DESIGN OF APPARATUS TO DETERMINE REYNOLDS' NUMBER OF VARIOUS LUBRICATING OILS UNDER VARYING CONDITIONS

by

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Fulfillment of the Requirements
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MASTER OF SCIENCE
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I. INTRODUCTION

For several years it has been the desire of the personnel of the Virginia Polytechnic Institute Mechanical Engineering Department to include among its hydraulic experiments an apparatus to determine the Reynolds' number of various liquids under varying conditions. It was generally believed that such an apparatus would greatly enhance the Hydraulics Laboratory by providing the students with a deeper understanding of the flow characteristics of liquids.

Before embarking upon the design of the apparatus it is necessary to investigate the behavior of liquids flowing through pipes. Flow through enclosed channels may follow either the pattern of laminar flow if the velocity is low or turbulent flow if the velocity is high. Laminar flow may be best visualized by thinking of the liquid as being made up of many concentric cylinders sliding each within the other. The outermost layer tends to adhere to the pipe walls and for purely theoretical purposes may be considered stationary. The innermost cylinder, however, moves at a maximum velocity and all intervening layers at a velocity proportional to their distance from the center cylinder. In the case of turbulent flow the orderly formation is broken, and many eddies are present in the pipe.

A very graphic picture of the phenomena of laminar and turbulent flow was presented in 1883 by Professor Osborne Reynolds in his classic experiment. Professor Reynolds injected a small stream of colored liquid axially into a glass tube through which water was flowing, and noticed that at low velocities the colored liquid was visible as a line, while at high velocities the particles of the colored liquid followed no regular path and soon became thoroughly mixed with the water. He also noticed that at a certain velocity the thread of colored liquid suddenly broke, thus showing an abrupt change from laminar to turbulent flow. This he termed the critical velocity. Further experimentation, however, showed that the real criterion of the critical point was a function not of the velocity alone, but more correctly a function of the ratio of the inertia forces and friction forces acting upon the liquid.

The ratio may be stated mathematically in the following manner:

A proportion representing the inertia forces may be obtained from Newton's second law,

$$F = k M A$$

By making the constant (k) equal unity the equation becomes

$$F = M A \qquad \text{- - - - - - - - - - -} \qquad (1)$$

where F = force in lbs.
M = mass in slugs
A = acceleration in ft. per sec. per sec.

The equation may also be expressed as

$$F = \frac{M(V - V_1)}{T} \qquad \text{- - - - - - - - - - -} \qquad (2)$$

where V = final velocity in ft. per sec.
V_1 = initial velocity in ft. per sec.
T = time in seconds

If $V_1 = 0$, then

$$F = \frac{MV}{T} \qquad \text{- - - - - - - - - - -} \qquad (3)$$

However, M may be expressed as the volume of a substance times its density. Therefore, $M = \rho L^3$ where L = any linear dimension in ft. and ρ = the density in slugs per cubic foot.

Therefore, $$F = \frac{\rho L^3 V_a}{T} \qquad \text{- - - - - - - - - - -} \qquad (4)$$

where V_a = average velocity in pipe.

In the case of a liquid standing in a pipe the volume of the liquid is equal to the cross-sectional area of the pipe times the length of the pipe.

$$\text{Vol.} = \frac{\pi D^2 L}{4} \quad \text{-------} \quad (5)$$

In the above expression it is seen that two of the linear dimensions involved in determining volume enter the expression as the diameter of the pipe (D) while the other is the length of the pipe (L).

Since the terms π and 4 are constants they may be dropped and the expression becomes:

$$\text{Vol.} \propto D^2 L \quad \text{----------} \quad (6)$$

and by substitution in equation(4)

$$F \propto \frac{\rho D^2 L V_a}{T} \quad \text{----------} \quad (7)$$

By letting $V = \frac{L}{T}$ it can be said that

$$F \propto \rho D^2 V_a^2 \quad \text{----------} \quad (8)$$

An expression for the viscous forces involved may be derived from the basic laws of fluid friction.

Referring to Fig. 1, the force (F) required to move one flat plate separated from another by a layer of a fluid may be expressed by the equation

$$F = \mu \frac{AV}{h} \quad \text{----------} \quad (9)$$

where F = force to overcome fluid friction in lbs.

A = area of surface of plate in contact with fluid in sq. ft.

V = velocity of plate of area (A) in ft. per sec.

μ = absolute viscosity of fluid in lb-sec. per sq. ft.

h = thickness of fluid layer in ft.

The flat plates mentioned in the above analysis may also be considered as a given layer of the fluid under investigation.

In the case of a fluid flowing through a pipe of circular cross-section (A) becomes the area of a cylindrical surface concentric to the pipe wall (Fig. 2) and (h) becomes the distance from the pipe wall to the surface (A). The force (F) is the force required to move the cylindrical surface through the fluid in the pipe. The radius of the surface (A) is (y) and the length may be called (L). Therefore, it is possible to express the area (A) as

$$A = 2\pi y L \quad \text{-----------} \quad (10)$$

and substituting into equation (9) the expression becomes

$$F = \frac{2\pi \mu y L V}{h} \quad \text{---------} \quad (11)$$

The velocity (V) of the surface in question may be represented by the vector a b (Fig. 3) and the velocity (Vm) of the fluid at the center of the pipe may be represented by the vector x y. The velocity (V) may be expressed in terms of (Vm) by the following proportion:

$$\frac{V}{h} = \frac{V_m}{R} \quad \text{-----------} \quad (12)$$

where R = the radius of the pipe

$$V = \frac{V_m \times h}{R} \quad \text{-----------} \quad (13)$$

Substituting in equation (11) we have

$$F = \frac{2\pi\mu y L \; V_m \times h}{R \times h}$$

$$= \frac{2\pi\mu y L \; V_m}{R} \quad \text{-----------} \; (14)$$

Referring to Fig. 4, dW is the total force acting on a cross-sectional differential strip (dy) (perpendicular to axis of pipe) of the fluid in the pipe.

$$d\,W = F\,dy \quad \text{-----------------} \; (15)$$

$$W = \int F\,dy \quad \text{-----------------} \; (16)$$

Inserting equation (14) for F, equation (16) becomes

$$W = \int \frac{2\pi\mu y L \, V_m}{R} \cdot dy \quad \text{---------} \; (17)$$

$$W = \frac{2\pi\mu L \, V_m}{R} \int_0^R y\,d\,y$$

$$= \frac{2\pi\mu L \, V_m}{R} \cdot \frac{y^2}{2} \Big]_0^R$$

Substituting the limits of R and zero,

$$W = \frac{2\pi\mu L \, V_m}{R} \cdot \frac{R^2}{2}$$

Since the diameter of the pipe (D) = 2R

$$W = \frac{\pi\mu L \, V_m \, D}{2}$$

The term (L) may be considered constant for all velocities, diameters, and viscosities, and may therefore be dropped. Likewise $\frac{\pi}{2}$ may be eliminated. Since the velocity determined by measuring the discharge will be the average velocity and is equal to one half the maximum velocity the expression becomes

$$W \propto \mu V_a D$$

where V_a = average velocity in pipe.

