HEAT TRANSFER OF CONDENSING FREON-12 INSIDE A HORIZONTAL TUBE

by

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INTRODUCTION

Freon-12 as a Refrigerant

In recent years, the development in refrigeration industries called for new types of refrigerant to meet their demands. For instance, sulfur dioxide, ethyl chloride and methyl chloride made their appearance as low-pressure refrigerants to permit lighter, smaller, and lower priced equipment. The need for a refrigerant having high vapor density, low ratio of compression, and other characteristics suitable for a centrifugal compressor led to the introduction of methylene chloride.

Finally, in exploring the field of the hydrocarbons of the methane and ethan series, physical chemists found that by substituting chlorine and/or fluorine atoms for hydrogen atoms, they could synthesize almost any type of refrigerant needed for a specific use or for a specific type of equipment. These halogenated hydrocarbons are supplied to the industry under the trade name Freon. Freon-12, as can be seen from its chemical formula, CCl_2F_2, is an organic synthesis from methane, CH_4, obtained by substituting two chlorine and two fluorine atoms for all hydrogen atoms in methane. Since its appearance to the refrigeration industry, Freon-12 has been playing an important role in this field.

Film Condensation

It is known that there are two modes of condensing vapors. If the condensate is formed in drops, the phenomena is called dropwise condensation. If, however, a continuous film is formed on a wettable cooling surface, it is known as film condensation. Dropwise condensation is the more desirable, because it yields in some cases up to 15 to 20 times more condensate (Jacob, 9) than film condensation does for the same temperature difference between vapor and surface. Film condensation, notwithstanding, prevails for all vapors except steam. The usual refrigerants of ammonia, methyl chloride, the freons, and other organic compounds are good wetting agents and always exhibit a pure film condensation.

Usually refrigeration condensers are classified into two types according to the tube orientation; horizontal and vertical. If a vertical arrangement is used, whether the vapor condenses on the outside or inside of a tube, the formed condensate flows down the tube under the action of gravity, or under the combined effect of gravity and the shear forces developed at the vapor-liquid interface as the result of high vapor velocity. The thickness and mean velocity of the condensate film increase in the direction of flow. The flow of condensate is initially laminar, but if the tube is sufficiently long, a transition to turbulent flow will occur. If, on the other hand, a horizontal-tube condenser is used, the vapor will condense in a film on either outside or inside surface of the tube. The thickness and mean velocity of the film increases as the condensate flows down the circumference of tube. If this occurs only under the action of gravity, the condensate downward flow in a horizontal tube is always laminar, since the condensate falls through a small distance, never greater than the diameter of the tube. It is to be noted that if the vapor condenses inside rather than outside, then the condensate must collect in the bottom of the tube and flow off in a horizontal direction. In this case, there exists two directions in which the condensate flows. For the same temperature difference between vapor and surface, the rate of condensation for this case will be less than that developed when the condensate forms on the outside surface. This is the consequence of the increased resistance to the flow of heat offered by the stream of condensate in the bottom of the tube.

The Problem

For the case of moderate and high vapor velocity, the condensation mechanism inside of horizontal tubes is complicated. The effect that the lateral shear forces produced by the moving vapor have upon the circumferential downward flow of the condensate film might differ much from that developed in a vertical tube in which the vapor and the condensate film flow in the same direction. In addition to this, the flowing vapor will affect the horizontal flow of the stream of accumulated condensate flowing along the bottom of the tube.

Although the problem of analyzing and predicting the performance when condensation occurs inside of horizontal tubes, it is often desirable to condense refrigerants in this manner. The air-cooled and evaporative condensers, common to the refrigeration industry, are examples. It is therefore, important to know more about the mechanism of heat transfer and the magnitude of the average coefficient of heat transfer realized in this type of condensation. A cooperative research project between The American Society of Refrigerating Engineers and the Engineering Experiment Station of Kansas State College was established to investigate condensation of Freon-12 in horizontal and inclined tubes. In this paper, a study of the performance of a horizontal tube is presented.

DESIGN OF THE TEST CONDENSER

Previous Experiments

Before going into the details of the test apparatus, it will be desirable to describe briefly about the previous experiments, which were carried out as a research project in Mechanical Engineering, Kansas State College.

A total of 142 runs was recorded in the previous experiments, (11 and 12). The test-section of a 2-foot length was located at the center part of a 10-foot counter flow, single tube-condenser with Freon-12 vapor flowing inside the inner tube. The increase in cooling water temperature was measured with five thermopiles at 6-inch intervals. The temperatures of the outside wall of the inner tube were measured at 6-inch intervals over a 2-foot length with five thermocouples installed in milled groove and covered with a solder of approximately the same conductivity as the brass tube. The arithmatic mean of these temperatures was taken as the average temperature of the outside surface of the inner tube. The Freon-vapor temperature was measured at both ends of the 10-foot condenser, and the arithmatic mean was taken as an average bulk temperature of the Freon-vapor.

The Freon-12 flow rates were measured with a Brooks Rotameter; then the vapor velocity and the mass velocity of flow of Freon-12 were calculated from these data. The heat transfer by conduction through the inner tube was calculated from the weight rate of flow and temperature rise of the cooling water through the test condenser. By utilizing the relation $Q=k_m A_m \frac{\Delta tb}{\Delta x}$, the tube wall temperature of the vapor-side, t_w, was calculated. Using the value of t_w and the measured vapor temperature, t_v, h_f was calculated from the equation $Q=h_f A_1 \Delta t$.

As a result of these experiments, using the vapor velocities of 10-20 ft/sec. Freon-side heat transfer coefficients of 300-600 B/hr ft^{2}F were reported, (Potter, 12). Patel (11) obtained a correlation using dimensionless groups and the physical properties of Freon-12.

$$\frac{h_f D}{k_f} = 10.17\left(\frac{DG}{\mu_f}\right)^{0.4013} \qquad (1)$$

It was noted also that the tube-wall temperature varied around the periphery as well as along the axis of the tube. He concluded that the Freon film coefficient, h_f, decreased with the increase in temperature drop across the film; and an increase in the mass velocity of Freon-12 condensate, G, would increase h_f.

Design and Construction of the Test Condenser

Considerations. In designing the test condenser, the following were considered:

1. The principle and the technique of the previous experiments were to be preserved, and their suggestions to be adopted.

2. Some parts of the apparatus such as the boiler and the after condenser used on the previous experiments were available as originally located, and the new design of equipment was to be based on their use.

3. The temperature variation of the tube wall around the perimeter was noted by many researchers (2, 4 and 11); and it was reported that no one location would satisfactorily give the perimeter mean temperature. Consequently the tube wall temperature was to be measured at a number of points around the perimeter as well as along the axis of the tube.

4. Sight glasses were to be installed for the visual study of the condensing mechanism.

5. The temperature of the flowing vapor was to be measured immediately before the entrance and after the exit of the test condenser.

6. A superheater was to be made to control the entrance conditions of the vapor.

7. The pressure drop of the flowing vapor across the test section was to be measured.

8. City water was to be used as the cooling medium. Thus, the maximum flow rate and the minimum entrance temperature of the cooling medium were almost fixed.

9. The temperature of the cooling medium was to be measured at several points along its path. This was to be done in an attempt to study changes of film coefficient of Freon-12 along the tube.

Inner Tube. In order to investigate the effect of vapor velocity upon the coefficient of heat transfer, the inside diameter of the tube was made small enough to give an appreciably high vapor velocity in the tube with the power input available. A brass-tube of 0.5 inch I. D. and 0.753 inch O.D. was selected as the inner tube of the test condenser.

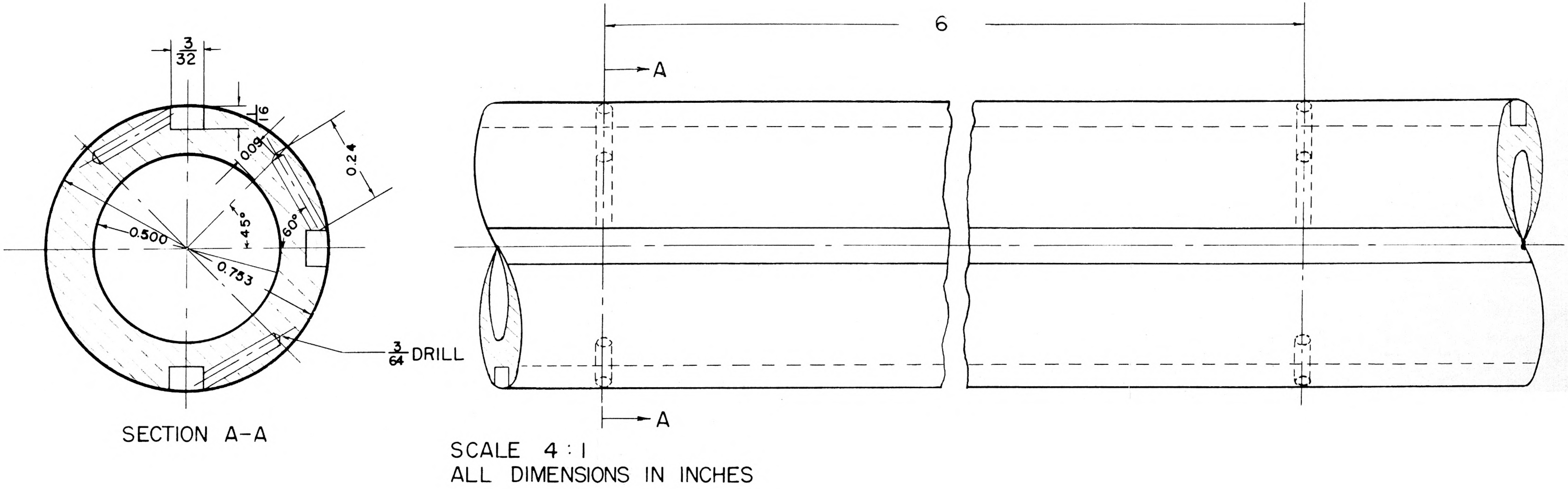
An attempt was made to determine the accurate tube wall temperature and from this the temperature drop across the condensate film, using for this purpose the measured value of the vapor temperature. By means of this temperature drop and the measured heat transfer rate to the cooling water, the film coefficient was evaluated.

As stated before, no one location of the thermocouple would give the mean tube wall temperature; therefore fifteen thermocouples were installed in the grooves on the tube. Thermocouple lead grooves each 1/16-inch deep and 3/32-inch wide were milled longitudinally on the tube. These grooves were deemed to be appropriate, since an increase in number of the grooves would reduce the heat transfer rate, and the heat flux through the tube wall would then be much distorted. In each groove, five thermocouples were installed at 6-inch intervals over a 2-foot length. The cross-sectional view of the tube is shown in Fig. 1. The grooves were 90-degrees apart from each other, over one half of the tube circumference. Holes, 3/64-inch in diameter, were drilled from the side walls of the grooves. These holes were drilled for thermocouples and were so formed that each thermocouple junction was located 45-degrees from the axis of its lead groove. The estimated distance of the thermocouple junctions from the inner wall of the tube was 0.09 inches.

Outer Tube and Flanges. The outer tube was a 15/16-inch I. D. and 1-1/2-inch O. D. micarta tube. Since the temperature of the cooling water was to be measured along its path, five thermocouple stations, the positions of which corresponded to the thermocouple stations on the inner tube wall, were built with 6-inch intervals over the 2-foot length of test section. Each of these stations consisted of three thermocouples 120-degrees apart around the perimeter. The thermocouples extended into the stream of cooling water a distance equal to one half the thickness of the annulus. Fifteen holes were drilled and tapped exactly at the location desired. Fig. 2 shows the details of the outer tube.

Flanges for the cooling water inlet and outlet were machined from 3-inch micarta rods, as shown in Fig. 3. A cavity was provided in each of the flanges for even distribution of the uniform temperature water and for mixing space to and from the annular path. Each flange had an 11/16-inch recess on one face to provide space for 1/8" "Garlock 115" packing, two caps for pressing the packing, which were machined from plates of bakelite. These were held in place by three screws.

Instrumentation. The instrumenting of the test-condenser involved the locating of thermocouples for measuring the temperatures of tube-wall, Freon-vapor, and cooling water; and the installing of a differential manometer for the measurement of the pressure drop across the test condenser.



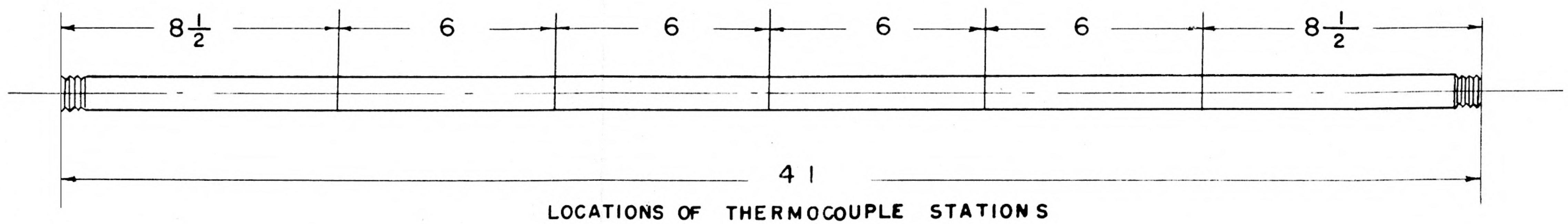


Fig. 1. Cross sectional view of inner tube and locations of thermocouple stations.

