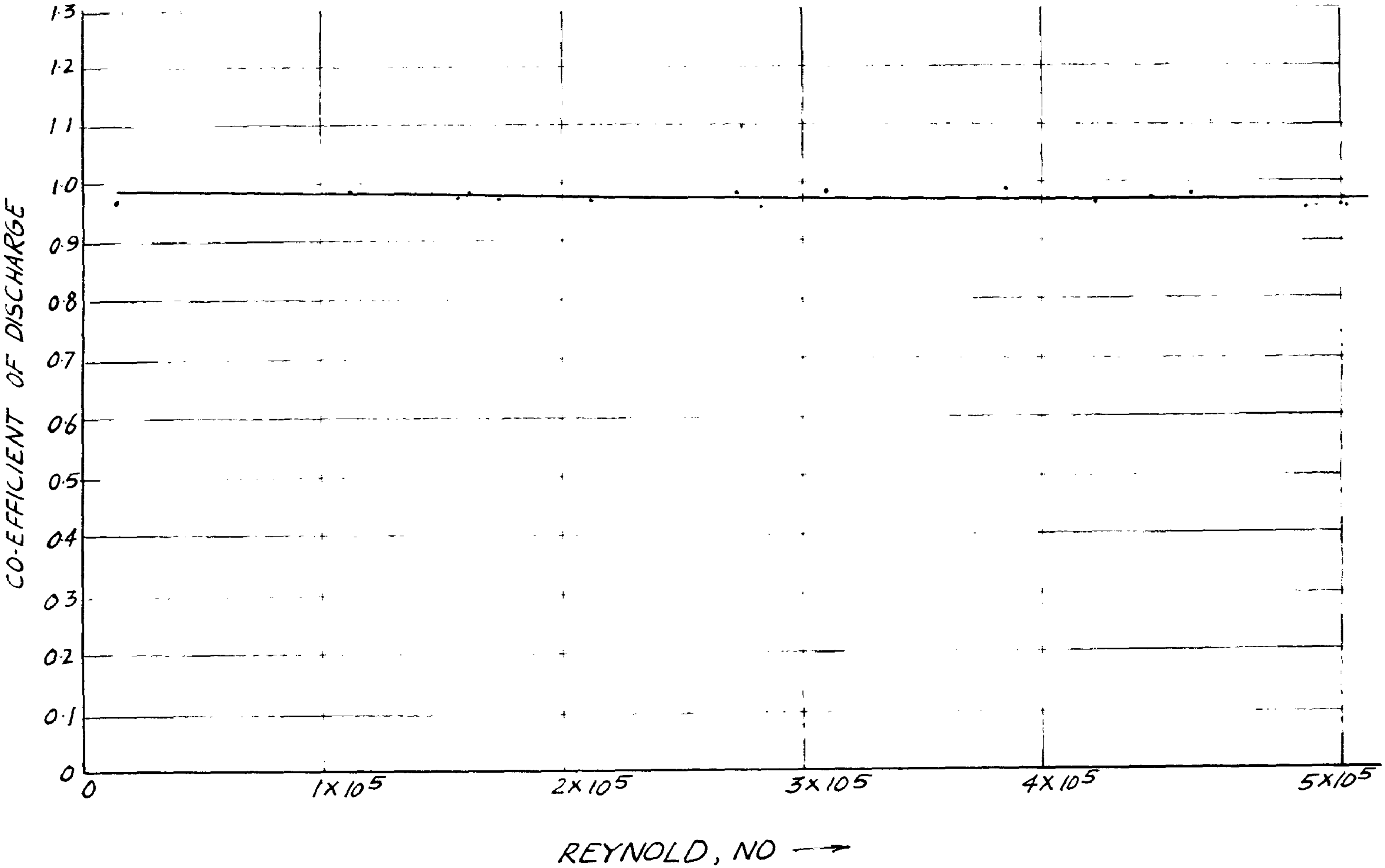
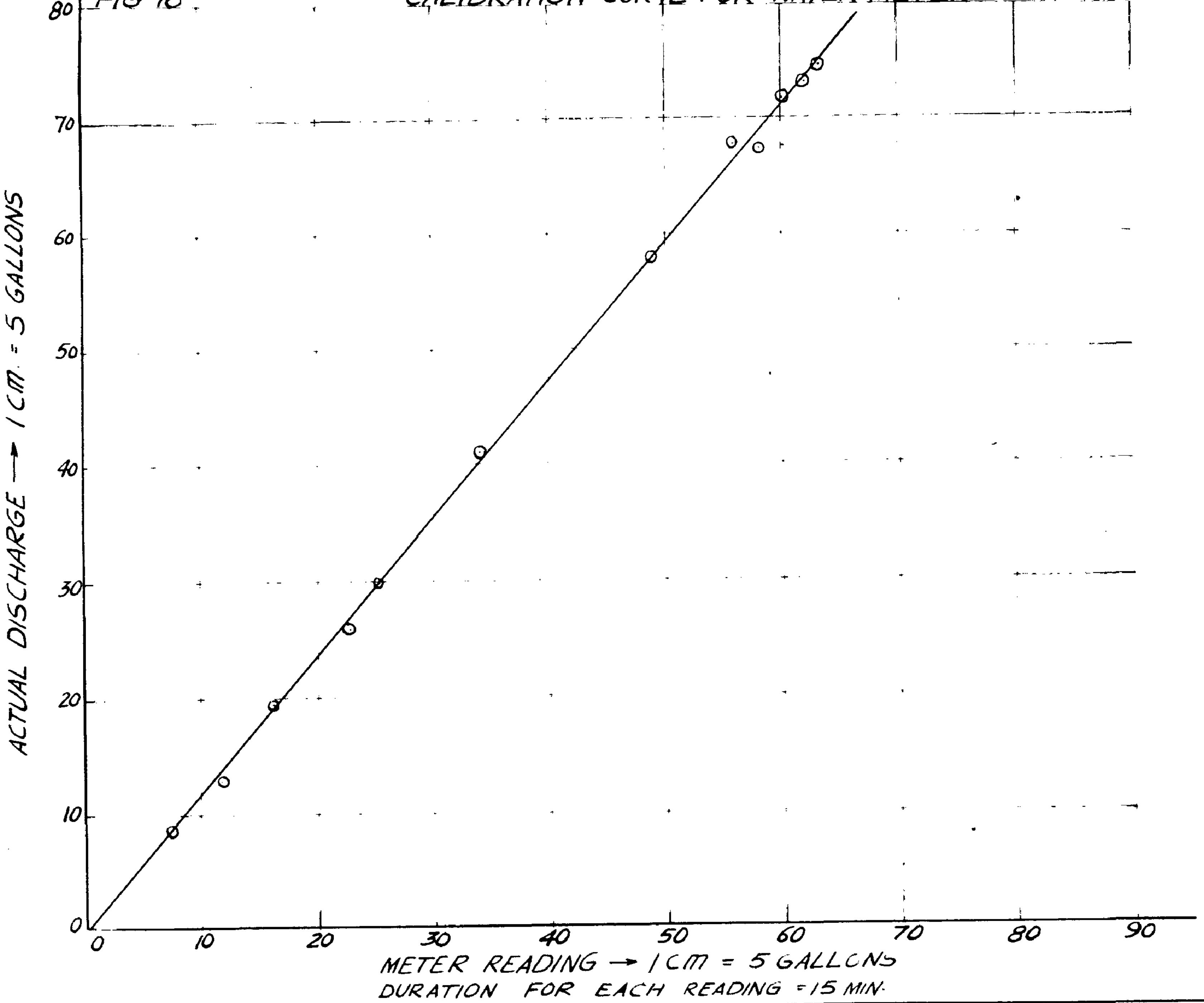


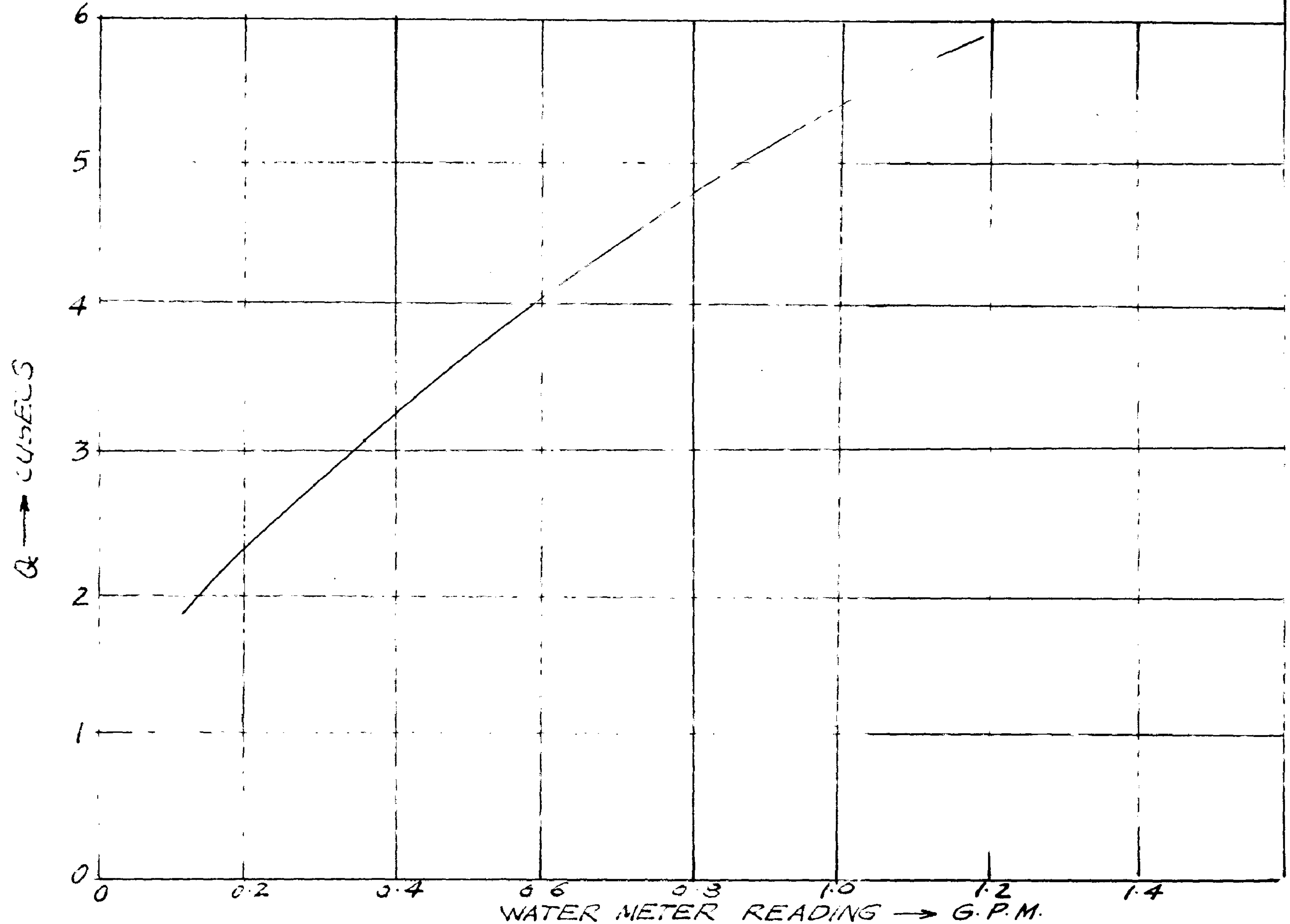
FIG 16

CALIBRATION CURVE FOR WATER METER



FIG·17

WATER METER READING VS DISCHARGE THROUGH MAIN PIPE



the coefficient of discharge C_D as shown in Fig. 15 .

CONCLUSION:

It is seen from this curve that the coefficient remains practically constant during the operating range of the meter. Moreover the flow is turbulent during the range of operation. The value of the coefficient of discharge falls slightly at high Reynold's number as it is expected.

THE SHUNT WATER METER :

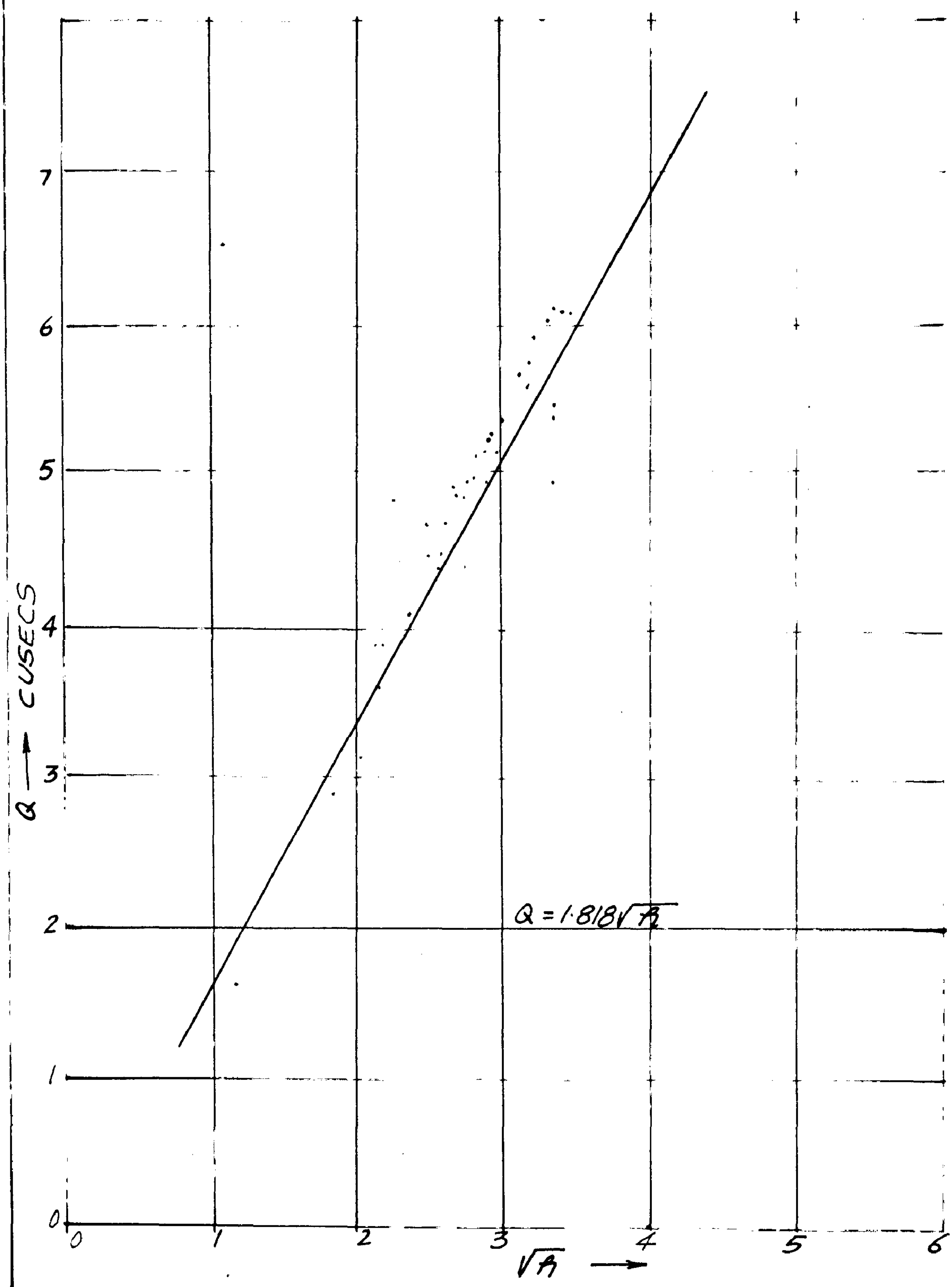
For the purpose of correlating the discharge between the main pipe line and the water meter - meter used parallel to the main line, a set of simultaneous observations are taken. Before using the water-meter, it is calibrated and a calibration curve is drawn as shown in Fig. 16. For each observation the time allowed is fifteen minutes. The values of water-meter readings are given in column 7 of the Table 1. The purpose of the use of the shunt water meter was that for a given water meter reading, the discharge in the main pipe could be obtained.

CONCLUSION:

A graph is drawn between the discharge through the water meter and the main pipe line. It will be seen from this graph Fig. 17 , the there is no good relationship

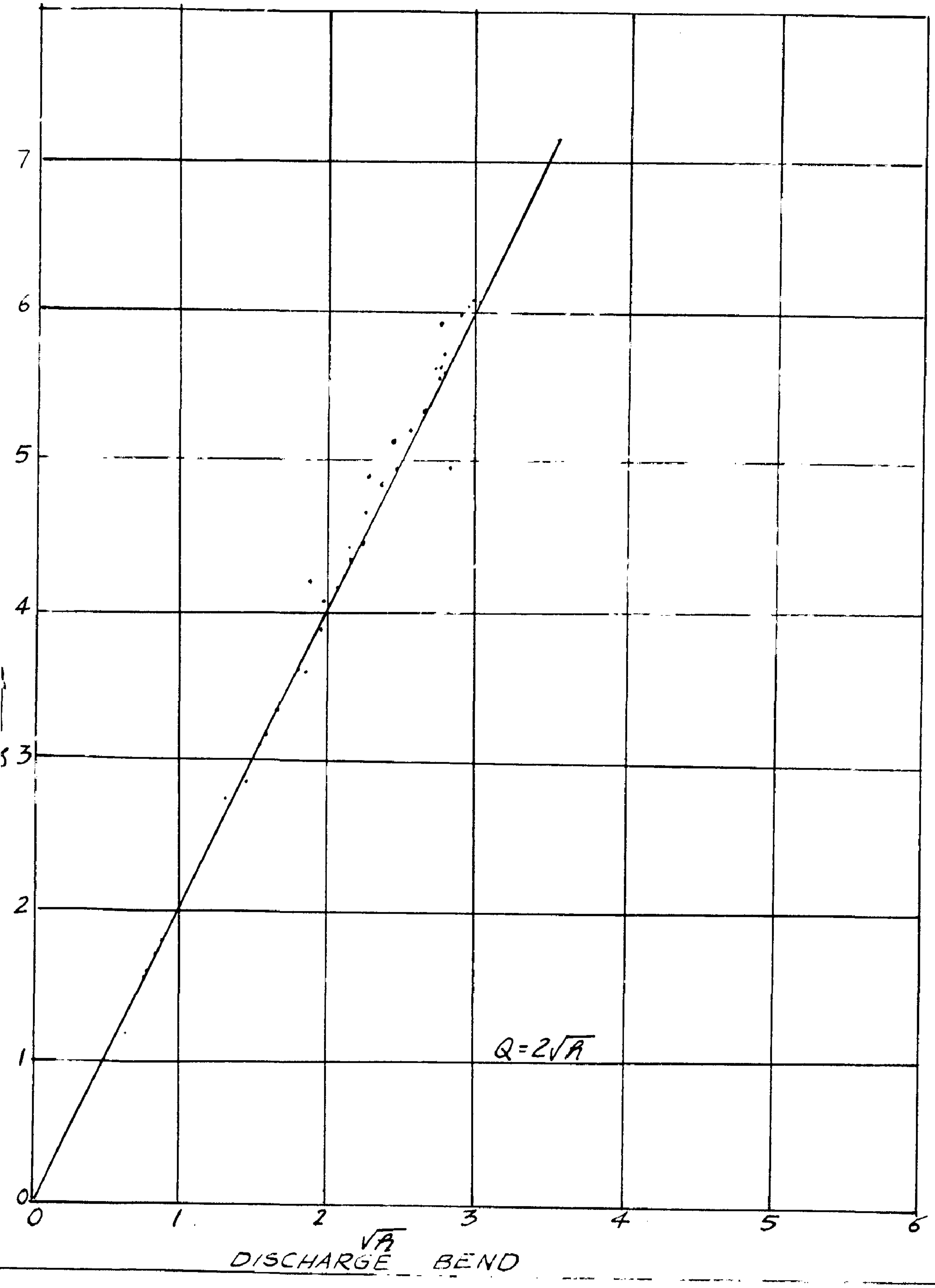
DISCHARGE THROUGH PIPE VS SQUARE ROOT OF SECTION BEND MANOMETER READING

FIG. 18



DISCHARGE THROUGH PIPE VS SQUARE ROOT OF DISCHARGE MANOMETER READING

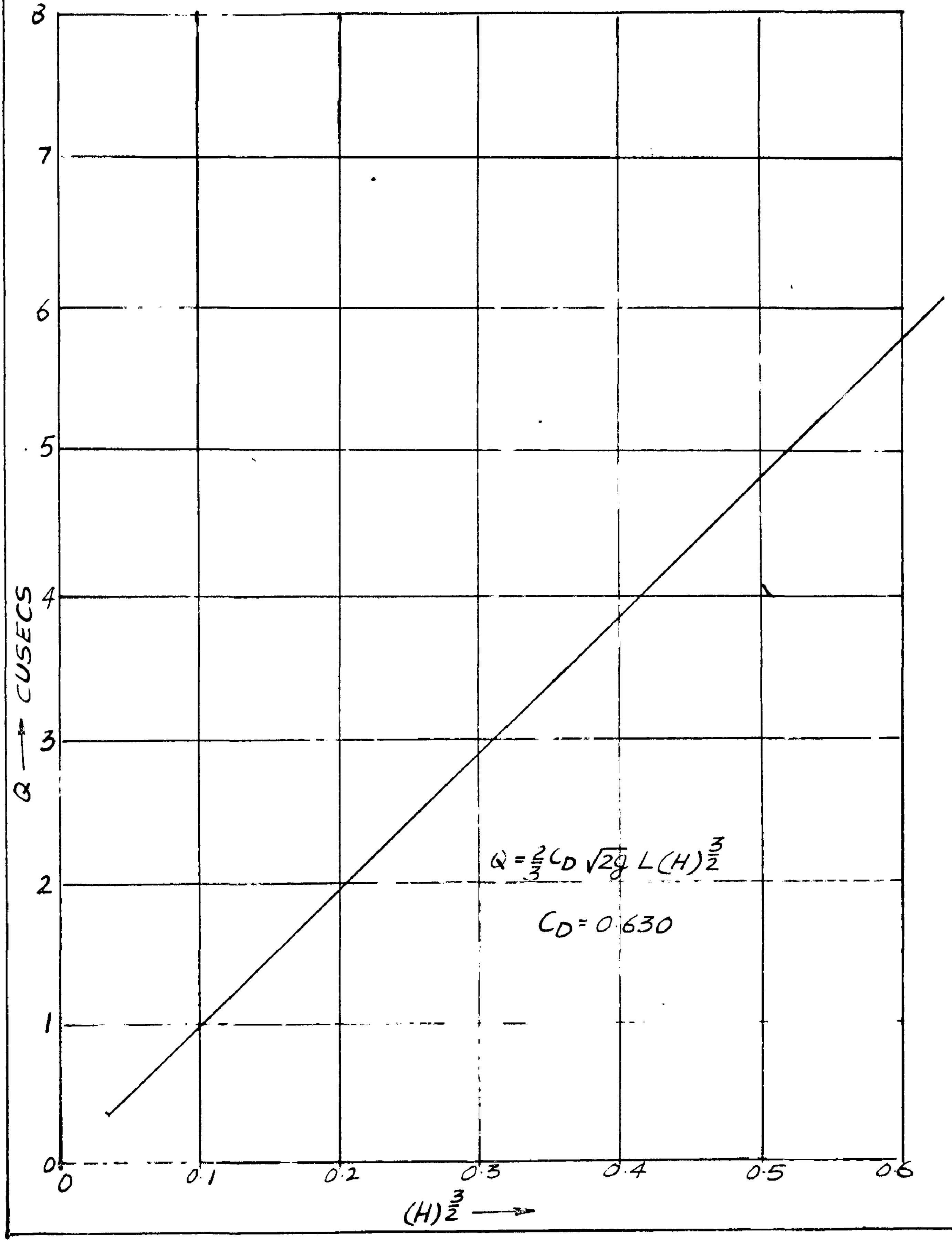
FIG. 19



Q DISCHARGE VS $(H)^{\frac{3}{2}}$ FOR RECTANGULAR WEIR

FIG 19(a)

EXPT. NO. 1



between the water meter reading and the discharge in the pipe line. But a smooth average curve is drawn to give the approximate discharge through the main pipe line.

