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UNIVERSITY OF LOUISVILLE

CONSTRUCTION AND TESTING OF A LABORATORY FLUID FLOW APPARATUS

A Dissertation

Submitted to the Faculty

Of the Graduate School of the University of Louisville

In Partial Fulfillment of the

Requirements for the Degree

Of Master of Science in

Chemical Engineering

Department of Chemical Engineering

By

Richard W. Dehoney

1936

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LIST OF SYMBOLS

Symbol	Quantity	Units
D	Pipe diameter	Ft.
V	Velocity	Ft. per sec.
p	Density	Lb. per cu.ft.
u	Absolute viscosity	LLb. per sec.per ft.
*X	Elevation head	Ft.
g	Acceleration of gravity	Ft. per sec. per sec
P	Pressure	Lb. per sq.ft.
F	Lost Head due to friction	Ft.
f	Friction factor	No units
*w	Work done on or by fluid	Ft. - Lb. per lb.
H	Differential head	Ft. of fluid
C _d	Coefficient of Discharge	No units
*N	Length	Ft.
*W	Weight rate of flow	Lb. per sec
P _f	Drop in pressure due to friction	Lb. per sq.ft.
Re	Reynold's Number	No units
o	Time	Seconds
r	Roughness factor	No units.

(All symbols except those noted * are in accord with the list of symbols adopted by the Standards Committee of the American Institute of Chemical Engineers.)

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CHAPTER I
INTRODUCTION

This investigation consisted in the construction and testing of an apparatus to be used for the laboratory study of the flow of fluids in pipes. The apparatus, illustration I, consists of the arrangement of pipes and instruments to measure flow by three standard methods, i.e., the orifice, the venturi, and the pitot meter. Each flow meter is connected to a pressure measuring instrument called a manometer. The rate of flow in a pipe is determined from the pressure reading on the manometer. Manometers are used to determine the loss of energy caused by the sudden enlargement or contraction of the channel through which the fluid is passing; the loss of energy caused by a combination of friction and the turbulence created when the fluid passes through a globe valve or around a bend; and the loss of energy caused by the friction of the fluid flowing in a long straight pipe.

The chemical engineer is concerned with the transportation of fluids of varying viscosity and density. A study of the effect of viscosity and density on the friction loss in the long pipe was made by employing water and blends of hydrocarbon oils. A light gas oil (Saybolt viscosity of forty-two seconds at twenty-six degrees Centigrade) and a heavy fuel oil (Saybolt viscosity of fifteen-hundred seconds at twenty-six degrees Centigrade) properly blended were used in this study. Water only was used in that section of the apparatus where the flow measuring instruments, the valves and the U-bends were located.

A laboratory fluid flow apparatus was constructed to measure the flow of fluids by the orifice, the venturi, and the pitot meters and to measure the loss of head in valves, U-bends etc., and to study the effects of viscosity and density on the loss of head caused by friction in a long straight pipe.

CHAPTER II
THEORETICAL

FLOW - VISCOUS AND TURBULENT

There are four general forms of Fluid flow; flow in a free jet, flow in an open channel, flow in a closed channel, and flow in a very small diameter tube or pipe which is termed capillary flow. Free jet flow is met with in the case of fluid issuing from the mouth of a nozzle into a gas filled space. Natural streams such as rivers are good illustrations of open channel flow. Flow in closed pipes or channels is met with in the transportation of fluids in pipes and conduits where the fluid is constantly under pressure. Capillary flow is essentially flow in minute closed channels at an extremely low velocity. This type of flow is exemplified by the gradual diffusion of water through a layer of pervious material such as gravel or sand. Flow in pipes only will be considered in this investigation.

Osborne Reynolds (1) in his classic experiments on the steady isothermal flow of a fluid through a long straight pipe showed that there are two different types of flow. These types of flow are known as viscous flow and turbulent flow. Under ideal conditions in viscous or streamline flow each particle of the fluid flowing will be followed by another particle, all moving along a definite path at a definite velocity. In other words the whole stream of the fluid flowing can be divided up into an infinite number of lines of flow, each line flowing parallel to the others. In contrast to this manner of flow is that called turbulent flow. Here the particles of fluid are in a constant state of agitation, surging backwards and forwards and from side to side.

It is rather difficult to tell just exactly where viscous flow stops and where turbulent flow begins. Reynolds developed a dimensionless group of factors, $DV\rho/u$, to be used as a criterion of the type of flow, where D is the pipe diameter in feet, V is the average velocity in feet per second, ρ is the density of the fluid flowing in pounds per cubic foot, and u is the absolute viscosity in English units, ($.000672 \times \text{centipoises}$). This group is known as the Reynolds Number. Experimental evidence indicates that below a Reynolds number of approximately 2100 viscous flow takes place, while above a Reynolds number of approximately 3100 turbulent flow takes place. Between the Reynolds numbers of 2100 and 3100 there lies a "transition zone" where the flow is gradually changing from viscous to turbulent.

In viscous flow, or streamline motion as it is termed by many investigators, the velocity at the wall of the pipe is zero gradually increasing to a maximum at the center of the pipe, the velocity distribution curve being parabolic in character and the ratio of the mean velocity to the maximum velocity at the axis of the pipe being approximately 0.5. For turbulent flow the velocity distribution curve rises more sharply near the wall and is flatter in the central section while the ratio of the mean velocity to the maximum velocity at the axis of the pipe is about 0.8.

V = average velocity

Viscous flow

$$V/V_{max.} = 0.5$$

Turbulent flow

$$V/V_{max.} = 0.8$$

In turbulent flow the velocity at the wall of a pipe is zero, very near the wall of the pipe the flow is streamline in character while as the distance from the wall of the pipe is increased toward the axis the flow changes gradually from a streamline motion to one of turbulence. This fact was very cleverly illustrated by the color-band experiment of Couch and Herrstrom (2) in which a color band was laid down near the wall of a glass tube in which water was flowing and was not disturbed. At the same time another color band introduced at the axis of the pipe was rapidly broken up by the turbulent character of the stream. This experiment indicates the presence of a layer of liquid or buffer zone in which the motion of the fluid is gradually changing from streamline to turbulent. Therefore an average thickness may be assigned to this buffer zone although it is thought that the layer decreases with an increase in velocity of the fluid flowing.

Work by Stanton (3) in 1911 on the radial distribution of gas velocity for the isothermal turbulent flow of air in smooth pipes shows that at a Reynolds number of 39,200 as the ratio of distance from wall to radius is increased the ratio of mean velocity to maximum velocity also increases. Three years later Stanton working with Pannell (4), in 1914 presented data on the ratio of mean velocity to maximum velocity plotted as a function of the Reynolds number. For viscous flow $V/V_{\max}$, (ratio of average velocity to maximum velocity) was found to be 0.5 but between Reynolds numbers of 2000 and 3000 the ratio rises

sharply to 0.726 and gradually higher in the range of turbulent flow.

The effect of roughness is clearly demonstrated by Stanton (3). Using different sizes of pipe which were artificially roughened by cutting a double screw thread in the inner walls, the depth of the threads in both pipes being about the same fraction of the diameter. Plotting $V/V_{\max}$ against ratio of distance from wall to radius of pipe on logarithmic paper he found the slope of the curve to be a constant value of 0.23 in the turbulent range instead of 0.14 which was obtained when smooth pipe was used. Thus the marked roughness of the pipe affected the velocity gradient in the turbulent zone showing that the local velocity increases at a greater rate as the distance from the pipe wall increases, thus increasing the turbulence.