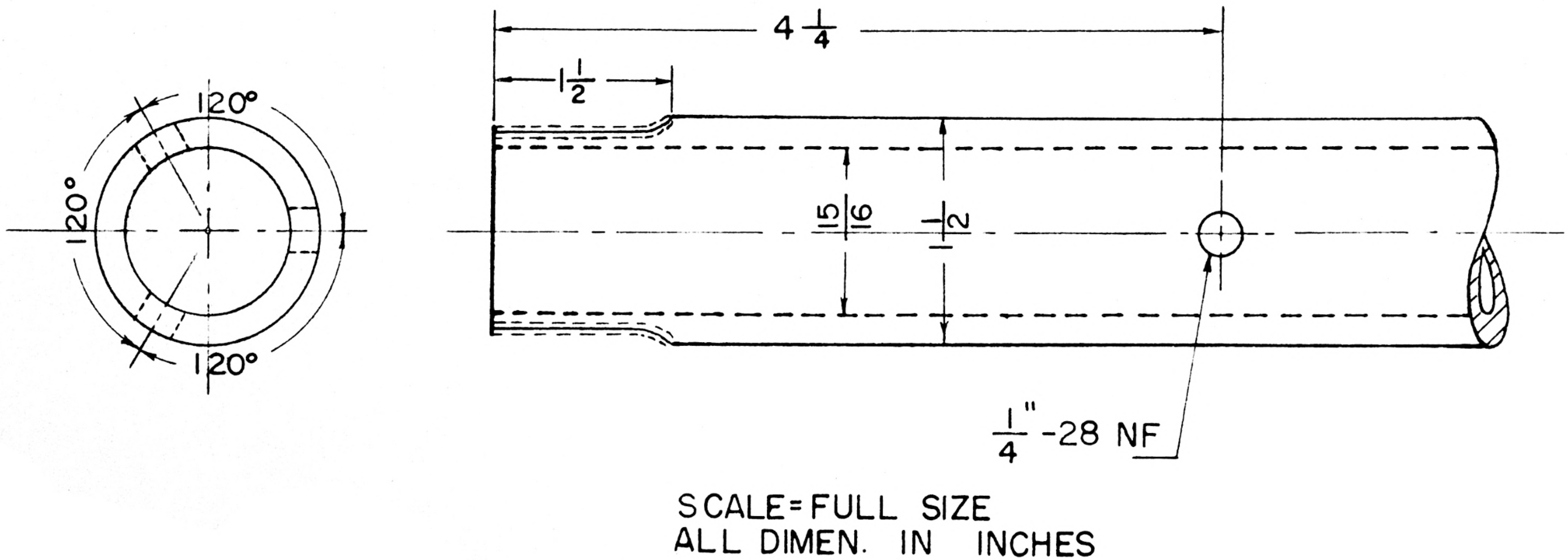


Fig. 2. Outer tube.

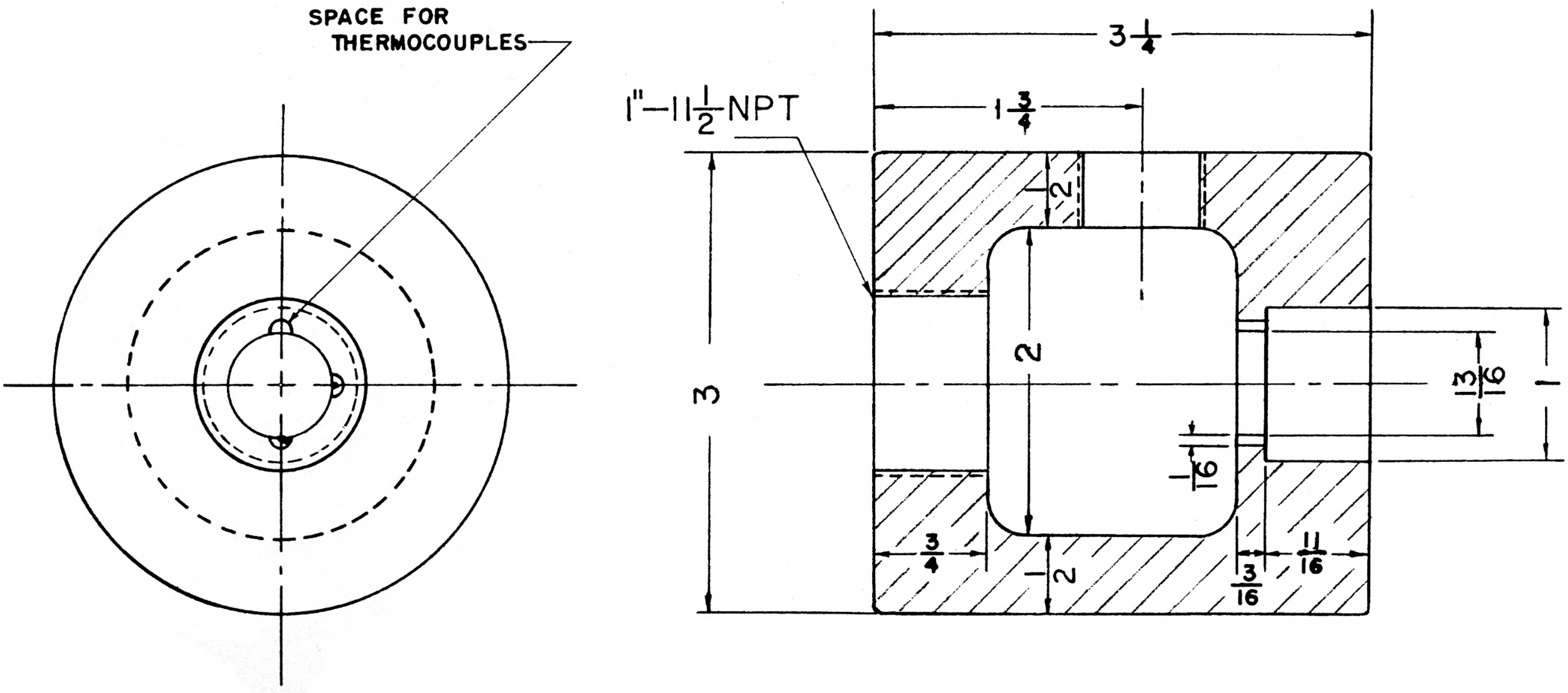

SPACE FOR
THERMOCOUPLES
1"–11 1/2 NPT
3 1/4
1 3/4
3
2
2
2
3/4
13/16
1
1/16
3/16
11/16

Fig. 3. Flange.

Number 30 (0.010-inch D) copper-constantan thermocouple wires made by Leeds & Northrup Company were used. These thermocouple wires were coated with "No. 450 G. E. clean Baking Varnish", and then baked in a furnace of about 220°F for 12 hours. The thermocouple junctions were spot welded to form beads; then they were calibrated from 40°F to the saturation temperature of steam at atmospheric pressure (212°F) by a set of standard mercury thermometers and a Leeds & Northrup semi-precision potentiometer. From these data, calibration charts were worked out.

In order to locate thermocouple junctions in the bottom of the drilled holes so as to make direct contact with the tube, the tube was heated above the melting point of rosin-core solder, and the solder was melted in the holes; then the junctions were inserted the required length. The leads of the thermocouples were brought to both ends of the tube along the grooves. At emerging portions of the leads from the grooves, each group of leads was covered with "spaghetti" to protect it from abrasion. When the leads were placed in the grooves, they occupied almost the entire cross sections of the grooves. "No. 450 G. E. clean Baking Varnish" was employed to fill the unoccupied spaces. Solder was not used because the application of heat during soldering was likely to spoil the insulations on the lead wires. The tube was put in a furnace and held at 220°F for 12 hours, then the surface of the tube was made smooth.

At each end of the test condenser, a thermocouple well was installed on the hexagonal brass coupling for the measurement of the vapor temperature. The detail of construction is shown in

Fig. 4. The thermocouple junctions were soldered to the bottom of the thermocouple wells, and the leads were covered with "spaghetti". Because it was expected that the inside tube of the test-section would be part-filled with condensate (Chaddock, 6), and its bottom temperature would be lower than that of the vapor, these thermocouple wells were screwed in from the top of the tube. The well projected into the inner tube a distance equal to 1/2-inch at the entrance and 1/8-inch at the exit of the test condenser. The level of the condensate at the exit was higher than that at the entrance and for that reason the thermocouple at the exit was allowed to project only 1/8-inch into the vapor stream, (p.843, 14).

For the measurement of cooling water temperature, fifteen thermocouple wells were from 1/4-inch I. D. polystyrene rod. A 3/64-inch hole was drilled through the rod and, on one end of the rod, threads were cut. A small amount of Duco cement was applied to each end of the polystyrene rod to fix the thermocouple in position. These thermocouple wells were to be screwed into the tapped hole on the outer tube. The junctions of these thermocouples were centered in the annular space. The average value of three temperature readings at one station was regarded as approximately the average temperature of the cooling water at this section. The construction can be seen in Plate I. The inlet and the outlet temperatures of the cooling water were measured by thermocouples installed at the mixing tees which were connected to the flanges. The installation of this part is shown in Fig. 5.

On first considering the measurement of the pressure drop across the test tube, there arose a difficult problem. From the equation

$$-\Delta p = \frac{1}{2g_c} \rho V^2 f \frac{\Delta x}{D} \tag{2}$$

it was realized that for Δx = 2 feet, Δp would be small, though it was pointed out (7 and 3), that the value of f is considerably higher for wetted walls than for a dry pipe. The U-tube type manometer with mercury was considered not accurate enough to measure this difference of pressure. It was decided to use a Freon-liquid manometer instead. Since vapor from the test section continuously condensed in this manometer, the fluid level in the manometer was not a constant, but was maintained within suitable limits by the periodic withdrawal of liquid from the manometer. An average reading of the difference of the liquid level was considered to be the difference in pressure between entrance and exit. The entire manometer and its piping were well insulated to minimize the heat transfer to the surroundings. When withdrawn, liquid Freon was drained to the point of lowest pressure in the system.

The pressure taps were located on the hexagonal couplings 7/8-inch from the thermocouple wells which were used to measure the Freon-vapor temperature. The tap for draining the Freon from the manometer was located on the return pipe near the boiler. Close to the taps, needle valves were provided to cut them off from the main system when they were not in use.

<u>Sight Glasses</u>. Two sight glasses of 0.5-inch I. D. one at each end of the test section, were connected to the brass couplings. These sight glasses were 1/2-inch high and 9/16-inch long. They were used to observe the flow condensate.

<u>Assembly</u>. Plate II shows the main components of the test condenser to be assembled. Before being assembled, every part of the test-condenser was cleaned, especially the surface of the inner-tube. Following this, the inner tube was fitted into the outer tube. In order to assure concentric alignment of the tubes, small spacers were used at three positions of the tube in the annular space. Next, two flanges were screwed into the outer tube. The packing and caps were then applied. Following this, the two brass couplings with sight glasses were coupled to the test tube. In this step, special care was taken to bring the pressure taps to the upper side of the tube, when in a horizontal position. The test section was then supported horizontally by two steel hangers at hexagonal couplings. At the exit and entrance sections of the test section a copper tube of 1/2-inch I. D., 5/8-inch O. D. was welded to the test section and the supply system to connect the two. The manometer circuit was completed by connecting it to the supply system with 1/8-inch copper tubes. Thermocouple leads were connected to a switch box, one thermocouple at 32°F served as reference junction. To minimize heat flow to the surroundings, the apparatus was well insulated. At the sight glasses, sheets of aluminum foil were used as reflectors of radiant heat. The set-up of the test condenser is shown in Fig. 5.