PRIMARY ELEMENT - CENTRIFUGAL TYPE:

There are two right angled bends in the pipe line, one is on the suction side of the centrifugal pump and the other is on the discharge side of its. The manometers are filled into these bends as shown in Fig. 1, to read the pressure differential between the two radii of the bends. Then the graphs are drawn between square root of the differential head and the discharge through the pipe. Fig. 18 gives such a graph for the suction side and Fig. 19 shows the relationship for the discharge side. The values of the differential pressures read in inches of water are tabulated in Column 1 and 2 of the Table 1.

CONCLUSIONS:

From Fig. 18 $\quad Q = 1.818\ (h)^{\frac{1}{2}}$

From Fig. 19 $\quad Q = 2\ (h)^{\frac{1}{2}}$

where h - differential pressure in inches of water.

Q - discharge in cubic ft/sec.

The graphs are straight lines passing through the origin. There is a little difference in the constants in the equations because, the suction and discharge bends are not of the same radius . More over there is some amount of water ~~through the~~ lost through leakages in bearings of the centrigugal force. Hence the discharge through the suction bend will be more than what passes through the discharge bend.

EFFECT OF DISCHARGE HEAD ON COEFFICIENT OF DISCHARGE:

During the experiment, the head causing flow was varied and different observations were taken by keeping the discharge head from the centrifugal pump constant. The purpose was to vetify it there could be any effect of discharge head on coefficient of the meter. But it is found that there is no appreciable effect of discharge head on coefficient of dischargeof the short venturimeter. The coefficientpractically remains constant and varies slightly with the Reynolds number.

CONCLUSION ON EXPERIMENT NO. 1.

The discharge coefficients found from the experiments are very high and there is a very slight variation of them with the Reynolds number. After getting the discharge by the method mentioned in Chapter III, the other meters are calibrated. A calibration curve for rectangular weir is also drawn, It is seen that the constants obtained are nearly equal to the constants given by Bazin's formula.

The bend meters are calibrated and hence with the help of them flow in the pipe line can be predicted with reasonable accuracy. The discharge coefficient of the meter is very high as is expected from this type of smooth entrance flow meters. Consequently the losses in the entrance portion are less.

The percentage pressure loss is very high and is nearly 30%. This is because the tappings for measuring pressure loss are very close to the venturimeter where the full recovery of the pressure is not attained. The normal flow pattern is not obtained just nearer to the venturimeter and hence the pressure loss as indicated by the meter is very high. Nevertheless the after sufficient length ahead of the meter flow becomes normal and the full recovery of pressure is obtained.

EXPERIMENTAL SET UP No. 2. (Fig. 20)

Two stage centrigugal pump.

Escher Wyss - Zurich. n - 2900 T/min.

H_m - 76 meters N - 25 ps No. 7932.

The cross-sectional view of the meter is same as the previous one. This short venturi is installed in the pipe line connecting the pelton wheel and the two-stage centrifugal pump as mentioned above. Here the same procedure of experiment is the same as described for experimental set up No. 1. The water-meter is filled directly in series with pipe line. The distance between the tappings for frictional head loss measurement is 10'-6". Since

FIG 21

CALIBRATION CURVE FOR SHORT VENTURI PELTEN WF

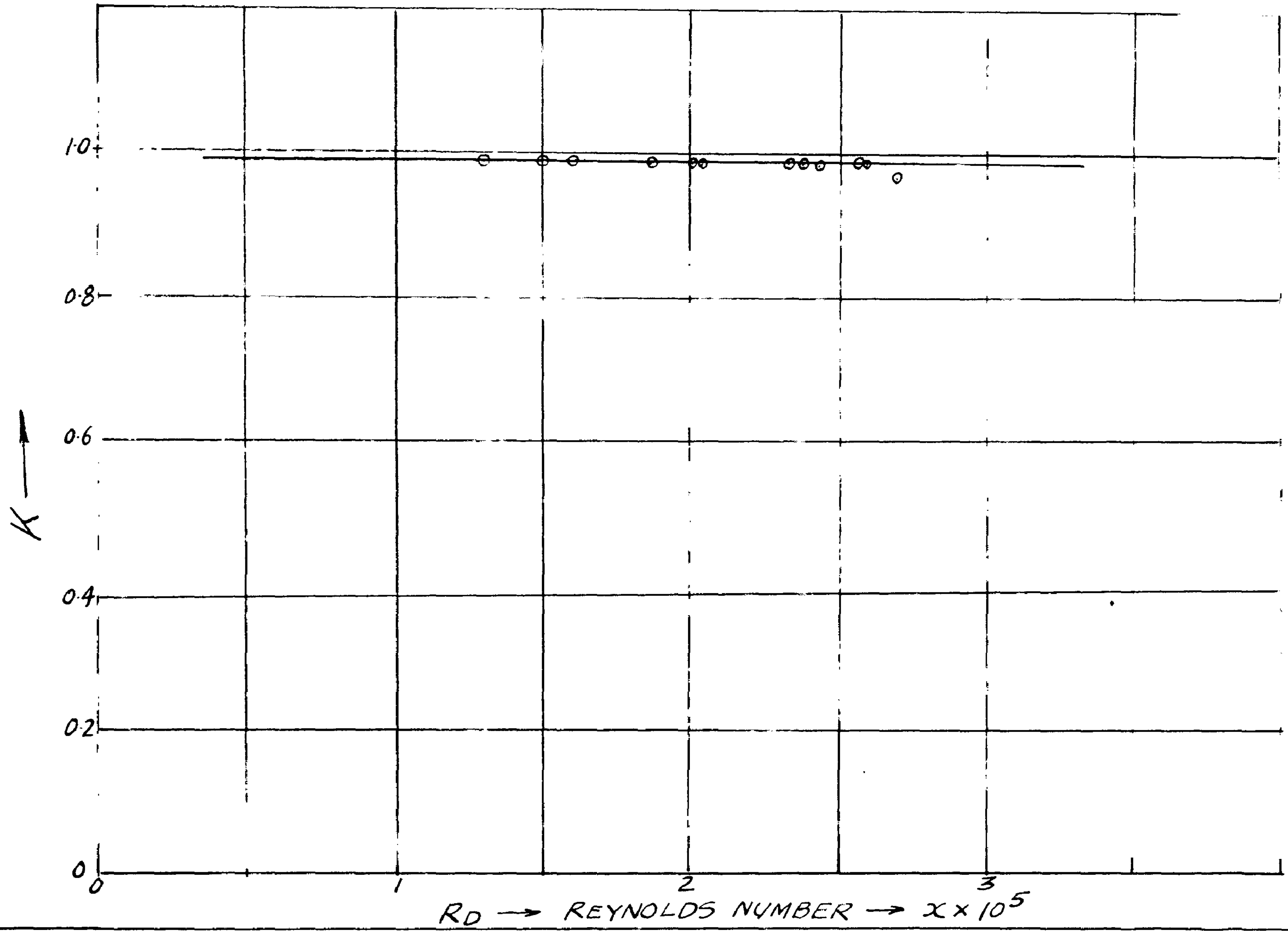
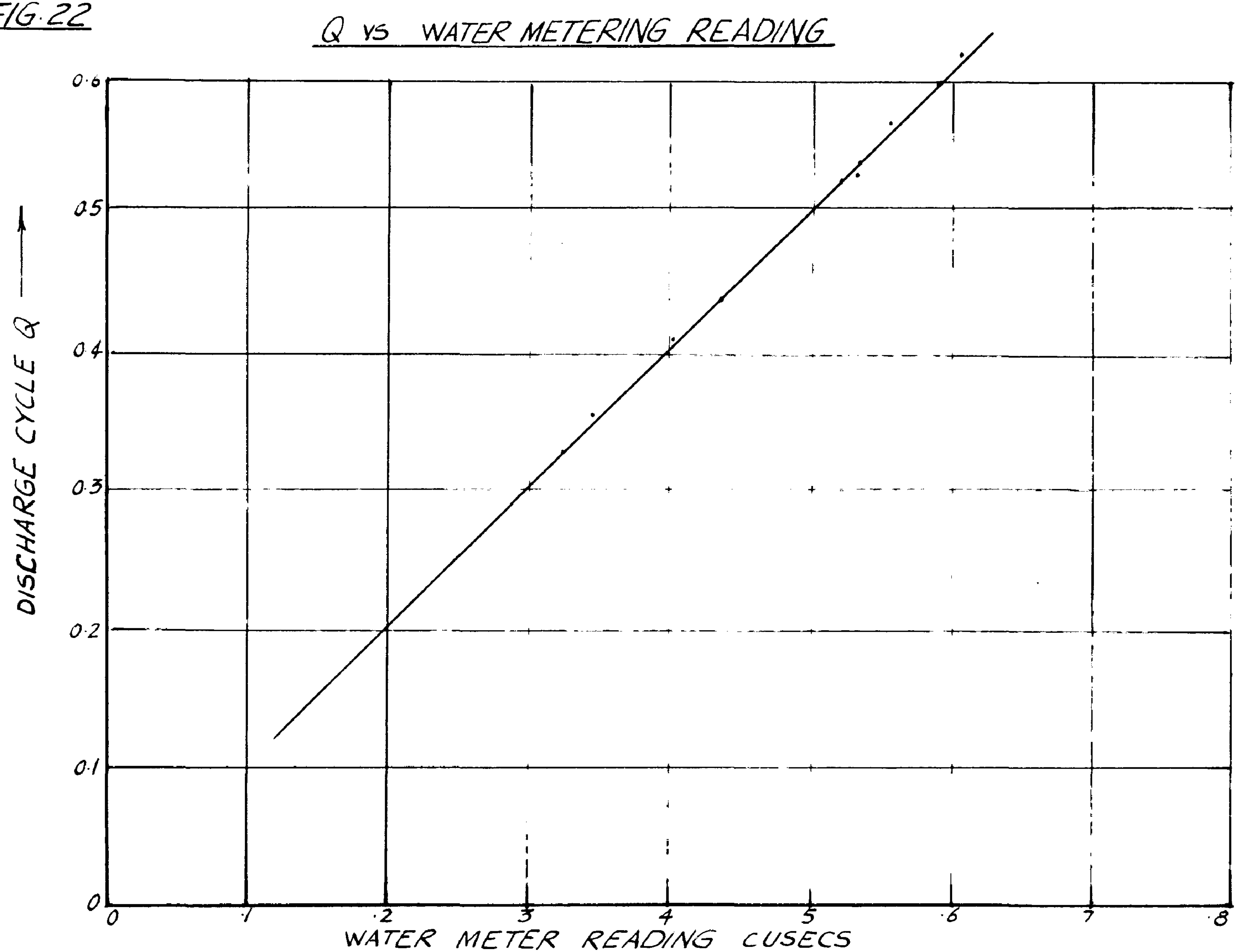


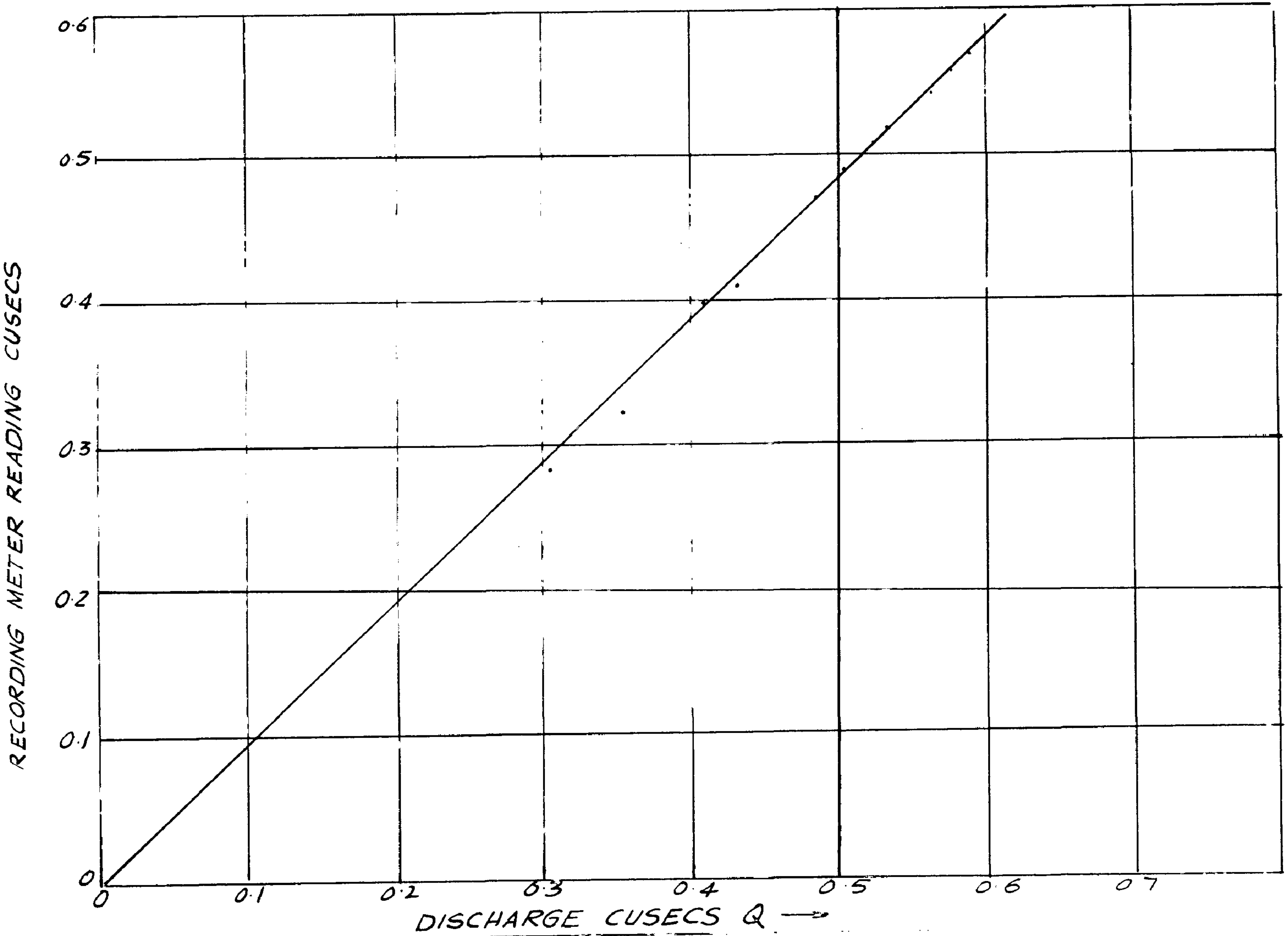
FIG. 22



Q VS WATER METERING READING
DISCHARGE CYCLE Q
WATER METER READING CUSECS
0·6
0·5
0·4
0·3
0·2
0·1
0
·1
·2
·3
·4
·5
·6
·7
·8

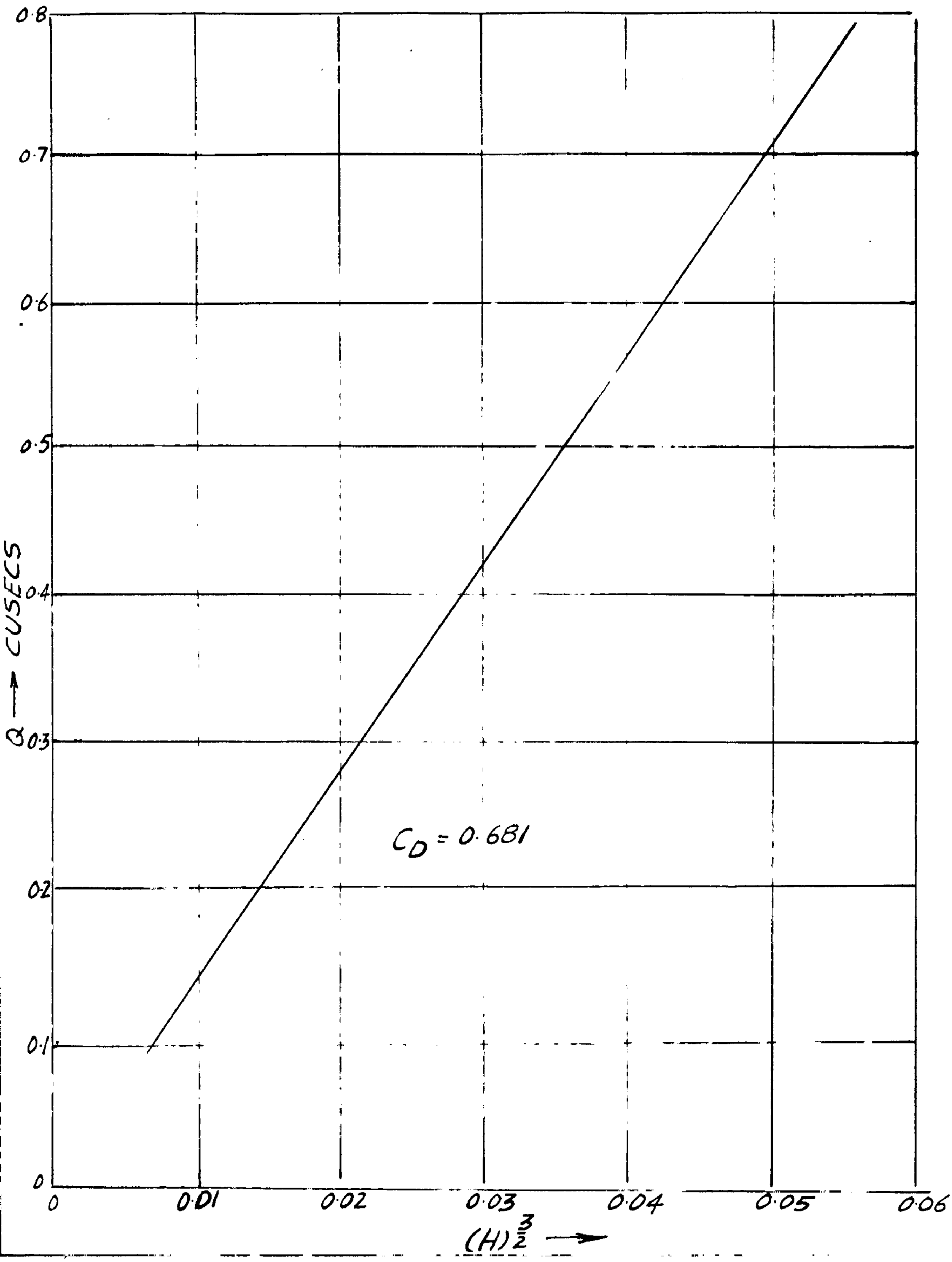
FIG 23

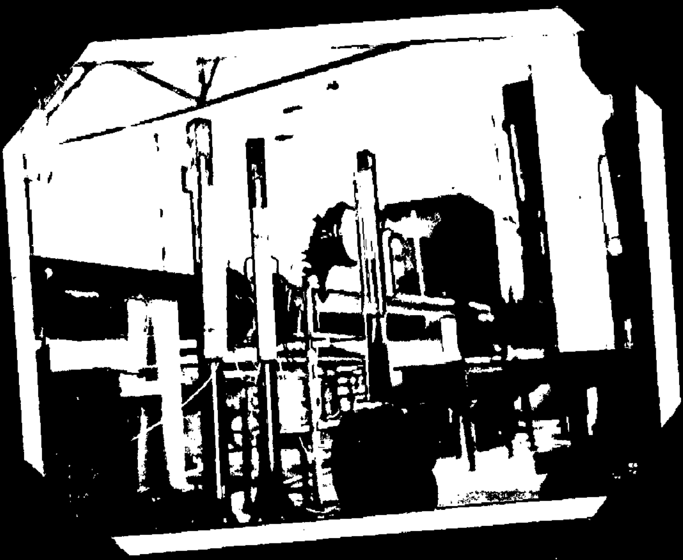
CALIBRATION OF RECORDING METER



DISCHARGE Vs $(H)^{\frac{3}{2}}$ FOR RECTANGULAR WIER EXPT·2

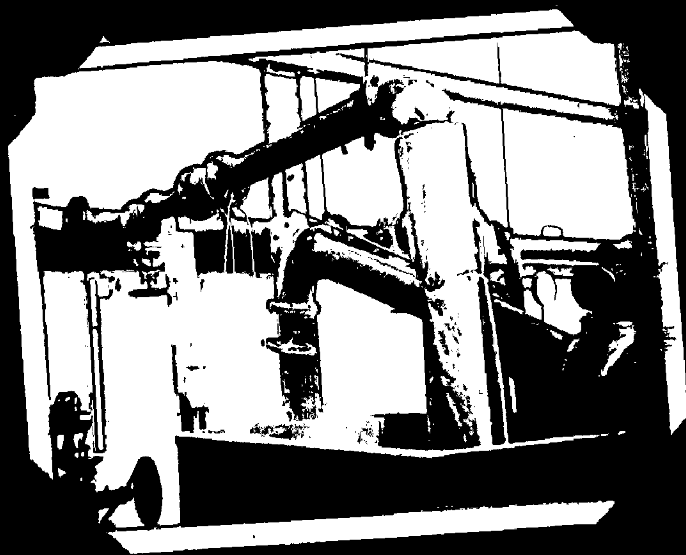
FIG. 23(a)





Expt. No.1 - Fig. 12

Expt. No.2 - Fig.20



Expt. No.3 - Fig.24

Expt. No.4 - Fig. 26_b

the short venturi meter is installed in between these tappings, the loss in the meter is substracted from the reading and the remaining head loss is the loss in the length 8.75 ft. Both the methods of calculationsare used here also and the values are tabulated in the Table No. 2.

CALIBRATION:

A curve is drawn Fig. 21 between the Reynold's number and the coefficient of discharge. A calibration curve for water meter is also drawn as shown in Fig. 22. and Fig. 23 shows the calibration of the recording meter attached to the venturi meter.