5-A

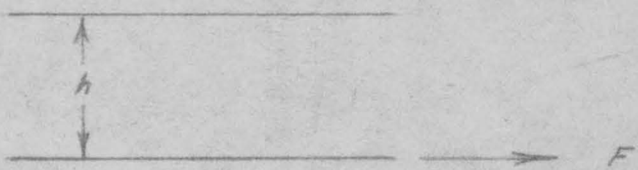


FIG. 1

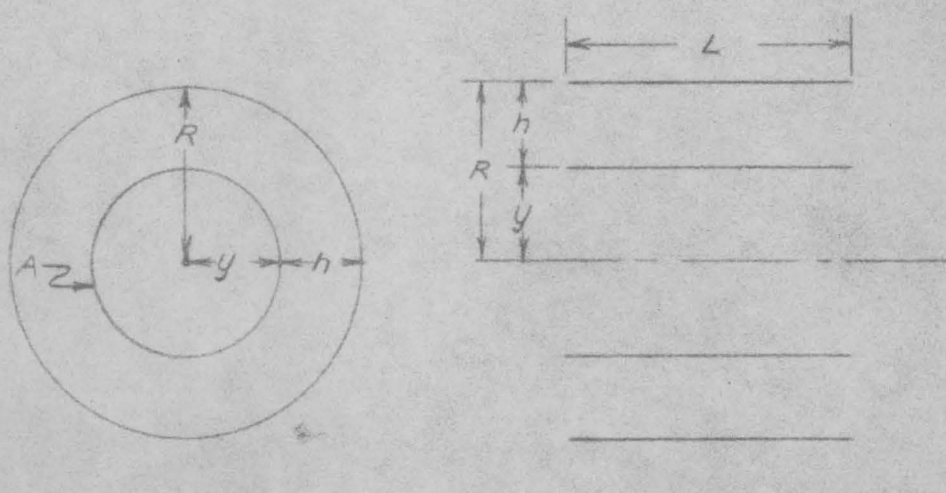


FIG. 2

R.W.B.
5-1-48

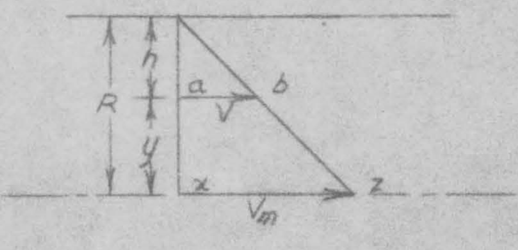


FIG. 3

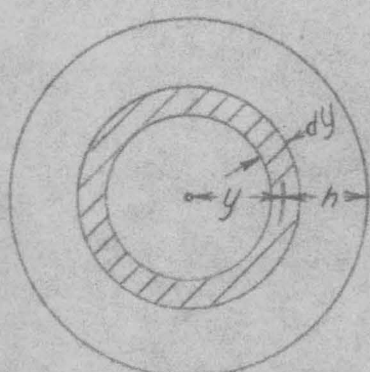


FIG. 4

R.W.B.
5-1-48

The ratio of the inertia forces to viscous forces (R_n) is

$$R_n = \frac{F}{W}$$

$$R_n = \frac{\rho D^2 V_a^2}{\mu V_a D}$$

Since the kinematic viscosity (ν) of the fluid is equal to μ/ρ

$$R_n = \frac{D V_a}{\nu}$$

where D = diameter of pipe in ft.
V_a = average velocity in ft. per sec.
ν = kinematic viscosity in ft^2 per sec.

The change from laminar to turbulent flow of a liquid in a pipe or tube is accompanied by a change in the force required to overcome frictional resistance of the pipe. If the pressure drop in a given length of horizontal pipe is measured at various velocities, it will be found that while laminar flow persists, the pressure difference will be directly proportional to the velocity and viscosity. When the critical Reynolds' number is reached, and flow becomes turbulent, the pressure differential increases abruptly and varies roughly as the square of the velocity, while being almost independent of the viscosity.

In the design of an apparatus to demonstrate flow characteristics of liquids the above mentioned variation in pressure differential is a very handy device for determining the critical Reynolds' number in that a sudden jump in the pressure difference will indicate the change from laminar to turbulent flow. The actual Reynolds' number may then be calculated by determining the velocity and kinematic viscosity of the liquid and substituting them along with the pipe diameter in the expression $R_n = \frac{D V_a}{\nu}$. It can thus be seen from the Reynolds' number equation that the essential features of the apparatus are several test lines, gauges a specified distance apart with which to measure the pressure differential, thermometers to determine the temperature of the liquid being tested, and a means of weighing the discharge so that velocity of flow may be calculated.

II. SCOPE AND PURPOSE

The original intention regarding the Reynolds' Number Apparatus was to design, construct, and operate extensively the experimental machinery. The extensive operation of the apparatus, however, would entail not only the investigation of the effects of various conditions on the Reynolds' number of liquids, but also the effects of laminar and turbulent flow on the heat transferred from the liquid to the pipe walls. It would also be possible to use the apparatus to determine friction factors of various diameter pipes. Any one of the above subjects contains enough material for a thesis. It was therefore decided to limit the scope of this investigation to the design, construction, and calibration of the apparatus. In view of the extremely slow delivery of equipment and the many delays encountered during the construction, this was indeed a wise decision.

It may therefore be stated that the purpose of this report is to present to the reader the Reynolds' Number Apparatus and to acquaint him with its design, construction, calibration, and experimental possibilities. Although only enough test runs will be made to calibrate the instruments required to determine the Reynolds' number of a given oil at one temperature, it is the author's belief that this report will serve as an adequate guide to the construction of similar apparatuses and will stimulate others to a more extensive investigation of the experimental possibilities.

III REVIEW OF LITERATURE

The need for the inclusion of the Reynolds' Number Apparatus among the experiments in the Hydraulics Laboratory was recognized during the summer of 1940 by Prof. J. B. Jones, Head of the Virginia Polytechnic Institute Department of Mechanical Engineering. At the suggestion of Prof. Jones, Mr. E. M. Simons assembled some preliminary data regarding the apparatus, but laid this work aside as more pressing duties arose.

Early in 1947 the author became interested in the Reynolds' Number Apparatus and a search for information concerning such an apparatus was instigated and carried out. Many textbooks and periodicals were consulted but no material of direct bearing was discovered. However, especially interesting accounts of Prof. Reynolds' early experiments and other remotely related material were found in the early 20th century editions of the English engineering periodical "Engineering". A complete list of literature examined is included in section two of the Bibliography.

Through the acquaintance of Prof. Jones with Prof. A. E. Bock of the United States Naval Academy it was learned that Prof. Bock had constructed an apparatus to determine Reynolds' number at the Naval Academy in 1946. This apparatus was different from any which could be constructed at the Virginia Polytechnic Institute in that Prof. Bock had ample room to run his test lines vertically.

Inquiries directed at a large number of other colleges and universities also brought negative answers in every case and led to the conclusion that there has been no material published which deals specifically with a device similar to the Reynolds' Number Apparatus.