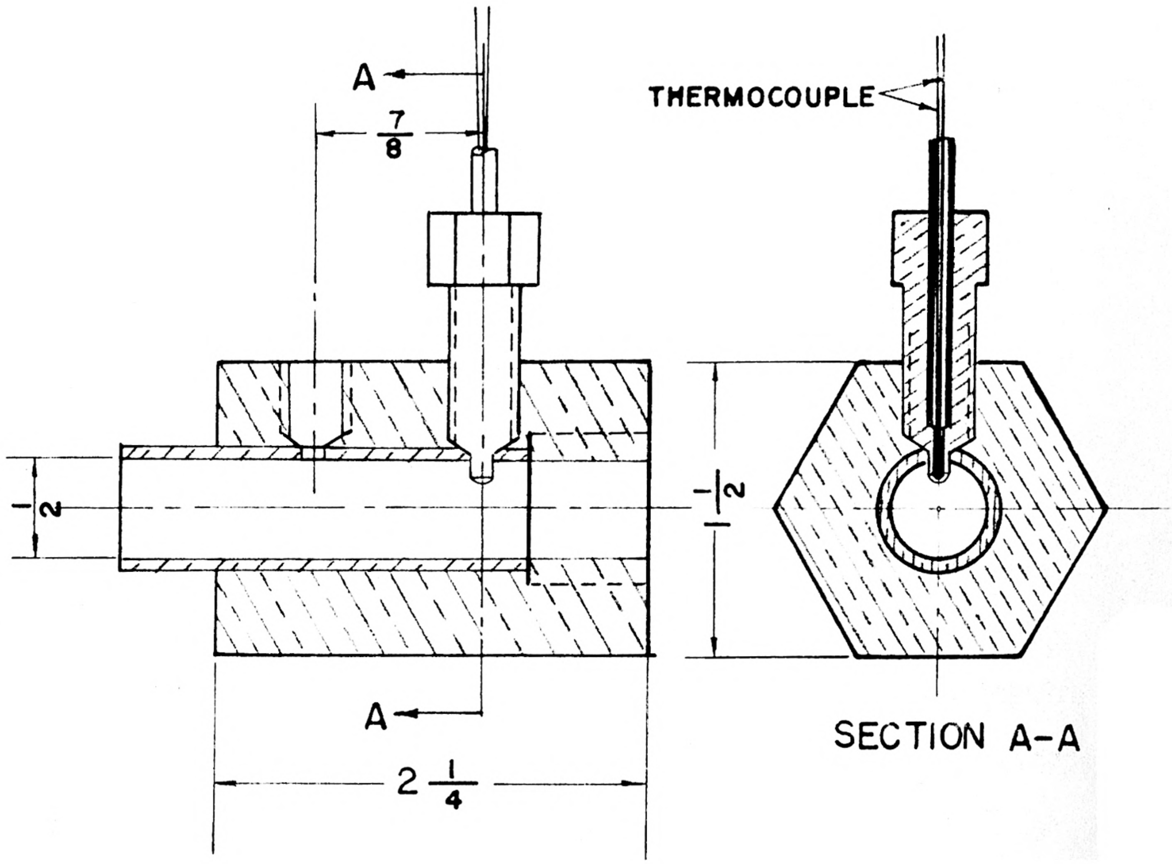
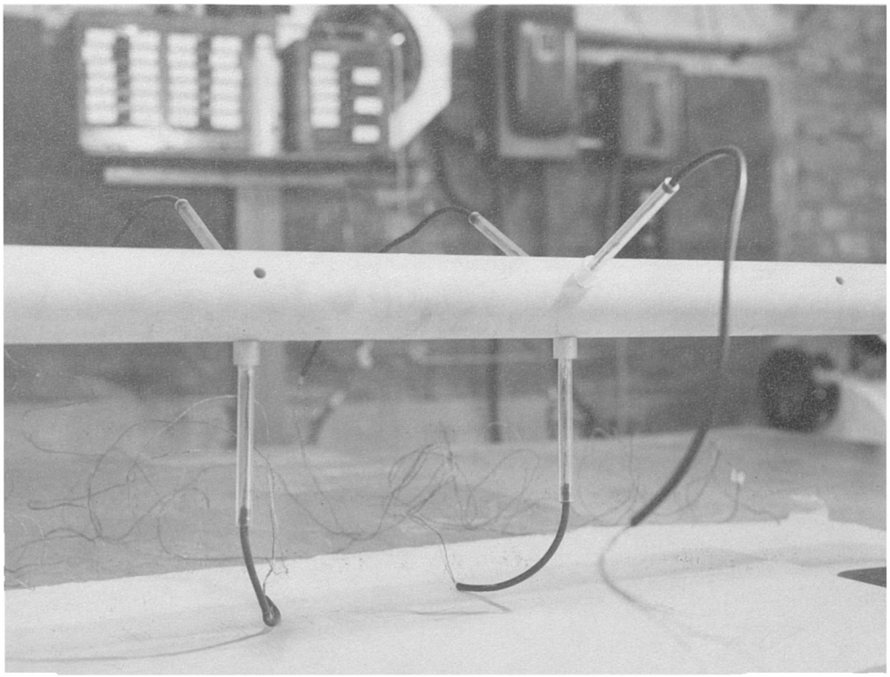


Fig. 4. Thermocouple installation for the measurement of Freon-12 vapor temperature.

EXPLANATION OF PLATE I

Photograph of the outer tube of the test-condenser, showing the method of installing the thermocouple wells for the measurement of cooling water temperature.

PLATE I



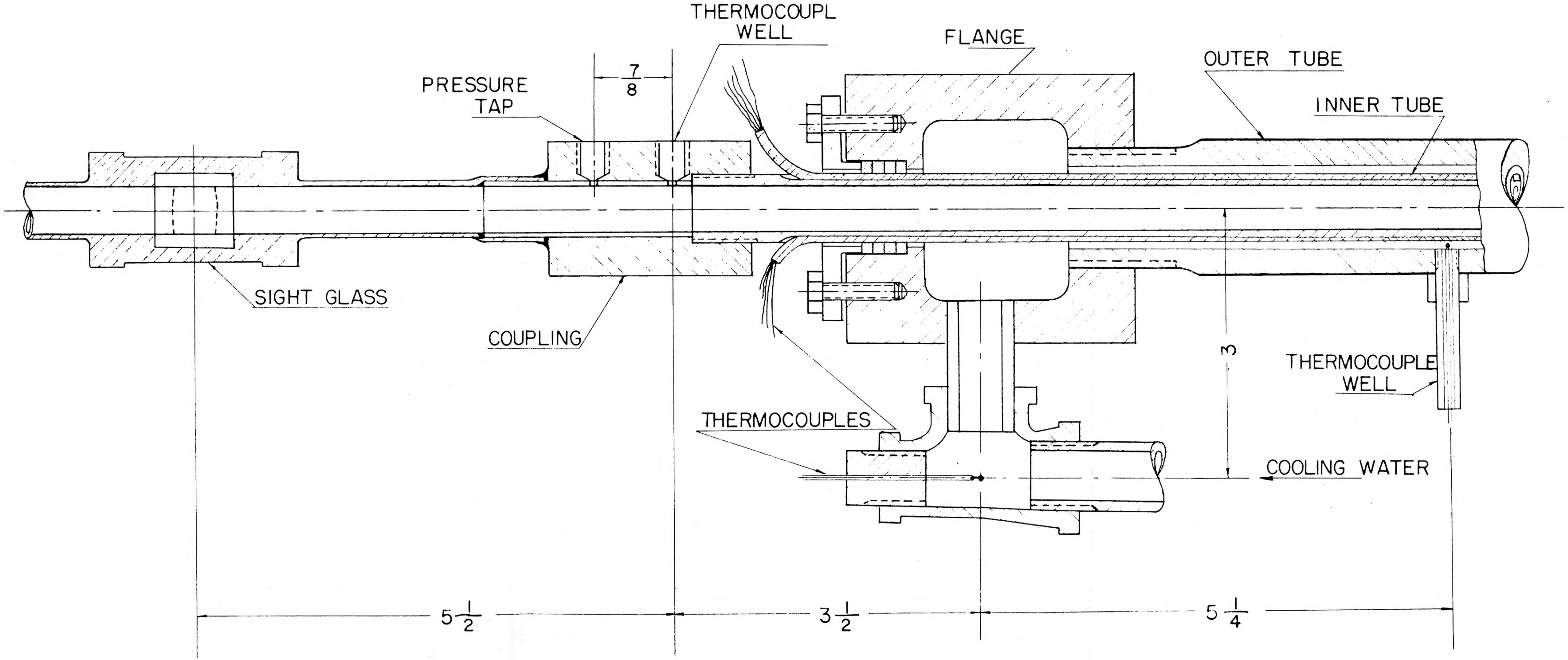
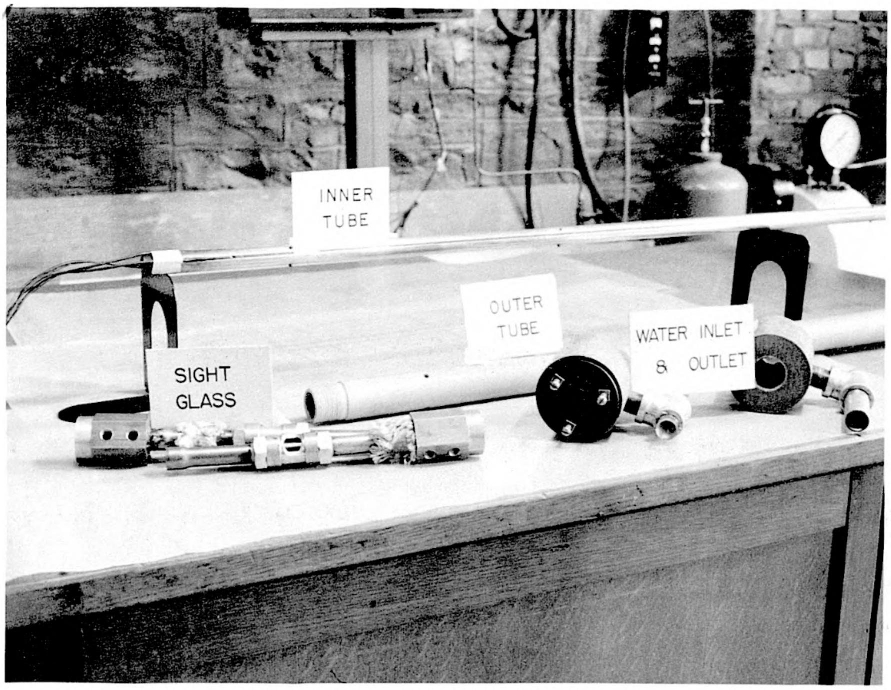


Fig. 5. Assembly drawing of the exit section of the test-condenser for studying the condensation of Freon-12 inside a horizontal tube.

EXPLANATION OF PLATE II

Photograph of the principal components of the test-condenser for studying the condensation of Freon-12 inside a horizontal tube.

PLATE II



Auxiliaries

Boiler. The boiler used was steel cylinder 6-1/32-inch diameter, 20-inch long. Two steel flanges of 1-5/16-inch thick were covered with eight bolts on each end of the boiler. Steel was used to withstand the corrosive addition of Freon-12. A 20-1/4-inch long dry pipe was attached near the top of the boiler in order to prevent the liquid Freon from entering the superheater. Through one flange, a 5-KW, 220-volt, A. C., immersion heater was screwed into the lower half of the boiler so that the liquid Freon in the boiler would always immerse the heater. The power input to the heater was regulated by a powerstat of 230-volts, 50/60 cycles, output 0/270 volts, 28 amperes, 7.5 KVA. Pyrex glass liquid indicator was used to check the level of the liquid Freon inside the boiler. The boiler assembly is shown as in Fig. 6.

Superheater. This was newly designed and constructed. A copper tube of 1/2-inch I. D., 5/8-inch O. D., 4-feet long was insulated electrically with two layers of asbestos paper. Then 79 feet of Number 20 Chromel-A wire was wound around the tube on the asbestos paper. This heater was estimated to utilize a maximum of 1-KW when connected to 220-volt source.

After Condenser. An after condenser was used to liquidify all the uncondensed vapor leaving the test condenser. It was a counter-flow, single tube condenser in a vertical position. Two thermocouples were installed to measure temperature rise of the cooling water passing through it.