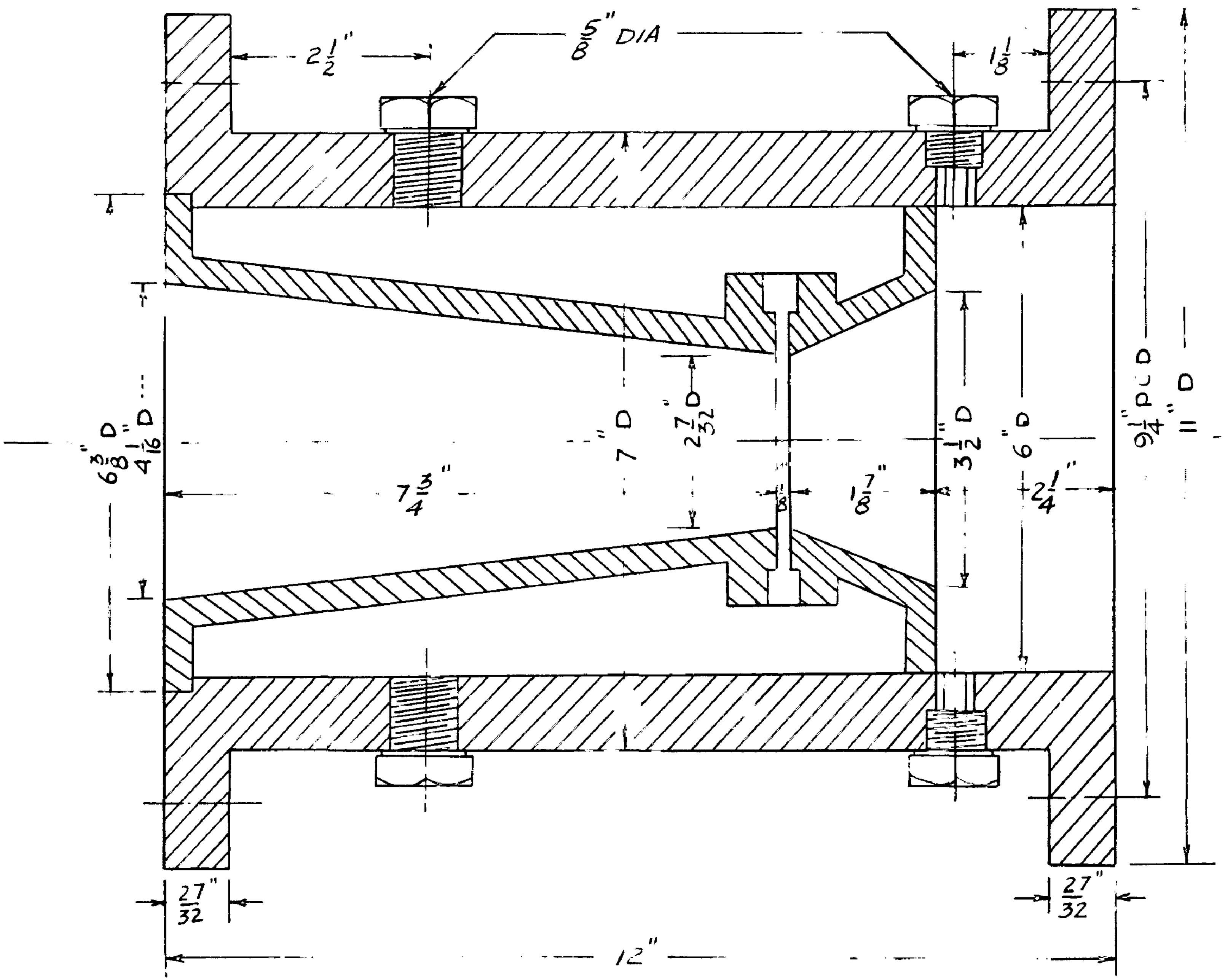
CONCLUSION:

After getting the discharge with the help of rounded entrance flow meter theory as mentioned earlier the rectangular weir is also calibrated. The coefficient of discharge from Bazin's formula agrees fairly well.

From the calibration curve Fig. 21, it is seen that the discharge coefficient is high and it remains constant during the operating range. Moreover there is slight fall of value at high Reynolds' number. The result is same as obtained for the previous set of experiments. Since the venturimeter is a smooth entrant one, the losses are small in the entrance cone, hence the coefficientof discharge is high.

FIG. 26(a)

DALL TUBE



The calibration curve for water meter is drawn and it is a straight line. Hence for any reading of the water meter. The direct reading of discharge can be obtained from this graph. Similarly the recording meter attached to the short venturi is also calibrated as in Fig. 23 and it can show the correct discharge for any reading exibited by the meter in the operating range.

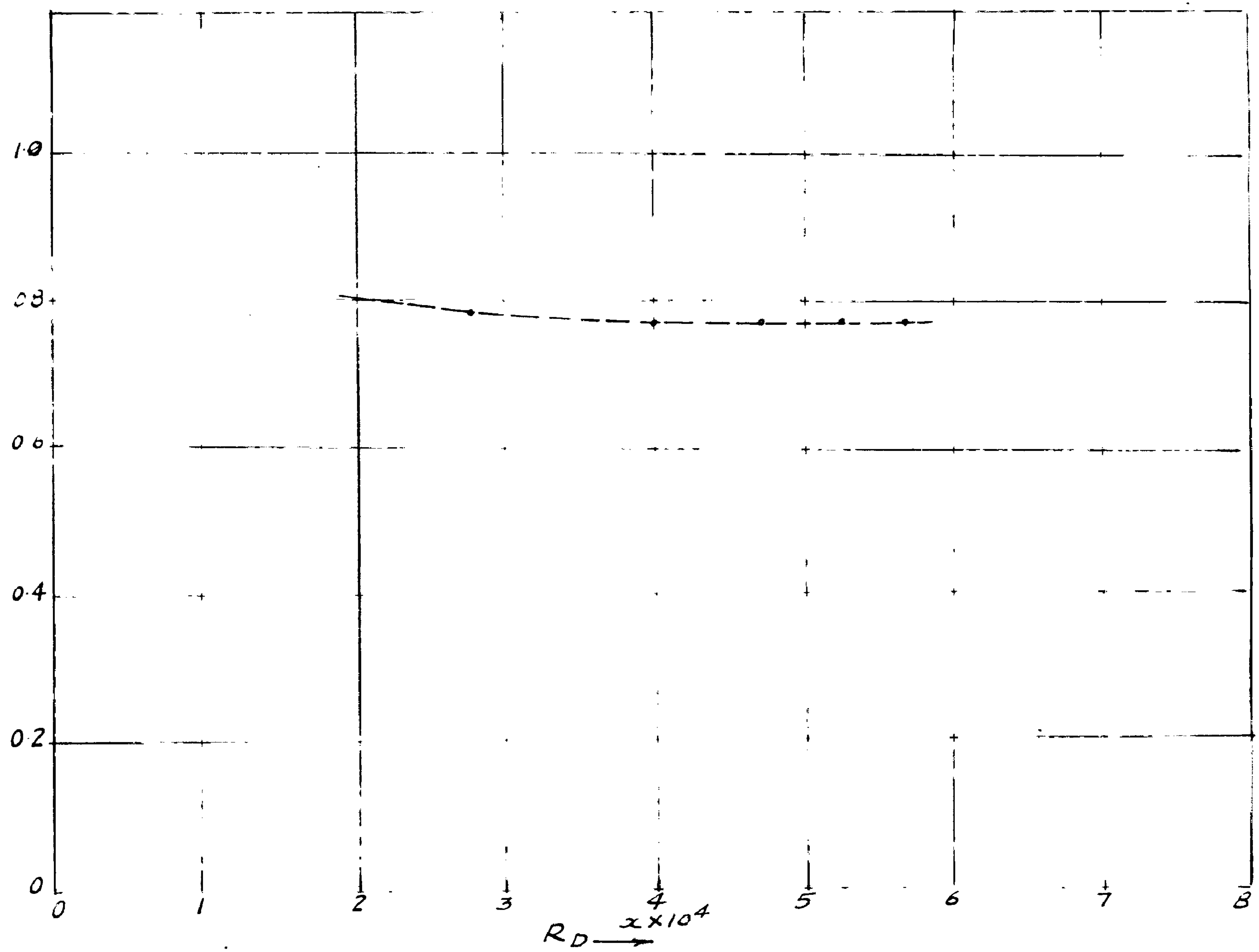
The percentage head loss is calculated and tabulated as in column 16 of Table 2. It will be seen that the percentage head loss in the meter is quite small. For a diameter ratio of 0.7031 , the percentage head loss as obtained for the short venturi is nearly 10.5% as will be seen later in Chapter IV. This is obtained by experiments. In this case of rounded entrance flow meter the head loss is less during the operating range and varies from 11 to 6.9% and it is seen from column 16 of Table 2 that the percentage head, is nearly same as is previously obtained. From the observations it is seen that most of the percentage head loss values are nearly equal to 9%.

EXPERIMENTAL SET UP NO. 3. (Fig. 24).

A Dall Flow tube is installed in the Junior Hydraulic laborator in a 6 inch pipe line as shown in Fig. 24. A measuring tank is used to know the correct discharge. A set of observations is shown in the Table 3. The purpose of this experiment was to calibrate the Dall Flow Tube

FIG 25

CALIBRATION CURVE FOR D.F.T ON 6 IN PIPE LINE



and to study its pressure loss characteristics.

CALIBRATION:

A calibration curve is drawn as in Fig. 25 between Reynold's number and the coefficient of discharge. It will be seen that the discharge coefficient starts to vary below a Reynold's number of 3.5×10^4 and at higher Reynold's number it remains practically constant. Moreover the shape of the calibration curve is nearly same as the orifice meter as will be seen in Chapter IV.

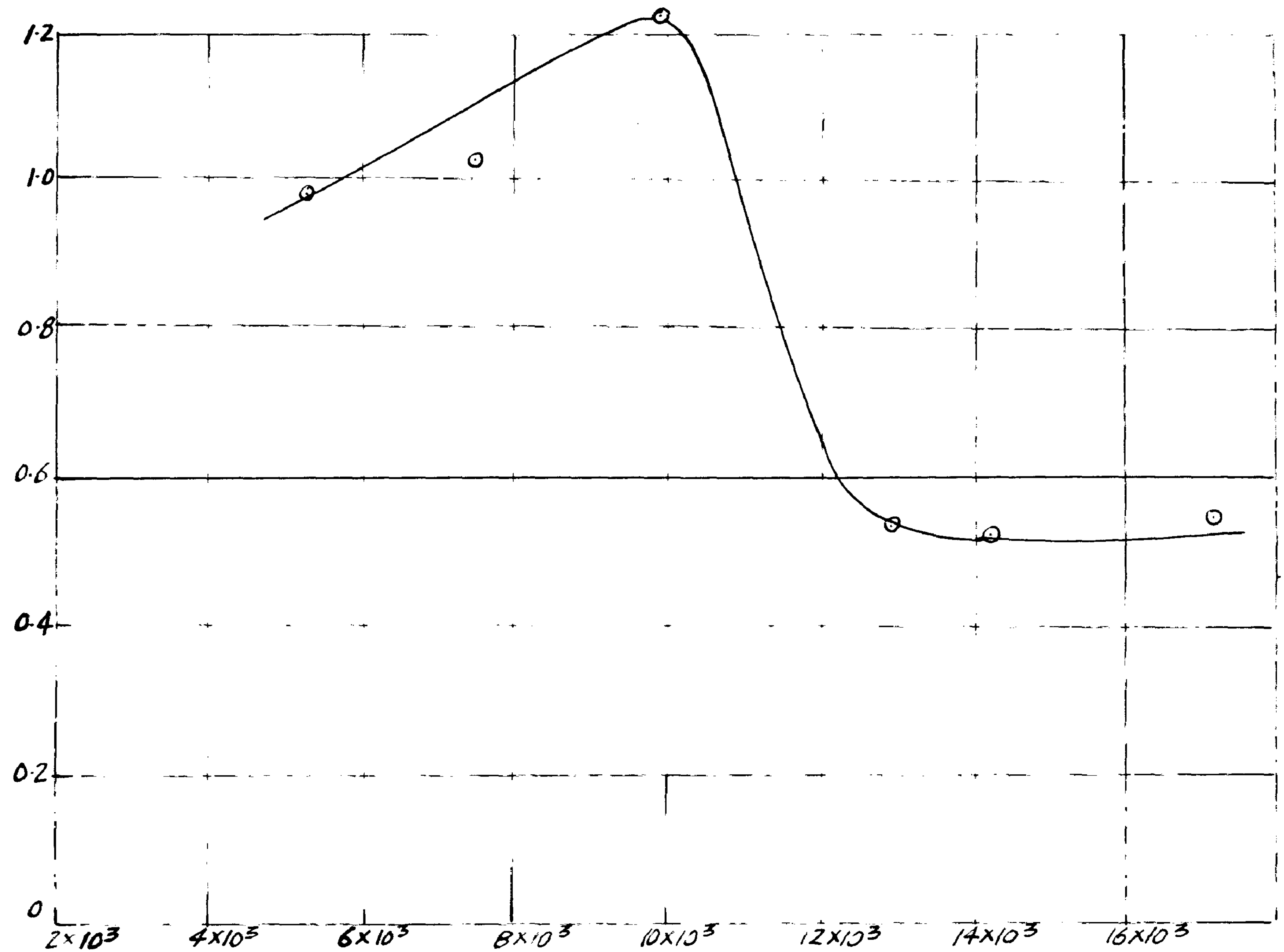
PRESSURE LOSS:

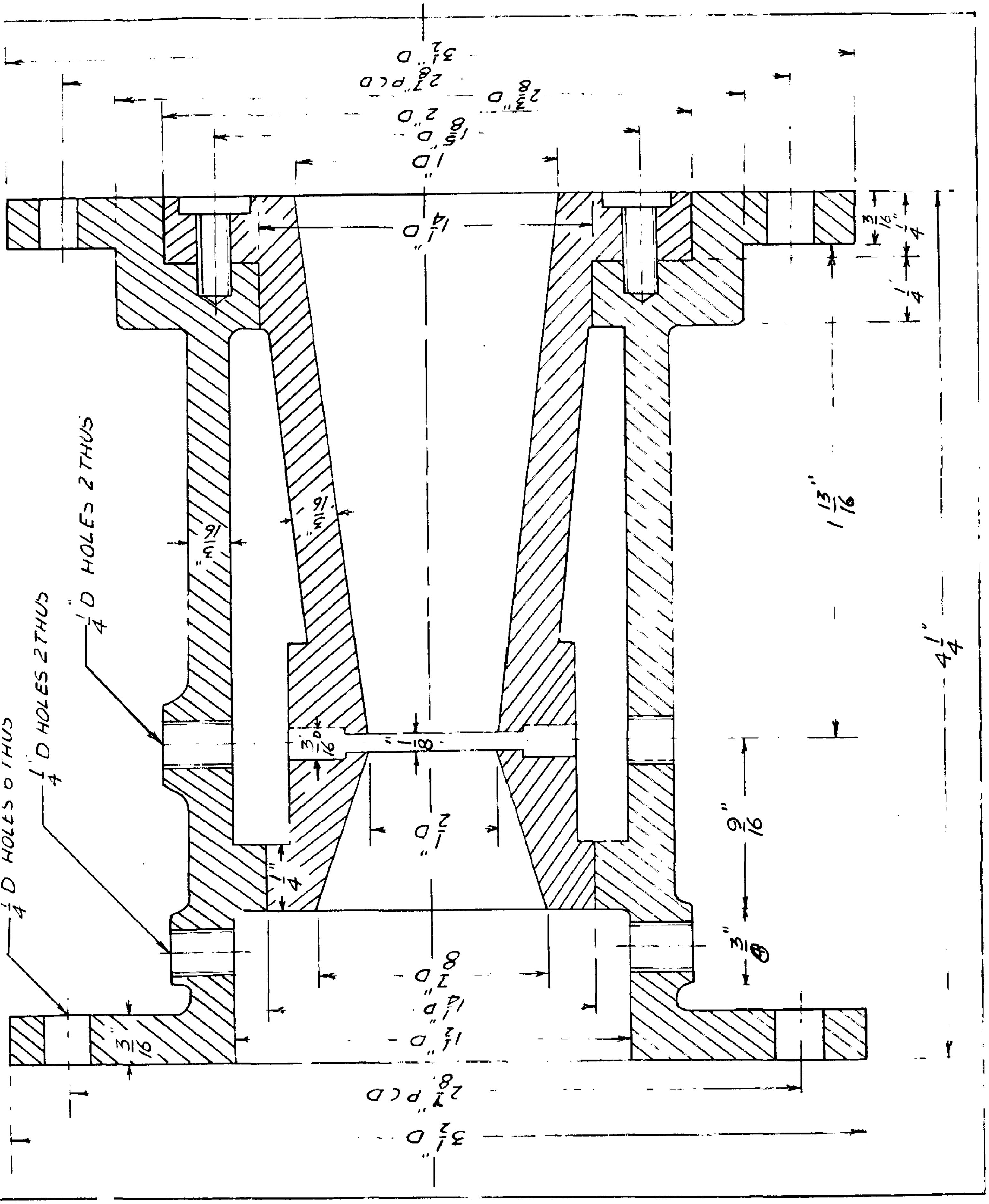
The friction factor for the calculation of frictional head loss in a length of 1'-9" between the pressure taps for measuring the pressure recovery is obtained from the graph given in (16). The friction factors are taken for different Reynold's number and for a pipe diameter of 6 inches. The calculated equivalent head loss is substracted from the readings of column 3 Table 3. The percentage head is calculated and tabulated in Column 8 Table 3.

CONCLUSION:

It will be seen that the head loss in the meter is very high. Of course it does not represent the full recovery of the head loss. Because the tappings for pressure loss measurement are very close to the Dull Flow Tube and hence full recovery of the differential head is not obtained.

FIG. 26 CALIBRATION CURVE OF FLOW TUBE EXPT. 4




1/4" D HOLES 6 THUS
1/4" D HOLES 2 THUS
1/4" D HOLES 2 THUS
3 1/2" D
2 7/8" P C D
2 3/8" D
2" D
1 5/8" D
1" D
1 1/4" D
1/2" D
1/8"
4 1/4"
1 13/16"
9/16"
3/8"
3/16"
1/4"
3/16"
1/4"
1/4"
3/16"
3/16"

More over the lengths of the straight pipe before and after the Dull Flow Tube are very small . Due to bends and gate valve nearer to the meter, the flow is highly turbulent and helical flow pattern exists. But nevertheless the differential pressure produced is quite high as is expected.

EXPERIMENTAL SET UP NO. 4. (fig. 26).

DESIGN OF A NEW DALL TUBE:

It was proposed to install a new Dall tube for a 1½ in diameter pipe line in the senior hydraulic laboratory For this purpose a Dall tube was designed and was manufactured in the workshop.

GOVERNING CONSIDERATIONS FOR DESIGN:

The process of designing the elements of an installation for a particular metering problem differ from calculation of flow rate in that it is basically an approximation. Thus for design purposes, a detailed evaluation of all terms in flow equations is not necessary and simplified methods may be employed in selection of meter size and manometer range. On the other hand subsequent calculation of meter factors or flow rates based on this design, require the use of complete flow equations with all variables determined as accurately as possible.

The factors to be considered are:

1. At the maximum flow rate, the pressure differential should not exceed a value which can conveniently be measured with a manometer. Some allowance should be made for surges in flow.

2. At the normal flow rate, the pressure differential should be large enough to permit it to be read with reasonable accuracy.

3. When metering gases under low pressure, the pressure differential should be kept low to minimise permanent pressure loss and to reduce the sensitivity of the meter to the gas expansion factor .

4. When metering compressible fluids, the meter range size (in inches of water) should not exceed the absolute upstream static pressure (in psi)

The design for the new Dall tube was done according to theprinciple geometrical similarity with that of the Dall tube for 6" pipe line installed in the Junior Hydraulics Laboratory. The following dimensions are obtained for this. Inlet cane angle = 19^{o} Diffuser angle = 7^{o}.

These angles were kept constant for both the Dall-tubes Other dimensions were obtained in proportion to the dimensions of 6" Dall tube. The relevant dimensions are given in Fig. 27. After preparing the drawing, it was given for manufacturing. The outer casing was made out of aluminium, because it is a light metal. The inner piece being brass. The brass was chosen because it can give better surface finish with less trouble while finishing. Valuable suggestions were received from Prof. H. Alvord (Guest Professor in Mechanical Engineering from U.S.A.) for designing the tube to ease manufacturing difficulties.

After getting it manufactured, it is installed in $1\frac{1}{4}$" diameter pipe line and was tested. A measuring tank was used for meauring the discharge through the meter The different quantity of discharge were measured for different runs and the observations were taken with all precautions as outlined in the end of this Chapter. The observations and calculated results are given in Table 4.

CALIBRATION:

A calibration curve is drawn between Reynolds number and the coefficient of discharge. The coefficient of discharge is more than unit at same readings. It becomes constant at Reynold's number of about 13×10^3 and after that remains a constant practically. The curve is similar to the calibration curve for a orifice meter. The coefficient number becomes variable at low Reynold's number. A more detailed discussion of this calibration curve will be given in Chapter IV, while comparing the flow meters.

HEAD LOSS:

The head loss in the meter is very high and consequently the pressure recovery is very poor. This low recovery may be due to:

1. Manufacturing defects are there in the Dall Flow Tube There are some blow holes in the casing of the outer aluminium casting so that water used to leak through them.

2. There are some internal threads in between the tapings to the manometer for reading recovery . It is greatly increased the pressure loss and hence low pressure recovery.

3. The size of the Dall Flow Tube being small, there is much of friction loss between the inlet and outlet of tube so that recovery is low.

4. Due to so many obstructions in between the pressure tappings such as abrupt changes of section and the reading portions, lot of eddies are formed and this gives a high pressure loss. However the differential pressure in the Dall Flow tube is quite high as is expected.

PRECAUTIONS:

1. A great care is to be taken to fill the manometers before filling into the pinch cocks.

2. In order to avoid trouble due to air bubbles which collected at the top of the pipe lines, the pinch cocks are fitted at the sides of the bottom of the pipe.

3. No air bubbles should be allowed to be entraped inside

the connecting lines of the manometers.

4. Manometers should be perfectly vertical and should contain sufficient amount of air initially so that under different pressure and velocity conditions of the liquid the full range of the manometers can be of use.

5. Readings are to be taken on the manometers while steady state is attained and both of the limbs of the manometer should be read simultaneously.

6. Connections to the manometers should be perfectly joined so that under high pressure conditions, they do not give away.

7. Consistency in reading the liquid level in the manometers shou'd be maintained.

8. Two hook gauges are provided to take the readings of the rectangular weir for greater accuracy.

9. The readings of the hook gauges are to be taken when the flow in the channel has attained a steady state. The mean value of the readings of the two gauges gives fairly accurate results. Sufficient layer should be taken to take the correct readings.

10. Sufficient time should be allowed between consequtive runs of the experiments to ensure theattainment of steady state. In these experiments fifteen minutes time internal is allowed between each observation.

11. The pump should be properly provided before starting and sufficient time should be allowed to run the pump to make sure that no air bubbles are entraped in the pipe line.

The manometers used in all these experiments were prespared in the laboratory and they are having bore of 1/4 in. diameter.

SUMMARY O RESULBS AND DISCUSSIONS:

From the foregoing observations and calculations of the experimental set ups some important conclusions are obtained. Through simple inverted U-tube manometers were used in all the experiments and there were inevitable errors in the readings due to fluctuations of water columns, even then the results obtained are quite satisfactory and conforms to the results already obtained. The following conclusions are drawn from these.

1. The discharge coefficients of short venturimeters having rounded entrance cone are higher than the discharge coefficients of the Dall Flow Tubes.

2. There exists a relationship between the quantity flowing in the bend and the differential pressure produced between outer and inner radii. From the graphs the relationship between the discharge and the square root of differential head is a straight line.

3. The discharge coefficients from Bazin's formula for discharge over the rectangular weir, taking into account the velocity of approach are sufficiently correct to obtain discharge over the weir.

4. The figure 19a and 23a show the relationship between discharge over the rectangular weir and the head measured by hook gauges. It will be noticed that the mean coefficient of discharge C_D is 0.68 at lower values of discharge (Fig. 23a) whereas it is 0.630 at higher values of discharge Figure 19a.

5. At high Reynold's number, the values of coefficient of discharge remain practically constant for all meters. For rounded entrance short venturis, the discharge coefficient slightly falls at higher Reynold's number.

6. The calibration curve for Dall Flow Tube is similar to the orifice meter calibration curve rather than that of venturimeter. Even the coefficient of discharge goes higher than unity at low Reynold's number. But at higher Reynold's number it remains practically constant.

7. Percentage head loss with tap-pings adjacent to the meters are high., with similar conditions o pressure tappings the percentage head loss is less than that of short venturi.

-:0:-

CHAPTER III

CHAPTER III

EVALUATION OF VARIABLES IN THE FLOW MEASUREMENT.