IV. DESIGN AND BILL OF MATERIALS

Design Requirements

Generally speaking, the requirements of the Reynolds' Number Apparatus were such equipment as would be necessary to control and measure each of the variables in the expression $R = \frac{DV}{\nu}$. This equipment was divided into the following four headings for the purpose of classification:

1. Pumping and Storage Equipment.

   Pumping equipment needed for the apparatus was a pump of sufficient capacity to deliver a large enough quantity of oil to insure velocities reaching into the critical range. To obtain a wide variation of velocities it was desirable to drive the pump by means of a direct current motor. Since exact alignment of pump and motor would be a laborious undertaking, it was decided to use a flexible coupling between the two units.

   Storage of the oil being tested required two large tanks. Covers for the tanks were desirable in order to prevent foreign matter from contaminating the oil.

2. Oil Temperature Control Equipment.

   Since both steam and water lines were readily available, a temperature control coil in one of the tanks could be constructed. Such an arrangement was an ideal means of varying the viscosity of the oil being tested, and would save considerable time that would otherwise be spent emptying the system and refilling it with oils of different viscosities.

3. Piping Layout.

The piping layout was limited by the amount of space available in the Hydraulics Laboratory which could be assigned to the apparatus. The floor area allotted was approximately 45 by 10 feet with an overhead clearance of seven feet - one inch. Test lines 30 feet in length between pressure gauges could easily fit into this space with ample length beyond the gauges in which disturbances set up by fittings would be damped out. Test line sizes decided upon were steel lines of one inch diameter, 3/4 inch diameter, 1/2 inch diameter and 3/8 inch diameter, and one copper line of 3/4 inch diameter.

4. Measuring Equipment.

To determine the velocity of the oil flowing through the test lines it was necessary to include in the design provisions for the discharge tank to rest upon a set of scales. The only other instruments to be a part of the apparatus were pressure gauges at both ends of the 30 foot test pipes and thermometers in each header. Knowing the temperature, the determination of the viscosity, either experimentally or from published data, would be an easy matter.

Design Details

1. Pumping and Storage Equipment.

The determination of the correct size and type of pump was based on the properties of S.A.E. 20 at a temperature of 130 deg. F and the discharge necessary to produce flow of a critical velocity in the one inch test line.

A. Pump Discharge Calculations:

Viscosity of S.A.E. 20 at 130 deg. F is 185 S.U.S.

Kinematic viscosity is 40 centistokes or .4 stokes.

Approximate Reynolds' number for critical velocity = 2500.

$$R = \frac{DV}{\nu}$$

where R = Reynolds' number

D = internal diameter of pipe in feet.

V = velocity in feet per second.

ν = kinematic viscosity in sq. ft. per. sec.

Since 0.4 stokes = 0.4 sq. cm per sec., the viscosity in sq. ft. per sec. = $0.4 \div (2.54)^2 \div 144$.

The internal diameter of a one inch standard pipe = 1.049 inches.

The critical velocity may be found by solving for V in the Reynolds' number equation.

$$V = \frac{R\nu}{D} = \frac{2500 \times .4 \times 12}{(2.54)^2 \times 144 \times 1.049} = 12.32 \text{ ft. per sec.}$$

Quantity of flow may be found from the expression

$Q = AV$

where $Q =$ quantity of flow in cu. ft. per sec.

$V =$ velocity in ft. per sec.

$A =$ cross-sectional area of pipe in sq. ft.

Since one cu. ft. = 7.48 gallons it is possible to determine flow in gallons per minute by multiplying the right side of the equation ($Q = AV$) by 7.48 x 60. Therefore, the equation becomes

$$Q = \frac{\pi D^2 V}{4} = \frac{\pi (1.049)^2 \times 12.32 \times 7.48 \times 60}{4 \times 144}$$

$Q =$ 32.2 gal. per min.

Pump should be rated at 40 gal. per min.

B. Pump Pressure Calculations:

Estimation of pipe lengths, fittings, and static head.

Static head = 10 feet

Straight pipe = 50 feet

4 Globe valves = (4 x 29.4) feet of straight pipe.(3)

Three 90 degree elbows = (3 x 2.6) feet of straight pipe.(3)

4 Tees = (4 x 1.8) feet of straight pipe.(3)

Equivalent pipe length = 50 + (4 x 29.4) + (3 x 2.6) + (4 x 1.8)

= 182.5 feet

(3) See Bibliography.

Velocity of 40 gal. per min. through a one inch pipe =
14.9 ft. per sec.

Friction factor = 0.0475.[3]

$$h_f = f\frac{L}{2g}\frac{V^2}{D} = \frac{0.0475 \times 182.5 \times (14.9) \times 12}{2 \times 32.2 \times 1.049}$$

= 341 feet of oil

where h_f = head due to pipe friction in feet

f = friction factor

L = length of pipe in feet

D = internal diameter of pipe in feet

V = velocity of flow in ft. per sec.

g = acceleration due to gravity in ft. per sec. per sec.

$$h_v = \frac{V^2}{2g} = \frac{(14.9)^2}{2 \times 32.2} = 3.45 \text{ feet}$$

where h_v = velocity head in feet of water.

V = velocity of flow in ft. per sec.

g = acceleration due to gravity in ft. per sec. per sec.

Total head = 10 + 341 + 3.45 = 354.45 feet

Specific gravity of S.A.E. 20 at 130 deg. F = 0.9[1]

Total head = 0.9 x 354.45 = 320 feet of water

= 0.9 x 354.45 x 0.433 = 138.6 lbs. per sq. in.

Since the fluid is viscous the pump should be of the rotary type and should deliver 40 gal. per min. at 138.6 lbs. per sq. in.

(1),(3) See Bibliography.

Pump chosen was a Worthington $1\frac{1}{2}$ G.R. Iron Internal Herringbone Gear Rotary Pump.

Speed: Up to 1160 R.P.M.

Capacity: 39 gal. per min.

Maximum Pressure: 200 lbs. per sq. in.

Maximum liquid temperature: 350 deg. F

C. Pump and Motor Horsepower Calculations:

$$\text{Pump Horsepower} = \frac{Q\,s\,h \times 8.33}{33{,}000}$$

where Q = discharge in gal. per min.

h = head in feet of oil

s = specific gravity of oil

$$\text{Pump Horsepower} = \frac{39 \times 0.9 \times 351 \times 8.33}{33{,}000} = 3.14$$

Assuming pump efficiency = 0.80

$$\text{Shaft Horsepower of motor} = \frac{3.14}{0.80} = 3.93$$

In view of the fact that motor deliveries were extremely slow during the post war period, it was decided to use a motor available in the Mechanical Laboratory. The unit chosen was a Westinghouse Direct Current Compound Motor equipped with starting box and field rheostat for speed control.

Motor.

Horsepower: 5.5

Volts: 230

Amperes: 23

Speed: 600 to 1800 rpm.

Westinghouse Starting Rheostat.

Style: 824958

Class: 7010

Westinghouse Field Rheostat.

Style: 1305695

Type: 2C

D. Storage Tank Calculations:

Basing capacity on maximum pump discharge over five minute period,

Capacity = 39 gal. per min. x 5 min. = 195 gal.

To avoid filling tank to capacity and to allow longer runs if desired, the capacity was increased to 400 gallons.

$400 \div 7.48 = 53.5$ cu. ft.

$$V = \frac{\pi D^2}{4} h$$

where V = volume in cu. ft.

D = diameter in ft.

h = height of tank in ft.

limiting h to 4 ft.

$$D = \sqrt{\frac{4V}{\pi h}} = \sqrt{\frac{4 \times 53.5}{3.14 \times 4}} = \sqrt{17} = 4.12$$

Use D = 4 ft.