MEASUREMENT OF FLOW IN PIPES BY THE ORIFICE METER

An orifice meter belongs to that class of flow measuring instruments designated as Head Meters in which the motion of the fluid produces between two parts of the apparatus an observable difference in pressure, or differential head, from which the rate of flow may be deduced. An orifice is considered to be a thin plate containing an aperture through which a fluid issues. If this plate is placed in a pipe line it constitutes a constricted portion, or throat, wherein, the mean linear velocity (and hence the velocity head $V^2/2g$) of the fluid must be greater than in the remainder of the channel. According to Bernoulli's theorem, assuming no heat exchange with the surroundings,

the total energy of the fluid must be the same at every cross section. Consequently any increase of the kinetic energy (i.e., the velocity head) at any section is necessarily accompanied by a corresponding decrease in some or all of the other terms in the Bernoulli formula. Thus applying Bernoulli's formula between point A before the throat at the entrance in the main pipe cross section and point B behind the throat at the point of minimum cross section known as the Vena Contracta, which is surrounded by swirling liquid,

$$X_a + V_a^2/2g + P_a/p_a - F + W = X_b + V_b^2/2g + P_b/p_b$$

where X is elevation head in feet, V is the average velocity in feet per second, P is the pressure in the pipe in pounds per square foot, and p is the density of the fluid flowing in pounds per cubic foot. Considering the two points to be at the same level the elevation terms X_a and X_b are equal and disappear. Since no work is done on or by the fluid the work term w in foot pounds per pound of fluid flowing also disappears from the equation. The friction term F in feet of lost head will be treated later by applying a coefficient of discharge which takes into consideration the friction in the orifice and the turbulence created by the sudden contraction in cross section of the stream. Since the density remains constant $p_a = p_b$ and the equation can be rewritten in the form,

$$(V_b^2 - V_a^2) / 2g = (P_a - P_b) / p$$

$$\text{or since } P_a - P_b = \Delta P, \quad \text{and } \Delta P/p = \Delta H$$

Then $(v_b^2 - v_a^2)^{\frac{1}{2}} = (2g\Delta H)^{\frac{1}{2}}$ where ΔH is the differential

head between the two points. Hence it is seen that the measured difference in pressure head at the two points is converted, theoretically, into an increased velocity head.

Now $A_a v_a = A_b v_b$ where A is the cross sectional area at a point in the channel

Or $v_b = A_a v_a / A_b = v_a (d_a/d_b)^2$ where d is the diameter in feet

Substituting this value of v_b in the above equation gives,

$$(v_a((d_a/d_b)^4 - v_a^2))^{\frac{1}{2}} = (2g\Delta H)^{\frac{1}{2}}$$

$$\text{Or } v_a^2((d_a/d_b)^4 - 1)^{\frac{1}{2}} = (2g\Delta H)^{\frac{1}{2}}$$

Multiplying both sides of the equation by the cross sectional area A of the pipe gives,

$$Q = A_a v_a = \frac{(2g\Delta H)^{\frac{1}{2}} A_a}{((d_a/d_b)^4 - 1)^{\frac{1}{2}}} \text{ where } Q \text{ is the theoretical discharge in cubic feet per second.}$$

Actual flow fails to fulfil the assumptions made in the theory; the real discharge differs from the theoretical value. This difference is contributed to by two causes, friction in the orifice and turbulence caused by the sudden contraction at the throat of the orifice. Therefore the right side of the above equation must be multiplied by a coefficient c_v , known as the velocity coefficient to allow for the effects of friction, and a coefficient c_c , known as the coefficient of contraction to allow

for the effects of turbulence. Therefore

$$Q = A_a V_a = \frac{c_v c_c A_b (2g\Delta H)^{\frac{1}{2}}}{[(d_a/d_b)^4 - 1]^{\frac{1}{2}}} = \frac{C_d A_b (2g\Delta H)^{\frac{1}{2}}}{[(d_a/d_b)^4 - 1]^{\frac{1}{2}}}$$

where C_d is known as the Coefficient of Discharge for the orifice meter and is the product of the coefficient of velocity and the coefficient of contraction. For the orifice the coefficient of velocity is approximately 0.98 and the coefficient of contraction is approximately 0.62 giving a value for the Coefficient of Discharge of 0.61. For a particular instrument, this coefficient C_d must be determined experimentally under the conditions in order to secure reasonable accuracy.

MEASUREMENT OF FLOW IN PIPES BY THE VENTURI METER

The venturi meter is another practical application of Bernoulli's theorem to the measurement of fluids flowing in pipes under pressure. It belongs to the same class of flow measuring instruments known as Head Meters as does the orifice meter. The Venturi meter consists simply of two truncated hollow cones inserted in a pipe line having the same internal diameter as the base of the cones. The upstream section is a contracting portion with a vertex angle of from 25 to 30 degrees. The downstream section is a portion of a cone diverging at a total angle of 7 degrees or less. The junction of the two truncated sections is known as the throat of the Venturi. Two pressure taps are located in the Venturi, one at the entrance to the upstream

section where the cross section is the greatest and the other tap at the throat of the meter which is the point of minimum cross section. The purpose of the long slowly widening downstream cone is to make possible a satisfactory reconversion into static pressure head the high velocity head at the throat. So well is this accomplished in a well made tube that the over-all loss in static head amounts to only about one-tenth to one-eighth of the differential reading.

Bernoulli's theorem can be applied between these two pressure taps in exactly the same manner as it was applied in the case of the orifice meter. Thus designating the tap at the entrance to the upstream section as point A and the tap at the throat as point B there results the same equation for the theoretical discharge as when the theorem was applied to the orifice. Thus

$$Q = \frac{A_a (2gH)^{\frac{1}{2}}}{[(d_a/d_b)^4 - 1]^{\frac{1}{2}}} \quad \text{where } Q \text{ is the theoretical discharge}$$

in cubic feet per second. However the actual discharge differs from the theoretical discharge just as was the case with the orifice. The effects of friction are allowed for by using a coefficient of velocity c_v but since there is no sudden contraction in cross section at the throat, instead there being a gradual contraction, the coefficient of contraction is unity. Therefore the Coefficient of Discharge, the product of the coefficient of velocity and the coefficient of contraction, is approximately 0.98 (7). Therefore

$$Q = A_a V_a = \frac{C_d A_a (2gH)^{\frac{1}{2}}}{[(d_a/d_b)^4 - 1]^{\frac{1}{2}}}$$

MEASUREMENT OF FLOW IN PIPES BY THE PITOT TUBE METER

The Pitot tube also belongs to the class of flow measuring instruments known as Head Meters, as do the Orifice and Venturi meters. However the pitotometer differs from the other two meters in that its differential reading is a direct measure of the velocity head of the fluid flowing. Its construction is really very simple and its operation obvious. It consists of a very small tube placed in a pipe line with the opening facing at right angles the direction of flow. The pressure registered on this tube is therefore the static pressure on the fluid in the pipe. Another tube known as an impact tube is inserted bodily within the pipe and has its opening facing squarely upstream. Obviously the pressure registered on this tube is composed of two parts, the static pressure on the fluid plus the pressure due to the impact of the particles of fluid rushing past the opening and colliding with other particles of fluid in the tube. The pressure created by the impact of the moving particles of fluid with other particles in the tube is known as the velocity head ($V^2/2g$) of the flowing fluid. Therefore if these two tubes are connected to a differential manometer, the one tube measuring the static pressure head plus the velocity pressure head, and the other tube measuring the static head only then the difference between them would be the velocity head direct which is the reading produced on the manometer. Writing this in the form of an equation gives;

$$V^2/2g = \Delta H \quad \text{or} \quad V = (2g \Delta H)^{\frac{1}{2}}$$

where V is the velocity in feet per second at the point where the impact tube is placed and ΔH is measured in feet of the fluid flowing. In order to obtain the average velocity over the entire cross section it is necessary to make several velocity explorations at different fractions of the radius of the cross section to find the velocity distribution in the pipe, and the ratio $V/V_{\max}$ is then calculated. Since this ratio varies with the Reynolds number it is necessary to make a plot of this ratio against the corresponding Reynolds number (8). Therefore knowing the Reynolds number and measuring the velocity at the center line of the pipe where it is a maximum the average velocity over the entire cross section follows directly. The average discharge is

$$Q = A V = A (2g\Delta H)^{\frac{1}{2}}$$

where A is the cross sectional area of the pipe and Q is the discharge in cubic feet per second, the area being measured in square feet, and ΔH is measured in feet of the fluid flowing.

ENERGY LOSSES DUE TO FLUID FRICTION INSIDE PIPES

When a fluid is transported under pressure through a long pipe there is a certain loss of mechanical energy which is consumed in overcoming the force of friction just as there is a certain amount of energy consumed in overcoming friction in the operation of any kind of an engine. The extent of this frictional loss of energy in a pipe line is dependant upon a number of factors. An expression can be derived by the use of Dimensional Analysis (9) which will show how the loss due to friction varies as a function of these factors which is known as the Reynolds number, when the type of flow is that known as isothermal turbulent motion.