INTRODUCTION:

First of all the theory which is used for calibrating the rounded entrance flow meter is dealt with in this Chapter. A brief outline of the theoretical analysis is given in order to appreciate the results that are obtained from it, further theoretical treatment is given[17]. The analysis for very low Reynold's numbers and for turbulentboundary layers are not discussed as they do not come into the present problem. During the operating range, the laminar boundary layer condition exists and the Reynolds numbers are moderately high.

Then the different variables that come into the flow meterin problems have been analysed and the assumptions that are taken for solving the problems are also outlined along with it. The variables are not the same for all kinds of fluids and for all meters. Same are important in one case whereas they have got less importance in other cases. Hence complete analysis of them is not within the scope of this Chapter. Only those, which are confronted for metering liquids at normal operating ranges, have been discussed. A brief outline of the factors which affect fluids like gases etc. have been given.

The effects of different parameters which affect the value of coefficient of discharge have been discussed; A brief discussion over pressure recovery cone (diffuser) is also given.

The reader may very well appreciate the difficulty for establishing all these factors but nevertheless some are quite logical and mathematically correct and some others have been established by experiments.

ROUNDED ENTRANCE FLOW METER THEORY :

The calculations of the coefficient of discharge for experimental set up No. 1 and No. 2 are based on the theory of rounded entrance flow meter. Because the meters used in these two sets of experiments are having rounded entrance cone as shown in Fig. 13 and hence it is fèlt that the theory of such type of flow meters should be given here.

This theory is based on the consideration of the potential and boundary layer flows in a converging nozzle. Curves are presented showingthe discharge coefficient as a function of diameter Reynold's number, with total equivalent length diameter ratio of the nozzle as parameter. The equivalent frictional length diameter ratio of the contraction section flow meter is presented. The theoretical curves of discharge coefficients v.s. diameter Reynold's number are in good agreement with experiments over a range of

Reynold's number from 1 to 10^6. The theory provides a rational frame work for corelating and extrapolating the experimental results; it shows the effects of contraction shape and location of pressure taps; it furnished values of discharge coefficient for untested designs, and it suggests precautions to be taken into design, installation and operation(17).

ASSUMPTIONS:

The rounded entrance flow meter in which there is no vena contracta will be considered here. Fig. 28 shows some of the configurations to which the present theory is applicable. In each case there is a smooth contraction from a large diameter to a smaller pipe, and following the contraction there is a short length of constant diameter pipe preceding the downstream static pressure tap. The solid surfaces are assumed to be smooth, more specifically the surface roughness dimensions are assumed to be negligible as compared with the boundary layer thickness. Only compressible and steady flow is considered.

DESCRIPTION OF FLOW IN ROUNDED ENTRANCE:

We now discuss the details of the flow in a rounded entrance meter and how these details influence the discharge coefficient.

High Reynold's Number:

Since, during the operating range of the differential pressure flow meters, the Reynold's numbers are high the flow characteristics in this range are to be fully considered . For such high Reynold's number, the viscous effects are, for practical purposes, limited to a boundary layer whose thickness is small compared to the nozzle dimensions. So we may investigate more or less separately the influences on discharge coefficient of the potential flow in the cone and the boundary layer flow near the wall.

Apart from viscous effects, the nature of the potential flow enters the problem primarily in connection with the non-uniformity of the pressure and velocity distributions in the plane of the down stream static pressure taps. In general the stream lines in this plane have a slight curvature and accordingly, there is a pressure gradient normal to the stream lines of such direction that the pressure at the wall is less than that on the centre line. Therefore the downstream static pressure tap measures a pressure less than the average over the cross-section, and by Bernoulli's equation, leads to inferred velocity greater than the average From this reasoning it is clear that the assumption of the flow being non uniform in the exit plane of the nozzle will lead to a discharge coefficient of less than unity even if viscosity were completely absent. However, the stream line curvature which generates this effect diminishes as

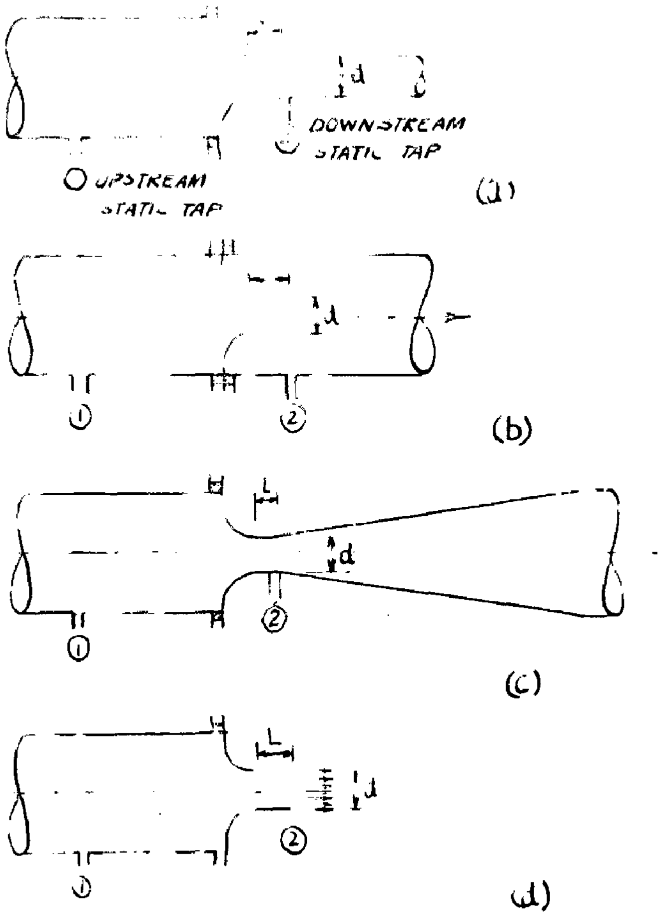


FIG 28 - FLOW METER CONFIGURATION

d - DIAMETER OF CYLINDRICAL SECTION OF ACCELERATED JET

L - LENGTH OF CYLINDRICAL SECTION

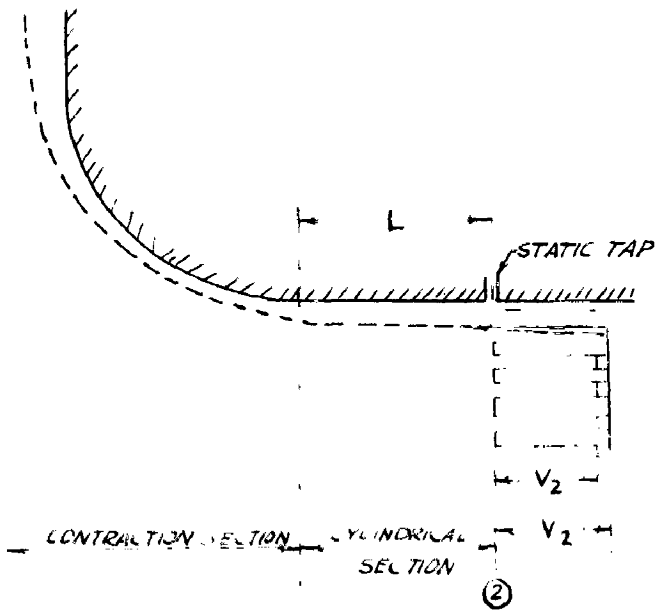


FIG 29 - ILLUSTRATIVE DETAIL OF FLOW

The length diameter ratio L/d of the straight portion increases. Indeed, when on both sides of the pressure taps there are straight sections having length of the order of ¼ diameter or more, the non uniformity of the potential flow in the plane 2 is negligibly small. It will be seen from Fig. 13 that L/d is nearly ½ and hence the assumption of uniform flow in the potential come at section 2 is valid.

The nature of the potential flow associated with a given contraction shape also influences the discharge coefficient indirectly, because of the boundary layer development is controlled in part by the longitudinal pressure gradient established by the potential flow. Turning no to the boundary layer, its growth depends upon:

1. Viscous effects.
2. Changes in radius of the contracted section which require changes in boundary layer thickness to accommodate to a given boundary layer flow, and
3. Pressure gradients.

For a typical contraction section, upto nearly the end of contractions the effect of falling pressure gradient is predominant, and the boundary layer becomes thinner. Shortly before the beginning of the cylindrical section, viscosity becomes predominent and the boundary layer becomes thicker Fig. 29.

~~VERY LOW REYNOLD'S NUMBER:~~

VERY LOW REYNOLD'S NUMBER:

When the Reynold's number is very small, theabove picture is modified in that the boundary layer occupies all or nearly all of the cross -sectional area. The conceptional picture of a boundary layer flow and potential flow which may be treated., Separately is invalid. The analysis of this flow is not given here, since it does not concern in the present analysis.

THE DISCHARGE COEFFICIENT:

If there is no friction whatsoever in the contraction, so that the boundary layer begins to form only at the beginning of the cylindrical section Fig. 3, then Bernoulli equation gives

$$p_o - p_1 = \rho / 2 \quad V_2^2 \qquad (17)$$

Between section 1 and 2 we may acount for friction by analogy with pipe friction in fully developed flow by writing.

$$p - p = 4 \bar{f}_{App.} \quad L/d \; \rho /2 \quad \bar{V}_2^2 \qquad (18)$$

where $\bar{f}_{App}$ is a mean apparent friction factor over the length L and is defined by this equation.

It is called apparent because it includes the effects of change in momentum flux as well as of the usual wall friction on the pressure tap, it is called means because the local apparent friction factor varies with axial distance Now adding equations 17 and 18.

$$p_o - p_2 = \rho/2 \; V_2^2 \; (1 + 4 \bar{f}_{App.} \; L/d) \qquad (19)$$

The discharge coefficient is defined as the ratio of the actual flow passing through the nozzle to the flow which would pass for the same pressure drop, if the flow were frictionless. Using the Bernoulli equation and the continuity equation $W = \rho AV$ together the equation 19. we get

$$C_D = (1 + 4 \bar{f}_{App} (L/d))^{-\frac{1}{2}} \qquad (20)$$

Analternate expression may be obtained for the case in which a potential core exists at section 2. Leting U_2 denote the velocity of the potential core at section 2, Bernoulli equation written for any stream line not entering the boundary layer is

$$p - p = \frac{\rho U_2^2}{2}$$ and the equation 20 shows

that

$$C_D = V_2 / U_2 \qquad (21).$$

In other words the discharge coefficient is equal to the ratio of the mean velocity to the potential core velocity at the section 2.

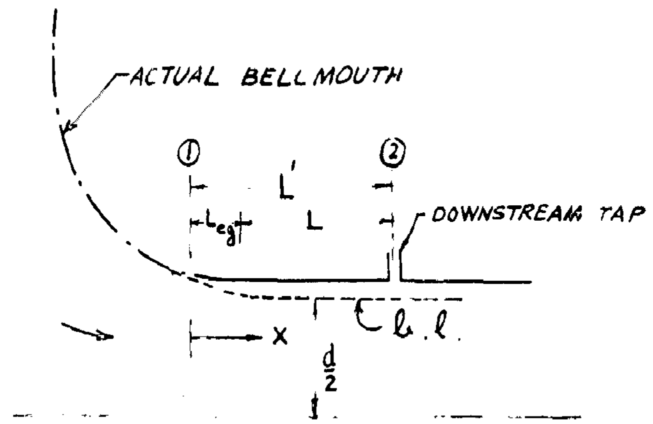


FIG 30:- SAMPLING MODEL IN WHICH CONTRACTION SECTION IS REPLACED BY EQUIVALENT FRICTIONAL LENGTH INTO WHICH FLOW ENTERS WITH UNIFORM VELOCITY AND NO. B.L.

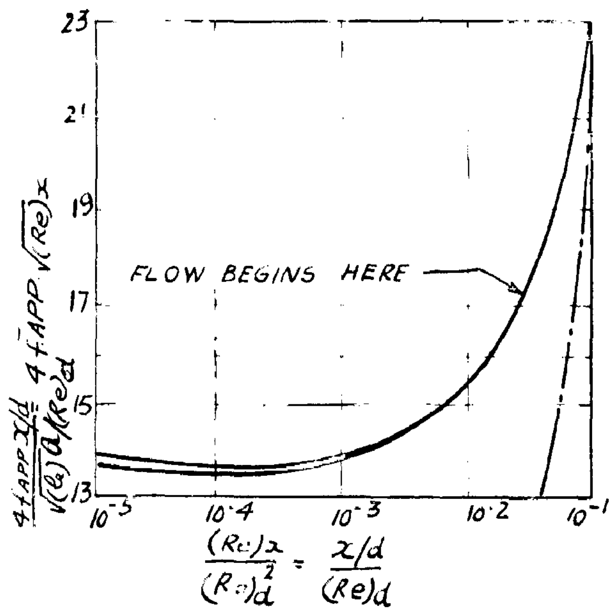


FIG. 31:- THEORETICAL CURVE FOR APPARENT FRICTION FACTOR FOR LAMINAR ENTRY TUBE

Thus far we have not considered friction in the contracted section. At high Reynold's number i.e. when potential cone exists, equation 21 for C_D is valid regardless of where the boundary layer begins to develop, whether in the contracted or in the cylindrical section. The equation 20, however, may be regarded as approximately valid when there is friction in the contraction section provided that L is replaced by L' where the later is interpreted as a pseudo length equal equal to the sum of L and equivalent frictional length Leq for the contraction.

Thus

$$C_D = (1 + 4 \bar{f}_{App.} L'/d)^{-\frac{1}{2}} \qquad (22)$$

ANALYSIS OF MODEL IN WHICH CONTRACTION SECTION IS REPLACED BY EQUIVALENT FRICTIONAL LENGTH:

In the model of Fig. 30, the contraction section is replaced by equivalent frictional length Leq and the flow is assumed to enter the tube at section 1 with uniform velocity and no boundary layer. If L/d is of the order of $\frac{1}{2}$ or greater this model will accurately give correct results, even with very rough estimates of Leq. Near $x = 0$, where the boundary layer is thin compared with the tube radius, the flow in the boundary layer is substantially like that on a flat plate, except that in the tube there is a falling pressure gradient as compared to with zero pressure gradient for the flat plate in an infinite stream, However the theoretical investigations of [18] concerning the boundary layer development in a tube demonstrate that in the region of thin boundary layers the pressure gradient has a virtually negligible effect on the velocity profile and rate of growth of boundary layer.

CONDITIONS FOR BOUNDARY LAYER TO BE LAMINAR.

Because of strong falling pressure gradient in the contraction section, the boundary layer beginning at $x = 0$ may be expected to be laminar. Furthermore, since the falling pressure gradient in the cylindrical section is favourable to the maintenance of laminar boundary layers, the longitudinal length required for the flat plate like boundary layer to become turbulent may be expected to be at least as great as in the case of a flat plate with no pressure gradient, For the latter it is known that in a stream of moderate initial turbulence, the length Reynolds number of transition $(Re)_x$ is of the order of 5×10^5 . Experiments in the entries of tubes demonstrates the validity of the foregoing argument.

Accordingly, the range of diameter Reynold's number $(Re)_d$ in which the flow upto section 2 may be expected to be laminar is of the order of

$$(Re)_d = (Re)_x \, d/L \doteq d/L \; 5 \times 10^5$$

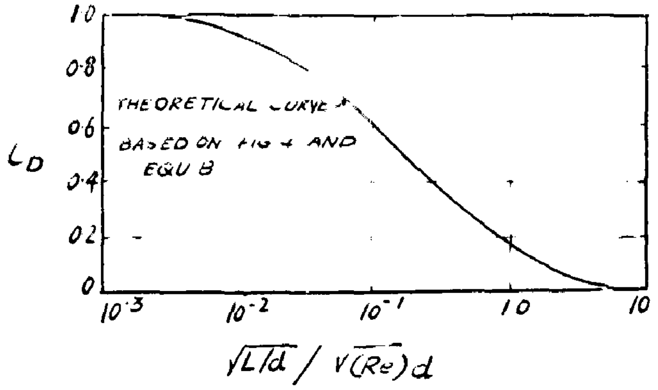


FIG 32 - UNIVERSAL CURVE SHOWING C_D AS FUNCTION OF SINGLE PARAMETS $\sqrt{L/d}\,/\,\nu(Re)_d$

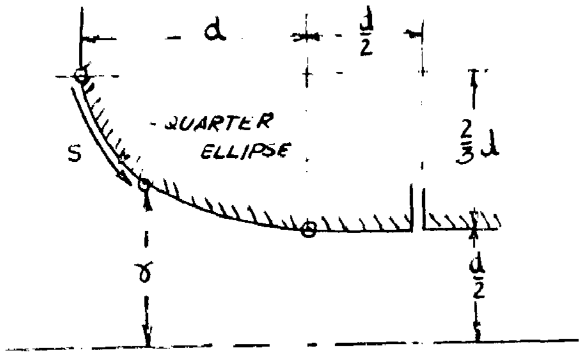


FIG 33:- A.S.ME ELLIPTICAL CONTRACTION SELECTED FOR SPECIFIC CALCULATION

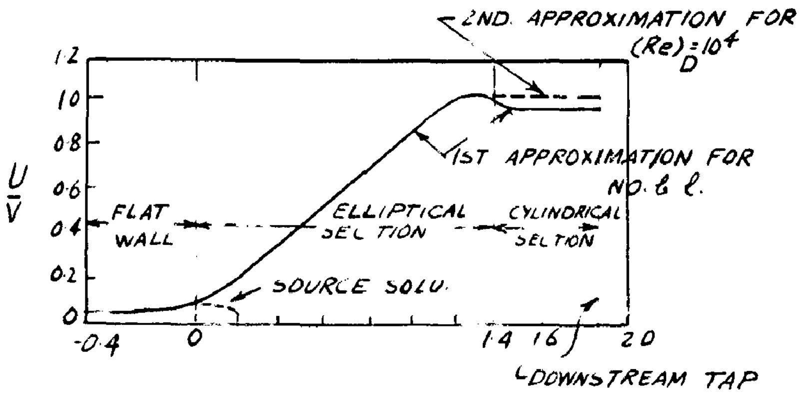


FIG 34:- POTENTIAL FLOW VELOCITY DISTRIBUTION ON NOZZLES WALL FOR NOZZLE NO.33

In the flow meters under consideration, L/d is of the order of ½ ; accordingly, the range for which the boundary layer in the length L may be expected to be completely laminar is of the order of 10^6 Many flow meters of interest have values of (Re)d less than this. Therefore we reach the importance conclusion that the assumption of laminar floor upto the downstream static pressure tap will cover many practical cases.

THEORETICAL SOLUTION FOR LAMINAR ENTRY:

The laminar flow in the entry of a tube has been investigated theoretically by Boussinesq, Schiller, Atkinson and Goldsstein, Langhaar, and Shapiro, Seigel and Kline. All these investigations led, by different methods, to result which agree within a few percent, and all may be considered to be substantially correct. The experimental results carried out in the range of $(Re)x / (Re)d^2$ between 10^{-5} and 10^{-3}, yielded a mean exponential curve expressed by

$$4\ \bar{f}_{App}\ (x/d) = 13.74\ ((Re)x / Re/d)^{\frac{1}{2}}$$

which agrees with Fig. 31.

BOUNDARY LAYER WITH PARTIAL TURBULENCE:

Experiments in the entries of tubes indicate that, other some peculiarities associated with the onset of turbulence upstream of section 2, the local apparent friction factor in the turbulent zone is generally slightly less than the average apparent friction factor in the laminar zone preceding transition, and that as x increases, the value of $\bar{f}_{A\,pp.}$ in the turbulent zone decreases, Slightly and gradually approaches a asymptotic value for fully developed turbulent pipe flow.

EQUIVALENT FRICTIONAL LENGTH OF CONTRACTED SECTION:

By the equivalent frictional length 'Leq' of the contraction section, we mean that frictions length of the straight tube of diameter of which, if placed before the actual length of cylindrical tube of length L, will lead to a discharge coefficient indentical with the discharging coefficient produced by a actual combination of bell mouth and the cylindrical section.

PROCEDURE TO FIND Leq.:

1. Boundary layer theory is applied to determine the boundary layer proper ties at section L any from equation 21 the discharge coefficient .C_D.

2. Inserting the latter in equation 22, the value of $\bar{f}_{App}$ L'/d is computed.

3. From Fig. 31, the value of L'/d may be found.

4. By substraction of L/d, the value of Leq/d is computed.

SPECIFIC CONTOUR CHOSEN FOR STUDY :

The equivalent length of contraction depends both

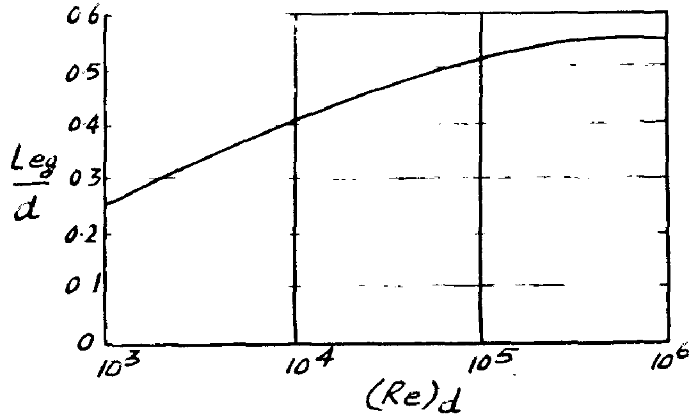


FIG 35 - EQUIVALENT LENGTH OF CONTRACTION SECTION Vs $(Re)_D$ FOR CONTOUR FIG 6

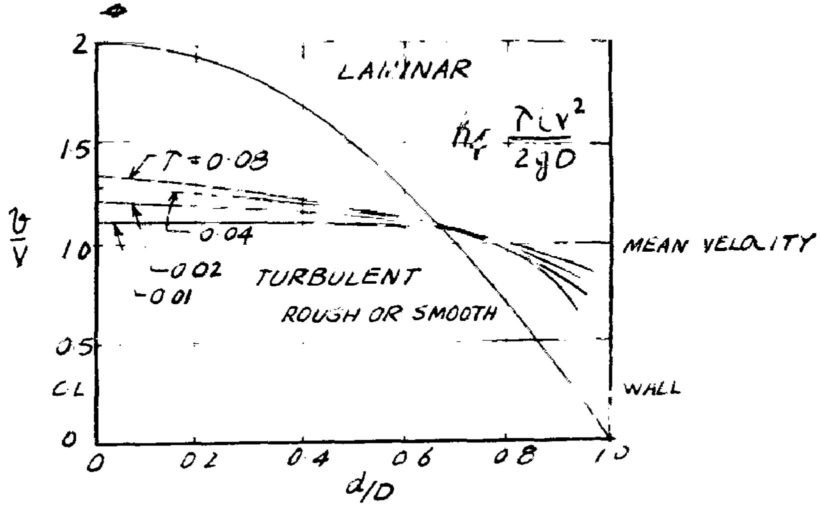
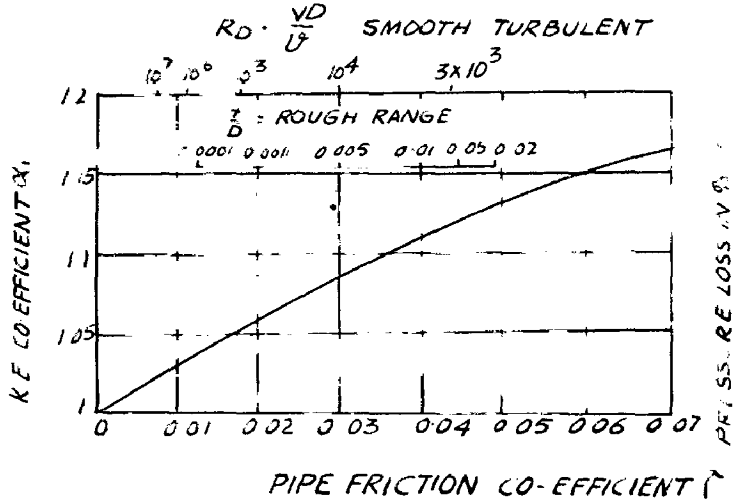


FIG 37 - VARIATION OF VELOCITY DISTRIBUTION IN A CIRCULAR PIPE WITH λ



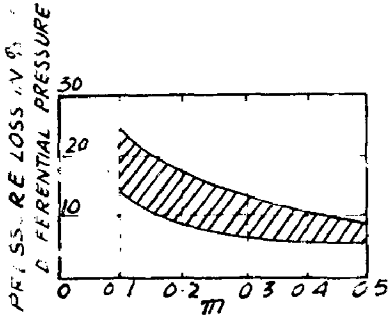


FIG 39

FIG 38 - VARIATION OF ENERGY FACTOR α_1 WITH λ FOR TURBULENT FLOW.

$$\alpha_1 = 2\int_0^{R_1} \left(\frac{v}{V}\right)^3 \frac{\gamma_1}{R_1} \cdot d\left(\frac{\gamma_1}{R_1}\right)$$

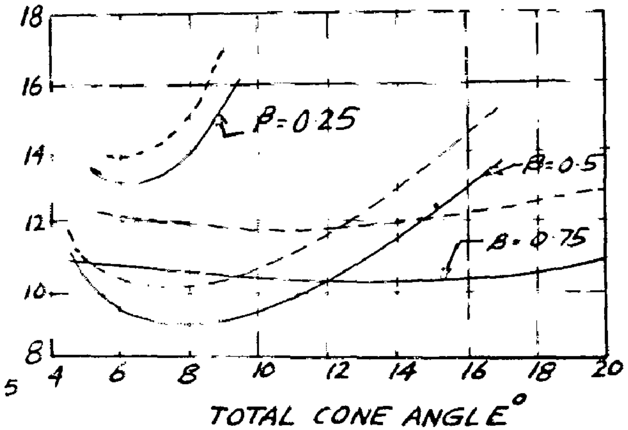


FIG 40 - EFFECT OF RECOVERY CONE ANGLE ON Y HEAD LOSS.