Actual tank capacity:

$D = 4$ ft., $h = 4$ ft.

$$V = \frac{\pi D^2}{4} h = 3.4 \times \frac{4^2}{4} \times 4 = 50.25 \text{ cu. ft.}$$

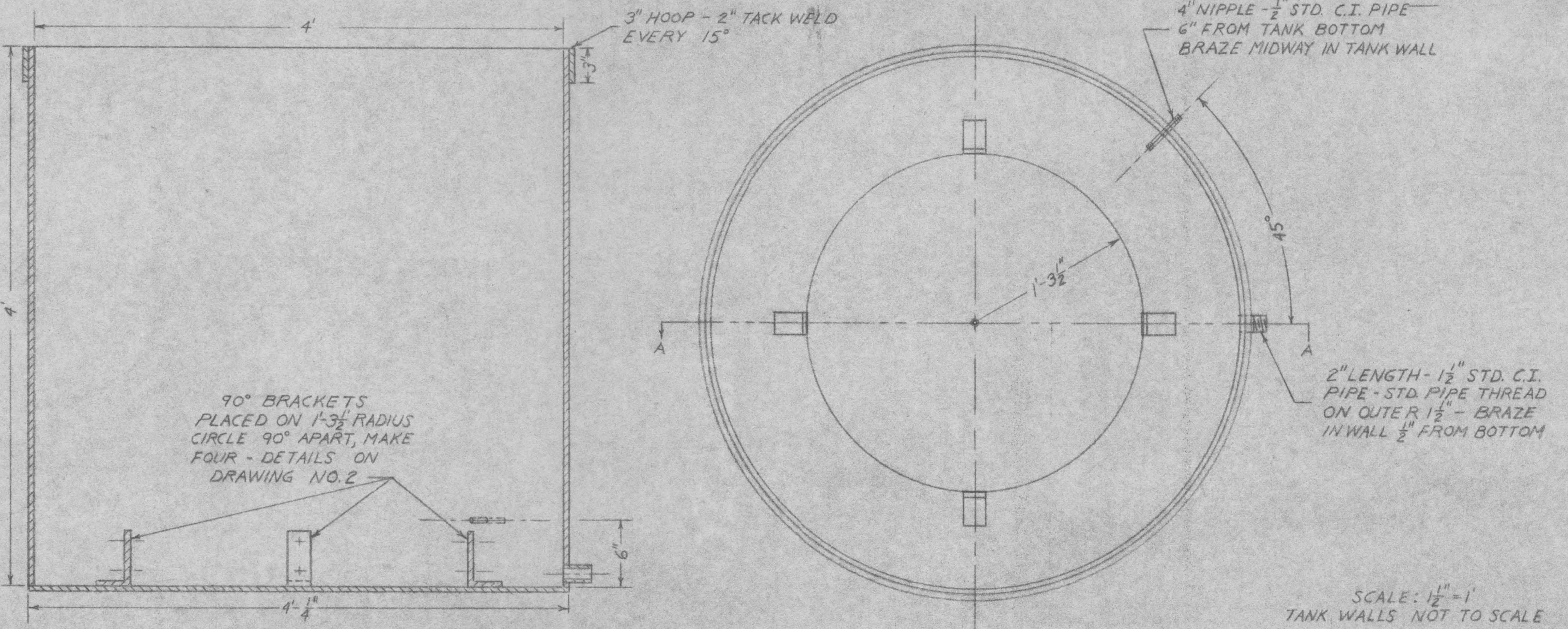
$$V = 50.25 \times 7.48 = 376 \text{ gal.}$$

For internal accessibility and simplified installation the tanks should be of the open cylindrical type and should have a 1½ inch nipple placed in the tank wall near the bottom. Furthermore, the design of one of the tanks must include brackets to hold the heating coil and an opening in the wall through which the discharge end of the coil could be run..

No attempt was made to locate holes in the covers for the various pipes and lines as such openings could be cut after delivery and there was the possibility of design revisions between fabrication and final installation.

Tank material specified was 1/8 inch wrought iron. Construction was to be entirely welded.

The drawings on pages 18 and 19 show complete tank details.



SECTION A-A
PROFILE VIEW

TOP VIEW

2 TANKS REQUIRED
MATERIAL - ⅛" WROUGHT IRON — WELD ALL SEAMS
4" NIPPLE AND BRACKETS ON ONE TANK ONLY
1½" PIPE ON BOTH TANKS