Such factors as weight M, length L, and time Θ may be selected as the fundamental mechanical quantities. In the table below is given a list of factors and their net dimensions appearing in the following derivation.

Symbol	Quantity	Net Dimensions
D	Diameter	L
g	Acceleration of gravity	L/Θ^2
N	Length	L
r	Roughness of pipe surface	L
V	Velocity	L/Θ
W	Weight rate of flow of fluid	M/Θ
μ	Absolute viscosity	$M/L\Theta$
ρ	Density	M/L^3
ν	Kinematic viscosity	L^2/Θ
Δp	Decrease in intensity of pressure, due to friction expressed as weight per unit area	M/L^2

The type of flow under consideration is that of isothermal turbulent

flow at constant weight rate W through a long pipe of constant diameter D . As is evident from a knowledge of the physical situation involved the controlling factors are Δp_f , g , N , D , p and u . For flow in long pipes, it is reasonable to assume that the force due to friction is directly proportional to the length N of the pipe. Also $g\Delta p_f$ gives the decrease in force per unit of cross-sectional area, since Δp_f has been expressed as weight per unit area. Then letting ϕ be any function involving the remaining factors (V , D , p , u) the equation may be written.

$$(g\Delta p_f)/N = \phi(D) \phi'(V) \phi''(u) \phi'''(p)$$

Replacing the above equation by an equivalent exponential relation

$$(g\Delta p_f)/N = \phi(D^a V^b u^c p^e) \quad (2)$$

wherein the dimensionless exponents a , b , c , and e , and the dimensionless proportionality factor ϕ may have any values required by the situation. Referring to the above table and substituting the dimensions of the various quantities in Equation (2), gives:

$$\frac{(L/\Theta^2)(M/L^2)}{L} = \frac{M}{L^2\Theta^2} = \phi L^a (L/\Theta)^b (M/L\Theta)^c (M/L^3)^e \quad (3)$$

Since equation (3) must be compatible with respect to each dimension the following condition equations may be written for each dimension.

$$\text{Summation } M; \quad 1 = c + e$$

$$\text{" } \Theta; \quad -2 = -b - c$$

$$\text{" } L; \quad -2 = a + b - c - 3e$$

Giving three condition equations and four unknowns. Arbitrarily retain-

ing c since c appears in each equation and solving the condition equations gives;

$$e = 1 - c \quad b = 2 - c \quad \text{and} \quad a = -1 - c$$

Substituting the numerical values of the exponents in Equation (2) gives

$$(g \Delta p_f)/N = \phi \frac{1}{D} \cdot \frac{1}{D^c} \cdot \frac{V^2}{V^c} \cdot u^c \cdot \frac{p}{p^c} = \phi \frac{p V^2}{D} \left(\frac{u}{DV_p} \right)^c \quad (4)$$

Rearranging Equation (4) gives

$$\frac{(g \Delta p_f D)}{N p V^2} = \phi \left(\frac{u}{DV_p} \right)^c \quad (5)$$

Since the exponential relation was employed for convenience, dimensional analysis has shown that

$$\frac{g \Delta p_f D}{N p V^2} = \phi \left(\frac{u}{DV_p} \right) \quad (6)$$

where the function ϕ of the dimensionless group (u/DV_p) remains to be determined experimentally.

If the pipe has an appreciable roughness r , then three dimensionless groups should enter, and the analysis gives

$$\frac{g \Delta p_f D}{N p V^2} = \phi \left(\frac{u}{DV_p} \right) \phi' \left(\frac{r}{D} \right) \quad (7)$$

wherein the functions ϕ and ϕ' remain to be determined from experiments.

Rewriting equation (6) in the form

$$\frac{\Delta p_f}{p} = \frac{N V^2}{g D} \cdot \phi \left(\frac{u}{DV_p} \right) \quad \text{or} \quad F = \frac{N V^2}{g D} \cdot \phi \left(\frac{u}{DV_p} \right) \quad (8)$$

where F is the energy loss caused by the friction of the fluid flowing in the pipe measured in feet of the fluid flowing. At the present time no one has been able to develop a satisfactory mathematical relation expressing the variation of the friction drop with a corresponding variation in the Reynolds number. In practice it has been found convenient to express this friction drop in the following manner;

$$\Delta p_F/p = F = (2 f N V^2)/(gD) \quad (9)$$

where f is called the friction factor, and then make a plot on log-log graph paper of the variation of the friction factor f with the Reynolds number, using f as ordinates with the Reynolds number as abscissas. The form in which equation (9) stands is commonly known as the Fanning equation is the one most commonly used. This formula and the variant of it, known as the Chezy formula $V = C(m s)^{\frac{1}{2}}$ where $C = (2g/f)^{\frac{1}{2}}$ are widely used forms of the theoretical expression derived above. The friction factor f is also found to depend upon the relative roughness of the wetted surfaces as predicted by the theoretical expression. The roughness factor, however, can be measured only by determining the resistance to flow in a duct having the wetted surfaces in question, hence it is impossible to give values of f except for pipes with surfaces which are commercially reproducible.

For isothermal streamline flow in a straight circular pipe, the pressure drop due to friction can be calculated from Poiseuille's law ($\ominus$), which is derived from the definition of absolute viscosity of the fluid, assum-

ing no slip at the wall of the pipe and constant viscosity.

$$F = \frac{P_f}{\bar{p}} = (32 \mu V N)/(g D^2) \quad (10)$$

By equating this formula to the Fanning formula gives the value of f to be used for streamline flow in circular pipes;

$$\frac{32 \mu V N}{g D^2} = \frac{2 f N V^2}{g D} ; \quad f = \frac{16 \mu}{D V \rho} = \frac{16}{\text{Re. no.}} \quad (11)$$

Therefore knowing the Reynolds number it is possible to calculate the value of f directly and use it in the Fanning formula for computing the loss in a straight pipe, when viscous flow exists.

CHAPTER III

APPARATUS

Illustration I

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 CHEMICAL ENGINEERING DE
 LABORATORY FLUID FLOW UN
 BUILT BY RICHARD W. DEHONEY
 DRAWN & TRACED BY R.W. DEHON
 JUNE 30, 1936

ARRANGEMENT OF PIPES IN APPARATUS

The apparatus described in this section was constructed primarily to be used for laboratory instruction on the flow of fluids in pipes. In its design an attempt has been made to include operations illustrating the more important factors that chemical engineers should be familiar with in the transportation of fluids through piping systems. Therefore provision has been made for the following operations;

1. Measurement of volume of flow by three methods; the orifice, venturi, and pitot meters.
2. Measurement of loss of head in a one inch globe valve.
3. Measurement of loss of head in a U-bend formed of two 90° ells separated by a nine and one-half inch spacer.
4. Measurement of loss of head caused by a sudden enlargement in duct cross section.
5. Measurement of loss of head caused by a sudden contraction of the duct cross section.
6. Measurement of loss of head caused by friction in the isothermal flow of fluids of varying viscosity and density in a long straight pipe.

The apparatus consists essentially of two sections, each section being supplied with fluid through a common discharge pipe from an over head tank. The two sections, their component parts and functions will be treated in detail in the following discussion.

Referring to Illustration I it will be seen that Section I is composed of three passes of one inch standard galvanized iron pipe and a fourth pass containing nine feet of two inch standard galvanized iron pipe, the four passes being connected in series. The first three operations listed above are carried out in this first section. Two gate valves are located in this section, one just at the entrance to the section from the discharge pipe and another at the exit from the section just before it joins the discharge pipe. These valves are so placed in order to permit independant operation of each section. In the first pass of this section, beginning at the top of the apparatus, approximately seven feet from the first union, is located the first flow measuring instrument known as the venturi meter. Two pressure lines lead from this meter to the center manometer located on the Manometer Panel which itself is situated at the bottom of the apparatus. This meter is of the same general type described in the previous chapter. It is twelve inches in length with a one-half inch throat at the center. One tap is located directly at the throat and the upstream tap is placed three inches distant from the throat. It was purchased from the Republic Flow Meters Co. where it was drilled and machined from bar steel. This venturi meter was located seven feet away from the nearest point of disturbance to provide a calming length of pipe for the purpose of letting the fluid obtain normal flow after passing through the tee at the entrance to the pass.