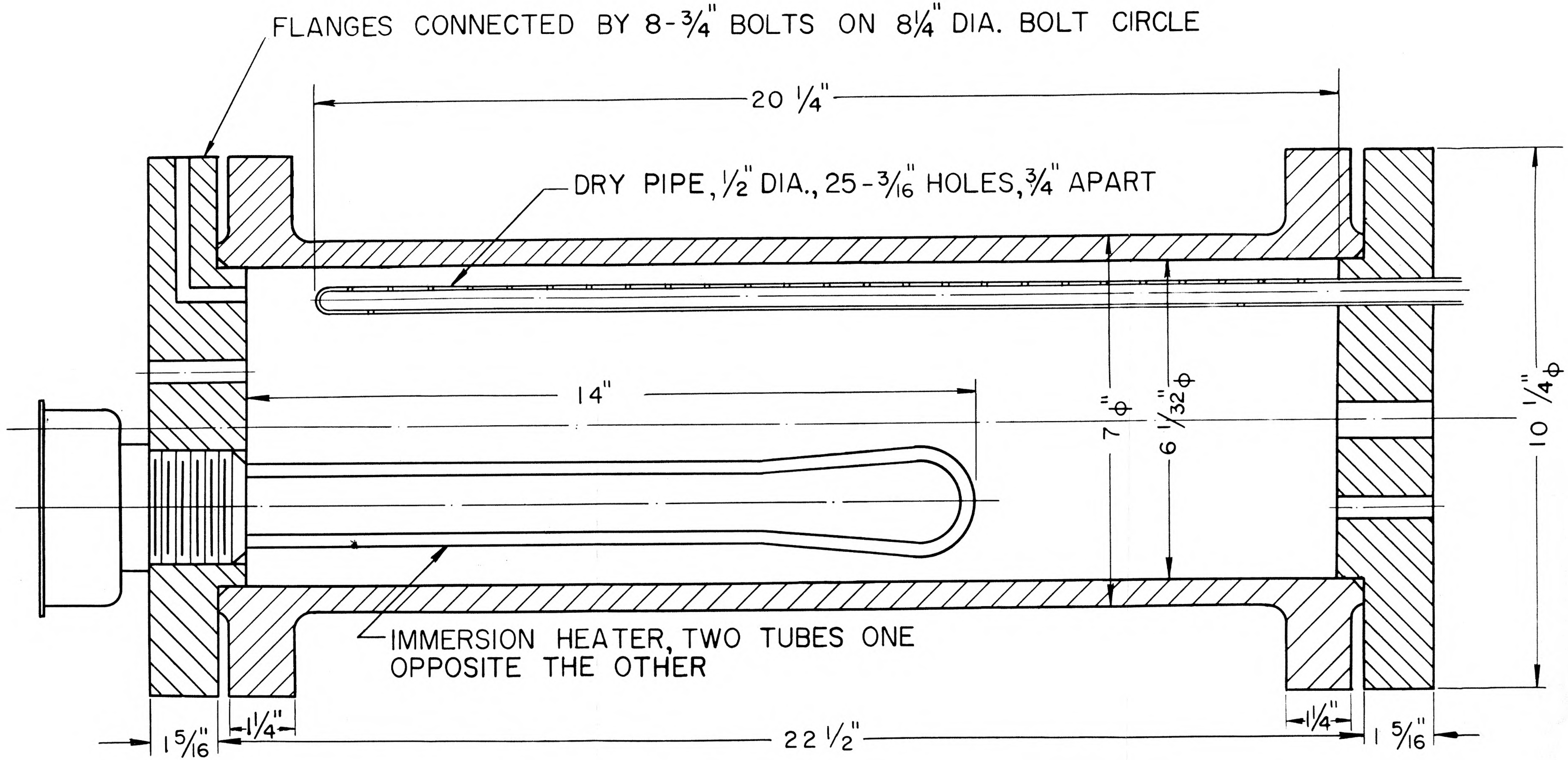


Fig. 6. Sectional view of the boiler assembly.

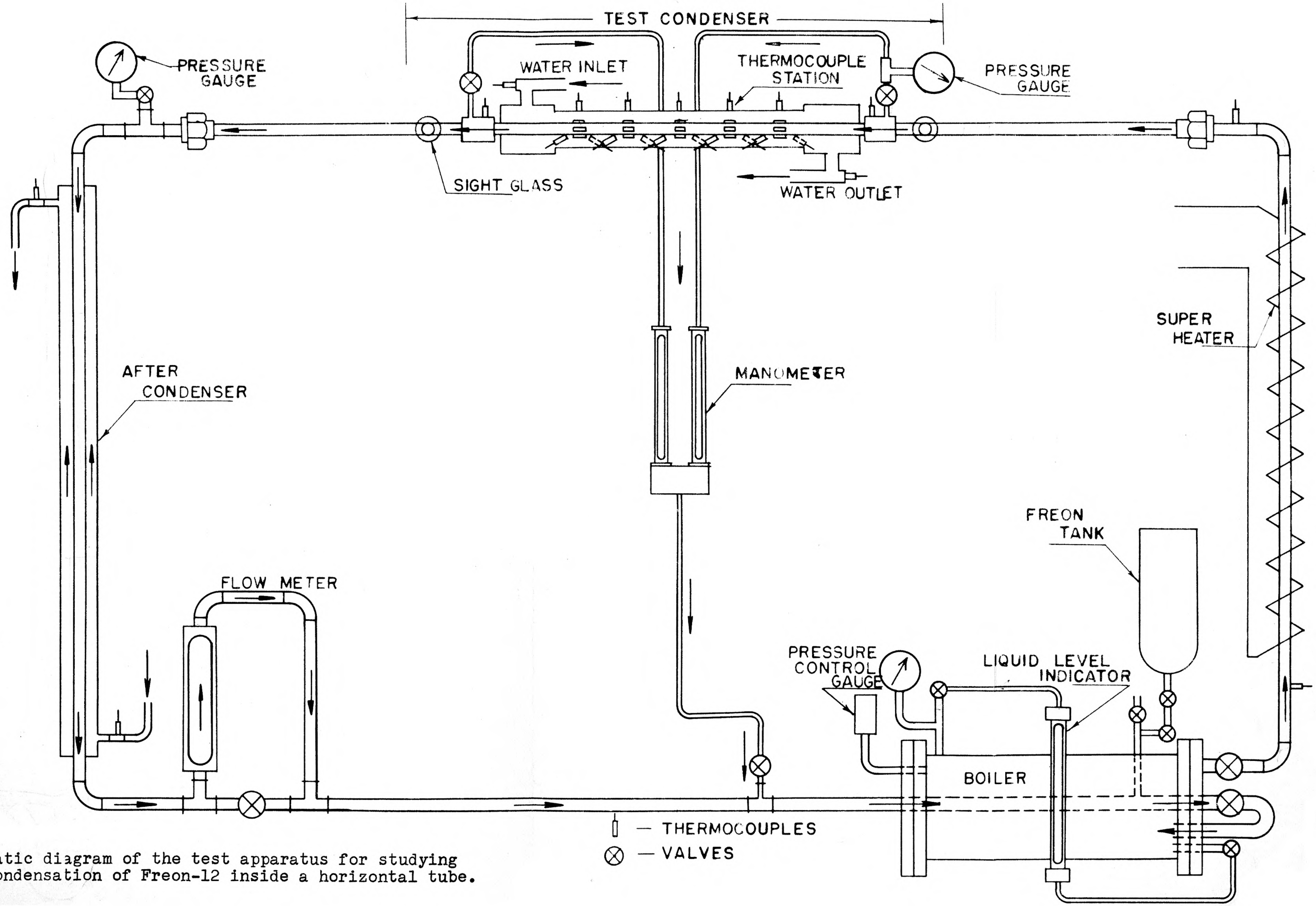


Fig. 7. Schematic diagram of the test apparatus for studying the condensation of Freon-12 inside a horizontal tube.

Flow Meter. A Brooks Rotameter was located right after the after condenser to measure the flow rate of liquid Freon in gallons per minute, with a minimum and maximum flow capacity of 0.05 and 0.55 gallons per minute, respectively. The specific gravity of the passing Freon was indicated to be 1.293. As this is the equivalent of 86°F saturated liquid Freon-12, the use of the after condenser was considered to be desirable, especially for high heat input at the boiler.

Pressure Gages. Bourdon-tube pressure gages with 0/300-psi range were used to measure the boiler pressure and the line pressure at the entrance of the after condenser. In order to set the boiler pressure within preset range, a bourdon-type, mercury operated pressure control with maximum setting of 220 psi was installed at the boiler. The minimum setting was to be about 60 to 70 psi, a little under the saturation pressure of Freon-12 at room temperature, so that the heat input to the boiler could be cut off in case of leakage or breakage at some parts of the system.

The set-up of the whole apparatus is shown by a schematic drawing in Fig. 7.

OPERATION OF THE APPARATUS

Testing Leakage

Because of its being a closed system any leakage from the system was deemed to be undesirable. The checking of leakage was carried out by the following methods:

1. Evacuation of the system with a vacuum pump: A Bourdon-type vacuum gage served as the indicator of leakage. It was not easy to find the location of leakages by this method, and the time for checking was unfavorably long.

2. Using compressed air: Compressed air of about 90 psi was available in the line. Soap water or glue was applied to connections in the apparatus to detect the leakage.

3. Special testing of the manometer: Because of its special shape in construction, the manometer was tested separately with 180 psi compressed air. Then the manometer was immersed in a water bath for checking.

Filling Freon-12

The entire apparatus was evacuated, and Freon-12 was supplied to the boiler from a tank. Air in connecting pipes was forced out by Freon before connecting them to the system. In doing this, the tank was heated and the condensers were cooled by water; thus Freon was forced into the system from the tank by reason of their temperature difference. After the desired amount of Freon was supplied to the boiler (this could be checked by means of the glass liquid indicator at the boiler) the pipe connection to the manometer circuit above the test condenser was slightly loosened to release some amount of Freon-12. This was done to remove air left in the apparatus. Since Freon-12 vapor is heavier than air this air was likely to accumulate at the highest portion of the apparatus, therefore, in the connecting pipe above the test-condenser.

Taking Data

Before each run, the test-condenser was made horizontal by the following procedure. The cooling water valve was opened to cool the test-condenser. This produced a thermal circuit which caused Freon condensate to accumulate along the lower part of the test-condenser-tube. This condensate was seen with the aid of the sight-glasses. When the flow ceased and a steady condition reached, the test-condenser-tube was so adjusted that the liquid level in both sight-glasses was the same. The measurement of levels was made with small scales through the glass window, with an estimated accuracy of 1/64-inches.

The flow rate of cooling water was so kept that the flow was turbulent. Two weigh tanks were provided to measure the flow rate of the cooling water. The heater load was set at a predetermined value by means of the powerstat. A separate powerstat was used for the superheater. The system was allowed to run for 3-hours to 6-hours to insure equilibrium conditions. Measurements of temperatures, Freon-12 flow rate, cooling water flow rate, and power input to the heaters were then made. Last, valves for the manometer circuit were opened and the manometer was read.
READING

Notes

During the experiments, some difficulties were experienced. The thermocouple installed in the tube wall on the top, 12-inch from the entrance, revealed unexplicable low readings. This is shown in Fig. 8 and 9 and Fig. 10 and 11 by the dotted line.

Thermocouple connections were checked and no short was found. The direction of the cooling water entrance was changed 180° from its initial direction, but the same temperature pattern continued to prevail. After taking data, for some runs, the heat input and the cooling water were shut off, and temperatures of the tube wall were checked at 10-minute intervals. These readings showed a decreasing temperature, but all of the thermocouple readings were almost the same. Hence the particular low temperature for this point was considered to be a feature of condensation for this size of tube. It seemed that the maximum resistance to heat flow existed some distance from the entrance section due to the slowing down of the vapor velocity. This phenomenan was reported in the paper of Patel (11), and Spencer (13). The experiment was continued till Run 42; furthermore, test runs were carried out to check all of the thermocouples in the test section. In these test runs, hot water was used instead of Freon vapor. When no cooling water was used, tube wall temperature was practially the same within the test section. (Table I)

Table 1. Order of agreement among thermocouples when hot water of uniform temperature was circulated instead of Freon-12, and cooling water was shut off.

Thermocouple position inches from entrance	0	6	12	18	24
Wall temperature, top, °F	154.8	154.5	154.4	154.3	154.8
Wall temperature, middle, °F	154.8	154.2	154.4	154.3	154.8
Wall temperature, bottom, °F	154.8	154.5	154.5	154.4	155.2

When cooling water passed through the test section, the top tube temperature pattern again was the same as that obtained when condensing vapor. (Table 2)

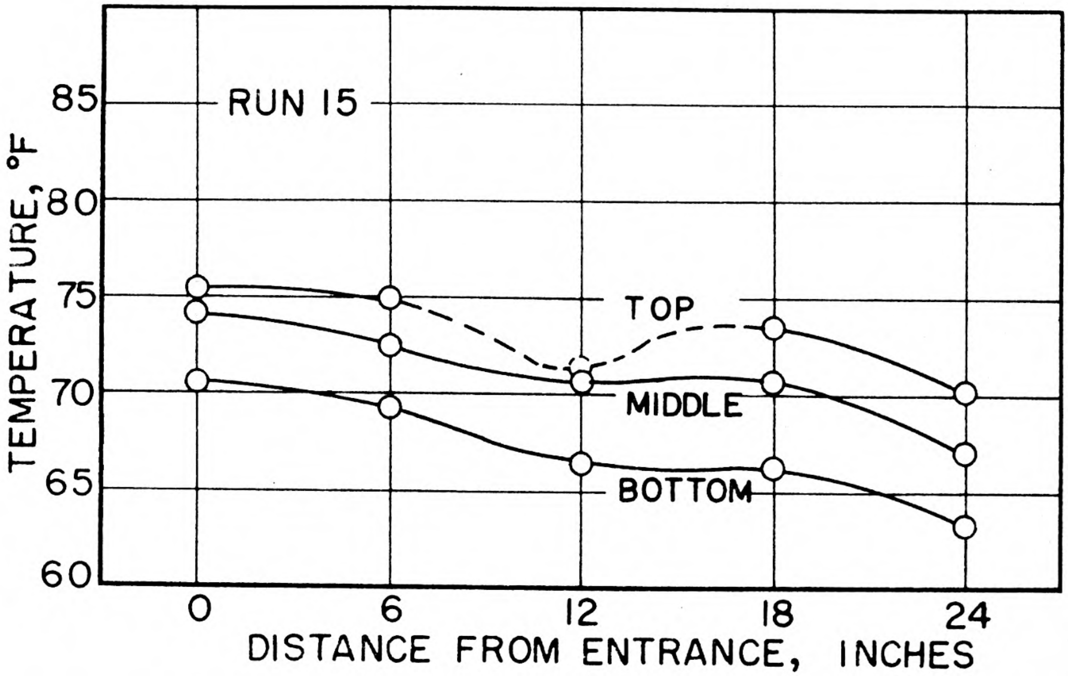


Fig. 8. Temperature distribution of the tube wall along the tube axis, Q/A = 18245 Btu/ft^2.