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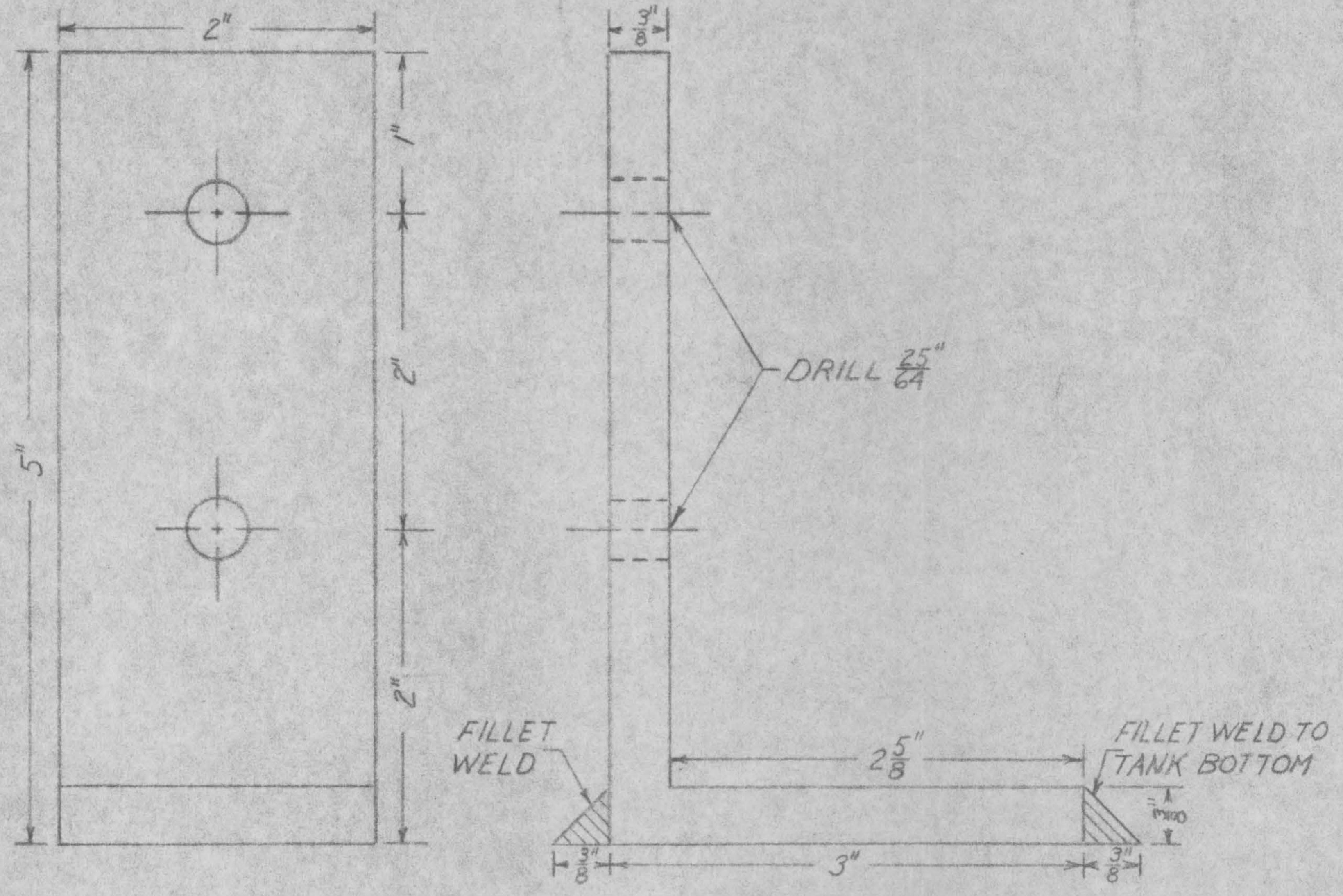
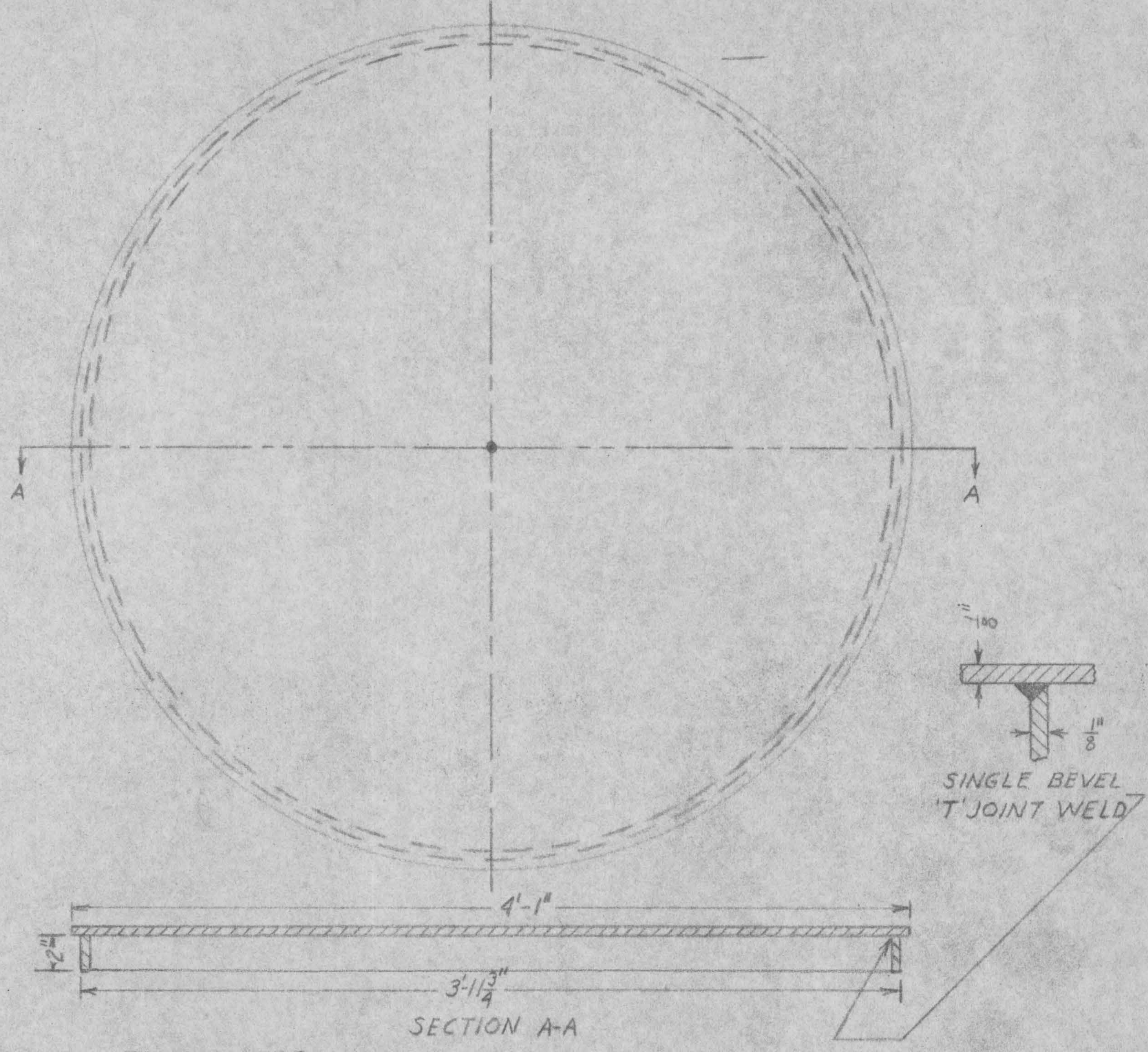
REYNOLDS NUMBER APPARATUS
TANK DESIGN - DRAWING NO. 1
1 MARCH 1947 R.W.B.



BRACKETS
4 REQUIRED
MATERIAL - WROUGHT IRON
SCALE: FULL SIZE



TANK COVER
2 REQUIRED
MATERIAL - 1/8" WROUGHT IRON
SCALE: 1 1/2" = 1'

VIRGINIA POLYTECHNIC INSTITUTE	
REYNOLDS NUMBER APPARATUS	
TANK DESIGN	DRAWING NO. 2
2 MARCH 1947	R.W.B.

2. Oil Temperature Control Equipment.

Steam was available from the high pressure mains in the Mechanical Laboratory at a pressure of 175 pounds per square inch and a temperature of 370° F. Calculations for the coil were based on the assumption that the average amount of oil in the tank during initial heating would be 300 gallons.

Quantity of oil to be heated = 300 gal.

Steam temperature = 370° F

Average specific heat of oil = 0.5 Btu. per lb., per °F[2]

Average specific gravity of oil = 0.9

Oil to be heated from 60° F to 140° F in period of 10 minutes

Thermal conductivity of copper = 220 Btu. per sq. ft. per hr. per °F.[1]

300 gal. per 10 minutes = 1800 gal. per hour.

lbs. of oil per gal. = lbs. of water per gal. x specific gravity of oil.

= 8.34 x 0.9 = 7.5

Temperature change of oil = 140 - 60 = 80° F

Btu. per lb. to heat oil = temperature change x specific heat of oil

= 80 x 0.5 = 40 Btu per lb.

Btu. required to heat 1800 gal. per hr. = 1800 x 7.5 x 40

= 540,000 Btu per hr.

Mean temperature difference between oil and steam = 370 - 100 = 270° F

(1), (2) See Bibliography.

Let heat transferred by coil under these conditions = Q

Q = Thermal conductivity of copper x mean temp. diff.

Q = 220 x 270 = 59400 Btu per sq. ft. per hr.