The second pass is connected to the first pass by a U-bend consisting

of two 90° ells separated by a $9\frac{1}{2}$ inch spacer of one inch pipe. Two pressure taps are located across this U-bend for the purpose of measuring the loss of head in the bend. The first tap is located in the first pass one foot upstream from the first ell and the second tap is located in the second pass approximately three feet downstream from the second ell. It has been previously determined that most of the loss caused by a U-bend occurs in the pipe downstream from the bend. Therefore it was necessary to place the second tap three feet downstream to be sure of measuring all of the loss occasioned by the U-bend.

The second flow measuring instrument known as the orifice meter is located in the second pass of the first section. This instrument is likewise provided with a calming section prededing it for approximately eight feet for the purpose of letting the fluid obtain normal flow after passing through the U-bend at the beginning of the pass. This instrument consists of a non-corrodible metal disk with a hole drilled it in three-quarters of an inch in diameter bolted between two cast iron flanges. Each flange is equipped with a piezometer ring connected to a $\frac{1}{4}$ inch pressure tap, the taps being located about $\frac{1}{2}$ inch from the center of the metal disk. This meter was purchased from the Builder's Iron Foundry Co. Two pressure lines lead from the taps in this instrument and are connected to the end manometer located on the manometer panel at the bottom of the apparatus.

The third pass of this section contains the nine foot length of two inch

pipe. There is no flow measuring instrument located in this pass at the present time. This length of two inch pipe was placed here so that a pitot tube could be inserted thereby providing the third important method of measuring the volume of flow in pipes that the engineer has at his disposal. The position of the pitot tube is shown on Illustration I approximately where it is to be placed. Two pressure lines lead from this instrument to the first manometer located on the manometer panel.

A one inch globe valve has been placed in the fourth pass of the first section. Approximately seven feet of straight pipe precede this globe valve providing a calming section for the purpose of letting the fluid obtain normal flow after passing through the U-bend at the beginning of the pass. Two pressure taps are placed across this globe valve for the purpose of measuring the loss of head occasioned by the fluid passing through the globe valve. In the case of a gate valve the loss of head is practically negligible when the valve is fully opened but with a globe valve the path followed by the fluid through the valve is that of a miniature S-bend which obviously causes an excessive turbulence to the fluid. The first pressure tap is located two feet upstream from the valve in order to measure all of the loss occasioned by the valve since most of the loss occurs downstream from the valve. According to Corp and Ruble (5) a valve or a U-bend caused disturbed flow in the pipe line downstream for some distance from it and a pressure tap must be located from 25 to 40 diameters downstream from the valve in order

that the true loss occasioned by the valve may be measured.

Into the design of the second section have been incorporated the last three operations named in the above tabulation. This section consists simply of two passes with an overall length of 54 feet - 2 inches. The first pass is made up of 20 foot lengths of standard one inch galvanized iron pipe connected by standard one inch couplings. As in the first section there are two gate valves, one at the entrance to the section and the other just preceding the exit where it rejoins the discharge pipe. Approximately 3 feet - 8 inches from the valve at the entrance to the first pass is located a pressure tap. Connected to this tap is an open manometer which registers the static pressure at that point. Further downstream at a point fifty feet distant from the first manometer which is designated as Manometer A on Illustration I is located another pressure tap which is designated as Manometer B. This manometer also registers the static pressure at this point which is always less than at point A. The difference between the static pressure readings at the points A and B is taken as the loss of head caused by fluid friction.

The second pass of the second section is connected to the first pass by a U-bend formed of two one inch ells separated by a five inch spacer, the total distance between the axes of the two passes being 8 inches. Near the end of the second pass approximately 4 feet - 8 inches from the point where it rejoins the discharge pipe, will be inserted a section of two inch pipe approximately 8 feet long. Two pressure taps will be

placed across the entrance to this section and two more across the end of this section. The first two taps will be connected to the inclined manometer indicated on the manometer panel and will measure the loss of head caused by the sudden expansion in the channel cross section. The second pair of pressure taps will also be connected to the inclined manometer and will measure the loss of head caused by the sudden contraction in the channel cross section. The intended arrangement of this last set up has been indicated on Illustration I.

The manometers used in conducting the first three operations listed above are of the differential type manufactured by the Meriam Company and have a capacity of fifteen inches of mercury differential pressure. The manometers used in measuring the loss of head caused by fluid friction were open manometers. Each manometer registers the static pressure only at each point. They were made in the laboratory by bending a piece of glass tubing and inserting a scale between the two columns of the tubing. The scale was graduated in eighths of an inch, 0 being at the bottom of the manometer. The manometer to be used in measuring the contraction and expansion losses is to be of the inclined type as indicated on Illustration I. The use of this type of manometer is necessitated by the fact that the contraction and expansion losses are so small that it would be impossible to measure them on the upright type of differential manometer.

The overhead tanks were made from ordinary 55 gallon alcohol drums with the tops removed. They were painted on the inside with an asphalt

preparation to prevent rusting. The overflow and discharge outlets . on the tanks were secured by drawing tight two locknuts, one on the outside and one on the inside of the tank and placing rubber gaskets next the tank as a seal.

The pressure tap connections were made by tapping the pipe with a standard one-eighth inch pipe tap and inserting brass connectors. The brass connectors were screwed in until the end was flush with the interior pipe wall. The pressure lines leading to the manometers were of 20 gage one-quarter inch outside diameter copper tubing. Tees were used in connecting the pressure lines to the manometers so that two pairs of taps could be used on one manometer. Shut-off cocks were placed in each pressure line to render independant usage.

The whole apparatus was supported on the side of a brick wall by pipe hangars screwed into 2 inch by 4 inch pine beams which were securely bolted to the wall. All the fittings were standard one and two inch galvanized iron fittings.

CHAPTER IV
PROCEDURE

In the testing of this apparatus the following procedure was followed: Section I was the first section to be tested. There are two gate valves in the discharge line from the main overhead tank as shown on Illustration I. The first gate valve, starting at the top of the apparatus, in the discharge line is closed, and the second one opened. The two valves in the first section as well as the globe valve that is to be tested are opened and the two valves in the second section are closed. Water is then admitted to the overhead tank from the city water supply line until the water level is up to the top overflow. The water supply valve is then adjusted until the water is flowing at a constant rate through the discharge pipe through Section I and also until there is a constant overflow through the two inch overflow pipe. The velocity through the first section is varied by closing the gate valve at the exit of the first section in the fourth pass.

In the operation of Section II the first valve in the discharge line, starting at the top, is opened and the second valve is closed. The two gate valves in the first section are closed and the two valves in the second section are opened. When this is done water will flow down from the discharge pipe and into the second section. The same operation as regards the flow of water through the second section and through the overflow pipe is followed in the operation of this section as in the operation of the first section. The velocity through this section is also varied by closing the valve at the exit of the second pass in the second section. Readings are made at each velocity on the manometers.

In every run made in each section the amount of water flowing through the section in a definite time interval is weighed. This is done by closing the quick opening gate valve on the discharge line and opening the valve emptying into the tank. This renders an accurate determination of the velocity in each run. This is obviously necessary to the calibration of the instruments.

In making the runs with oil it was necessary to employ the Auxiliary Tank indicated in Illustration I. The oil was pumped from the drum up to the auxiliary tank until it was full. Oil was allowed to flow from the auxiliary tank into the main tank until the level of the first overflow was reached. In the runs on oil Section II was the only section used. The same setting of the valves was used as described above. By regulating the rate of the flow from the auxiliary tank into the main tank it was possible to maintain a constant head in the main tank. The oil would be caught and weighed, to determine the velocity, and then pumped back up to the auxiliary tank. To obtain oil mixtures of different viscosities, after each run was completed about ten gallons of heavy fuel oil would be thoroughly stirred into the oil mixture previously used. Oil runs at three different viscosities were made.