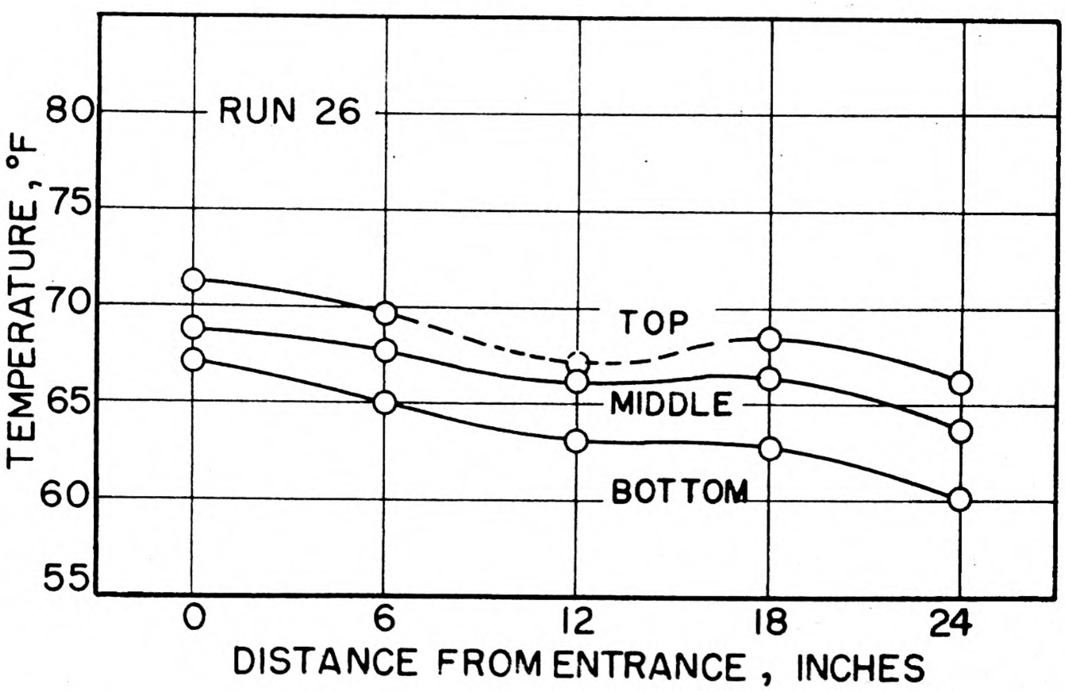


Fig. 9. Temperature distribution of tube wall along the tube axis, Q/A = 11131 Btu/ft^2.

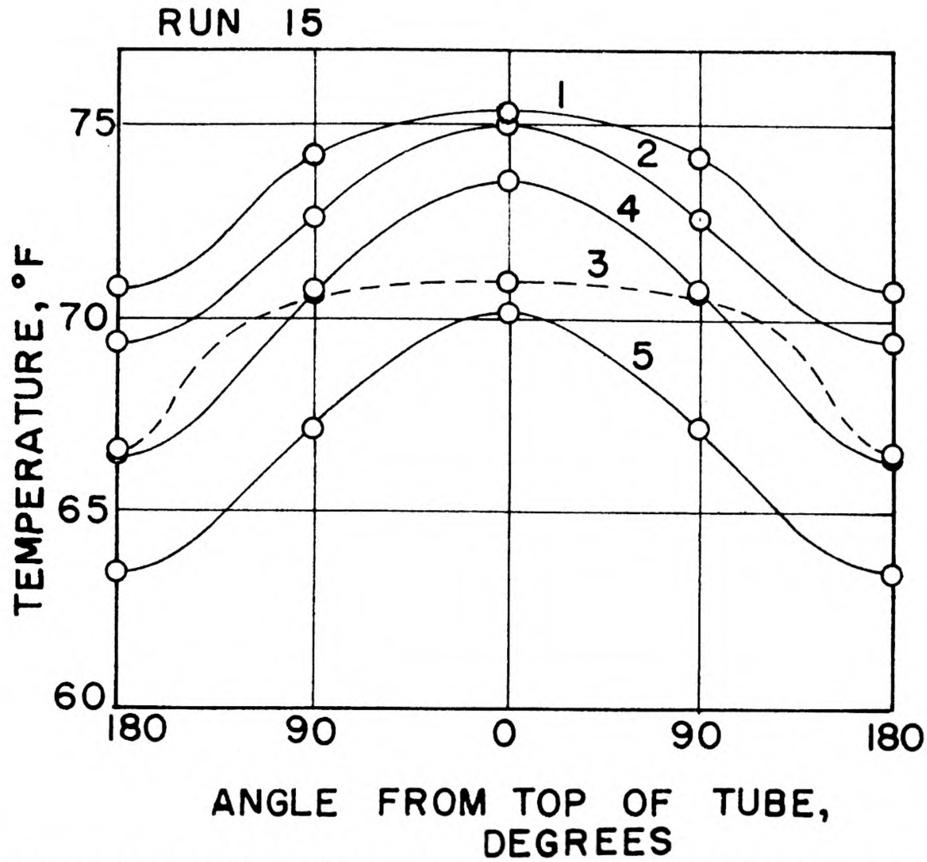


Fig. 10. Temperature distribution of tube wall at the cross-section 1, 2, 3, 4 and 5, corresponding to the distance from the entrance 0, 6, 12, 18 and 24 inches, respectively.

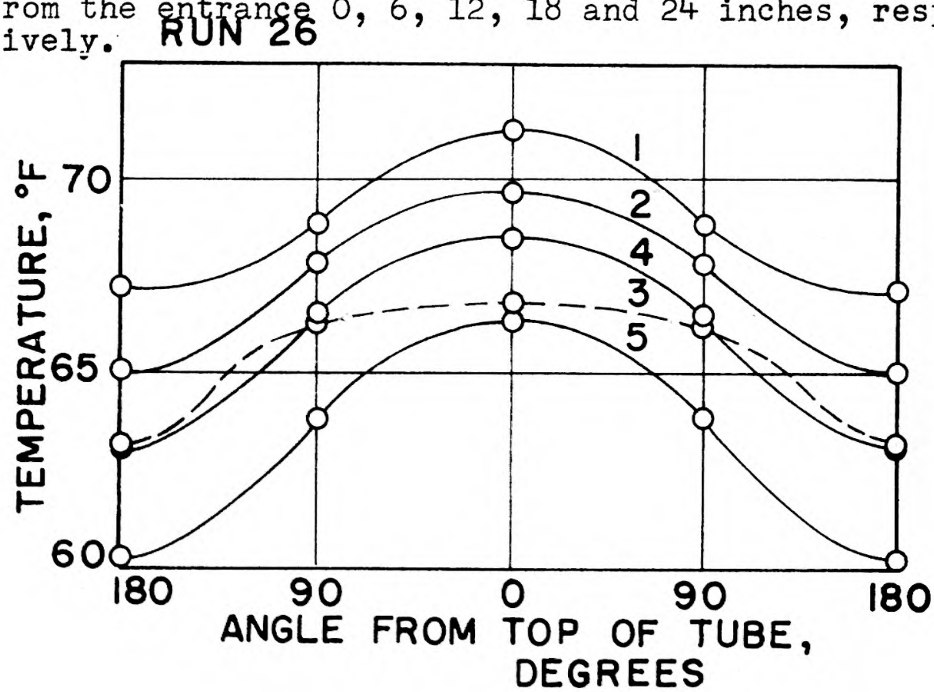


Fig. 11. Temperature distribution of tube wall at the cross-section 1, 2, 3, 4 and 5, corresponding to the distance from the entrance 0, 6, 12, 18 and 24 inches, respectively.

Table 2. Thermocouple calibration readings using hot water instead of Freon-12 vapor, with cooling of the test condenser tube.

Thermocouple position, inches from entrance	0	6	12	18	24
Wall temperature, top	160.4	150.8	132.3	135.7	117.7
Wall temperature, middle	158.5	150	142.3	129.5	115.1
Wall temperature, bottom	160.8	148.8	139.0	127.8	114.3

It had been made clear by these test-runs that something was wrong with the thermocouple located on the top of tube, 12-inch from entrance. As the last check, the whole test tube was turned upside down, and the top thermocouples were brought to the bottom position. Thus, if the thermocouples originally on the bottom side, which was considered to function well, would show the same temperature pattern as the top thermocouples did, then all the thermocouples would have been sound, and the particular temperature pattern having prevailed would be considered to be a feature of film condensation inside horizontal tubes. When the original top thermocouples were moved to the bottom they showed the same temperature pattern as that obtained when they were on top. The special behavior of the single thermocouple in position 12 was charged to short circuits inside the condenser tube. When the entire tube was at the same temperature, the emf reading was all the same. While there was temperature gradient in the tube, there occurred short circuits or opposing emf to render a wrong reading. Thus an accurate top wall temperature distribution was not obtainable with this experiment; it seemed most likely that the temperature patterns for all three grooves were the same and parallel with each other.

Freon-12 condensate could be seen by means of sight-glasses on both ends of the test condenser. It was found that the measurement of condensate levels was practically impossible because of wave motion and periodical fluctuations of the condensate flow. The test condenser was checked to assure its being horizontal. The valve which was located next to the line pressure gage and on the same level as the test condenser was taken away because this valve was considered to offer a high resistance to the condensate flow. The fluctuation of the flow still prevailed. No final explanation has been reached for the pattern of condensate flow under considerable vapor velocity, but it seemed to be reasonable to attribute the wave motion of condensate in the bottom of the tube to the combined effect of vapor shear force and the down-coming condensate around the inside tube surface.

In measuring temperatures of the cooling water, it was found that there was no suitable point to represent cooling water temperature at a section. The buoyant force caused a circulation, and as a result set-up a forward-moving spiral flow in the annular space. The thermocouple readings were taken individually. Along the same row, thermocouples did not show a steady increase of temperature in the direction of flow but fluctuated considerably. An average of three temperature readings at a section seemed unable to represent the average bulk temperature at the section.

EXPLANATION OF PLATE III

Photographs showing the sight-glasses at the entrance and the exit of the test condenser.

Fig. 1. Sight-glass at the entrance, showing no condensate stream in the bottom of the tube.

Fig. 2. Sight-glass at the exit, showing the accumulated condensate in the bottom of the tube 5/32 inches high.

PLATE III

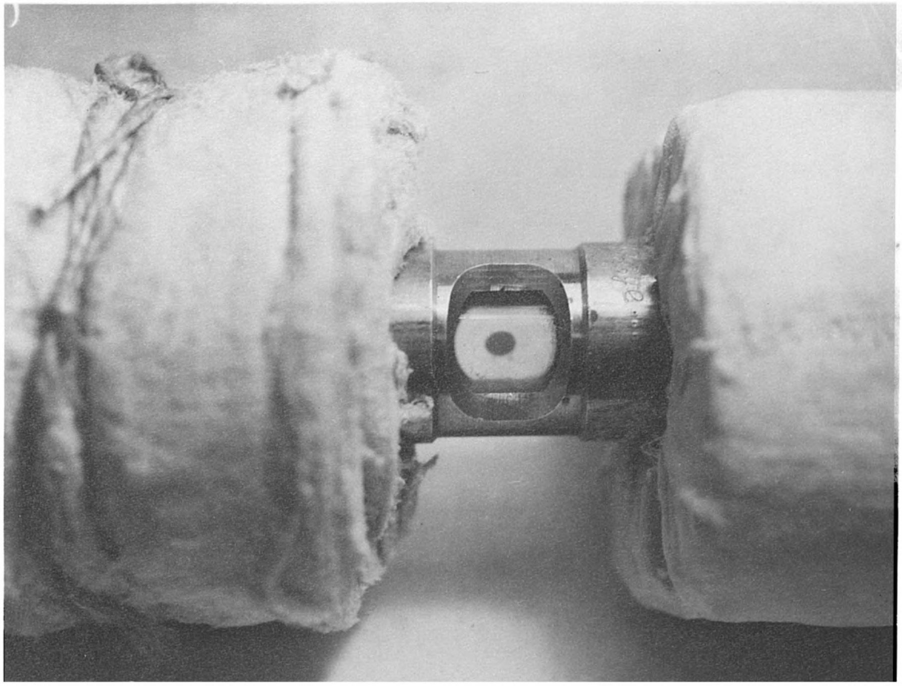


Fig. 1.