Let tube surface area required = A

$$A = \frac{\text{Btu per hr. required}}{\text{Btu per sq. ft. per hr. transferred}}$$

$$A = \frac{540,000}{59,400} = 9.1 \text{ sq. ft.}$$

Using $\frac{1}{2}$ inch copper tubing, outside diameter = $\frac{5}{8}$ inch.

$$\text{Surface area per ft. of tube} = \frac{\pi D}{12} \times 1$$

$$= \frac{3.14 \times 5}{12 \times 8} \times 1$$

$$= 0.1635 \text{ sq. ft. per ft.}$$

$$\text{Coil length} = \frac{\text{Surface Area Required}}{\text{Surface Area per foot of length}}$$

$$= \frac{9.1}{0.1635} = 55.6 \text{ ft.}$$

Making coil diameter $2\frac{1}{2}$ ft., the length of one turn

$= \pi D = 3.14 \times 2.5 = 7.85$ ft.

$$\text{Number of turns} = \frac{55.6}{7.85} = 7.1$$

If coil pitch is 3 inches the entire coil will be submerged when tank is half full.

In order to obtain close temperature regulation it was necessary to include in the design a needle globe valve on the discharge end of the coil. After passing through the needle globe valve the discharge from the coil was to enter a $\frac{1}{2}$" Armstrong Inverted Bucket Steam Trap.

An agitator to circulate the oil around the coil during the heating process was a desirable feature of the temperature control system. Such an arrangement would aid greatly in obtaining an even temperature throughout entire quantity of oil. Several types of agitators were investigated. The model chosen was a Type D, Pneumix Air-Driven Agitator, Colloid Equipment Company, New York. This unit has a geared drive which makes it especially suitable for viscous materials.

3. Piping Layout.

The piping layout of the apparatus was divided into two systems. The group of lines through which oil was to flow was called the oil cycle, while all temperature control piping was labeled the control cycle.

The drawing on page 26 shows all details of both systems.

The main header in the oil cycle was to be a seven foot length of standard six inch steel pipe with circular discs welded or brazed in the ends. The fittings specified for the header and the test lines proper were threadolets. Such specifications made construction of the header possible in the shop of the Mechanical Laboratory and decreased the possibility of turbulance caused by fittings to exist in the test lines. The discharge header was to consist of a $1\frac{1}{2}$ inch pipe with the ends plugged and threadolets to receive the test lines.

Return of the oil to the head end tank from the weighing tank was to be effected by closing the valves marked "Y" on the drawing on page 26 and opening the valves marked "Z". Such an arrangement would put the weighing tank on the suction side of the pump and the heating tank on the discharge side.

4. Measuring Equipment.

Measuring equipment for the Reynolds Number Apparatus was divided into the following classifications: Flow measuring equipment, pressure measuring equipment, and temperature measuring equipment.

A. Flow Measuring Equipment:

In the Mechanical Laboratory were several sets of large platform scales. It was decided to use one of these scales for measuring quantity of flow as it was believed that this method would prove more dependable over a long period of time than would a flow meter.

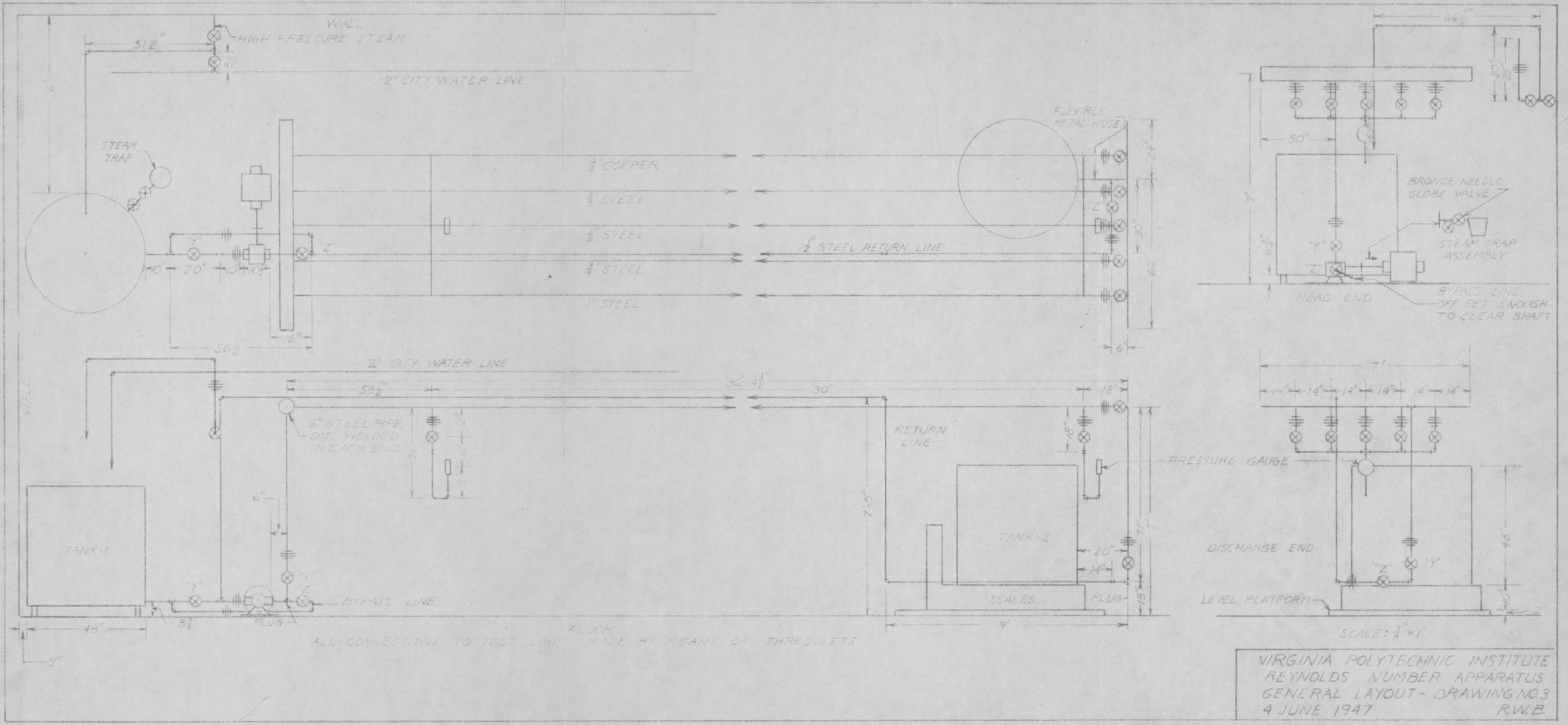
From the quantity of flow it is possible to determine velocity of the liquid by dividing the flow by the cross sectional area of the pipe through which the liquid is passing.

B. Pressure Measuring Equipment:

The difficulty of erecting a manometer to measure a pressure differential between points 30 feet apart made it necessary to select pressure gauges as the instruments with which to measure the pressure drop through the test lines. Since the pump was rated at 200 lbs. per sq. in. the gauges were chosen with the scale ranges of zero to 300 lbs. per sq. in.

C. Temperature Measuring Equipment:

In order to determine the viscosity of the oil in the test lines it was necessary to include provisions for temperature measurement. Since thermometer wells at any point along the test lines would disturb the flow of the liquid it was decided to measure the temperature in the headers. The thermometers chosen were of the standard glass tube mercury type.