BROKEN LINES INCLUDE PIPE FRICTION

on the shape of the contraction and on the diameter r Reynold's number (Re)d. The results obtained are for standard ASME elliptic contour shown in Fig. 33.

Since the contour of Fig. 33 is not very different from that used for other flow nozzles and venturis, and in as much as the coefficient of discharge does not usually depend decissively on the equivalent length of the contraction, the equivalent length found from the contour of Fig. 33. may offer with little error be used for contraction contours.

For our present purpose the significant result of the potential flow solution is the velocity distribution at the wall shown in Fig. 34 in dimensionless terms. The description for the solution of equivalent length with the help of potential flow solution and boundary layer flow solution is not given here. The reference for such procedure is (17). The results of these calculations are presented in Fig. 35 where Leq/d is plotted against (Re)d for the 5th approximation . The equivalent length decreases for a value of about 0.56 at (Re)d 10^6 to value of 0.25 at (Re)d = 1000.

DISCUSSION OF VARIABLES:

Shape of Contour:

The contour of the contraction section influences the equivalent length of the latter. Most important, however is ~~the way in which the pressure distribution~~

on the contour determines whether the flow in the nozzle is laminar or turbulent and whether the flow is fully attached a separated. If the wall pressure fell monotonically, then there could be no boundary layer separation and in addition, boundary layer will be entibely (a minor in the flow nozzle except at avery high Reynold' number when the adverse pressure gradient , which is almost inevitably present is sufficiently large, however, there may be boundary layer separation, there may be tranition to turbulence or there may be both. Faithfuly production of the prescribed contour is essential, otherwise the adverse pressure gradient might inadvertently be accentuated, leading possibly to boundary layers transition and a radial change in discharge coefficient.

POSITION OF DOWN STREAM TAP :

It would seem desirable to place the downstream static tap in a region where the stream lines no longer have curvature. According to fig. 34, This sets the minimum value of L/d at about 1/4 . In addition, there should be about 1/4 diameter of straight section downstream the static tap . Accurate location of pressure tap is very important. For a change in value of L/d, the coefficient of discharge is also changed.

ROUGHNESS OR INITIAL TURBULENCE:

Either wall roughness or initial turbulence will decrease the value of (Re)d at which point a laminar transition moves into the flow nozzle. Either of these occurences, by changing the friction factor, would alter the discharge coefficient.

Examination of the equation 20 shows that the accuracy required in 4 $\bar{f}_{App}$ L/d, the term computed by flow theory, depends to a large extent to the value of C_D. If an error of less than one percent in C_D is required, and the value of C_D is 0.9 , the error which can be tolerated in 4 $\bar{f}_{App}$ L/d is 11 percent and in Leq 22 percent.

These considerations suggest that the calculations should be particularly accurate and the effect of minor changes in contour and initial boundary layer is very small when

1. the value of (L'/d) / (Re)d is small.
2. the value of L' is large compared to Leq.

Obviously neither of these applies beyond the point where the transition to a turbulent boundary layer might begin. If conditions 1 and 2 are met, then it also follows that the curve of Fig. 35 can be used to predict discharge coefficient for other contours with excellent accuracy provided transition does not occur.

USE OF ABOVE THEORY FOR SOLUTION OF DISCHARGE COEFFICIENTS IN EXPERIMENTAL SET UPS No. 1 and No. 2.

VALIDITY OF USING THE THEORETICAL ANALYSIS -

Experiment Set Up No. 1.

It is seen from Fig. 13 that the length of the cylindrical portion after contraction upto the low pressure tap 2 is 6". and the diameter of the throat section d is 9¼" . Hence L/d = 6/9¼ = 0.65 . Hence since L/d ratio is nearly ½ , the above analysis is applicable and the boundary layer flow may be assumed laminar.

The length of the cylindrical section beyond the pressure tap position is nearly 2 inches. So L/d ratio beyond low pressure tap is 2/925 and it is nearly equal to ¼.

The maximum Reynolds number (ReOD in the working range is found to be nearly 6.25×10^5 and correspondingly the throat diameter Reynold's number (Re)d is nearly 9.3×10^5. Th-is number is obtained by assuming the coefficient of discharge to be unity. Hence in the working range of the meter the Reynolds number cannot exceed thin value. It is seen that this value is less than 10^6 which sets a limit for the application of the above theory.

From the above considerations it is quite evident that the theoretical analysis can be very conveniently used to obtain the discharge coefficient for experimental setup No. 1.

EXPERIMENTAL SET UP No. 2.

Let us now see whether the above theoretical analysis will hold good for experiment No. 2 . It will be seen from diagram 20 that the L/d ratio before the low pressure tap 2 is nearly ½ and the L/d ratio down stream the tap is nearly ¼. Again the maximum Reynolds number (Re)d is nearly 4×10^5 and this is much less than the limit of 10^6. Hence from the above consideration the flow theory discussed will apply without any doubt.

PROCEDURE FOR CALCULATING DISCHARGE COEFFICIENTS:

It really becomes a difficult affair to predict discharge coefficient of the meter when it is not possible to measure the discharge directly. Because the coefficient of discharge varies more or less with thecharacteristic dimensionless factor called Reynolds number which is dependent on the quantity of flow. Due to this interconnecting relationship of both the unknown quantities that a dial and error method using successive approximation is applied. The procedure followed is as given below :

1. First of all it is assumed that the discharge coefficient is unity and the velocity V_2 is obtained from Equation 2. From this the Reynolds number (Re)d is calculated.

2. Since the flow theory for rounded entrance flow meters, can be applied in this case Fig. 35 is then used and corresponding to (Re)d as found in step 1, the value of

Leq/d is read and from that Leq is found out.

3. To this value of Leq, the value of the length of the cylindrical portion is added and total equivalent length L' is found out.

4. The value of L' found out above corresponds to the throat diameter df. The frictional head loss corresponding to the equivalent length L' and diameter ~~df~~ d is the loss between the high and low pressure taps. Since the measurement of frictional head loss is done on the upstream side where the diameter is D, it is now essential to find the equivalent length L_1 for the pipe of diameter D such that the head loss in both the cases remains constant.

5. Head loss in the cylindrical section of diameter d is given by

$$h_f = \frac{4\ f_2\ L'\ V_2^2}{2\ g\ d.}$$

and the head loss due to friction in pipe of diameter D is

$$h_f = \frac{4\ f_1\ L_1\ V_1^2}{2\ g\ D}$$

In order that the head loss be same, equating both we get

$$L_1 = \frac{f_2}{f_1} L' \left(\frac{D}{d}\right)^3 \qquad (23)$$

FIG. 36

FRICTION FACTOR f vs REYNOLDS NUMBER

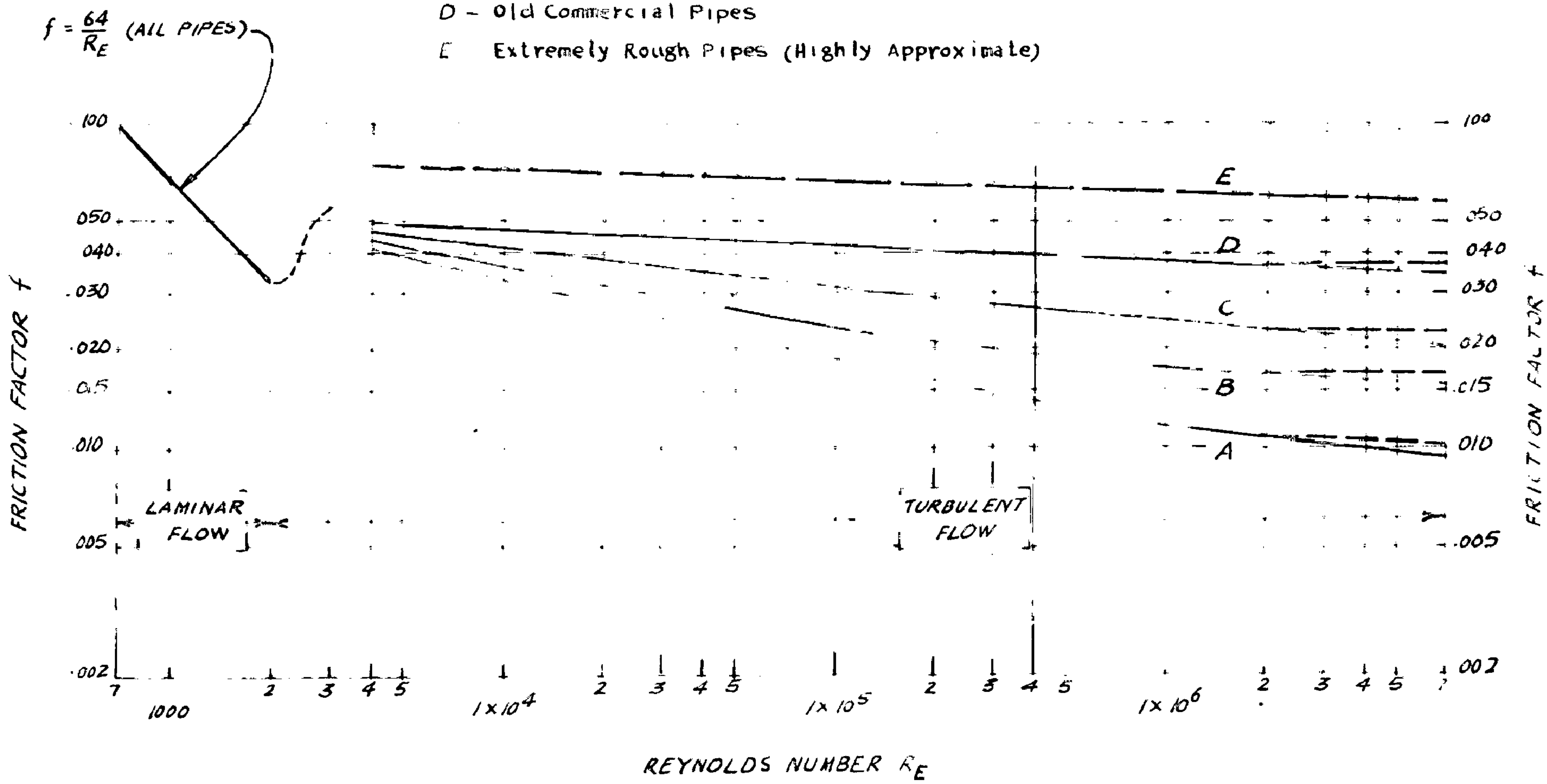
A – Very Smooth Pipes (Brass, Glass, Copper Tubing)

B – New Commercial Pipes

C – Average Commercial Pipes

D – Old Commercial Pipes

E – Extremely Rough Pipes (Highly Approximate)



6. The values of f_2 and f_1 depend on (Re)d and (Re)D respectively. Hence with the values of Reynolds number obtained earlier, these values are read from Fig. 36.

<u>Note on Figure 36:</u> The valuesof friction factor obtained from figure 36 corresponds to four times the values of used in the above equations. For turbulent flow a simple mathematical relation does not exist for the variation of the friction factor with Reynolds number. The condition of the pipe walls, a difficult thing to estimate with reasonable accuracy, causes complex degrees of turbulence and different values of f for the same Reynolds number. For various types and conditions of pipes the friction factor f can be expressed as

$$f = \frac{a}{(Re)^b}$$

where a and b are numerical constants.

However, in calculations, values of f are usually selected from a set of curves as shown in Fig. 36. The accuracy of pipe friction calculations is decreased by unpredictable changes in these roughness factors. Hence the general average values shown in Fig. 36 are considered of satisfactory accuracy. The values of f are taken from curve c of Fig. 36 for commercial pipes.

7. When the equivalent length L_1 is calculated from

equation 23, the head loss corresponding to this length is calculated from the column 3 Table 1 of Appendix for the experimental set up No. 1 and from the column 4 Table 2 of Appendix.

8. After getting the head lost in the meter, the discharge coefficient is obtained from

$$C_D = \left(\frac{h - h_f}{h}\right)^{\frac{1}{2}} \qquad (24)$$

9. But this is not the correct value f coefficient of discharge, because we have assumed the Reynolds numbers corresponding to no friction conditions which are higher than the real conditions. Hence, again this value of C_D as obtained from equation 24 is put in equation 2 and the similar steps are followed as mentioned above. This process can be repeated till the convergence in the value of C_D is obtained. For the present case three successive calculations give quite satisfactory results.

EVALUATION OF VARIABLES :

Before a flow equation can be applied to a specific metering problem, each of its variable terms must be evaluated. The basic field data that are necessary are given below:

Installation Details:

1. Make of flow meters and principle of operation.

2. Maximum differential range of meter.
3. Type and exact location of pressure taps.
4. Position of pressure tap at which static pressure is measured.
5. Internal diameter of the pipe D.
6. Throat diameter d.
7. Sloping of meter connecting lines.
8. Nature of internal surface.

OPERATING CONDITIONS:

1. Temperature of the flowing fluid.
2. Temperature of meter fluid and sealing fluids.
3. Barometer pressure.

PHYSICAL PROPERTIES:

1. Specific gravity for manometer and sealing fluids.
2. Viscosity at operating temperature.

CONDITIONS AND FACTORS WHICH AFFECT THE SOLUTION OF FLOW EQUATIONS:

METER AREA MULTIPLIER :

If the meter under consideration is designed for a particular condition, but it operates under a condition far away from the designed conditions, then it will not give the same results. The factors which bring such troubles are primarily temperature and meter metal. Hence in order to accommodate such inevitable changes a multiplying factor called Meter area multiplier is used in the flow equation. Under normal operating conditions it is not required.

DISCHARGE COEFFICIENT C_D

The discharge coefficient is a function of

1. Location of pressure taps.
2. Diameter ratio $\beta = d/D$
3. Pipe size D.
4. Reynolds number $(Re)_D$ which depends on pipe diameter D, flow rate W and viscosity μ.
5. Viscosity depends on flow temperature and nature of fluid.

EXPANSION FACTOR:

In our case it is unity, because we face water as incompressible at ordinary temperature. However, where the flow temperature and pressure are relatively high with respect to the critical temperature and pressure, the liquid becomes significantly compressible, Casey[23] cites an example of the metering of liquid propane (at 1800 psi a and 160°F) in which failure to correct for compressibility would have resulted in a calculated flow 8.8 percent lower than the actual flow. For gas flow measurement it is an important factor. It is a function of

a. Location of pressure taps.

b. Flow pressure.

c. Differential pressure.

d. Diameter ratio

e. Specific heat ratio k which depend on

i) Flow temperature.

ii) Nature of fluid.

FLUID DENSITY

It depends on flow temperature, Flow pressure and Nature of fluid.

In the present cases of experiments the density of water is taken as 62.25 lb/ft^3 at 74 ^{o}F.

TOLERANCES FOR FLOW CALCULATIONS:

No two meters can be expected to give exactly the same readings for identical flow rates because of unavoidable miner differences in construction and operation of the equipment.

In order to show the probable magnitude of these differences for well established meters, the limits of errors or tolerances of the various factors entering into the measurement of flow rates are given below[8].

Factors	Tolerance percent.	Qualifications.
Differential pressure	$\pm$ 2 to $\pm$ 0.2	Decreasing as

			differential pressure increases.
Static pressure	$\pm$	2 to $\pm$ 0.2	Decreasing as pressure increases.
Fluid temperature	$\pm$	0.5	
Diameter of Throat	$\pm$	0.1	
Fluid density	$\pm$	0.1 to $\pm$ 0.2	Except near critical region.

These tolerances do not include accidental and systematic errors of observations. They give either 1, the most probable differenfe that could be expected between obsensibly duplicate installations or 2. the probable accuracy of a particular set of data.

Since there is a slight probability that the tolerance of each factor will affect the final result in the same direction, the overall tolerance can be calculated by the law of Combining Errors (i.e. multiply each tolerance by the power to which the factor it represents enter into the flow equation. Then take the square root of the sum of the squares of the each factor). By this means the most porbable departure from 100 percent accuracy can be predicted, although in certain cases the actual error may be smaller or greater.

TOLERANCE FOR COEFFICIENT OF DISCHARGE:

In order to predict the tolerance of C_D a large number of different diameter pipes are required and the meters to be used with them should have same area ratio m. Since these meters are not available, experimental results are not possible. But a theoretical consideration for it seems to be necessary, because thetolerance figure for a single meter installation gives the meter user a little idea as to how it will operate and with what accuracy.

According to Dr. Witte, the basic tolerance for C_D is calculated in the following manner[(24)].

For a given value of area ratio 'm' the coefficient of discharge coefficients were obtained as function of Reynolds number for several series of tests, each series corresponding to a given pipe diameter. A average curve was drawn for each series of tests, thus compensating accidental errors. For all normal curves, a normal curve was obtained, valid for the given value of 'm' and all diameters. The tolerance was taken equal two twice the standard deviation. It seems to be general consensus that the 'tolerance' defined as twice the standard deviation is acceptible both to the manufacturer and to user of flow metering devices. It should not be forgotten that the statistical analysis should include only tests made on equipment constructed and installed in accordance with modern standards.

At relatively large Reynolds number, the basic tolerance is equal to one percent for orifices and varies from 1 to 1.8 percent (higher figures being for larger values of area ratio m) for flow nozzles.

For Herschel type venturi tube, the tolerance for coefficient of discharge is 0.75 percent for pipe diameters between 4 in to 32 in and Reynolds number (Re)D above 200,000.

INFLUENCE OF FACTORS ON COEFFICIENT OF DISCHARGE:

Coefficient of discharge is function of

1. External factors: α_1 - entry velocity, distribution, dependent on installation pipe roughness.
2. Internal factors - m meter geometry, construction ratio.
   - K_L Entry losses (shape, roughness etc.).
   - α_2 throat velocity distribution, (shape and area ratio m).

Variation of C_D by external effects ( ) -

The coefficient α_1 depends on the velocity distribution at inlet to the meter, which is governed by the flow conditions upstream in the main pipe. In practice, the effect of upstream bends , valves, transitions etc., upon the velocity distribution, will have to be allowed for and this is usually best done empirically. Some times the error caused by such fittings are as much as 5% if the contraction ratio 'm' is large. The turbulent velocity distribution and hence α_1 are functions of the friction coefficient λ (4 f) Typical velocity distributions are shown in Fig. 37 for various values of λ . It will be seen in Fig. 38 that α_1 increases with λ approximately linearly.

Because λ is a function of either (Re)D or E/D for smooth and rough turbulent flow regimes respectively, alternative scales of (Re)D and ϵ /D have been added in Fig. 38. For normal range of (Re)D and roughness ratios encountered in practice λ lies between 0.91 and 0.4 with corresponding values of α_1 ranging from 1.03 to 1.11 as ϵ/D increases, hence C_D increases. Thus roughening the upstream piping increases C_D at first sight a rather paradoxical finding but neve theless borne out in practice.

Variation of C_D caused by internal effects :

It is found that the changes in C_D caused by variation in 'm' should be very small (because $1- C_D^2 = 0$) This explains why it is usually possible to manufacture meters to within close limits on 'm' that associated errors in C_D are negligible.

Provided m is small (0.3) and the nozzle is smoothly faired into the throat, the practical value of is very close to 1.

Prediction of K_L for venturi - It is sum of

a. Cylindrtical pipe diameter D, length D/2.

b. Conical tapering from D to d.

c. Cylindrical pipe, diameter d, Length d/2.

It is difficult to calculate accurately the losses in parts (b) and (c) . If the total head lost is written as h_f

$$\text{or} \quad \sum_{1}^{2} \frac{\lambda \, \Delta x \, v^2}{2gd} = h_f = \lambda \frac{x}{d} \frac{V_2^2}{2g}$$

$$K_L = \frac{h_f}{V_2^2/2g.} = \lambda \frac{x}{d}$$

Similar result can be obtained by dimensional analysis. This assumes that the friction and Kinetic energy losses can be included under friction coefficient λ . By analogy with pipe flow it would be expected that K_L and therefore C_D both vary with (Re)d and roughness ratio ($\frac{\epsilon_v}{d}$) where ϵ denotes the effective roughness of the meter.

Variation of C_D with Roughness and Time

Although the 'internal' and external effects of roughening act in opposite directions, the latter is usually negligible compared to the former and so the net result in practice is usually that C_D decreases with time.

Cole brook and White have shown that the increasing surface roughness of water pipes caused by the deposition of lime or other nodules can be represented by an empirical formula.

$$\epsilon_T = \epsilon_0 + \gamma T$$

Where γ is a constant depending on material and PH value of water. γ = 0.025 in/annum for the roughening of cast iron pipes in water of average alkalinity (pH = 7).

RELATIVE INFLUENCE OF FACTORS INCLUDENCING C_D (4).