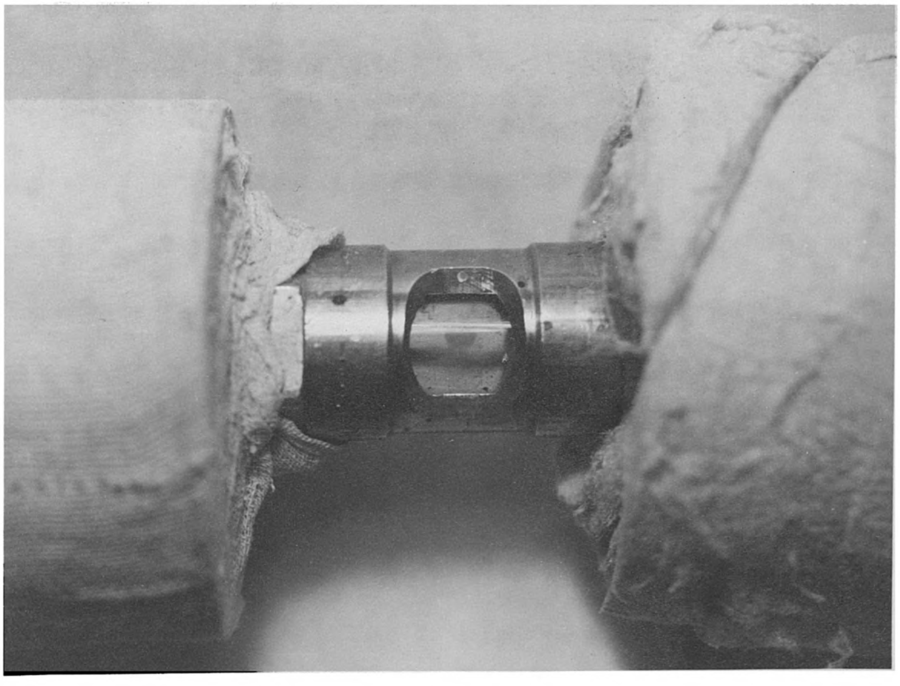


Fig. 2.

RESULT AND DISCUSSION

Data

A total of 42 runs was made, and recorded on data sheets. They were rearranged and listed in Table 4, in Appendix. A study on the data revealed that the boiler pressure is some function of heat input, and the heat input was limited by the setting of the pressure control. Tube surface temperatures were based on the data taken from thermocouples in the middle groove, since a thermocouple in the top groove was proved to be in error. It was obvious that a high value of the film coefficient would be obtained if the top wall temperature was taken as the average tube surface temperature; and that the coefficient of heat transfer would not appropriately be called "film coefficient", if the bottom wall temperature was used. The temperature rise of the cooling water was rather small because of the high flow rate of the cooling water.

Facts Observed

The following observations were made during the test:

1. From visual studies through the sight-glasses, there seemed two categories of flow patterns at the entrance section of the test-condenser. One was that the Freon-12 flow at this section was all dry; the other, partially condensed. The first case indicated that dry vapor entered the test condenser, and that condensate was carried forward by vapor friction. The second condition was considered to be partly due to the low

quality of the vapor from the boiler, and partly due to the pre-condensation which took place on the cold wall of the tube 3-feet long ahead of the test-condenser. The second case was observed when the boiler was nearly full; it seemed true therefore that moisture laden vapor was carried into the test section from the boiler.

For the later case, liquid Freon occupied the lower part of the tube cross-section, and its flow was turbulent with more or less pulsation. When more heat was added at the boiler or the superheater, the flow rate increased, and the condensate surface at entrance became concave, probably due to the interfacial friction of the vapor.

2. At the exit section of the test-tube, there was no appreciable difference in the flow pattern such as existed at the entrance. Generally speaking, however, wet entrance of Freon-12 indicated higher liquid level than that of dry entrance for the same heat input. Common to both cases, slugging as well as wave motion occurred in the condensate flow. Sometimes the slugging was so severe that the liquid level reached the top of the tube.

3. On account of pulsation in condensate flow, U-type differential manometer used in author's experiments was not reliable. Except in the case of dry entrance of vapor and small heat input, the pulsation in the manometer liquid level made it impossible to take accurate readings.

4. It was also noted that in the case of wet entrance, there was a much larger flow rate and much more fluctuation in the Rotameter readings than in the case of dry entrance. In view of

this and the other observations already stated, the calculated velocity for the wet entrance based on the measured flow rate was considered unreliable.

Table 3. Flow and heat transfer data for condensing Freon-12. Case 1 - Liquid present at entrance. Case 2 - Freon at entrance in form of dry vapor.

Types of Entrance	Run No.	Condensate Level, Entrance, inch	Condensate Average Level, Exit, Inch	Heat Input K. W.	Freon Flow Rate, lb/hr.	Calculated Vapor Velocity ft/sec
Wet	38	3/32	1/4	1.50	239.4	13.74
Dry	6	0	5/32	1.53	61.5	3.68

From Table 3, it is seen that, with the nearly same heat input, the calculated vapor velocity for wet entrance is about four times that of dry entrance. If there is no way of telling what the height and bulk velocity of the bottom condensate are, the vapor velocity can not be calculated without bold assumptions. As has been stated the wave motion and fluctuation of the stream of condensate at the bottom of the tube introduced difficulties in the way of measuring the condensate height and pressure drop across the test section with the present set-up. The vapor velocity could not be calculated from the measured flow rate of Freon-12 when condensate accumulated at the bottom of the tube.

5. Freon vapor temperature was constant in the test section for most of runs. In case of wet entrance there appeared a slight temperature drop across the test section. Splashing of bottom condensate seemed to have some effect on this temperature drop. On the other hand, the temperature drop can be attributed to the total condensing of vapor, due to low quality of vapor at entrance.

For dry entrance, Freon-12 was partially condensed. There were descrepancies in saturation temperature and pressure readings, with pressures higher 0-3 psi.

Calculations

A sample calculation is shown in the Appendix. Arithmatic mean of temperature based on thermocouples in the middle groove was used as the tube wall temperature at a distance 0.09-inch from the vapor side tube surface. From this and the measured bulk temperature of Freon-12, the temperature drop across the tube was calculated. The heat transferred in the test section was determined by the cooling water rate and its corresponding temperature rise. Mass velocity was calculated by the equation of continuity,

$$W_{(flow\ meter)} = W_{(entrance)} \qquad (3)$$

$W_{(flow\ meter)}$ was determined by the measured value of the flow rate of Freon-12 and a Freon-liquid density with specific gravity of 1.293. The temperature of the film with which physical properties of Freon-12 was evaluated was

$$t_f = t_v - 3/4\ (t_v - t_w)$$

Throughout the calculations, the one-dimensional heat flow equation $Q = -KA \frac{dx}{dt}$ was used. A table calculator was used in most of the calculations; limited physical properties of Freon-12 were plotted in charts and interpolation was made from the curves.

Correlation of Data

As there were two types of flow at the entrance, the correlations of calculated data were made for both dry and wet entrances. A correlation between Q/A and ΔT in logarithmic coordinates showed a straight line, as shown in Fig. 12. The ranges of Q/A and ΔT were 5700 to 21000 Btu/(hr)(ft^2) and 17 to 53^oF respectively. At a high value of ΔT, Q/A for wet entrance indicated lower value than that for dry entrance. In the h_m vs. ΔT correlation, dry entrance showed a decreasing h_m with an increasing ΔT for values of ΔT larger than 35^oF (Fig. 13); wet entrance shows an almost constant value of h_m for all values at ΔT. Generally speaking, the trend of the curves indicates decreasing h_m. The correlation between h_m and G, mass velocity of the Freon-12 condensate, indicated a slightly increasing h_m with increasing G for wet entrance (Fig. 14); dry entrance shows an optimum value of G approximately 50000 lb/(hr)(ft^2) for which h_m becomes a maximum. For wet entrance a large part of the heat transferred from Freon-12 is conducted through the condensate stream, since an increase of G increases the Reynolds number and hence the turbulance of the condensate stream. The results in this case is an increase in h_m with an increase in G. For dry entrance, on the other hand, an increase in G will increase the vapor velocity and so the rate of heat transfer through the condensate film. It is to be noted that an increase in G will accompany an increase in the depth of the condensate stream, the heat transfer rate through which is lower than that through the condensate film.

Hence overall heat transfer rate decreases when G passes the optimum value. As a result, h_m increases at first, then decreases with increasing G; finally, h_m approaches asymptotically to the value for the case of wet entrance.

For film condensation of the outside of a single horizontal tube, the following dimensionless equations can be used to calculate the film coefficient of heat transfer (McAdams, 10):

$$h_m = 0.725\ (K_f^3 \lambda \rho_f^2 g / D \mu_f \Delta t)^{\frac{1}{4}} \tag{5}$$

and

$$h_m \left[\frac{\mu_f^2}{K_f^3 \rho_f^2 g}\right]^{\frac{1}{3}} = 1.51 \left[\frac{4\Gamma}{\mu_f}\right]^{\frac{1}{3}} \tag{6}$$

For the case of condensation inside horizontal tubes, the tube is partially filled with flowing condensate. Film condensation takes place on the upper part of the inner tube wall. Chaddock (6) assumed that a film which formed over an angle ψ inside the tube would be identical with a film formed over the same angle ψ on the outside of the tube. Chaddock treated a case of negligible vapor velocity, and made the following assumptions for the derivation of equations:

1. Streamline motion exists throughout the thickness of the continuous film of condensation.

2. The temperature difference between vapor and wall is constant at all points.

3. The total thermal resistance is in the film of condensate, through which the latent heat of condensation is conducted.

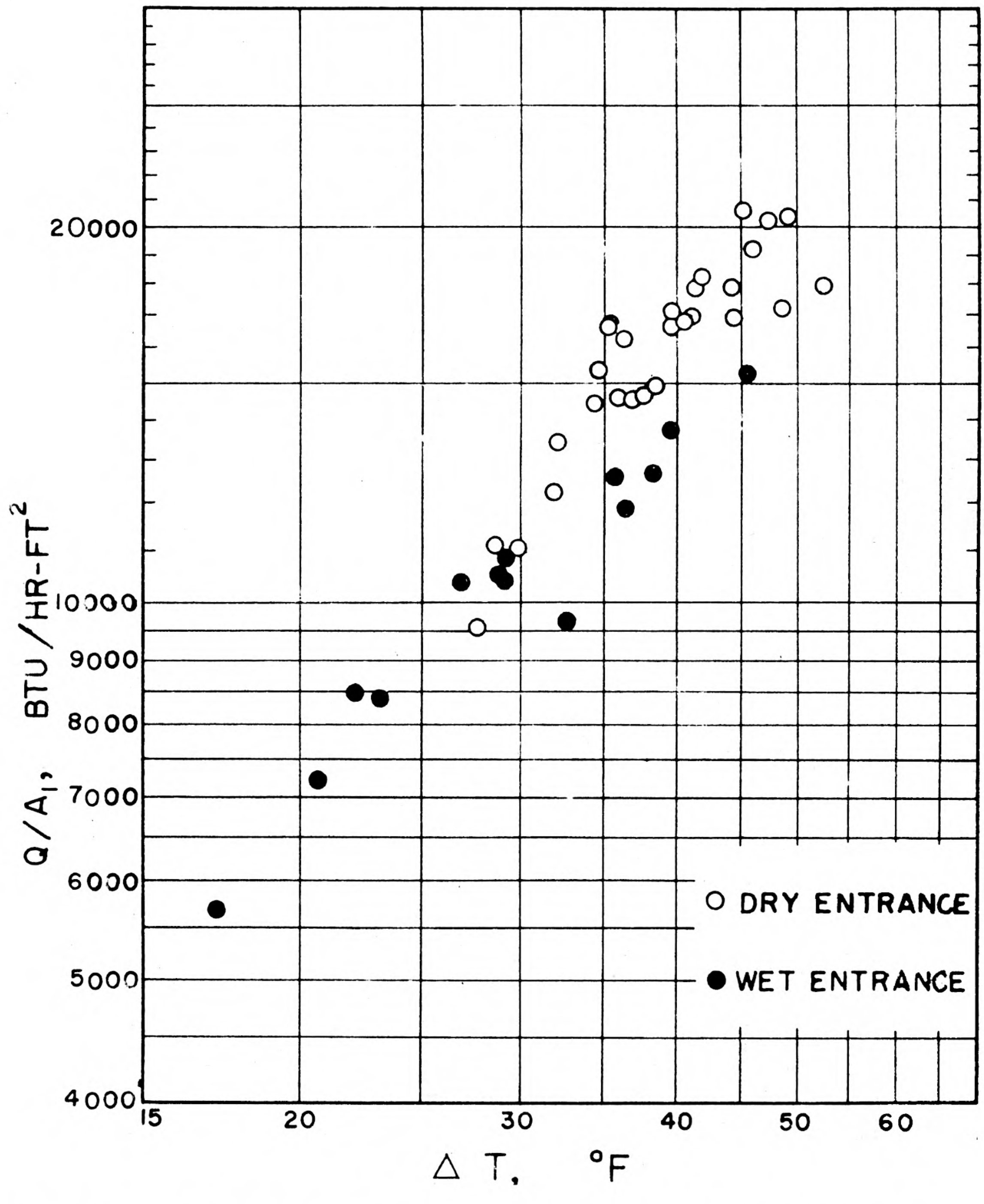


Fig. 12. Relation of heat flux, Q/A_1, to the temperature difference between the condensing surface and Freon-12 vapor, ΔT, during the condensation of Freon-12 inside a horizontal tube.