VIRGINIA POLYTECHNIC INSTITUTE
REYNOLDS NUMBER APPARATUS
GENERAL LAYOUT - DRAWING NO. 3
4 JUNE 1947 R.W.B.

Bill of Materials

The Bill of Materials for the Reynolds' Number Apparatus was divided into the classifications of (1) Oil Piping System, (2) Temperature Control System, and (3) Miscellaneous Equipment.

(1) Oil Piping System

Number	Quantity	Description
1	7 ft.	6" Std. Steel Pipe
2	80 ft.	1½" Std. Steel Pipe
3	8	1½" Std. Steel Tee
4	8	1½" Std. Steel 90 deg. Elbow
5	2	1½" Std. Steel Cross
6	6	1½" Globe Valve
7	5	1½" Std. Steel Union
8	4	1½" Std. Steel Plug (square)
9	4	1½" Std. Steel Coupling
10	40 ft.	1" Std. Steel Pipe
11	2	1" Std. Steel Union
12	1	1" Globe Valve
13	2	1" Std. Steel Coupling
14	50 ft.	3/4" Std. Steel Pipe
15	4	3/4" Std. Steel Union
16	2	3/4" Std. Steel Coupling
17	2	3/4" Globe Valve
18	36 ft.	3/4" Seamless Copper Pipe
19	40 ft.	½" Std. Steel Pipe

Number	Quantity	Description
20	2	½" Std. Steel Unions
21	1	½" Globe Valve
22	40 ft.	3/8" Std. Steel Pipe
23	2	3/8" Std. Steel Union
24	1	3/8" Globe Valve
25	28 ft.	¼" Std. Steel Pipe
26	10	¼" Std. Steel Union
27	8	¼" Std. Steel 90 deg. Elbow
28	4	¼" Std. Steel Tee
29	2	¼" Std. Steel Cross
30	10	¼" Globe Valve
31	1	6x1½ Bonney Thredolet
32	1	6x1 Bonney Thredolet
33	2	6x3/4 Bonney Thredolet
34	1	6x½ Bonney Thredolet
35	1	6x3/8 Bonney Thredolet
36	2	1x¼ Bonney Thredolet
37	2	3/4x¼ Bonney Thredolet
38	1	1½x1 Bonney Thredolet
39	2	1½x3/4 Bonney Thredolet
40	1	1½x½ Bonney Thredolet
41	1	1½x3/8 Bonney Thredolet

(2) Temperature Control System

Number	Quantity	Description
42	40 ft.	½" Std. Steel Pipe

Number	Quantity	Description
43	6	½" Std. Steel 90 deg. Elbow
44	60 ft.	½" Copper Tubing
45	1	½" Bronze Needle Globe Valve
46	2	½" Std. Steel Union
47	25 ft.	½" Pipe Insulation
48	2	½" Std. Steel Tee
49	3	½" Globe Valve
50	1	½" Armstrong Inverted Bucket Steam Trap No. 211.
51	2 ft.	1½" ID, LT RW-91 Bronze Flexible Metal Hose with coupling brazed on each end. Chicago Metal Hose Corp., Maywood, Ill.

(3) Miscellaneous Equipment

Number	Quantity	Description
52	2	Open Cylindrical Tank. Consult drawings for details.
53	2	Pressure Gauge. Range of zero to 300 lbs. per sq. in.
54	2	Thermometers. Glass tube mecury type. Range of zero to 200 deg. F.
55	1	Platform Scales. Platform dimensions 4 ft. x 4 ft.

Number	Quantity	Description
56	1	Worthington 1½ G.R. Iron Internal Herringbone Gear Rotary Pump. Speed: Up to 1160 rpm. Capacity: 39 gal. per min. Maximum Pressure: 200 lb. per sq. in. Maximum Liquid Temperature: 350 deg. F.
57	1	Westinghouse Type SK Direct Current Motor. Horsepower: 5.5 Volts: 230 Amperes: 23 Speed: 600 to 1800 rpm.
58	1	Westinghouse Starting Rheostat Style: 824958 Class: 7010
59	1	Westinghouse Field Rheostat Style: 1305695 Type: 2C
60	1	Pneumix Air Driven Agitator, Type D, Colloid Equipment Company, New York.

V. CONSTRUCTION

In constructing the Reynolds Number Apparatus it was found to be quite impractical to attempt to follow exactly the dimensions on the drawings. Small deviations in nipple and pipe lengths, while in no way changing the general scheme of the lay out, often saved considerable time by allowing the use of standard nipples and pipe lengths on hand rather than requiring the cutting of special and exact lengths. The drawings were strictly adhered to, however, in installing the test lines.

Construction was begun by placing the head end storage tank in position. The elevation of the tank above the floor was determined by the elevation of the pump inlet which in turn was determined by the height of the motor shaft. To obtain the proper tank elevation a small frame was made of standard 2 inch channel iron.

The pump was supported by a frame similar to that upon which the tank was placed. This frame was of the proper height to allow the pump shaft and motor shaft to be coupled when the motor was bolted directly to the floor. Holes one inch in diameter were drilled in the floor and the pump frame secured by $\frac{5}{8}$ inch bolts grouted in the holes by lead plugs.

The $1\frac{1}{2}$ inch pipe from the tank to the pump and the fittings immediately following the pump outlet were installed. The bypass line running from the pump outlet to the tee near the tank outlet was not installed as it would interfere with placing the motor.

Holes were drilled to receive the motor, and the $\frac{5}{8}$ inch bolts grouted into place. The motor was then lowered onto the bolts and the shaft aligned with the pump shaft. Slight shimming was necessary to achieve satisfactory alignment.

The copper heating coil was formed and put in the tank. It was secured by pipe hangers bolted to extensions of the brackets which were welded to the bottom of the tank. The discharge end of the coil was connected to the $\frac{1}{2}$ inch nipple in the tank wall, and the valves and steam trap placed on the outlet side of the nipple as shown on the drawing on page 26.

The two inch water line and the high pressure steam line were tapped and the common water and steam line run to the inlet of the heating coil.

Frames made of angle and flat sections were hung from the over head beams to within a few inches of the level at which the test lines would run. These frames were spaced at ten foot intervals and from them were hung loops of $\frac{1}{32}$ inch sheet steel. The loops could be raised and lowered by means of the bolts holding them to the frames. Such an arrangement provided a means of accurately leveling the test lines.

The large header was fabricated from a seven foot section of six inch pipe by brazing the ends and threadolets in position. After completion the header was tested for leaks by filling it with air at 175 lbs. per sq. in. and applying a soap solution to all brazed joints. The header was then hung from the overhead frames and the $1\frac{1}{2}$ inch line from the pump outlet was connected to the inlet fitting.

At this point in the construction the starting box for the motor arrived. The box was installed in the 220 volt direct current line and the motor tested at no load. After the motor tests were complete the bypass line from the pump discharge to tank outlet was installed, and the shafts of the motor and pump were connected by means of a flexible coupling.

The test lines were assembled and the threadolets to receive the pressure gauge lines were brazed in place. In joining the two sections that made up each test line special care was taken to see that the pipe ends met and fit tightly together inside the coupling. This method of assembly would reduce the turbulence set up at the coupling to a minimum. The lines were tested for leaks and then hung from the loops provided for them.