Considering the variation of C_D with the factors in equation

$$C_D = \left(\sqrt{\frac{1-m^2}{\alpha_2 + K_L - m^2\alpha_1}}\right)$$

and differentiating together with the approximation that C_D = 1 gives

$$\frac{\partial C_D}{\partial \alpha_1} = \frac{C_D^3}{2}\frac{m^2}{1-m^2} \simeq \frac{1}{2}\frac{m^2}{1-m^2} > 0$$

$$\frac{\partial C_D}{\partial m} = -\frac{mC_D}{1-m^2}\left(1-\alpha_1 C_D^2\right) \simeq -\frac{m}{1-m^2}\left(1-\alpha_1 C_D^2\right) > 0$$

Usually $\alpha_1 C_D^2 > 1$

$$\frac{\partial C_D}{\partial K_L} = -\frac{C_D^3}{2}\frac{1}{1-m^2} \simeq -\frac{1}{2}\frac{1}{1-m^2} < 0$$

$$\frac{\partial C_D}{\partial \alpha_2} = -\frac{C_D^3}{2}\frac{1}{1-m^2} \simeq -\frac{1}{2}\frac{1}{1-m^2} \simeq 0$$

CONCLUSION:

Hence as α_1 increases C_D decreases uneven velocity distribution increases C_D

as m increases C_D increases (effect is small).

as K_L " " decreases (increasing nozzle loss decreases C_D)

as α_2 " " decreases (uneven velocity distribution decreases C_D)

~~To~~ minimise all these effects m should be less than 0.3

Comparing the relative importance of various effects, denoted by E with appropriate subscript

$$\frac{E_m}{E\alpha_1} = -\frac{2}{m}(1-\alpha_1 C_D^2)$$

$$\frac{E_{K_L}}{E\alpha_1} = -\frac{1}{m^2}$$

$$\frac{E\alpha_2}{E\alpha_1} = -\frac{1}{m^2}$$

For m = 0.25 , α_1 can be between 1.02 to 1.04

C_D 0.975 to 0.990 depending on size and $(Re)_D$

Hence

$$-0.16 < \frac{E_m}{E\alpha_1} < +0.16$$

$$\frac{E_{K_L}}{E\alpha_1} = \frac{E\alpha_2}{E\alpha_1} = -16$$

Thus it is seen that the effect of m is less than α_1, which is in its turn is much less than the effects of K_L and α_2 (which are 16 times α_1, and in the opposite direction).

The above consideration gives us an insight as to why the coefficient of dischharge decreases in case of Dall Flow Tube. Because in this case the velocity distribution at section 2 is very much peaky as compared to velocity distribution of the venturi meters at the section 2.

DIFFUSER:

The conical diffuser is the characteristic part of all differential pressure flow measuring devices (except pitot tube). It allowa a substantial part of the difference pressure to be regained. In thefirst venturi tubes built according to specification given by Herschel, the angle of opening of diffuser was 5 to 7°. This small value led to fairly long devices. The choice of small value was justified by the necessity of maintaining low pressure loss. These losses are primarily due to eddies that are created when the flow separates from the wall of the diffuser.

A few years before the war, German manufacturers suggested cutting the diffuser at a downstream diameter smaller than the diameter of the pipe (truncated) Beckmann has shown that such a truncated diffuser may allow the same amount of recuperation as a non truncated type. This is also confirmed by some of Rupel's experiments. The truncated diffuser, of course, is more advantageous both for space and economy . The length of standard truncated diffuser may not be smaller than the throat diameter d.

CONCLUSION:

From the above considerations it is clear that the pressure recovery is not affected much by truncating the diffuser cone. This truncating which is also used in Dall Flow Tubes is not a new idea due to Dall as this was known before. Hence the high pressure recovery in the Dall tube is quite evident.

THE RATING OF DIFFUSERS (4)

The rating of various diffusers is largely dependent on the residual pressure loss. In Germany a differential pressure is measured between piezometer openings ~~xxxx~~ situated at a distance D upstream and a distance of 6, 7 or 4 times D downstream from downstream standardized pressure connection for m = 0.1 , 0.25, 0.50 respectively for truncated venturi's and D down stream for non truncated venturis. The ratio of differential pressure thus obtained to the differential pressure between the two standardised connections is conveniently , called 'ratio of permanent loss'. It seems more rational to adopt the method recommended by

Marchetti and Ferroglio. The pressure difference is measured between a section situated upstream and a section downstream from the venturi. These sections are chosen far away from the device so that its disturbing influence is not felt and normal flow conditions may be expected to prevail. From this differential pressure is substracted the pressure loss in a pipe of diameter D and length equal to distance between two sections. The difference is called the 'local venturi loss' H'. This loss is generally expressed, not in absolute value, but by its ratio to the differential pressure h of the venturi as h'/h .

The venturi pressure loss is a function of Reynold's number; according to Marchetti its value is

$$\left[0.052 - 0.25\ m^2 + \frac{1.25}{(Re)\ D^{0.25}} \right] (1/m^2 - 1) \frac{V_1^2}{2g}$$

for diffusers of 7° opening and non truncated type. For truncated ones, the recommendations of Helsinki give Figure 39 (function of m).

SUMMARY :

The experimental evidences which have been established for diffuser are outlined below:

1. The minimum percent head loss for recover cones in venturi meters depends on both the cone angle and the ratio of the cone entrance diameter to exit diameter (β).

2. Truncating conical diffuser may result in increased or decreased percentage head loss dependent upon the cone angle, the prospects for decreased loss seeming to be greate at larger values of cone angles.

3. For a certain length of recovery cones including truncated cones, there is a certain cone angle that will give minimum percentage head loss.

4. Diffuser of curved walls instead of cones, bent of the same length, can be expected to give different percentage head loss.

5. Unmachined cast cones may in the vicinity of 8° total cone angle, be expected to give but a slightly higher percentage loss than smooth s urface rolled steel cones of the same total length.

6. Grease on the walls of the certain recovery cones tends to increase the percentage head loss.

7. Fig. 40 shows the effect of recovery cone angle on percentage head loss. It will be seen from the figure that for lower values of β , the percentage head loss increases more rapidly for increase in the value of cone angle.

EFFECT OF ROUNDING SHARP EDGES OF DALL FLOW TUBE:

Tests were made to determine the effects of rounding the sharp edges in a Dall Flow tube[4]. An 8 in x 5.85" tube was used for the test. First the tube was calibrated as manufactured. Following this test, sharp edge at a Fig. 3 was slightly rounded with emery cloth. The coefficient dropped 0.4 percent. Sharp edge at c at the leading edge of the throat slot was next slightly rounded. There was no further change in the calibration. The edge at the downstream, side of the slot was then slightly rounded and again there was no further change in the calibration. In view of the small change affected by imparting a slight radius to three sharp edges, edge a was machined to 1/16 inch radius. The coefficient dropped by one percent, below its original value (0.6 percent more drop than for slight rounding) A 1/64 in radius was then machined at edge c, and the coefficient increased 0.4% making it 0.6% lower than the originally. Finally, a 1/64 inch. radius was machined on edge ~~of~~ d and the coefficient remained 0.6 percent below its original value when all edges are sharp.

CONCLUSION:

The rounding of sharp edge at a helps the velocity distribution to be smooth and hence the value of α_1 decreases. The decrease of value of α_1 necessitates increase in the value of C_D but there is increase in the value of K_L due to boundary layer growth. Since the effect

K_L is more prominen than the effect of α_1 , the value of C_D decreases finally. The fact becomes more prominent when the edge is machined to 1/16 in. radius. The rounding off of edges b, d and e do not change the coefficient of discharge. The rounding off of edge c increases the coefficient discharge which in otherwards means that the value of is decreased.

SUMMARY OF CHAPTER III.

1. The theory of rounded entrance flow meters is used to calibrate the short venturis installed in the senior Hydraulic Laboratory. It has been shown that this theory is fully applicable in the present case.

2. The steps for calculation of coefficient of discharge are given and this method of calculation involves successive approximation . This method is adopted because it is not possible to have direct measurement of discharge as has been already stated.

3. The values of friction factors which have been taken from reference 16 are quite satisfactory. Because the values of friction fa tors obtained after calculations compare favourably with those taken from Fig. 36. This gives an indirect indication that the theoretical and experimental investigations agree to a great extent.

4. The different variables that enter into the flow calculations have been outlined. It will be seen that the downstream low pressure tap conditions have got more effect than others.

5. The variations of co-efficient of discharge have been discussed . It is been that the effect of velocity distribution on low pressure tap section has not more effect than the effect of velocity variation of upstream pressure tap sections. Hence in the case of Dall flow tube, due to the presence of slot in the throat section and having no cylindrical throat section, the velocity distribution is quite peaky and hence the coefficient of discharge is low as is seen from experiments of Chapter II.

6. Choice of diffuser plays a vital role in getting favourable press-ure recovery. For a particular area ratio m, there is a particular opening which will give minimum pressure loss.

7. Truncating the diffuser cone also increases pressure recovery and hence this fact is utilised in Dall Flow Tube. Moreover for a smaller values of area ratio m, the change in cone angle results in a greater change in the pressure recovery as is seen from Fig. 40.

-:0:-

CHAPTER IV.

CHAPTER IV.

COMPARATIVE STUDY OF DIFFERENT DIFFERENTIAL PRESSURE FLOW - METERS WITH DALL FLOW TUBE.

INTRODUCTION:

In this Chapter the essential difference in features of different types of differential pressure flow measuring devices are outlined. A brief discussion over calibration curves of different types of flow meters are examined and the possible explanations for their difference also outlined . Then a comparision on the basis of head loss characteristics of these meters is also given. Because this fact is of some importance to meter users, a full discussion over it is given. Then the advantages and disadvantages of these meters are shown, It is a concern primarily of meter manufacturers, because if any meter is more advantageous to the meter user, it will have better market and the manufacturer makes a profit. Since the discussion regarding the use of the meters is given fully in some test books on fluid meters, only a brief outline of it is given here.

Essential Difference Between Dall Tube and Classical Venturi (1)

1. The steepness of both the converging cone and of the

diverging cone or pressure recovery cone is different. The Dall flow tube has an included angle of appropimately 40° and the later an included angle of approximately 15°.

2. The presence of a dam at Fig. 3 and the Characteristic location of the pressure connection, similar to the type commonly used in Europe and known as corner tapes.

3. The presence of a sharp edges at slots instead of any smooth curve in the Dall Flow tube is also a characteristic feaute as against long venturi meter.

4. The sharp edge slot at the throat at which down stream pressure connection is there in the former.

5. The sudden enlargement by truncating the recovery cones a compact instrument is obtained , the length of thewh ole device being of the order of two diameters only.

CALIBRATION CURVES:

It is seen from the Chapter II that the coefficient of discharge of short venturis having rounded entrance cones, is very high. This is due to low loss in the converging cone as has been already described in Chapter III.

The calibration curve of Dall Flow Tube resembles more like the calibration curve of orifice meter. A representative calibration curve of orifice meter is shown in the Fig. 41. The calibration curve for Dall Flow tube of experimental set up no. 3 is also similar with the exception that there is no hump in this curve. Because this meter is designed such that under normal operating conditions the Reynold's number are quite high and calibration curve remains flat during that range. Below a critical Reynolds' number, thecoefficient of discharge increases as is expected. This critical Reynold's number for experiment No. 3 is nearly 3.5×10^{4}. Below this value the coefficient of discharge increases and hump portion of the curve is expected.

For the experiment no. 4 the calibration curve Fig. 28 shows perfect resemblance to Fig. 41 of orifice meter calibration curve. The Reynolds number below which the coefficient of discharge becomes variable is nearly 1.3×10^{4} and beyond this the normal working range exists.

EXPLANATION FOR HUMP PORTION OF THE CALIBRATION CURVE:

From the curves Fig. 41 and Fig. 28 it will be seen that below the critical value of Reynold's number, the discharge coefficient rises as the value of Reynolds number decreases, reaches a maximum value and then decreases rapidly, so that hump results in a coefficient of discharge. For orifice plate, the height of hump is related to the value of β (the diameter ratio) and increases as β is increased there is to say an increase in the throat bore results in the increase in height of the hump. The reason for hump is explained by an analysis of the various factors which determine the value of Coefficient of discharge. For the orifice plate, these factors are the expansion of the jet on the downstreamside of the orifice plate due to the reduction in the momentum of the fluid after it has passed through the plate, the change in the velocity distribution, the viscous shear forces in the fluid tending to suppress eddies on the downstream side of the orifice plate. This last effect is due to predominating effect of viscosity at very low Reynolds number. Below the critical value of Reynolds number the expansion of the jet of fluid and the alteration in the velocity distribution both reduce the differential head and hence the value of discharge co-efficient increases. On the other hand the viscous forces tend to increase the differential head and therefore reduce the coefficient of discharge. But the effect of viscous force is less than the effect of expansion of jet and the velocity distribution and the combined result is that the co-efficient of discharge increases. In addition when the orifice plate has a large bore, owing to the parabolic nature of velocity distribution less force is required to accelerate the jet of fluid through the bore and a larger increase in the discharge coefficient is obtained then with small orifice bores. In all cases the coefficient of contraction of the jet will be less than in the turbulent flow region. For very low Reynolds numbers the viscous forces predominate and hence there is a tendency for the differential head to be increased. This reduces the discharge coefficient.

For venturimeters very low Reynolds number are not obtained in the present experiments but it has been seen that at very high value of β , there is a tendency of a hump to be produced in the 'mixed' flow region i.e. in the region mjust below the critical value of Reynolds number (Fig. 41). This hump is due to predominating effect of alteration in the velocity distribution. It will be well appreciated that the jet is unable to expand, as it is confined by the walls of the tube.

For Dall tubes and Dall orifices the hump in the discharge coefficient curves at low Reynolds numbers is due to reduction in the additional differential head resulting from the streamlined curvature at the throat. The change in value of discharge coefficient is more pronounced for larger value of .

The relationship between the discharge coefficient and Reynolds number in case of Dallorifices is similar to Dall tubes, but the difference is that in the case of Dall orifices the discharge coefficient is appreciable higher in the region of low value of diameter ratio β . The discharge coefficient of Dall orifice lies between 0.6 and 0.8.

In the case of Double throat venturi tube the discharge coefficient is low. Because the parallel stream lines in the first throat are given a curvature in the second contraction as shown in figure, which lowers the pressure measured in the second throat. The discharge coefficient of nozzle type differential pressure producing approximates to that of a standard venturi tube. Because the entrance portion is similar to the venturi meter. In the nozzle there is no recocery cone as in the venturimeter but because there is practically no effect on discharge coefficient due to recovery cone, the discharge coefficient of nozzle approximates to that of venturi tube. Test results show little difference in discharge coefficient for the various shapes of nozzles.

It has been stated [5] as one of weaknesses of Dall Flow tubes that the discharge coefficient starts vary at a higher Reynolds number as compared so that of venturi meters. For Dall flow tubes, the critical value being given as 2×10^5 and for venturi meters it is 4×10^4 But from the present experimens No. 3 and No. 4 it will be seen that the critical Reynolds number is much less than 2×10^5. In case of Experiment no. 3 it is 3.5×10^4 and in case of experiment no. 4 it is 1.6×10^4 nearly . This means that the coefficient discharge variation for Dall tubes is at less critical Reynolds number than the venturimeter case.

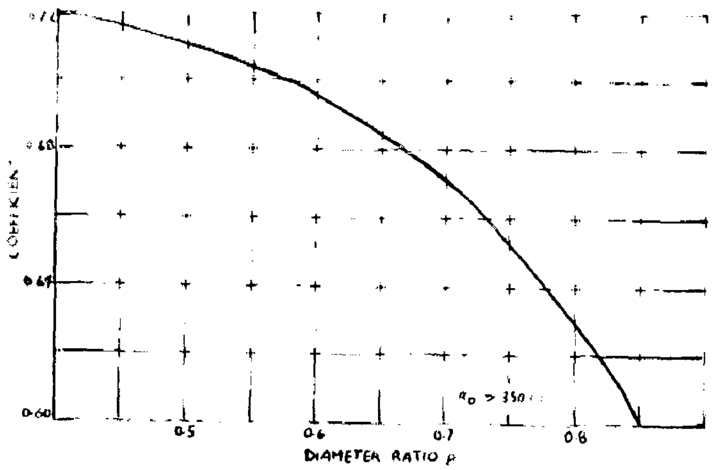


FIG 42 - DALL-FLOW-TUBE COEFFICIENT VERSUS DIAMETER RATIO

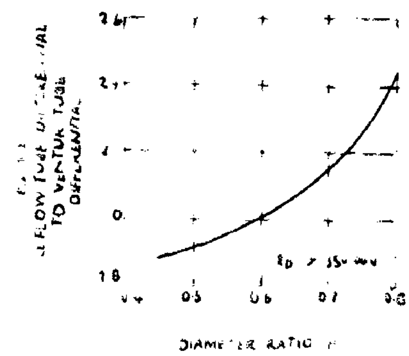


FIG 43 - RATIO OF DALL FLOW TUBE DIFFERENTIAL TO HERSCHEL TYPE VENTURI TUBE DIFFERENTIAL

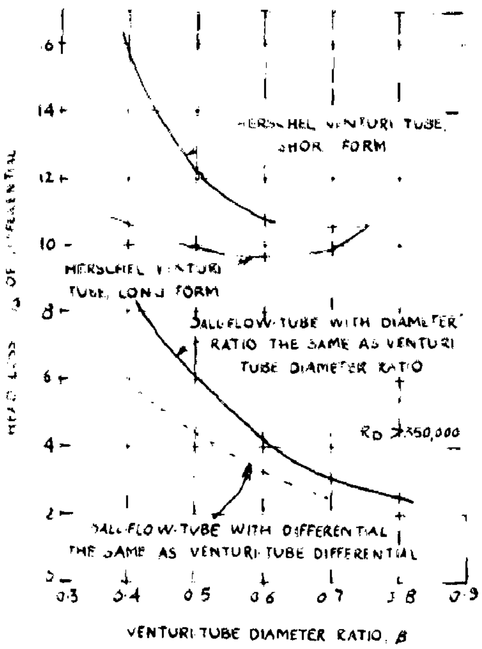


FIG 44 - HEAD LOSS COMPARISION

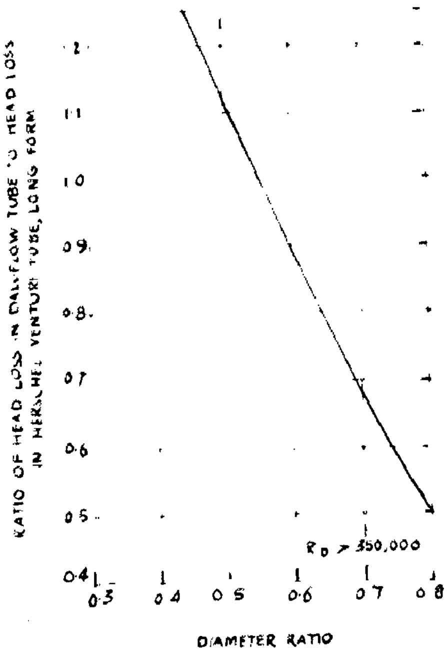


FIG. 45 - HEAD LOSS RATIO

In order to get a curve relating the coefficient of discharge of Dall tube and the diameter ratio, a number of test pieces are requiree. But this is not available at present case to establish such a curve. However a representative curve is given in Fig. 42. It will be seen from the above curve that as diameter ratio increases, the coefficient of discharge decreases.

Similarly in Fig. 43 it is shown that as the diameter ratio increases the ratio of Dall Flow Tube differential to venturi tube differential increase.

HEAD LOSS COMPARISION:

In fluid metering problems accuracy and reliability of fluid measurement are important factors. Besides them the other important factors which warrants consideration is the head loss in the matter . In cases where fluid is being discharged under gravity, it may not be a vital factor to be considered, but in cases where fluid is to be delivered against gravity or to a destination where pressure of fluid plays a vital role, the pressure loss in the meter is to considered fully, Because if pressure loss in measuring device will be more, then the power required to deliver the fluid becomes more and finally it becomes uneconomical.

As a rule, in order to obtain lower pressure loss additional ~~and~~ capital expenditureis required, as the differential pressure producing device is more costly thus the orifice plate form of detecting element is the least expensive but produces the greatest head loss, while the venturi tube with its conical downstream section, costs for more but saves energy other factors being equal before deciding on the form of differential pressure producing device to install, on the basis of ave-rage working conditions, the continuous expense caused by the loss through the various devices available should be capitalised, thus obtaining a guide as to whether the extra cost of low pressure loss device is warranted. In many instances it will be found that, when dealing with large volumes of even low pressure gas, a simple orifice plate is more expensive in the long run than a device designed to recover some of the differential pressure produced since the farmer calls for additional power requirements.

From the Chapter II it is seen that for short venturis the pressure loss is less at small diameter ratio, This fact can be well verifi d from Fig. 44. More over the pressure loss in the Dal Tubes, Gould be expected to be low but due to reasons already stated, they are little high. The follow ng discussion gives a clear picture of the head loss conditions for different meters.

Fig. 44 shows the head loss through a Dall flow tube expressed in percentage of differential pressure evolved . For the purpose of comparision , the head losses for short form Herschel typ venturi tubes are of long form Herschal type variation tubes have been included in the same figure. These curves are plotted against diameter ratios β . The dotted curve shows the percentage head loss through a Dall flow tube which produces the same differential as a venturi tube with a diameter ratio indicating in horizontal scale. The head loss figures given in figure 44 indicate the loss caused by the differential produced when it is followed by a sufficient straight pipe to provide full recovery. This is about 5 diameters. In other words, the loss is the increase in pressure drop which would occur between the ends of a straight piece of the pipe if the pipe were cut into two, the pieces separated, and the metering device inserted between them. For the Dall flow Tube this loss was determined by measuring a pressure drop between a tap 1 diameter upstream and a tap 5 diameters downstream of the differential producer and rubstracting the loss in a piece of pie 6 diameters long . This pipe loss was determined by test. The increased loss caused by inserting a differential producer in a section of pipe by removing a section of the pipe and inserting measuring device without separating the remaining pipe sections (as would normally be done in an existing piping system) is less. In the case of the Dall Flow Tube the difference is substantial.

Fig. 45 shows the ratio of head loss in Dall Flow Tube to head loss in Herschel type venturi meter (at Reynolds number greater than 350,000 where the ratio becomes constant).

It will be noted from Figure 44 that the head loss of the Dall Flow tube varies from about 0.5 to 1.3 times the head loss in a venturi tube of a corresponding diameter ratio β. In other wards, part of the phenomenal performance of the Dall Flow Tube is dependent on the increase in differential. However, thehead loss even with a given β, is less for the Dall flow tube when the diameter ratio is greater than 0.552.

INVESTIGATING HEAD LOSS IN DALL FLOW TUBE:

Some investigations has been made, and it sheds considerable light on the reason for the high differential, but it does not indicate why the head loss is lower than that for a venturi tube of the same diameter ratio. provided that this ratio is above 0.552. A test probe was used to explome the variation in static pressur e in the plane of the throat slot of 6 in. Dall flow tube with 0.354 β ratio. A U-tube manometer was connected to the test probe and to the inlet tap of the Dall Flow Tube. The probe was moved across a diameter in the plane of the throat slot. The probe was fitted with a slot in the side and was kept parallel to the axis of the Dall flow tube since it was desired to pick up the approximate static pressure rather than an impact pressure.

Fig. 46 shows the ratio between the manometer indication with the probe in various positions and the differential pressure when the probe was on the axis of the tube. It will noticed that the static pressure decreases to provide 2.4 times as much as differential at the wall as at the centre the Dall Flow Tube . Further investigation is necessary to find out more about this gradual increase in differential as the probe is moved from the axis of the tube, but it is easy to see that a change in the di-rrection of flow lines can be largely responsible.

An attempt is being made by Dr. John R. Weske at the University of Maryland to photograph streamlines through a Dall flow tube made of methyle methacrylate. It is hoped that the extremely low loss of head through a Dall flow tube may be evdentually explained.

From thehead-recovery efficiency point of view the order in which differential pressure producing devices should be placed in

1. Dass Tube.
2. Venturi Tube.
3. Dall Orifice Plate.
4. Orifice Plate.

From the head loss point of view, for all practical purposes the efficiency of the nozzle is identical with that of theorifice plate.

ADVANTAGES AND DISADVANTAGES:

DALL FLOW TUBE

Advantages:

1. Lowest head loss of all known velocity increasing differential pressure producers. The pressure loss owing to Dal tube varies between half and one third (depending on throat ratio) of that of long pattern venturi tube. At the higher throat ratios the loss is as low as 2½% of differential head and is no more than 7½ % at the lowest ratios. These low losses represents a very considerable saving in pumping or compressor costs.