CHAPTER V
DATA AND RESULTS

SAMPLE CALCULATIONS

The following sample calculations will serve to illustrate the methods used in compiling the data and computing the results of the runs made in the final testing of the apparatus. The values appearing in these calculations can be located by referring to the Tables from which they were taken as designated.

DISCHARGE

Table I; Vel.# 3.

$$\begin{array}{rclclcl} \text{Lb./min.} & \times & \text{cu. ft./Lb.} & \times & \text{gal./cu.ft.} & = & \text{gal./min.} \\ 108/2 & \times & 1/62.4 & \times & 7.48/1 & = & 6.48 \end{array}$$

VELOCITY

Table I; Vel.# 3.

Internal cross sectional area of a standard 1" pipe = 0.864 sq.in.

$$\text{Gal./min.} \times \text{cu.ft./gal.} \times 1/\text{sq.ft.} \times \text{min./sec.} = \text{Ft./sec.}$$

$$6.48 \times 1/7.48 \times 1/(\text{.864}/144) \times 1/60 = 2.41$$

VENTURI COEFFICIENT OF DISCHARGE

Table I; Vel.# 2

The following formula is used in the calculation of this coefficient

$$\begin{aligned} C_d &= \frac{V_a [(d_a/d_b)^4 - 1]^{\frac{1}{2}}}{(2 g \Delta H)^{\frac{1}{2}}} & \text{where } V_a &= \text{vel. in pipe, ft./sec} \\ & & d_a &= \text{pipe diameter, ft.} \\ & & d_b &= \text{throat " " } \\ & & g &= \text{acc. of gravity,} \\ & & H &= \text{Venturi differential} \\ & & & \text{reading, ft. of fluid.} \\ & & C_d &= \text{coefficient of discharge} \end{aligned}$$

$$\begin{aligned} &= \frac{2.39 [(1.049/0.5)^4 - 1]^{\frac{1}{2}}}{(2 \times 32.2 \times 12.6/12 \times 1.60)^{\frac{1}{2}}} \\ &= 0.987 \end{aligned}$$

ORIFICE COEFFICIENT OF DISCHARGE

Table II, Vel.# 2.

The same formula is used in the calculation of this coefficient as used in the preceding example.

$$\begin{aligned} C_d &= \frac{2.39 [(1.049/0.75)^4 - 1]^{\frac{1}{2}}}{(2 \times 32.2 \times 12.6/12 \times 0.63)^{\frac{1}{2}}} = 0.618 \end{aligned}$$

REYNOLDS NUMBER

Table III; Vel. # 1.

The Reynolds Number is a dimensionless group of factors calculated

$$\text{as: } Re. = (DVp)/u$$

$$Re. =$$

$$\frac{1.049/12 \times 1.04 \times 62.4}{.000672 \times 1.1}$$

$$= 7,540$$

where D = pipe diam., ft.
 V = vel. in pipe, ft./sec.
 p = density, lb./cu.ft.
 u = absolute viscosity =
 .000672 x centipoises
 (vis. of H₂O = 1.1 centi-
 poises at 15° C.)

LOSS OF HEAD IN A ONE INCH GLOBE VALVE

Table III; Vel. # 1.

$$\text{Average loss in valve and pipe} = 0.15'' \text{ Hg.} \times 12.6 = 1.89'' \text{ H}_2\text{O}$$

Loss in pipe between pressure taps: is calculated from the Fanning Equation.

$$F = 2 f L V^2 / g D \quad \text{where - } f = \text{friction factor, no units (Fig. 5)}$$

$$F = \frac{2 \times .0097 \times 5.25 \times 1.04^2}{32.2 \times 1.049/12}$$

$$F = .039'' \text{ H}_2\text{O} \times 12 = .47'' \text{ H}_2\text{O}$$

L = Pipe length, = 5.25 ft.
 V = velocity = 1.04 ft./sec.
 g = 32.2 ft./sec²
 D = pipe diam = 1.049/12 ft.
 F = lost head in feet.

$$\text{Therefore loss of head in valve only} = 1.89 - 0.47 = 1.42'' \text{ H}_2\text{O}$$

LOSS OF HEAD IN A ONE INCH U-BEND

Table IV; Vel. # 1.

$$\text{Average loss in U-bend and pipe} = 0.10'' \text{ Hg.} \times 12.6 = 1.26'' \text{ H}_2\text{O}$$

Loss in pipe between taps is calculated as in the previous example

$$F = 2 f L V^2 / g D$$

$$F = \frac{2 \times .0097 \times 1.04^2 \times 4.79}{32.2 \times 1.049/12}$$

$$= .0358'' \text{ H}_2\text{O} \times 12 = 0.428'' \text{ H}_2\text{O}$$

Re. = 7,540
 f = .0097 (Fig. 5)
 L = length of pipe in
 bend = 4.79 ft.

$$\text{Therefore loss in bend only} = 1.36 - 0.43 = 0.93'' \text{ H}_2\text{O}$$

FRICTION FACTOR f FROM FANNING EQUATION

Table V; Vol.## 1.

The friction factor is calculated from the Fanning Equation

$$f = F g D / 2 N V^2$$

where F = observed pressure drop, ft.
 g = acceleration of gravity, 32.2 ft. per sec. per sec.
 D = Pipe diameter, ft.
 N = pipe length, = 50 feet
 V = velocity, feet.

Average static pressure at A = 10.83 in. Hg.

Average " " " B = 10.74 in. Hg.

Difference = 10.83 - 10.74 = .09 in. Hg. $\times 13.6/12 = F = 1.02$ ft. H_2O

Correction Factor:

In calculating the true differential loss between points A & B in section II it is necessary to apply a correction factor to the observed differential pressure taking into account the fact that the mercury in the two open manometers at these points will not be at the same level, i.e., the mercury in the left column of the manometer where the pressure is applied. When the mercury level in the left column of manometer B is lower than the corresponding level in manometer A there is a greater pressure on B than there would be if both levels were the same. This tends to make the static pressure at B higher than it actually is which in turn tends to make the loss appear lower than it actually is. When this is true, therefore, the correction factor must be added to the observed loss. Thus,

$$\begin{aligned} \text{Average reading on left column of manometer A} &= 1/3 (3.25 + 3.44 + 3.44) \\ &= 3.38 \text{ in. Hg.} \end{aligned}$$

$$\begin{aligned} \text{Average reading on left column of manometer B} &= 1/3 (1.68 + 1.81 + 1.81) \\ &= 1.83 \text{ in. Hg.} \end{aligned}$$

In this run the scales used on the manometers were graduated in one-eighth inch divisions, the readings beginning at the bottom and reading up, both scales being at exactly the same level. Here manometer B has the lower reading which indicates that the pressure in B is greater than it would be if the manometers had the same mercury levels. Therefore the correction factor must be added in this case to the recorded differential.

$$\begin{aligned}\text{Correction Factor} &= 3.38 - 1.83 = 1.55 \text{ in. water} \times 1/13.6 \\ &= 0.118 \text{ in. Hg.}\end{aligned}$$

$$\begin{aligned}\text{Therefore the true loss} &= .09 + .118 = .208 \text{ Hg.} \times 13.6/12 \\ &= 0.236 \text{ ft. of water}\end{aligned}$$

$$\begin{aligned}\text{Therefore the true loss per foot of test length} &= .236/50 \\ &= .00472 \text{ ft. water.}\end{aligned}$$

$$\text{Therefore } f = \frac{.00472 \times 32.2 \times 1.049/12}{2 \times 1.04 \times 1.04} = .00615$$

However in the runs on oil a different type of scale was used, graduated in inches and tenths of inches, the readings beginning at 0 in the center of the scale and going from 0 to 7.5 at the bottom and from 0 to 7.5 at the top. It is obvious that the manometer having the greater depth in the left column is that one having the greater left column reading from 0 down. On the contrary when the left column reading at A is lower than the left column reading at B the correction factor must be deducted from the recorded loss to obtain the true loss.