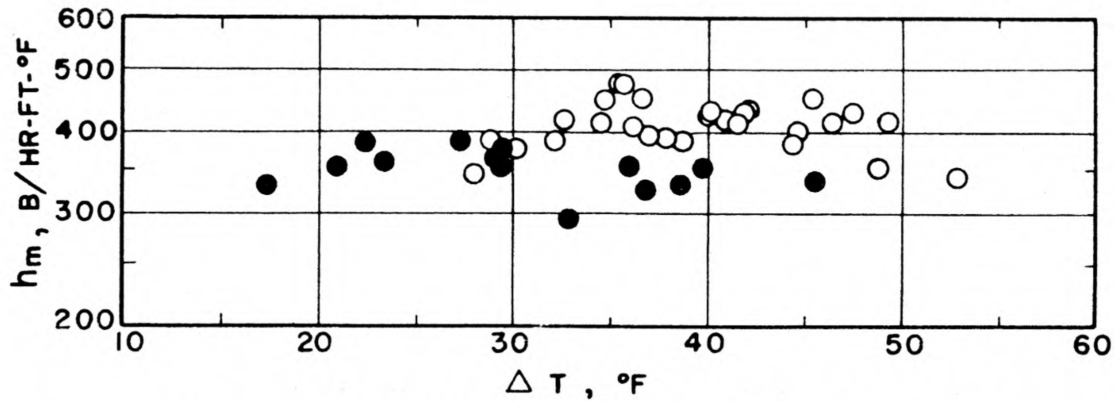


Fig. 13. Relation of the average coefficient of heat transfer, h_m, to the temperature difference between the condensing surface and Freon-12 vapor, ΔT, during the condensation of Freon-12 inside a horizontal tube.

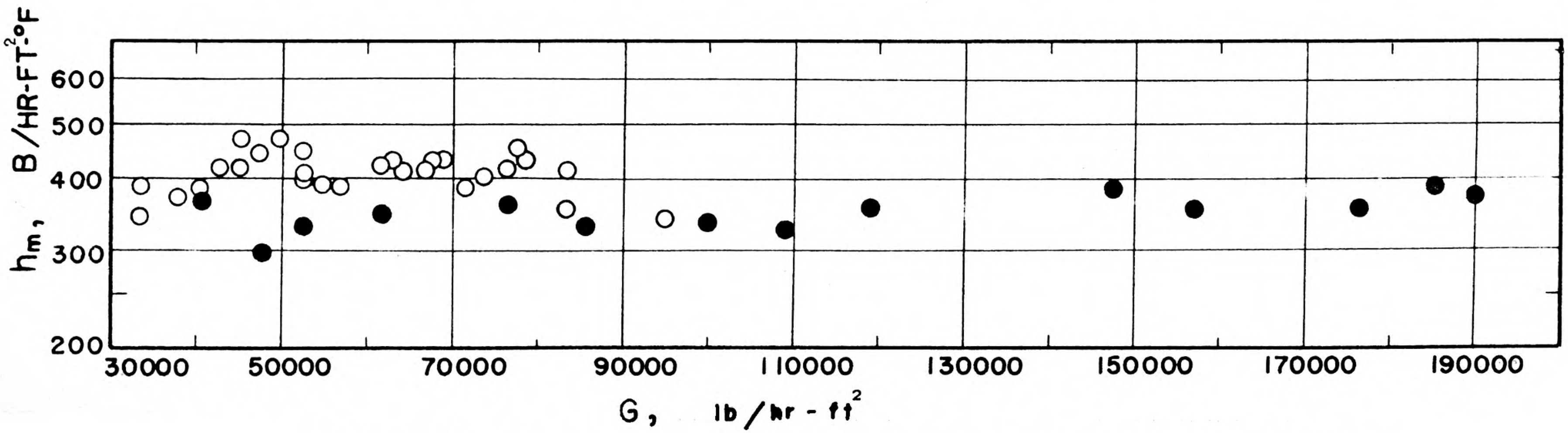


Fig. 14. Relation of the average coefficient of heat transfer, h_m, to the mass velocity of condensate, G, during the condensation of Freon-12 inside a horizontal tube.

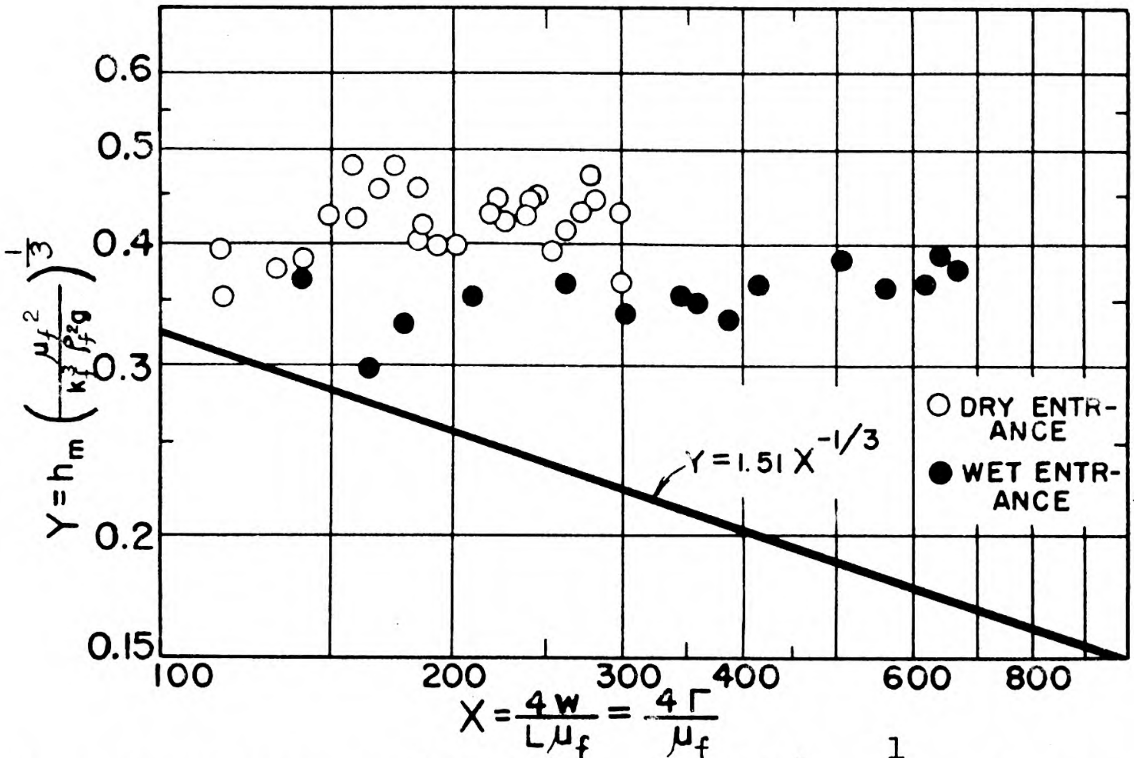


Fig. 15. Relation of $\frac{4\Gamma}{\mu_f}$, and $h_m \left(\frac{\mu_f^2}{K_f^3\ \rho_f^2\ \ g}\right)^{\frac{1}{3}}$, using the observed data during the condensation of Freon-12 inside a horizontal tube.

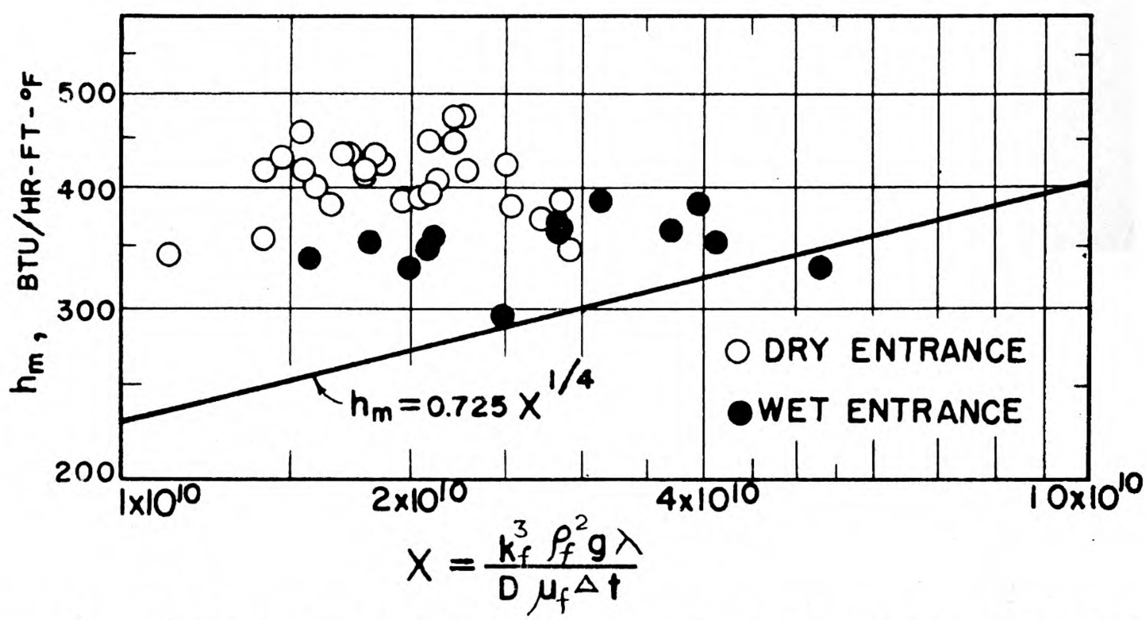


Fig. 16. Relation of the average coefficient of heat transfer, h_m, $\left(\frac{K_f^3\ \rho_f^2 \lambda g}{D\mu_f\ \ \Delta t}\right)$, using the observed data during the condensation of Freon-12 inside a horizontal tube.

4. The physical properties of the condensate are independent of temperature.

5. The force of gravity alone causes the flow of condensate over the surface.

If one could find the angle ψ covered by the condensing film, the average heat transfer coefficient over this angle could be determined. (Appendix)

$$h_{m\psi} = \beta \left[\frac{K_f^3 \rho_f^2 \lambda g}{\mu_f} \right]^{\frac{1}{4}} \left[\frac{1}{D \Delta t} \right]^{\frac{1}{4}} \tag{7}$$

where $$\beta = \frac{0.9036}{\psi} \int_0^{\psi} \frac{d\theta}{S^{\frac{1}{4}}} \tag{8}$$

and $$h_{m\psi} = (2\psi)^{\frac{1}{3}} \beta^{\frac{4}{3}} \left[\frac{K_f^3 \rho_f^2 g}{\mu_f^2} \right]^{\frac{1}{3}} \left[\frac{4\Gamma}{\mu_f} \right]^{\frac{1}{3}} \tag{9}$$

If one inserts the value of $\psi = 2\pi$ in the above equations, the resulting expressions are the same as those for condensation on the outside of a horizontal tube, equations (5) and (6). In the author's experiments, there existed a non-negligible vapor velocity at all times. For the case of wet entrance, the condensate stream filled about half of the tube. Although it was expected that the foregoing equations for condensation on a horizontal tube would not be applicable to the author's experiments, they are of comparative interest as representing the limiting case wherein there is no horizontal flow, therefore no vapor friction and also no accumulation of condensate at the bottom of the tube. Hence, in Fig. 16, h_m is correlated with $X = \frac{K_f^3 \rho_f^2 \lambda g}{D \mu_f \Delta t}$ for both "dry" and "wet" entrance of Freon-vapor. Equation (5) is represented by a straight line in the Fig. 16, for comparison. The following are to be noted:

(1) In $X = \frac{K_f^3 \rho_f^2 \lambda g}{D \mu_f \Delta t}$ the variations of the physical properties are relatively small with respect to the temperature change. The temperature drop across the film condensate, Δt, may be regarded as the controlling factor in this group. In the author's experiment, Δt varied from 17 to 53°F. As a result, the correlation resembles that of h_m vs. Δt, with large Δt corresponding to small X, and small Δt to large X.

(2) Small values of h_m for the case of wet entrance are of the order expected, since the angle ψ covered by the condensate film is the smaller for the wet entrance.

(3) The average heat transfer coefficient which includes the heat transfer through the condensate stream has a value 0 to 70 per cent higher than the predicted value of the average heat transfer coefficient for the condensation on a horizontal tube.