2. Easier and cheaper installation than a venturi tube. The length and weight of Dall tube are between one third and one sixth of those of the long pattern venturi tubes and are about half those of short pattern venturi type Handling and installation are thrus easier , and often a saving is achieved in the size of any pit or chamber.

3. Lowe-r first cost than a venturi tube owing to economy in material and machining. Manufacture is facilitated by the short length and the absence of longitudinal curvature. Fine maching limits are possible and the tubes are

always manufactured as a single unit. The design lends itself to east or fabricated constfuction.

APPLICATION:

The Dall tube is suitable for most application where pressure loss is important or costly, it can be used with any 6 in (150 mm) or larger pipes or circular ducts and for most fluids. For water, sewage, or other liquids tubes are usually made from castings, while for measurements of air, gases and steam fabricated tubes are normal. For cold water duties, the inner and oter surfaces are protected from corrosion by a standard Epidocose finish.

DISADVANTAGES:

Like other known flow metering elements, the Dall tube fails to be outstanding in all respects. Its weaknesses are:

1. A standard Dall Flow Tube is not suitable for measuring the flow of fluids containing solids which are apt to settle out in the throat of the slot.

2. A more straight pipe is required than for some primary devices which have been used for a few decades.

3. Some cavidation has been experienced at unusually high velocities and low pressures. However, no tests have been made to determine the comparative cavidation in venturi

tubes, orifices or nozzles under the same conditions.

4. It has been stated that the coefficient of discharge becomes variable below a Reynolds number considerable higher than that at which a venturi tube coefficient starts to vary In the present investigation the contrary is found to be true.

VENTURI TUBES:

Advantages:

The main advantage is that the overall loss of pressure is less than for nozzles and orifice plates. This is important in small. The loss is usually from 10- 20 per-cent of the differential pressure and decreases as the size of the venturis throat is increased. At high speeds, however, the total pressure recovery may not be attained until the fluid has flowed to a considerable distance beyond the end of the venturi tubes. A venturi tube is useful in measuring the flow of slurries and suspensions of solids in liquids where as Dall Flow tube fails to serve this purpose. A piezometer ring should not be used in such application.

DISADVANTAGES:

The main disadvantages of venturi tubes are:

1. Its high initial cost.
2. It cannot be easily installed in an existing pipe line

because of its length.

3. Once the tube is manufactured and installed it is impossible to change the range of flow installation except by modifying the differential pressure measuring instruments or replacing the venturi tubes.

ORIFICE METER:

It has already been discussed that the calibration curve of orifice meter resembles that of Dall Flow tube. Its head loss is highest as compared to other meters since there is no recoverable part in it. As regards its cost, its first cost will be less than that of Dall of flow tube.

BEND METER:

The losses in a bend are:

1. A loss at entrance to the band due to a tendency for changing from rectilinear to vortex motion.

2. A loss in the bend greater than the normal pipe loss because of increased turbulence.

3. An excess loss in the straight portion of the pipe following the bend due to the re-establishment of normal pipe flow. This loss occurs over a considerable length of straight pipe beyond the bend.

Since these losses in a bend are permanent bosses, it is not desirable to use a bend for metering problem. But where there exists a bend in the layout of a pipe, then it is advantageous to calibrate it so that no other meter will be required for measuring purpose. But its reliability in operation i.e. indicating the flow rate may not be so accurate.

FLOW NOZZLE:

Owing to the smooth entrance cane, fluids flow more easily through nozzle than an orifice so that a small value of 'm' can be used for a given rate of flow. The nozzle is therefore used in high velocity mains.

Its main disadvantage is that owing to its having no exit cone it produces a large overall pressure loss, although this loss is slightly less than that produced by an orifice plate. This loss is usually about 50% of the differential pressure so that this type of installation cannot be used where the available pressure head is very small. It cannot be used satisfactorily when the value of the ratio m lies outside 0.2 and 0.55. In gas or steam service the ratio is limited to values below 0.40.

PITOT STATIC TUBES :

Advantages :

1. It produces no appreciable pressure loss in the main

unless it is made large in comparision with the size of the main.

2. It can be inserted through a comparatively small hole into the main without the necessity for shutting down the main. It is, therefore very useful for estimating the flow through a main in order that a more permanent type of flow measuring device such as orifice type may be designed for the main.

3. It can be used to find the distribution of velocities in the main or flue. This is useful, for example, when it is required to site a sample tube in a flue for waste gas analysis.

4. Its cost is low.

DISADVANTAGES:

One of the principal limitations to the use of pitot static tube is the fact that the gas or liquid must be moving at high velocity, if an appreciable differential pressure is to be produced. For this a modified form of pitot tube, such as the pitot venturi tube and pitot Dall tubes is used. It is essentially an exploratory device and is rarely used permanently in industrial work.

The pressure loss produced by the devices like pitot venturi and pitot tube is usually negligible, especially in the case where the size of the pipe is large in comparision with the venturi or Dall tube used.

SUMMARY:

From the above considerations it is clear that Dall flow tube has got superiority over other meters in many respects.

1. It produces a high differential.
2. Its pressure recovery is very high i.e. the loss in the meter is low contrary to the expectation. Because due to sharp edges, it is expected that the losses will be more.
3. Its dimensions is smallest amongst venturimeters of Long form and short forms . Hence installation trouble is minimised.
4. As regards calibration, it resembles more like orificemeter calibration than venturimeter calibration.

-:0:-

CHAPTER V

CHAPTER V

INSTALLATION, TESTING AND MAINTENANCE OF METERS.

INTRODUCTION:

This Chapter deals with the installation, testing and maintenance of differential pressure flow measuring devices and the recording units, Generally all these factors are furnished by the manufacturers and the meter user has to follow the instructions of the manufacturers. The disturbances in the upstream side of the Dall Flow Tube and the effect of ambient water temperature on the discharge coefficients have been discussed also. Then the precautions that are necessary to be followed during installation, testing and maintenance have been outlined too. A full description of these facts are not given here, as they can be found in many text books. Only the factors which came during the experiments have been outlined.

INSTALLATION:

Before considering fully the installation conditions for Dall flow tube, the effect of upstream disturbances is given first.

UPSTREAM DISTURBANCES:

Dall flow tube is sensitive to upstream disturbances, like other detecting elements. In many cases it is more sensitive than the most commonly used primary devices, but it is by no means more sensitive than all other differential producers under all installation condition

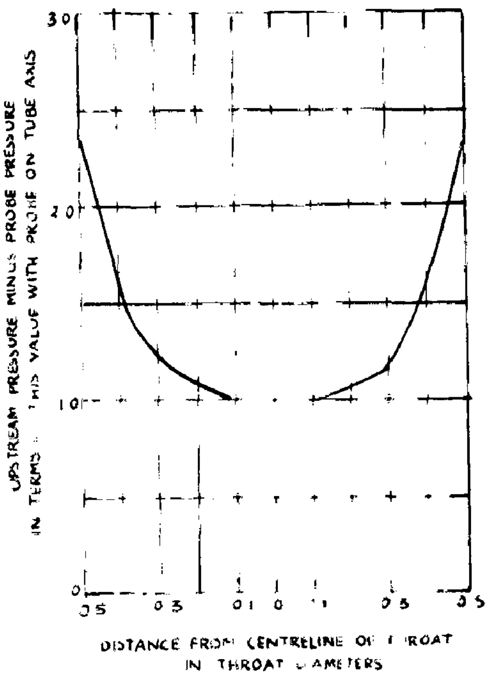


FIG 46 - DISTRIBUTION OF STATIC PRESSURE AT THROAT OF DALL FLOW TUBE

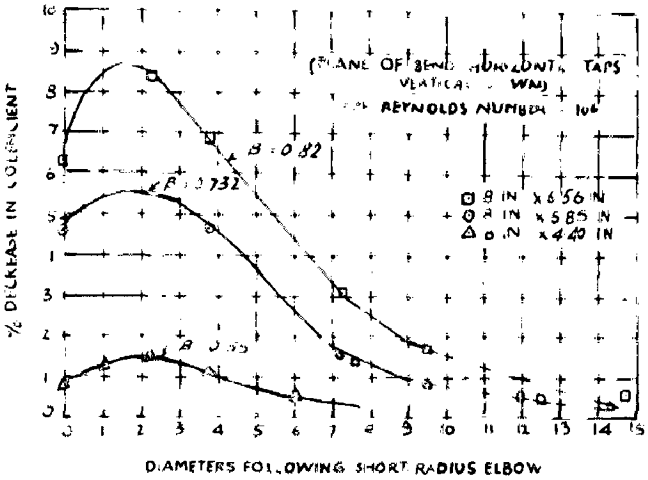


FIG 47 - EFFECT OF SHORT RADIUS ELBOW UPSTREAM FROM DALL FLOW TUBE

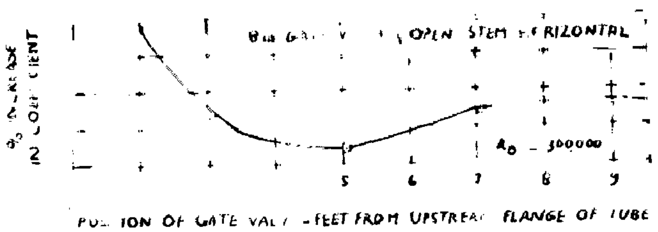


FIG 48 - EFFECT OF GATE VALVE UPSTREAM

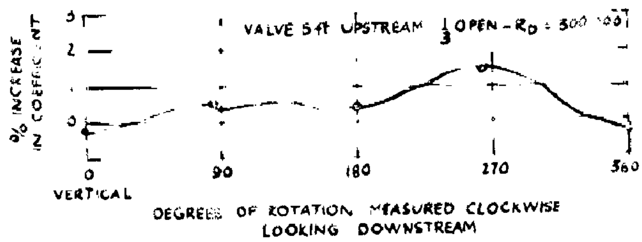


FIG 49 - GATE VALVE 5 ft UPSTREAM

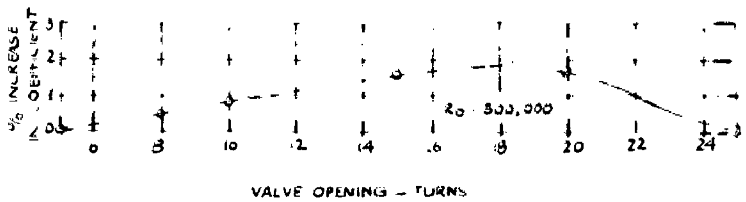


FIG 50 - GATE VALVE 5 ft UPSTREAM, STEAM HORIZONTAL

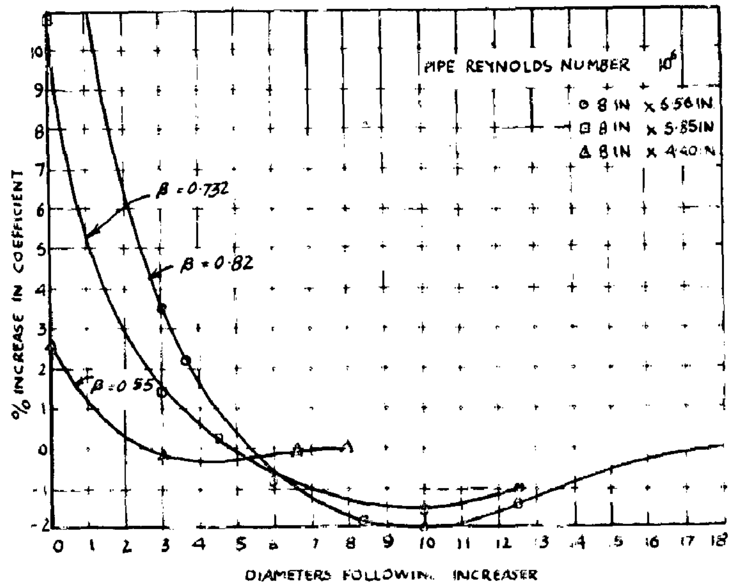


FIG. 51:- EFFECT OF 6"× 8" INCREASER 11" LONG PRECEDING DALL-FLOW TUBE

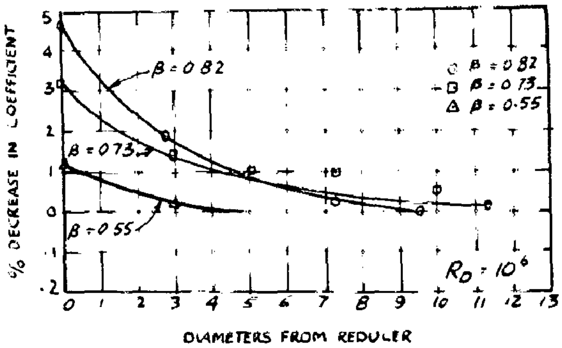


FIG. 52:- EFFECT OF 10"× 8" REDUCER 12" LONG UPSTREAM FROM FLOW TUBE

The error caused by various upstream disturbances, when allowance is made for coefficient of discharge is shown in Fig. 47, 48, 49, 50 and 51 and 52.

Fig. 47 shows the effect of a single elbow upstream for three different β ratios. Due to elbows in the up- stream, the coefficient of discharge decreases . Fig. 48 shows the effect of a gate valve, opened onethird, with its stem parallel to the inlet tap, and the portion of the gate in the stream on the same side as the inlet tap in the Dall flow tube. This figure shows that the coefficient of discharge increased with a gate valve in the upstream side of the Dall flow tube.

Fig. 49 shows the effect of rotating the valve so that its stem took the position shown by abscissa. The valve remained one third open, it was located 5' ($7\frac{1}{2}$' pipe diameters) upstream and the inlet pressure tap was at 90° position. Here also there is a increase in the coefficient of discharge. Fig. 50 shows the effect of a gate valve with various openings when located 5 ft. upstream, with the stem parallel to the inlet tap. The coefficient of discharge increases with valve opening-turns.

Fig. 51 shows the effect of a 6 in x 8 in increaser ahead of Dall flow tubes of three different β ratios. The coefficient of discharge increases here also. In all cases the error reverses. The coefficient starts off too

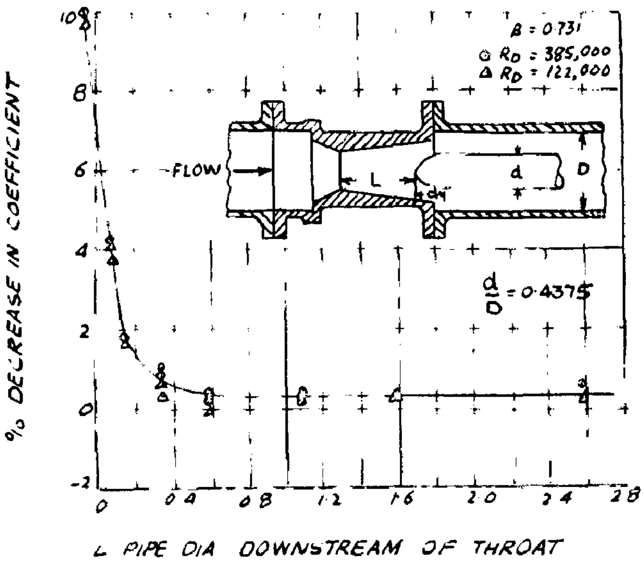


FIG. 53 INTERFERENCE TEST

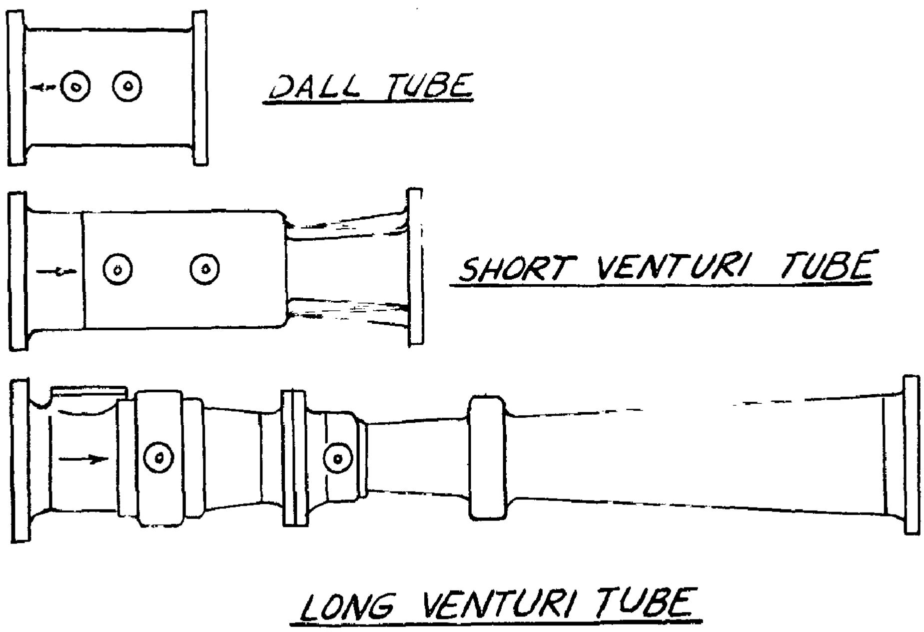


FIG. 54a

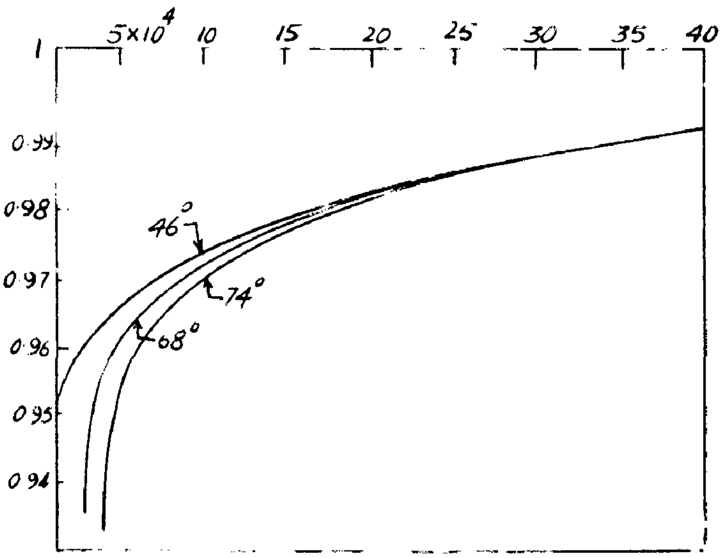


FIG.54:- TEST OF VENTURI METER AT VARIOUS TEMPERATURES CO-EFFICIENT PLOTTED AGAINST REYNOLDS NUMBER

high with no straight pipe between the increaser and the Dall flow tube, then the coefficient becomes too low as the straight pipe ahead of the Dall tube is increased and finally the coefficient rises to its standard value.

Fig. 52 shows the effect of a 10" x 8 in reduces upstream from the Dall flow tube. The coefficient also increases in this case.

A question arose as to how close a particular type of disturbance could be located to the throat when placed in the downstream cone. Fig. 53 indicates the type of disturbance and its effect. It will be noted that there is no observable change when the position of the disturbing object is increased beyond one throat diameter downstream from the throat slot. The coefficient was about 0.3% higher than standard, but this is believed to be due to test inaccuracies rather than to the object in the diffuser.

CONCLUSION:

From the above experimental investigations it will be seen that the coefficient of discharge is affected by upstream disturbances. The reason for these changes being the change in flow pattern caused by the disturbing elements placed ahead of Dall flow tube . In all the above cases except the effect due to short radius elbow, it is seen that the coefficient of discharge increases which means that the differential pressure read for the same discharge as without disturbance is less. As the position of disturbance is placed away from the Dall flow tube, the effect reduces and the increase in coefficient of discharge starts falling in almost may cases. For higher a β ratios the increase or discrease of coefficient of discharge is more than smaller, diameter ratios.

EFFECT OF AMBIENT TEMPERATURE:

If a venturi meter or Dall flow tube is installed and is calibrated, after some days it will be found that the calibration curve does not tally. This discrepancy is due to the ambient water temperature (26) Fig. 54 shows the calibration curve for a venturimeter at three different temperature, of water, ie at 46° , 63° and 74° F.

DISCUSSION:

The rate of heat transfer through the walls of the meter tube is sufficient to produce eratic coefficients at low throat velocities, presumably because of time element. Due to the fact that the heat transfer coefficient for turbulently flowing water and still air are of different orders, changes ofwall temperature from that of stream are ordinarily insignificant except at low velocities. Errors are expected with un insulated hot water meters, supercasted liquid ammonia meters and gas meters which are heated to prevent the formation of deposits at the throat. A rise of gas temperature should lead to a negative instead of positive error, since the viscosity of gas increases with temperature, while that of water then fall; No effect as to be effected on the flat portion of the coefficient curve for any differential producer. Since a large value of venturi coefficient exists on the flat portion of the curve and starts to fall off at a relatively high value of the throat velocity and usual temperature and hence of the peynolds number such a venturi is relatively sensitive to this error.

But this effect of ambient water temperature on the coefficient of meters is only at low Reynolds number and hence usually in the working range this does not alter the values. The Dall tube can be installed at any angle. For water, flow measurement in horizontal, or sloping mains, the connections are normally at the sides; for air or gas flow measurement the connections are normally at the top. An arrow, cast on the case, indicates the direction of flow.

As for other differential pressure producers Dall tubes should normally be installed with at least ten pipe diameters of straight pipe on the upstream side- more for large ratios (i.e. large throat bones) less for small ratios. Special consideration must be given to installations where unavoidable disturbances occur upstream of the Dall tubes. Any form of control should be fitted downstream, preferably a few diameters or more away.

PROCEDURE:

Before proceeding to fit the detecting element in position is the pipe, the pipes should be flushed out to remove any debris and other foreign matter. Then, in the case of venturi and Dall tubes, nozzles and orifice plates, the necessary washers should be prepared. For low and medium pressure work, washers of rubber composition are used, and the internal diameter of these should be cut slightly larger than the bore of the main; it is essential that the washers do not project into the main. Care must be taken to ensure that the detecting element is bolted into the main the correct way round. In the case of orifice plates, there are marked to indicate which side faces upstream, and it is necessary to check this point, as sometimes the plate is chamfered on the downstream side.

In the case of pitot tubes the pipe should be drilled and tapped in the position specified and the static pressure connection also drilled and tapped in the correct position, if this connection is required. The pitot tube must be inserted into the main to the correct distance, and the upstream orifice must be carefully located to point truly upstream.

In all cases isolating valves are provided for fitting at the tapping points. These enable the pipe and detecting element to be isolated, from the remainder of the metering system. When an orifice plate with pipe or flange tappings are being installed, particular care is necessary to ensure that the upstream and downstream tappings are located in the exact positions specified. After drilling and tapping it is essential to remove all burrs from the interior of the pipe, and the screwed ends of the valves or connections must mot project into the pipe.

The two pressures from the differential pressure producing device to the measuring unit are transmitted through pipes, in the majority of cases the metbred fluid being allowed to enter the pressure pipe-s and the instrument. The installation of pipes should receive particular attention; faulty installation of these pipes is a source of more errors than all other factors combined.

For a gas metering it is essential that the pipes are run in such a manner that water locks cannot occur in the connecting pipes, conversly , when the metered fluid is liquid, air and gas locks must be prevented. It was observed during the experiments that any discontinuity in the column of liquid gave rise to spurious head, the magnitude of which will be equal to the keen of the vertical heights of the breaks. The resultant meter error is not always revealed by equalising the two pressures nor by the rate of flow falling to sero.

To ensure satisfactory installation, the following ~~rules should be observed :~~

For Liquids:

When the instrument is above the dete-cting element the two pressure pipes should dip down from the detecting element to minimise the possibility of air from the main entering these, and then gradually rise to the instrument at a slope not less than are in twenty. When the instrument is below the detecting element the two pipes should gradually fall to it at not less than the minimum slope.