The discharge header was made from two sections of $1\frac{1}{2}$ inch pipe joined together by a tee. The outlet of the tee was to receive the line running down to the weighing tank. The ends of the header were plugged, and the threadolets brazed in place. The brazed joints were tested for leaks and the header connected to the test lines.

The weighing tank was placed upon the scales and the line from the discharge header to the tank was installed and connected.

The return line from the weighing tank to the suction side of the pump was next installed. This line was supported by resting it upon the frames which had been previously hung from the overhead beams. It was impossible to support the line in this manner for its entire length, however, due to interference of a large steam line. At the point of interference the return line was brought to the level of its connection with the weighing tank and continued at that level to the connection. As the point of interference of the steam line and the return line was near the weighing tank, it did not obstruct movement about the floor to have the return line near the floor for the short distance required.

Thermometer wells were placed in each header and pressure gauges were mounted on the test lines as shown on the layout drawing.

The final steps in the constructing of the Reynolds Number Apparatus consisted of bolting the agitator to the cover of the heating tank, placing the cover on the tank, and connecting the steam and water line to the temperature control coil.

VI CALIBRATION

Calibration of the Reynolds' Number Apparatus was a relatively simple matter as the group of instruments involved included only the platform scales, pressure gauges, and thermometers.

The calibration of the scales was checked by adding known weights in increments of fifty pounds and checking the scale reading against the sum of the weights on the platform. The scale reading was found to be slightly low. This error was corrected by placing several lead shot in the cup on the end of the weight hanger, thus making it possible to use direct scale readings and eliminating the need for a calibration curve.

The pressure gauges were tested on a Shaeffer and Budenberg Dead-Weight Pressure Gauge Tester. Both gauges were found to be accurate to within one pound. Such accuracy was deemed sufficient for all practical purposes.

The thermometers were checked by comparison with other thermometers at several different temperatures. Differences in temperature readings were very small, and it was decided to use the thermometer readings without correction.

The one inch line and the $\frac{3}{8}$ inch line were chosen as the lines upon which preliminary tests would be run, for it was reasonable to assume that the other lines would present no difficulties if the critical point could be reached in both the largest and smallest pipes.

The temperature of the S. A. E. 20 Lubricating Oil in the heating tank was raised to 120°F. The oil was then pumped through the one inch test line, the speed of the pump being slowly increased until the needle of the pressure gauge began to vibrate violently. The discharge was measured at this point, and the velocity of the oil was calculated from the discharge. This velocity was found to be 12.1 ft. per sec. A Saybolt Universal Viscosity of 217 S. U. S.

was determined by means of a Tag Viscosimeter. The corresponding kinematic viscosity was 0.0004945 sq. ft. per sec. and the resulting Reynolds' number was 2142.

The above procedure was repeated for the $\frac{3}{8}$ inch test line. The temperature, however, was raised to 126°F. This change in temperature lowered the kinematic viscosity to 0.00043 sq. ft. per sec.. The critical velocity for the $\frac{3}{8}$ inch pipe was 21.8 ft. per sec.. These test results gave a Reynolds' number of 2072.

Although the tests run on the one inch and $\frac{3}{8}$ inch lines were by no means exhaustive and did not demonstrate the complete possibilities and flexibility of the apparatus, they did very clearly indicate the value of the apparatus as an experimental device.

Reynolds' number calculations.

$$V = \frac{Q}{62.4 \times S \times A}$$

where Q = discharge in lbs. per sec.

S = specific gravity

A = cross sectional area of pipe in sq. ft.

V = velocity in ft. per sec.

62.4 = lbs. of water per cu. ft.

$$R = \frac{DV}{\nu}$$

where R = Reynolds' number.

D = diameter of pipe in ft.

V = velocity in ft. per sec.

ν = kinematic viscosity in sq. ft. per sec.

VII. SUMMARY

The design of the Reynolds' Number Apparatus was concerned chiefly with the control and variation of the test line diameter, velocity of flow, and viscosity of the liquid. Briefly, it may be said that these three aims were achieved by providing several test lines of various diameters, a variable speed pump driven by a direct current motor, and a temperature control coil. Another important element of the design was the inclusion of pressure gauges at both ends of the test lines so as to observe the transition from laminar to turbulent flow.

Before any of the above factors could be considered, however, it was necessary to make a survey of all possible locations in the laboratory where the apparatus could be constructed. In the Virginia Polytechnic Institute Mechanical Laboratory the only space available had a low overhead clearance, thus making it necessary to run the test lines horizontally. Had it been possible to erect the test lines in a vertical position several advantages could have been realised. Among these advantages is the possibility of obtaining such low velocities that water could be used rather than oil as the liquid being tested. This is possible because pumping the liquid upward through a vertical pipe will always keep the pipe full while the flow is laminar. In the case of horizontal test lines the velocity required for laminar flow is often so low that the pipe is only partly full unless the liquid is very viscous. The advantage of using water is greatly offset, however, by the small change in viscosity that accompanies a temperature change. If it is desired to show the effect of viscosity on Reynolds' number it is necessary to use oil or some other liquid with a wide viscosity range. The other advantages of vertical test lines are the need for less floor area and the use of gravity to return the liquid to the lower tank.

The construction of the Reynolds' Number Apparatus was unable to follow a logical plan or schedule due to the erratic and undependable arrival of materials. It is the opinion of the author that considerable time could be saved by arranging the work in such a manner as to allow greater concentration of effort than was possible in the construction of the apparatus at Virginia Polytechnic Institute. One example of such a plan would be the scheduling of all brazing and welding at one time rather than interrupting such an operation to install each part of the apparatus as it is fabricated.

The fact that it was possible to observe the effects of the variables involved in Reynolds' number and the transition from laminar to turbulent flow establishes the Reynolds' Number Apparatus as a practical and useful piece of laboratory equipment. Undoubtedly there are numerous refinements and improvements applicable to the design which will be brought forth in subsequent operation of the device. The author feels, however, that the fundamental concepts of the design are sound, and that the apparatus will well serve the purpose of demonstrating to the student at Virginia Polytechnic Institute the factors concerned in the determination of Reynolds' number.

VIII RECOMMENDATIONS

The immediate problem confronting the staff of the Mechanical Laboratory is that of working up in experimental form a series of tests on the Reynolds' Number Apparatus that will show the effects of various test line diameters , velocities, and viscosities on Reynolds' number. Since this is the express purpose of the apparatus, no design modifications will be required. It is the earnest hope of the author, however, that interest will not stop here, and that the subject of heat transfer will be investigated. It is a known fact that less heat is transferred when flow is laminar than when flow is turbulent. This is because the layer of liquid next to the pipe wall acts as an insulator. The Reynolds' Number Apparatus, with the addition of insulation on the test lines and accurate temperature recording instruments at several points along the lines , appears to be an ideal device for demonstrating this phenomenon.

It is the author's belief that sometime in the not too distant future, the rapid growth of the Mechanical Engineering Department will require a new and modern building. Such an e vent will provide an excellent opportunity for furthur development of the Reynolds' Number Apparatus. It is strongly recommended that the possibility of using vertical test lines be investigated. It is also suggested that the use of glass or a clear plastic be considered as the material for one of the test lines. This would greatly extend the flexibility of the apparatus by allowing the duplication of Prof. Reynolds' experiment.

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