AN ALTERNATE METHOD OF COMPUTING THE TRUE LOSS BETWEEN A AND B

Static pressure at A = $10.83 \text{ "Hg.} \times 13.6/12 = 12.29 \text{ ft. H}_2\text{O}$

Static pressure at B = $10.74 \text{ "Hg.} \times 13.6/12 = 12.18 \text{ ft. H}_2\text{O}$

Correcting these pressures to what they would be at the center line of the pipe gives;

Center Line of the pipe at 10.50 inches on scale.

Average reading on left column of Manometer A = 3.38

Water pressure on Manometer A below center line of pipe

$$= (10.50 - 3.38)/12 = 0.58 \text{ ft. H}_2\text{O}$$

Therefore true pressure at A = $12.29 - 0.58 = 11.71 \text{ ft. H}_2\text{O}$

Average reading on left column of Manometer B = 1.83

Water pressure on Manometer B below center line of pipe =

$$(10.50 - 1.83)/12 = 0.71 \text{ ft. H}_2\text{O}$$

True pressure at B at center line of pipe = $12.18 - 0.71 = 11.47 \text{ ft. H}_2\text{O}$

Therefore the True Loss between A & B = $11.71 - 11.47 = 0.24 \text{ ft. H}_2\text{O}$

VISCOSITY

The following will illustrate the conversion of viscosity in Saybolt

Seconds to Centipoises. The formula used is

$$u = p (.22 \times \theta - 180/\theta)$$

$$= .835 (.22 \times 42 - 180/42) = 4.13$$

Therefore viscosity in centipoises
= $4.13 \times 1.01 = 4.18$

where u = viscosity relative to H_2O at $60^\circ\text{F.} = 1.01^2$ centipoises
 θ = Saybolt viscosity in seconds consumed in collecting 60 cc. of oil through the Saybolt Standard Universal Viscosimeter = 42 seconds.
 p = specific gravity of the oil = 0.835

Table I gives the results of a calibration test of the venturi meter in which runs were made at eight different velocities. The average coefficient for discharge for this meter is, omitting the first run as unusually low, 0.962. The generally accepted value for this type of meter is 0.98 (7). Figure I is a Calibration Curve for this meter in which the discharge in gallons per minute is plotted against the differential reading in inches of mercury as abscissas.

Table II gives the results of a similar test on the orifice meter involving the same number of runs. The first reading on this meter was also unusually low, so the average coefficient of discharge for the last seven runs was 0.612, the generally accepted value being 0.61 (7). Figure 2 is also a Calibration Curve for the orifice meter plotted to the same coordinates as Figure I.

TABLE I

CALIBRATION OF VENTURI METER

Vel. No.	Read- ing No.	Venturi Lt. (in. of Hg.)	Reading Rt. - Dif. ("Hg.)	Ave. Dis- charge (Gal/min)	Ave. Dis- charge (Ft/sec)	Vel- ocity (Ft/sec)	Ave. Vel- ocity (Ft/sec)	Coef. of Dis- charge
1	1	.20	-.20	.40	.36	2.58	.959	
1	2	.15	-.15	.30		2.92	1.08	
1	3	.20	-.20	.40	.36	2.92	2.70	1.08
							1.04	.900
2	1	.75	-.80	1.55		6.43	2.39	
2	2	.80	-.85	1.65	1.60	6.43	6.43	2.39
							2.39	.987
3	1	.85	-.90	1.75		6.48	2.41	
3	2	.85	-.90	1.75		6.48	2.41	
3	3	.85	-.90	1.75	1.75	6.42	6.46	2.38
							2.40	.949
4	1	1.00	-1.05	2.05		7.00	2.60	
4	2	1.00	-1.00	2.00	2.03	7.00	7.00	2.60
							2.60	.954
5	1	1.40	-1.45	2.85		8.11	3.02	
5	2	1.35	-1.40	2.75		8.20	3.05	
5	3	1.35	-1.40	2.75	2.78	8.15	8.15	3.03
							3.03	.949
6	1	1.600	-1.60	3.20		9.00	3.34	
6	2	1.65	-1.60	3.25	3.23	9.04	9.02	3.36
							3.35	.975
7	1	2.10	-2.15	4.25		10.15	3.77	
7	2	2.05	-2.15	4.20		10.05	3.74	
7	3	2.10	-2.15	4.25	4.23	10.05	10.08	3.74
							3.75	.952
8	1	2.85	-2.95	5.80		12.40	4.61	
8	2	3.20	-3.25	6.45		12.55	4.66	
8	3	3.20	-3.25	6.45	6.24	12.40	12.45	4.61
							4.63	.967

Average coefficient of Discharge = .962

Generally accepted value of Coefficient of Discharge for the Venturi Meter = 0.98 (7).

VENTURI METER
CALIBRATION CURVE

FIG. 1

TABLE II
CALIBRATION OF ORIFICE METER

Vel- ocity No.	Read- ing No.	Orifice Reading (in. of Hg.)			Ave. Dis- Dif. charge		Ave. Dis- charge	Ave. Vel- ocity	Ave. Vel.	Coefficient of Discharge	
		Lt.	Rt.	Dif.		Gal/min.		Ft/sec.			
1	1	.05	-.10	.15		2.58		.96			
	2	.10	-.10	.20		2.92		1.08			
	3	.10	-.10	.20	.18	2.92	2.70	1.08	1.04	0.50	
2	1	.30	-.35	.65		6.43		2.39			
	2	.30	-.30	.60	.63	6.43	6.43	2.39	2.39	0.618	
3	1	.30	-.35	.65		6.48		2.41			
	2	.30	-.35	.65		6.48		2.41			
	3	.35	-.35	.70	.67	6.42	6.46	2.38	2.40	0.603	
4	1	.35	-.40	.75		7.00		2.60			
	2	.35	-.40	.75	.75	7.00	7.00	2.60	2.60	0.617	
5	1	.50	-.55	1.05		8.11		3.02			
	2	.50	-.55	1.05		8.20		3.05			
	3	.50	-.55	1.05	1.05	8.15	8.15	3.03	3.03	0.606	
6	1	.60	-.65	1.25		9.00		3.34			
	2	.65	-.65	1.30	1.28	9.04	9.02	3.36	3.35	0.608	
7	1	.75	-.80	1.55		10.15		3.77			
	2	.75	-.75	1.50		10.05		3.74			
	3	.75	-.80	1.55	1.53	10.05	10.08	3.74	3.75	0.622	
8	1	1.10	-1.15	2.25		12.40		4.61			
	2	1.20	-1.25	2.45		12.55		4.66			
	3	1.15	-1.20	2.35	2.35	12.40	12.45	4.61	4.63	0.620	

Average Coefficient of Discharge = 0.614

The generally accepted value for the Coefficient of Discharge for the Orifice Meter is 0.61 (7).

FIG. 2

Table III gives the results of a test on the loss of head in a one inch globe valve. In this test the loss registered on the differential manometer was the sum of the loss of head in the globe valve and the loss of head due to the pipe between the pressure taps. Hence to obtain the loss due to the valve only it was necessary to deduct the loss due to the friction in the pipe between the taps. The loss in the pipe was calculated by taking a value of the Reynolds number from and getting the corresponding value of the friction factor from Figure 5, and calculating the loss in the pipe from Fannings Equation. Figure 3 is a curve showing how the loss of head in the globe valve varies with the fluid velocity. Data obtained by Corp and Ruble (5) in a similar test is plotted on the same curve.

Table IV gives the results of a test on the loss of head in a U-Bend formed from two one inch ells separated by a nine and one-half inch spacer. The loss in the U-Bend alone was obtained by deducting the loss due to friction in the pipe between pressure taps from the total registered loss. The loss in the pipe was calculated as in the case with the globe valve. Figure 4 is a curve showing how the loss of head in a U-Bend varies with the fluid velocity.