When metering dirty liquids special precautions have to be taken to prevent the pressure pipes from becoming blocked and to prevent foreign matter entering the differential pressure measuring instrument, the diameter of the pressure holes at the throat should not exceed onetenth of the throat diameter.

With regard to the thermometer pockets and similar projections into the main, whenever possible there should be located downstream for the detecting ~~monitmg~~ element when it is essential for a projection to the situated upstream of the detecting allotment and it should be about fifteen pipe diameters ~~dimands~~ away; the minim. distance is influenced to a considerable degree by the relationship between the area of projection and the cross sectional area of the pipe.

LOCATION OF METER:

The meter should be located in a pipe section where the flow exibits the symmetrical velocity distribution commonly known as 'normal turbulence' If interferences (i.e. swirls, Cross currents, or eddies) in the flow paytern occur close to the meter, particularly upstream as has been discussed earlier, flow measurements are apt to be erroneous. For a given installation, the magnitudes and direction of erros will depend upon the specific combination of conditions existing in the flow. The following general criteria should be satisfied in selecting the meter at location:

1. The meter should be located where the flow is the most uniform, publsations and surges should be avoided.

2. A pipe section with the maximum available length of stroight pipe upstream and downstream from the meter should be selected.

3. The fluid should remain in single phase when passing through the meter , no vaporisation of liquid or condensation of vapour should occur.

4. When metering liquids at low Reynold's number a location should be chosen ~~highxinxmmdox~~ where temperatureis higher. in order to minimise viscosity effects on discharge coefficient.

5. When metering a gas, the meter should be, installed

at a point where the pressure is lowest and temperature highest in order to minimize gas low deviations.

POSITION OF LINE:

Differential meters can be installed in any position. For liquid metering, location of meter in a vertical line with downward floor is not advisable, since under some conditions the liquid may fall free and may not fill the line. However, when gases contain condensable contitituents, an installation of this type may be desirable in that it allows any condensed liquid to be blown through the meter. The line should be concentric to the meter.

STRAIGHTENING VANES:

Swirls or eddies upstream of the primary element caused by partially closed valves, by regulators, or by combination of the elbows in different planes, will lead to inaccuracy in flow meter reading. The flow meter will usually read low.

The use of straightening vanes will eliminate, or very greatly reduce , such measurement errors if they are due to helical motion of the liquid. When placed after irregularities where helical flow does not exist, they may do more harm than good, by preserving abnormal velocity distribution due to the bend, which would otherwise normalise itself.

Straightening vanes should be securely fixed and should be accessible for periodic inspection, if there is any danger of their damage. A length of 5 diameters of uninterrupted pipe is sufficient downstream of primary element in all cases. Straightening vanes should preferably not be used downstream from single elbows, since they lend to prolong the distorted velocity traverses which is produced by this type of fitting.

TESTING AND CHECKING OF METERS:

Even though all recommended procedures for setting up the differential pressure produces meter installations are carefully adhered to in any given case, it is constantly necessary to guard against faulty operation of the system. This is one of the functions of the instrument maintenance groups in refineries and other industrial plants and makes possible the consistent accuracy and reliability of flow measurements. Their work involves the applications of a rautined schedule of tests and checks which experience has shown are essential for revealing, preventing and eliminating commonly encountered errors. Although these errors are not inherent in the primary or secondary element, they develop with time under the influence of service conditions and can only be recognised and remedied through continuous inspection.

Instructions for adjustment and repair of instruments are furnished by equipment manufacturers so that the difficulties which develop may be readily corrected by the user. To this end, maintenance scheduled are set up with provision for frequent zero adjustment and periodic calibration of the flow meter itself so that operating troubles may be uncovered. The frequency of checking will depend on such considerations as the importance of the instrument, past experience with the type of instrument being used and how after there are indications of error. Usually when meters are calibrated, a detailed inspection of the conditions of the detector, connecting lines, and attendant facilities is made in addition to the required adjustment, cleaning, and servicing of the instrument. In this respect, it should be noted that all elements of the system should be carefully inspected any time that a meter is employed for test purposes.

The tests of meters serve two purposes:

1. In the case of a meter which has been in operation, they reveal the magnitude of past errors so that corrections may be made in meter records obtained for lost accounting requirements or for studies of unit or plant performance. Under these circumstances, no adjustment of the meter mechanical are made unfit after each test has been completed.

2. In the case of a meter which is to be placed in service, the tests ensure that dependable results will be provided in ensuring operation of the meter . In this situation, all possible adjustments of the meter mechanism are made before several tests are under taken.

In addition to the foregoing, attention should be given to the possibility of leaks in the system, concentric rotation of the chart etc. For the case of meters recording at a renote point through a pneumatic or electrical transmitter, it is important that the receiver be checked frequently, to see that it reads the same as transmitter. Seal pots, where used, should be vented fr m time to time, and sealing fluids should be replenished, and or drained and replaced at regular intervals particul where emulsification with the flowing fluid may occur,

CONCLUSIONS:

1. As has been seen the ~~xxxxx~~ upstream disturbances affect the coefficients of of discharge of Dall flow tube and in order to minimise the effects there should be a straight portion of the pipe preceding the meter. The length of this tube should be minimum 15 diameters.

2. Where the disturbing elements can be eliminated, their effects can be to a great extent controlled by the provision of straightening vanes.

3. The ambient temperature does not affect the discharge coefficient in the working range of the meter.

4. As regards a installation of the meters, the nmanufacturer instructional should be followed.

5. Periodical checking and checking of members are mevessary to obtain good service from them.

D I S C U S S I O N S

1. In Chapter 1 the different types of differential pressure producer members given with a short description on each of them. A brief theoretical analysis of the metering problems has been dealt with.

2. In Chapter II, the experimental set ups are given. There are four experimental set ups, 3 in senior hydraulic Laboratory and onein Junior Hydraulic Laboratory. The calibration curves for all of them have been drawn from the experimental results.

In experimental set up no. 1, after calibrating the short venturi, the calibration curves for bend meters, shunt water meter and rectangular weir have been drawn. The coefficient of discharge of this short venturi is very high as is expected to be. The relationship between the discharge and square root of head in the bend meter is found to be a straight line. The calibration curve for this meter is nearly a straight line and slightly decreases with high Reynolds number. The operating range of this meter is inturbulent flow region only.

The co-efficients of rectangular weir has been calculated and it is seen that the coefficient is nearly equal to the Bazin's coefficient. There is sufficient straight portion before and after the meter, hence the disturbing effects are small.

3. In the experimental set up no. 2, the short venturi is calibrated and then the calibrations of water-meter in series with the meter, rectangular weir and shunt meter have been drawn. Here again the discharge coefficients are very high and remains practically constant during the operating range. The pressure recovery is very high also.

4. Dall tube is installed in Junior Hydraulic Lab. A calibration curve for the meter is drawn. It is seen that the calibration curve is nearly similar to the calibration curve of orifice meter. The coefficients of discharge are higher because there is upstream disturbance due to gate valve and the bends nearer to the meter.

5. A Dall tube was designed and manufactured. It was fitted in a 1½ inch pipe line in Sr. Hydraulic lab. This design is done geometrically similar to the Dall tube in the experiment no. 1. A calibration curve is drawn for the meter and it is seen that the curve is similar to the calibration curve of the orifice meter but the head loss is higher due to the region explained in Chaper II.

6. The theory of rounded entrance flow meters is dealt with in Chapter III. The calculations of discharge coefficients for the short ~~vehicles~~ venturis of experiment no. 1 and 2 is done with the help of this theory. Because the limitations of this theory apply quite favourably to

these meters. Hence the trouble of measuring the large quantity of discharge is overcome. This also helps in calibrating other meters used during experiments as has been already stated.

7. The different variables that come into play in the flow calculations have been discussed also in Chapter III. It has been found that out of all factors the velocity distribution at the low pressure tap i.e. at the throat section plays a vital role in the coefficient of discharge. Other factors have been discussed also.

The diffuser of the meter does not affect the coefficient of discharge. The diffuser angle of opening should not be very high to produce eddies etc and hence the maximum recovery depends upon the angle of opening of the diffuser and its length for a particular meter.

8. Possible explanations for the causes of various factors that come in the flow metering problems and the manner in which they affect the coefficient of discharge have been put forth.

9. In Chapter IV, the comparative study of different types of differential pressure flow meters has been given. It will be seen from this Chapter that the Dall Flow tube has got superiority over other types as regards differential pressure and pressure recovery are concerned. Moreover, the instrument is compact and being small in size, is cheap

and it is easy to install also.

10. The Chapter IV deals with the installations, checking and maintenance of meters in general. The effect of upstream disturbances on the discharge coefficient of Dall Flow tube is also discussed. From all these considerations, the limitations for the installation conditions have been outlined.

11. POSSIBLE EXPLANATION OF LOW LOSS IN DALL FLOW TUBE:

A vortex ring was observed at the discontinuity of the diameter at diffuser discharge of the Dall tube and the suggestion that this annular vortex contributes to the measured loss appear confirmed by investigations of Perkins and Hazen who found that a corresponding vortex in a wind tunnel increased the pressure recovery and imperoved the uniformity of velocity distribution downstream. The low head loss of the Dall Tube may be attributed to the fact that in retarding flow as in boundary layers and diffusers, the fluid shear stress at the walls drops to very low values even to zero and negative values, from a maximum which occurs away from the wall. This is as may be expected since the central stream will entrain the fluid near the wall previously retarded by adverse pressure gradients. It occurs in the downstream portion of any diffuser; in the Dall tube strong adverse pressure gradients are established directly downstream of the throat by abrupt change of slope in the wall; hence the reduction of wall stress is effective in the region where ordinarily very high friction losses occur.

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APPENDIX.

TABLE 1.

OBSERVATIONS FOR EXPERIMENT NO.1.

Sl. No.	Suction	Dis-charge bend dif. head "of water	Fric-tion head loss inch of water	Venturi Press. loss inch. of water.	Venturi Diff. head inch. of water.	Hook gage reading inches of water.	Water meter reading gallon/min.	Recording meter reading Cu. sec.
1	1	2	3	4	5	6	7	8
Head - 12 ft.								
1.	10.9	7.9	2.5	23.5	7.2	6.7430	0.92	6.30
Head - 15 ft.								
1.	8.2	6.2	2.2	20.2	7.0	6.3125	1.133	6.20
2.	10.9	9.0	2.5	25.6	9.5	6.5462	1.20	6.90
Head - 20 ft.								
1.	6.7	5.0	1.5	15.25	5.0	5.8745	0.933	5.60
2.	7.9	6.0	2.0	19.2	6.0	6.2805	1.05	6.18
3.	9.8	7.5	2.3	23.5	6.5	6.7008	1.10	6.70
4.	11.3	8.7	2.6	27.1	8.5	7.0313	1.20	7.10
Head - 25 ft.								
1.	5.3	3.6	1.0	11.7	4.1	5.3065	0.65	5.00
2.	6.9	5.2	1.5	16.2	6.0	5.9287	0.85	5.80
3.	7.8	5.9	1.7	19.0	6.4	6.1875	0.90	6.15
4.	10.0	7.7	1.8	23.7	7.5	6.7412	1.05	6.80
5.	11.0	8.9	2.4	28.8	8.5	7.0468	1.10	7.20
Head - 30 ft.								
1.	4.8	3.4	0.8	9.1	3.1	4.8675	0.58	4.60

1	1	2	3	4	5	6	7	8
2.	6.2	4.5	1.2	14.1	4.5	5.6535	[illegible].65	5.50
3.	8.2	6.5	1.9	20.7	7.5	6.3478	0.80	6.35
4.	9.7	7.7	2.0	24.8	8.1	6.7495	0.85	6.80
5.	11.2	9.0	2.5	28.8	9.8	7.0430	1.10	7.30
	<u>Head - 35 ft.</u>							
1.	4.1	2.3	0.6	6.7	2.3	4.4225	0.40	4.2
2.	5.4	3.9	0.9	12.2	4.0	5.3257	0.55	5.2
3.	7.3	5.5	1.4	17.8	5.6	6.0000	0.75	5.9
4.	8.7	6.9	2.0	22.3	8.2	6.4628	0.8	6.5
5.	10.5	8.3	2.5	27.0	8.5	6.9332	0.9	7.0
	<u>Head - 40 ft.</u>							
1.	2.4	0.8	0.4	2.2	0.7	2.9300		2.8
2.	4.4	2.8	0.7	9.5	2.6	4.6338	0.40	4.5
3.	6.3	4.6	1.3	15.0	5.0	5.6875	0.55	5.1
4.	7.6	6.0	1.4	19.0	5.8	6.1325	0.60	6.1
5.	9.4	7.6	1.8	24.5	7.0	6.6950	0.80	6.75
	<u>Head - 45 ft.</u>							
1.	2.1	0.4	0.1	0.8	0.2	2.1016		2.35
2.	2.8	1.0	0.4	3.2	1.0	3.660	0.25	3.25
3.	5.0	3.7	1.0	11.0	3.6	5.1032	0.60	4.90
4.	6.5	5.0	1.2	15.2	4.8	5.6778	0.65	5.60
5.	7.9	6.0	1.5	19.6	5.8	6.1825	0.85	6.20
6.	9.4	7.3	1.9	23.8	7.0	6.6330	0.933	6.70

1.	1	2	3	4	5	6	7	8
<u>Head - 50 ft.</u>								
1.	2.3	0.7	0.2	1.7	0.5	2.7300		2.70
2.	3.4	2.1	0.6	6.0	2.4	4.1365	0.30	4.00
3.	4.8	3.8	1.0	10.8	3.2	5.1094	0.50	4.90
4.	6.0	4.9	1.2	16.5	4.1	5.6533	0.70	5.50
5.	7.1	5.5	1.5	18.0	6.1	6.0703	0.80	6.00
<u>Head - 55 ft.</u>								
1.	1.2	0.1	0.1	1.0	0.7	0.3595		2.70
2.	3.1	2.5	0.6	7.1	4.2	4.4217	0.60	4.50
3.	4.4	3.6	0.9	10.8	5.7	5.1327	0.75	5.20
4.	6.0	9.2	1.1	18.4	8.5	5.8273	0.85	5.80
5.	7.4	6.1	1.6	19.5	9.00	6.1853	0.90	6.30
<u>Head - 60 ft.</u>								
1.	8.3	6.5	1.7	20.9	7.3	6.3805	1.0	0.60
2.	5.3	4.3	0.9	12.9	3.6	5.4390	0.70	5.50
3.	2.7	1.7	0.3	5.1	1.7	3.9750	0.25	4.10
4.	1.4	0.6	0.2	1.5	1.1	2.7385		3.20

A_1 = 1.03 sq.ft. A_2 = 0.467 sq.ft.

β = diameter ratio = 9.25/13.75 = 0.673

Opening ratio m = 0.4545

Bezins coefficient n = 0.405 + 0.000984 / H

Sl. No.	Bazins Coef.	Velocity ft/sec.	Renold's no.(Re)$_D$ x x 10^5	Discharge Q cusec.	Coef.of discharge.C_D	Friction factor	% friction loss.
	9	10	11	12	13	14	15
Head - 12 ft.							
1.	0.4209	5.65	5.84	5.81	0.991	0.00550	28.9
Head - 15 ft.							
1.	0.4237	5.29	5.38	5.4	0.991	0.00590	31.0
2.	0.42305	5.32	5.51	5.6	0.9915	0.00600	35.0
Head - 20 ft.							
1.	0.42505	4.55	4.71	4.68	0.9925	0.00506	32.0
2.	0.4238	4.99	5.16	5.14	0.9920	0.00558	32.0
3.	0.4226	5.52	5.72	5.59	0.9818	0.00550	26.2
4.	0.42178	5.90	6.12	6.08	0.9915	0.00554	27.0
Head - 25 ft.							
1.	0.4272	3.93	4.07	4.20	0.992	0.00454	30.3
2.	0.42492	4.58	4.75	4.90	0.991	0.00500	31.7
3.	0.4241	4.80	4.96	5.11	0.973	0.00516	29.3
4.	0.4225	5.55	5.75	5.93	0.991	0.00407	26.3
5.	0.42178	5.92	6.13	6.10	0.988	0.00478	32.7
Head - 30 ft.							
1.	0.42925	3.46	3.59	3.56	0.978	0.00465	32.1
2.	0.42502	4.29	4.45	4.42	0.983	0.00470	31.3
3.	0.4236	5.05	5.24	5.20	0.985	0.00530	30.8
4.	0.4225	5.55	5.75	5.72	0.983	0.00458	31.9
5.	0.42177	5.91	6.12	6.09	0.990	0.00498	27.8

	9	10	11	12	13	14	15
Head -35 ft.							
1.	0.4316	3.01	3.12	3.10	0.989	0.00465	33.1
2.	0.4271	3.95	4.009	4.06	0.988	0.00404	30.2
3.	0.4227	4.71	4.88	4.85	0.990	0.00440	28.8
4.	0.42328	5.20	5.38	5.36	0.993	0.00516	34.2
5.	0.42205	5.82	6.03	6.00	0.990	0.00532	28.9
Head -40 ft.							
1.	0.4453	1.682	1.742	1.732	0.989	0.00748	31.2
2.	0.4305	3.22	3.34	3.32	0.988	0.00470	29.3
3.	0.4258	4.22	4.37	4.34	0.991	0.00510	32.8
4.	0.42425	4.84	5.01	4.98	0.988	0.00392	29.4
5.	0.42265	5.46	5.66	5.63	0.987	0.00418	26.9
Head - 45 ft.					~~0.998~~		
1.	0.4637	1.07	1.11	1.103	0.993	0.00698	24.8
2.	0.4400	2.04	2.115	2.10	0.989	0.00697	28.2
3.	0.4281	3.74	3.88	3.85	0.988	0.00510	31.9
4.	0.4258	4.31	4.46	4.45	0.989	0.00646	30.7
5.	0.42415	4.80	4.97	4.95	0.987	0.00455	29.8
6.	0.4228	546	5.65	5.63	0.988	0.00432	29.3
Head - 50 ft.							
1.	0.4483	1.52	1.565	1.565	0.983	0.00600	30.2
2.	0.4334	2.74	2.84	2.82	0.980	0.00558	34.1
3.	0.4281	3.74	3.88	3.85	0.988	0.00465	33.1
4.	0.42585	4.32	4.47	4.45	0.990	0.00432	24.3
5.	0.42445	4.70	4.80	4.84	0.988	0.00175	32.2

	9	10	11	12	13	14	15
Head - 58 ft.							
1.	0.833	0.136	0.141	0.14	0.983	0.0376	59.8
2.	0.4334	3.05	3.16	3.14	[illegible]	0.0045	51.2
3.	0.4280	3.74	3.88	3.85	0.987	0.00418	30.2
4.	0.9272	4.50	4.66	4.64	0.983	0.00380	29.8
5.	0.4241	4.80	4.97	4.95	0.980	0.00485	50.1
Head - 60 ft.							
1.	0.4235	5.12	5.3	5.27	0.980	0.00187	33.1
2.	0.4267	4.05	4.2	4.17	0.973	0.00384	27.8
3.	0.4347	2.63	2.73	2.71	0.992	0.00304	32.8
4.	0.4482	1.54	1.595	1.59	0,988	0.00588	59.2.

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TABLE 2.

OBSERVATIONS OF EXPERIMENT NO. 2

Observation.	Press ft of water.	Diff. press. inch of water.	Friction head loss inch of water.	Venturi Press. loss inch of water.	Hook gauge reading in of water.	Water meter reading Gpm	Recording Meter reading
1.	2	3	4	5	6	7	8
1.	235	23.2	8.2	3.3	1.4800	216.5	0.560
2.	225	26.4	11.5	4.3	1.4177	230	0.600
3.	230	24.8	9.0	3.5	1.3933	221	0.570
4.	238	22.2	8.8	3.2	1.3320	208.5	0.545
5.	240	20.7	7.3	2.9	1.3050	201.0	0.520
6.	295	19.6	6.7	2.7	0.295	198.5	0.510
7.	355	18.3	6.5	2.6	1.1540	190.	0.490
8.	260	15.8	5.8	2.3	1.0800	180	0.475
9.	265	14.6	5.0	1.9	1.0400	264.5	0.410
10.	27T	13.5	4.1	1.6	0.9480	151.0	0.400
11.	275	11.7	3.2	1.2	0.8870	133.5	0.325
12.	280	10.0	2.2	1.0	0.7500	112.0	0.285

Observation	Bazins Cof. n	Velocity ft/sec.	Discharge Q cusecs	Rynold no (Re) $X \times 10^5$	Coef. of Discharge C_D	Friction factor f.	% head loss.
	9	10	11	12	13	14	15
1.	0.4606	7.73	0.620	2.59	0.982	0.00402	10.55
2.	0.4586	8.07	0.647	2.72	0.971	0.00544	11.0
3.	0.4595	7.78	0.623	2.62	0.983	0.00447	9.66
4.	0.4536	7.30	0.585	2.46	0.979	0.00516	9.41
5.	0.455	7.14	0.575	2.405	0.985	0.00516	9.75
6.	0.4562	7.00	0.562	2.362	0.984	0.00400	9;68
7.	0.4570	6.12	0.491	2.06	0.983	0.00510	9.96
8.	0.4540	6.06	0.468	2.04	0.984	0.00468	10.5
9.	0.4582	5.64	0.449	1.89	0.985	0.00478	8.77
10.	0.4580	4.83	0.387	1.628	0.986	0.00515	8.10
11.	0.4578	4.51	0.362	1.52	0.981	0.00461	6.90
12.	0.4625	4.08	0.300	1.32	0.989	0.00352	7.6

B = 0.7031 A_1 = 0.0802 sq. ft. A_2 = 0.0431 sqft.

TABLE 3.

OBSERVATIONS FOR EXPERIMENT NO. 3.

Obs. No.	Diff. head in. of water	Press. loss inches of water.	Discharge Q Cusec.	Velocity ft/sec.	Reynold no. $(Re)_D$ x $x\ 10^4$	Coef. of dis-charge	% head loss
1.	2	3	4	5	6	7	8.
1.	27.0	6.0	0.222	1.13	5.7	0.760	21.5
2.	6.3	2.5	0.109	0.58.7	2.8	0.782	38.3
3.	12.8	3.4	0.157	0.802	2.04	0.773	25.0
4.	17.0	4.0	0.184	0.938	4.72	0.772	23.5
5.	22.0	4.7	0.205	1.045	5.26	0.770	21.3

Dimensions of Tank 6'-10" x 3'-7.5"

D = 6" A_1 = 0.196 sq. ft.

β = 0.367

TABLE 4.

OBSERVATIONS FOR EXMPERIMENT NO.4.

Obs. No.	Diff. head inches of water.	Press. loss in.of water	Friction head loss in. of water.	Discharge Q cusecs X x 10^{-3}	Velocity ft/sec.	Reynold no. $(Re)_D$ x $x10^3$	Coef.of discharge C_D.
1	2	3	4	5	6	7	8
1.	2.10	1.6	1.6	4.675	0.295	4.20	1.035
2.	3.55	3.1	3.1	5.840	0.369	5.25	0.98
3.	5.65	5.1	5.4	8.340	0.528	7.52	1.11
4.	8.10	7.8	7.9	11.000	0.626	9.90	1.22
5.	13.70	13.1	13.2	14.400	0.912	13.00	0.53
6.	17.20	16.2	18.0	15.750	0.996	14.20	0.514
7.	22.70	21.8	24.5	19.100	1.210	17.20	0.545

D = 1.5 inch. $A_1 = 1.225 \times 10^{-2}$ sq. ft.

$A_2 = 0.136 \times 10^{-2}$ sq. ft.

$\beta = 0.333$

Area of cross section of measuring tank

= 6.88 sq.ft.

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