TABLE III

LOSS OF HEAD IN A ONE INCH GLOBE VALVE

Vel- ocity No.	Read- ing No.	Loss in Pipe & Valve Lt. - Rt. - Dif.	Ave. Dif.	Re. No.	Fric- Factor	Pipe Fric. ("H ₂ O)	Valve Loss ("H ₂ O)	Vel- ocity Ft/sec	Ave. Vel.
1	1	-.05 .10 .15	.15	7540	.0097	.47		.96	
	2	-.05 .10 .15						1.08	
	3	-.05 .10 .15					1.42	1.08	1.04
2	1	-.30 .40 .70						2.39	
	2	-.30 .35 .65	.68	17300	.0085	2.17	6.40	2.39	2.39
3	1	-.35 .35 .70						2.41	
	2	-.30 .35 .65						2.41	
	3	-.30 .35 .65	.68	17400	.0085	2.19	6.42	2.38	2.40
4	1	-.40 .45 .85						2.60	
	2	-.35 .45 .80	.83	18830	.0084	2.54	7.91	2.60	2.60
5	1	-.55 .60 1.15						3.02	
	2	-.55 .60 1.15						3.05	
	3	-.55 .60 1.15	1.15	21930	.0083	3.42	11.08	3.03	3.03
6	1	-.65 .70 1.35						3.34	
	2	-.65 .70 1.35	1.35	24250	.0083	4.08	12.94	3.36	3.35
7	1	-.80 .90 1.70						3.77	
	2	-.80 .85 1.65						3.74	
	3	-.80 .90 1.70	1.68	27200	.0083	5.24	15.95	3.74	3.75
8	1	-1.20 1.25 2.45						4.61	
	2	-1.25 1.30 2.55						4.66	
	3	-1.25 1.30 2.55	2.52	33500	.0081	7.79	24.00	4.61	4.63

FIG. 3

TABLE IV

LOSS OF HEAD IN A ONE INCH U-BEND WITH A NINE & ONE-HALF INCH SPACER

Vel- ocity No.	Read- ing No.	Loss in Pipe & U-bend (¹ / ₂ in. - Rt. - Dif.	Ave. Dif.	Vel- ocity Ft/sec	Ave. Vel.	Rey- nolds No.	Fric- tion Factor	Pipe Fric. (¹ / ₂ in. - Rt. - Dif.)	Loss in U-bend (¹ / ₂ in. - Rt. - Dif.)
1	1	-.05	.05	.10	.96				
	2	.00	.10	.10	1.08				
	3	.00	.10	.10	1.05	1.04	7540	.0097	0.428
2	1	-.15	.20	.35	2.39				
	2	-.15	.20	.35	2.39	2.39	17300	.0085	1.98
3	1	-.21	.20	.41	2.41				
	2	-.15	.20	.35	2.41				
	3	-.15	.20	.35	2.38	2.40	17400	.0085	2.00
4	1	-.15	.25	.40	2.60				
	2	-.20	.25	.45	2.60	2.60	18830	.0083	2.32
5	1	-.25	.30	.55	3.02				
	2	-.25	.30	.55	3.05				
	3	-.25	.30	.55	3.03	3.03	21930	.0083	3.11
6	1	-.30	.40	.70	3.34				
	2	-.30	.35	.65	3.36	3.35	24250	.0083	3.75
7	1	-.40	.45	.85	3.77				
	2	-.40	.45	.85	3.74				
	3	-.40	.45	.85	3.74	3.75	27200	.0083	4.77
8	1	-.60	.65	1.25	4.61				
	2	-.60	.65	1.25	4.66				
	3	-.60	.70	1.20	4.61	4.63	33500	.0081	7.01

FIG. 4

Table V gives the results of a test on the loss of head due to friction in a long straight pipe in which water was the fluid used. Tables VI, VII, and VIII give the results of similar tests in which blends of two hydrocarbon oils were used as the fluid, each blend being of a different viscosity and a different density. Figure 5 is a curve showing how the friction factor as calculated from the Fanning equation varies with the Reynolds number. In the range of Reynolds from 10,000 to 35,000 the values of f obtained from these tests are slightly higher than those recommended by Drew, Koo and McAdams (6).

TABLE V

LOSS OF HEAD DUE TO FRICTION IN A LONG PIPE

Water Data

Velocity No.	Reading No.	Pressure at Manometer A (in. Hg.)			Pressure at Manometer B (in. Hg.)			Discharge Gallons per min.	Average Discharge Gal/min.
		Lt.	Rt.	Diff.	Lt.	Rt.	Diff.		
1	1	3.25	14.25	11.00	1.68	12.63	10.94	2.58	2.70
	2	3.44	14.19	10.75	1.81	12.50	10.69	2.92	
	3	3.44	14.19	10.75	1.81	12.44	10.63	2.92	
2	1	3.06	14.50	11.44	1.63	12.63	11.00	2.92	3.69
	2	3.06	14.50	11.44	1.63	12.63	11.00	3.65	
3	1	3.25	14.25	11.00	2.13	12.19	10.06	5.48	5.51
	2	3.25	14.15	11.00	2.13	12.19	10.06	5.54	
4	1	3.25	14.31	11.06	3.48	12.25	9.88	5.57	5.69
	2	3.31	14.19	10.88	2.38	12.19	9.81	5.72	
	3	3.31	14.19	10.88	2.44	12.19	9.75	5.78	
5	1	5.13	12.50	7.38	4.19	10.00	5.81	6.43	6.43
	2	5.13	12.50	7.38	4.19	10.06	5.87	6.43	
6	1	3.81	10.44	6.63	4.75	9.50	4.75	7.00	7.00
	2	3.81	10.44	6.63	4.75	9.50	4.75	7.00	
7	1	4.63	9.50	4.87	6.00	8.25	2.25	8.11	8.15
	2	4.56	9.56	5.00	6.00	8.25	2.25	8.20	
	3	4.69	9.44	4.75	6.00	8.25	2.25	8.15	
8	1	5.38	8.69	3.31	6.94	7.25	.31	9.00	9.02
	2	5.38	8.75	3.37	7.06	7.31	.25	9.04	
9	1	3.94	13.63	9.69	3.88	10.69	6.81	9.08	9.29
	2	3.94	13.56	9.62	4.06	10.56	6.50	9.44	
	3	3.94	13.56	9.62	4.06	10.63	6.57	9.28	
10	1	4.19	13.25	9.06	4.94	9.75	4.81	10.81	10.79
	2	4.19	13.31	9.12	4.88	9.75	4.87	10.76	
11	1	4.56	13.00	8.44	5.50	9.06	3.56	11.74	11.76
	2	4.56	13.00	8.44	5.50	9.06	3.56	11.77	

TABLE V (Continued)

LOSS OF HEAD DUE TO FRICTION IN A LONG PIPE

Water Data

Vel- ocity No.	Ave. Vel. Ft./sec.	Ave. Dif. on Man. A (in.Hg.)	Ave. Dif.on Man. B (in.Hg)	Loss Bet. A & B (in.Hg)	Correc- tion Factor (in. Hg)	Ture Loss per Ft. of pipe (Ft. H ₂ O)	Fric- tion Fac- tor f.	Rey- nolds No. Re.
1	1.04	10.83	10.74	0.09	0.118	.0047	.00615	7530
2	1.38	11.44	11.00	0.44	0.106	.01236	.00914	10000
3	2.05	11.00	10.06	0.94	0.083	.0232	.00776	14,800
4	2.14	10.95	9.82	1.13	0.066	.0272	.00834	15,500
5	2.39	7.38	5.84	1.54	0.069	.0365	.0090	17,380
6	2.60	6.63	4.75	1.88	-.069	.0410	.00853	18,820
7	3.03	4.88	2.25	2.63	-.100	.0572	.00878	21,920
8	3.35	3.34	0.28	3.06	-.119	.0666	.00839	24,250
9	3.45	9.64	6.62	3.02	-.005	.0684	.00805	25,000
10	4.00	9.09	4.84	4.25	-.053	.0950	.00836	29,990
11	4.37	8.44	3.56	4.88	-.069	.1129	.00832	31,600

TABLE VI

LOSS OF HEAD DUE TO FRICTION IN A LONG PIPE

Oil Data - Pure Gas Oil

Viscosity = 42 Saybolt Seconds (26.°C.)
 Specific Gravity = 0.8355

Vel- ocity No.	Read- ing No.	Pressure at Manometer A (in. of Hg.)			Pressure at Manometer B (in. of Hg.)			Discharge Gallons per min- ute	Average Discharge Gallons per min.
		Lt.	Rt.	Diff.	Lt.	Rt.	Diff.		
1	1	-5.50	2.80	8.30	-5.90	2.15	7.05	2.40	
	2	-5.50	2.80	8.30	-5.90	2.15	7.05	2.58	2.50
2	1	-5.20	2.50	7.70	-5.00	1.25	6.25	5.74	
	2	-5.20	2.50	7.70	-5.00	1.25	6.25	5.88	5.81
3	1	-4.80	2.10	6.90	-3.95	0.15	4.10	8.33	
	2	-4.80	2.10	6.90	-3.90	0.10	4.00	8.60	8.47
4	1	-4.70	2.00	6.70	-3.70	-.10	3.60	9.04	
	2	-4.70	2.00	6.70	-3.65	-.10	3.55	8.90	8.97
5	1	-4.45	1.75	6.20	-2.95	-.75	2.20	10.05	
	2	-4.45	1.75	6.20	-2.95	-.75	2.20	10.05	10.05

Vel- ocity No.	Ave- Vel. Ft/sec	Ave. Diff. on Man. A. (in.Hg.)	Ave. Diff. on Man. B. (in.Hg.)	Loss Bet. A & B (in.Hg.)	Correc- tion Factor (ft.oil)	True Loss per Ft. pipe. (ft. Oil)	Fric- tion Factor f	Rey- nolds Number Re.
1	0.93	8.30	7.05	1.25	.033	.0345	.0106	1,515
2	2.15	7.70	6.25	1.45	-.017	.0390	.01186	3,490
3	3.15	6.90	4.10	2.80	-.072	.0744	.01050	5,120
4	3.33	6.70	3.58	3.12	-.085	.0829	.01054	5,410
5	3.73	6.20	2.20	4.00	-.125	.1059	.01072	6,060

TABLE VII

LOSS OF HEAD DUE TO FRICTION IN A LONG PIPE

Oil Data

. Blended Gas Oil & Fuel Oil

Viscosity = 58 Saybolt Seconds (26 °C)
Specific Gravity = 0.8555

V	Vel- ocity No.	Read- ing No.	Pressure at Manometer A (in. of Hg.) Lt. - Rt. - Diff.			Pressure at Manometer B (in. of Hg.) Lt. - Rt. - Diff.			Discharge Gallons per minute	Average Discharge Gallons per min.
1	1		-5.70	3.00	8.70	-1.85	6.20	7.05	1.12	
	2		-5.70	3.00	8.70	-1.85	6.20	7.05	1.33	1.24
2	1		-5.40	2.80	8.20	-1.10	5.35	6.45	5.44	
	2		-5.40	2.75	8.15	-1.10	5.35	6.45	4.67	
	3		-5.40	2.75	8.15	-1.05	5.35	6.40	5.04	5.05
3	1		-5.00	2.25	7.25	-0.15	4.20	4.35	7.80	
	2		-5.00	2.25	7.25	-0.15	4.15	4.30	7.95	7.88
4	1		-4.70	2.00	6.70	-2.90	0.00	2.90	8.80	
	2		-4.70	1.95	6.65	-2.90	0.00	2.90	8.65	8.74
5	1		-4.65	1.90	6.65	-2.60	-.30	2.30	9.36	
	2		-4.65	1.85	6.45	-2.60	-.30	2.30	9.50	9.43

Vel- ocity No.	Ave - Vel. Ft/sec	Ave. Dif. on Man. A (in.Hg.)	Ave. Dif. on Man. B (in.Hg.)	Loss Bet. A & B (in.Hg.)	Cor. Fac. (ft. of oil)	True Loss per ft. pipe (ft.oil)	Fric- tion factor f	Rey- nolds Number Re.
1	0.42	8.70	7.05	1.65	-.321	.0372	.0458	349
2	1.88	8.16	6.43	1.73	-.360	.0386	.0153	1,569
3	2.93	7.25	4.33	2.92	-.429	.0694	.0114	2,442
4	3.24	6.68	2.90	3.78	-.150	.0971	.0130	2,705
5	3.50	6.55	2.30	4.25	-.136	.1098	.0126	2,920

TABLE VIII

LOSS OF HEAD DUE TO FRICTION IN A LONG PIPE

Oil Data

Blended Gas Oil & Fuel Oil

Viscosity = 83 Saybolt Seconds (26 °C.)

Specific Gravity = 0.8683

Velocity Number	Reading Number	Pressure at Manometer A (in. of Hg.)			Pressure at Manometer B (in. of Hg.)			Dis- charge Gallons per min.	Average Dis- charge Gal/min.
		Lt. -	Rt. -	Diff.	Lt. -	Rt. -	Diff.		
1	1	-3.65	4.50	8.15	-3.70	3.05	6.75	5.36	
	2	-3.65	4.50	8.15	-3.70	3.00	6.70	4.14	4.75
2	1	-3.35	4.15	7.50	-3.04	2.40	5.44	6.62	
	2	-3.35	4.20	7.55	-3.00	2.35	5.35	5.86	6.24
3	1	-1.65	5.45	7.10	-2.55	1.85	4.40	7.30	
	2	-1.65	5.40	7.05	-2.55	1.85	4.40	6.55	6.93
4	1	-1.60	5.30	6.90	-2.35	1.65	4.00	7.03	
	2	-1.55	5.30	6.85	-2.35	1.65	4.00	7.80	7.42
5	1	-1.70	5.10	6.80	-2.38	2.22	4.60	7.24	
	2	-1.70	5.10	6.80	-2.38	2.22	4.60	7.24	7.24
6	1	-1.15	4.85	6.00	-1.30	1.20	2.50	7.36	
	2	-1.15	4.85	6.00	-1.25	1.21	2.46	8.28	7.82

Velocity No.	Ave. Vel. Ft/sec.	Ave. Diff. on Man. A (in.Hg.)	Ave. Diff. on Man. B (in.Hg.)	Loss Bet A & B (in.Hg.)	Cor. Fac. (Ft. oil)	True Loss per Ft. pipe (Ft.Oil)	Fric- factor f	Rey- nolds Number Re.
1	1.77	8.15	6.72	1.43	0.00	.0373	.0154	870
2	2.32	7.52	5.40	2.13	-.0275	.0549	.0143	1,140
3	2.57	7.08	4.40	2.68	.075	.0715	.0153	1,264
4	2.75	6.88	4.00	2.88	.063	.0765	.0143	1,350
5	2.69	6.80	4.60	2.20	.057	.0585	.0114	1,321
6	2.91	6.00	2.48	3.52	.011	.0923	.0153	1,430

FANNING FRICTION FACTORS FOR A LONG STRAIGHT PIPE

CHAPTER VI

CONCLUSIONS

A laboratory fluid flow apparatus was constructed which allowed the following measurements on fluids of varying densities and viscosities

1. Measurement of flow in pipes by the venturi meter.
2. Measurement of flow in pipes by the orifice meter.
3. Measurement of flow in pipes by the pitot meter.
4. Measurement of loss of head in a globe valve.
5. Measurement of loss of head in a U-bend.
6. Measurement of loss in head caused by a sudden enlargement.
7. Measurement of loss of head caused by a sudden contraction.
8. Measurement of loss of head caused by friction in a long pipe and the determination of the variation of the friction factor with the Reynolds Number.

Calibration curves of the discharge rate versus the differential manometer reading were prepared for the venturi and orifice meters. The coefficients of discharge of 0.96 for the venturi meter and 0.61 for the orifice compare favorably with the accepted values of 0.98 for the venturi meter and 0.61 for the orifice meter (7).

A curve showing how the loss of head in a globe valve varies with the velocity checks exactly the results of Corp & Ruble (5) in a similar experiment.

A curve was prepared showing how the loss of head in a U-bend formed of two ninety degree ellis separated by a nine and one-half inch spacer varies with the velocity.

A curve showing the variation of the Fanning friction factors for a long straight pipe with variations in the Reynolds number was prepared within the range of Reynolds numbers from 1,000 to 32,000. In the turbulent range from 3,000 to 10,000 the curve coincides with the curve recommended by Drew, Koo, & McAdams (6) while from 10,000 to 32,000 the curves is approximately from 0 to 20% higher than the other curve. This indicates that in the small sizes of galvanized iron pipe the friction factors fall into the category of rough pipes.

CHAPTER VII

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