(4) There seems to exist an optimum value of X for which h_m becomes a maximum. For the values $X = 5 \cdot 10^{10}$ the heat transfer coefficient of condensation inside a horizontal tube seems to fall down below the predicted value by the equation (4).

In the correlation between $h_m \left[\frac{\mu_f^2}{K_f^3 \rho_f^2 g} \right]^{\frac{1}{3}}$ and

$\frac{4\Gamma}{\mu_f} = \frac{4w}{L\mu_f}$, the author used L = 2.5 feet, the effective length of the test tube, and the values of w calculated from the flow-meter reading of Freon-12 (Appendix).

As stated before, for the case of wet entrance the calculated value of w based on the flow-meter reading does not indicate the true amount of condensation which takes place in the test condenser but includes some condensate carried into the condenser from the boiler. Hence for the case of wet entrance, the observed value of $\frac{4w}{L\mu_f}$, was greater than the Reynolds number of the film condensate at the bottom of the condensing surface of tube (level of contact with condensate stream). In the correlation it is noted that $Y = h_m\left[\frac{\mu_f^2}{K_f^3\rho^2{}_f\ g}\right]^{\frac{1}{3}}$ increases with increase in $x = \frac{4\Gamma}{\mu_f}$ for both dry and wet entrances. This trend is opposite to that indicated by equation (6), for which Y decreases as x increases. It is of interest to compare this result with the data of Carpenter (5) and Colburn (7) for condensation in the presence of high vapor velocities. The Fig. 17 (Appendix) is reproduced for the purpose of comparison. Carpenter's data were based on the condensation of pure vapors of stream, methanol, ethanol, toluene, and trichlosethylene inside a vertical tube 0.459 in I. D. and eight feet long with inlet vapor velocities up to 500 feet per sec. Carpenter et al (5) obtained an equation for the local value of the condensing film coefficient, h_f,

$$\frac{h_f\mu_f}{K_f\rho_f^{\frac{1}{2}}} = 0.043\ \left(\frac{C_{p_f}\mu_f}{k_f}\right)^{\frac{1}{2}}\ F^{\frac{1}{2}} \qquad (10)$$

by semi-theoretical reasoning, and as a conclusion, they stated:

> Whereas the flow of the condensate layer tends to be essentially laminar in the absence of vapor velocity up to Reynolds numbers of 2000, in the presence of high vapor friction the layer becomes turbulent at a Reynolds number around 240. It is thus found that the hydrodynamics of a free flowing layer, and one with a large frictional force are quite different.

Although in the case of condensation in a horizontal tube, the mechanism of the condensation differs from that in vertical tubes, the similarity between the trend of Carpenter's data and that of the author's data seems to indicate that there exists some similarities between condensation in vertical and horizontal tubes in the presence of high vapor velocity. As stated before, when the interfacial friction of the vapor increases, the surface of the condensate stream, which is a plane when there is no vapor shear force, tends to become concave. In the case of the horizontal tube, if the vapor velocity continues to increase, and the shear stress becomes big compared with the gravity force, the cross-section of the condensate stream assumes the shape of a crescent moon, and in the extreme case the cross-section of the interface of condensing vapor and the horizontal stream of condensate approaches a circle concentric with the tube cross-section. Thus with high vapor velocity, the condensation in the horizontal tube approaches to that in vertical tubes. The author considers this is an explanation for the similarity between Fig. 15 and Fig. 17.

CONCLUSIONS AND SUGGESTIONS

Conclusions

When the vapor velocity cannot be neglected in the condensation of vapor inside a horizontal tube, the mechanism of condensation seems to differ from that for the case wherein vapor velocity is negligible. The difference in the level of the condensate stream between the entrance and the exit of the condenser in the former case is likely to be greater than that in the latter case. For the case of negligible vapor velocity, it was noted (6) that the difference in the level of the condensate stream at the exit and entrance of the condenser is quite small. A greater difference in this level can be attributed to the vapor shear force on the interface of the condensing vapor and the condensate stream. With an extremely high vapor velocity, the gravity force is small compared with the shear force, and the case approaches that of condensation in a vertical tube.

The average rate of heat transfer for condensation inside a horizontal tube within the range of the author's experiments, boiler pressure 85 to 181 psi, and Freon flow 0.065 to 0.230 G.P.M., indicates a value 0 to 70 per cent higher than that predicted by equation (5). Equation (6), which is for the laminar flow of
p46
film condensate, seems inapplicable because the author's data gives an entirely different trend from that given by the equation (6). On the other hand, the correlation of the data indicates a similarity with the case of turbulent condensate flow in the vertical tube. The average heat transfer coefficient for the

condensation of Freon-12 inside a horizontal tube was found to be 300 to 500 Btu/(hr)(ft^2)(^oF).

There were two kinds of flow patterns at the entrance section: in one, dry vapor entered the condenser; in the other, wet vapor entered the condenser. Wave motion accompanied the flow of accumulated condensate as the velocity wasincreased from a low velocity to a very high velocity, this flow assumes a sequence of patterns which can be classified according to Bergelin, et al (3), as "stratified flow", wave flow" and "slugging flow", respectively.

A considerable temperature difference existed around the circumference and along the axis of the test condenser. The maximum temperature was on the top of the entrance section to the condenser; the minimum, at the bottom of the exit-section from the condenser. There seemed to be no consistent positions in the tube which represented the average tube-wall temperature. The temperature drop across the film condensate of Freon-12, Δt, which is considered to be one of the controlling factors of condensation, was rather high, ranging from 17 to 53^oF, the heat flux, Q/A, ranged from 5600 to 21000 Btu/(ft^2).

Suggestions

With the installation of the boiler liquid indicator, it seems that an error is introduced in the reading of the liquid level when the boiler is under operation. The Freon-12 in the apparatus is saturated, therefore, heating or cooling by the outside agent will change the Freon from liquid to vapor, or from vapor to liquid. During the run, the temperature inside the boiler was higher than that in the liquid indicator. Hence, continuous condensation of Freon-12 occurred in the liquid indicator. Sometimes it was not certain whether the indicator showed the liquid level in the boiler or not because cooling of the indicator increased the level of the liquid column in the indicator. Shortening the lead tubes connecting the boiler to the liquid indicator, or better, installing a sight glass on the boiler wall might be the solution.

In the calculation of the heat transfer coefficient in author's present apparatus, two factors were controlling. The one was the temperature rise of the cooling water; the other was the temperature drop across the condensate film. The temperature rise of the cooling water was the more important. Hence, it would be desirable to use thermocouple piles to enlarge the emf readings of the temperature rise of the cooling water through the test condenser. Especially with a high rate of cooling water flow and so a small temperature rise, an error in the temperature reading had a considerable effect on the calculated data.

It would be of great interest to investigate further the differences in the two types of flow at the entrance to the condenser. For the wet entrance, a measurement of the vapor velocity is desirable. At the same time, the measurement of the pressure drop across the test-section needs to be improved. Since the pulsations of pressure accompanied by the wave motion of the condensate stream prevent the pressure drop from being accurately measured, the installation of a damper for pulsations or the use of a different kind of manometer are recommended.

With the same temperature drop across the condensate film a higher value of the average heat transfer coefficient is apparently obtainable with an inclined tube than with a horizontal tube. If the condensation takes place in an inclined tube, the bulk velocity of the condensate stream will be higher than that in a horizontal tube; and the level of the condensate stream will be low; hence the area for the transfer of heat through the film of condensate rather than through the condensate stream will be the greater in the first case. As a result, there will be higher average heat transfer coefficient for an inclined tube than for a horizontal tube.

It was said that if one could measure the film thickness of the condensate accurately, the experiments would be simplified, and the prediction of the results could be made accurately. For this purpose, it might be possible to utilize a radioactive isotope. So far the utility of such an isotope in this field is still unknown, but radioactive isotopes have been used in industry to accurately control the thickness of metal sheets, and in the refinement of oil. If Freon-12 can be made radioactive, then the measurement of the thickness of the film on the tube wall seems not impossible. The strength of the radioactivity, will be proportional to the film thickness and measurements of the local intensity of radioactivity will render the information on this thickness.

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LITERATURE CITED

(1) Air Conditioning Refrigerating Data Book--Design Volume 1955-56.

(2) Baker, E. M. and A. C. Mueller.
Condensation of vapors on a horizontal tube. Trans. Amer. Inti. of Chem. Engg., 33:531-558. 1937.

(3) Bergelin, O. P., P. K. Kegel, F. G. Carpenter and Carl Gazley, Jr.
Co-current Gas-liquid flow. Heat Transfer and Fluid Mechanics Institute, A.S.M.E. 1949.

(4) Bromley, L. A.
Effect of temperature variation around a horizontal condenser tube. Indus. Engg. Chem., 44:2966. 1952.

(5) Carpenter, E. F. and A. P. Colburn.
The effect of vapor velocity on condensation inside tubes. Proceeding of the General Discussion of Heat Transfer, Institution of Mechanical Engineers, London, and American Society of Mechanical Engineers. New York: 1951.

(6) Chaddock, J. B.
Film condensation of vapor in horizontal tubes. Sc.D. Thesis. Massachusetts Institute of Technology. 1955.

(7) Colburn, A. P. Proc 1951 Vol 164 p 448. 620.6 I59p
Problems in design and research on condensers of vapours and vapour mixtures. Proceeding of the General Discussion on Heat Transfer, Institution of Mechanical Engineers, London, and American Society of Mechanical Engineers. New York: 1951.

(8) Data Sheet, Air Conditioning, Heating and Ventilating. Oct. and Nov. 1956.

(9) Jacob, Max.
Heat transfer, Vol. I. New York: John Wiley, 1949.659 p.

(10) McAdams, W. H.
Heat transmission, 3rd ed. New York and London; McGraw-Hill, 1954.

(11) Patel, S. P.
Film coefficient of heat transfer of Freon-12 condensing inside a single horizontal tube. M. S. Thesis 1956.

(12) Potter, R. C. and S. P. Patel.
Condensation of "Freon-12" in a horizontal tube. Preprint of paper, Project 222, Mechanical Engineering, Kansas State College.

(13) Spencer, H. C.
Film coefficient of vapors condensing,inside horizontal tubes. M. S. Thesis, Kansas State College. 1950.

(14) Temperature, Its Measurement and Control in Science and Industry. Vol. 1. American Institute of Physics. Reinhold Publishing Corporation.

APPENDICES

APPENDIX A

Sample Calculation

Run No. 15

Arithmatic mean of measured tube wall temperature, middle groove,

0.09-inch from vapor side surface = $71.3^{\circ}F$

Weight rate of flow of cooling water = M = 1028.6 lb/hr

Heat transferred to cooling water in test section

$= Q = M \cdot Cpw \cdot \Delta t = 1028.6 \times 1 \times 5.8 = 5966$ Btu/hr

Heat transfer by conduction through the metal = $Q = Km\ Am \dfrac{\Delta t_b}{\Delta x}$

Area of inside surface of inner tube, assuming effective heat transfer length of the tube be 2.5 feet

$$= A_1 = \frac{\pi(0.5)2.5}{12} = 0.3273\ ft^2$$

Area of the cylindrical surface where thermocouple junctions are located = $\dfrac{\pi(0.68)2.5}{12} = 0.4451\ ft^2$

$$A_m = \frac{A_2 - A_1}{\ln \dfrac{A_2}{A_1}} = \frac{0.4451 - 0.3273}{\ln \dfrac{0.4451}{0.3273}} = 0.3830\ ft^2$$

$K_m = 56.9$ (p. 445, 10)

$$\Delta t_b = \frac{0.09 \times 5966}{12 \times 0.3830 \times 56.9} = 2.05^{\circ}F$$

Average temperature of inside surface of inner tube

$= 71.3 + 2.05 = 73.35^{\circ}F$

Average bulk temperature of Freon-12 $= \frac{115.4 + 115.2}{2} = 115.3^{o}F$

Temperature difference between inside surface and Freon- 12 vapor = 115.3 - 73.35 = 41.95°F

From $Q = h_m A_1 \ (t_v - t_w)$

$$h_m = \frac{5966}{0.3273 \times 41.95} = 435 \ Btu/(hr)(ft^2)(^{o}F)$$

$$\text{Heat flux} = \frac{Q}{A_1} = \frac{5966}{0.3273} = 11940 \ Btu/(hr)(ft^2)$$

Freon rate of flow = 0.145 gpm

$\rho_f = 80.671 \ lb/ft^3$, for Freon liquid with specific gravity of 1.293 (at 86°F)

Weight rate of Freon-12 $= W_v = 0.145 \frac{gal}{min} \times \frac{60 \ min.}{hr} \times 0.1337 \frac{ft^3}{gal} \times 80.671 = 93.8 \ lb/hr$

Area of cross-section of the inner tube $= \frac{\pi}{4} \left(\frac{0.5^2}{12^2} \right) = 0.001364 \ ft^2$

Mass velocity of flow of Freon-12 $= G = \frac{W_v}{A} = \frac{93.8}{0.001364} = 68,970 \ lb/(hr)(ft^2)$

In this run, the Freon-vapor was dry at entrance to the test section, vapor velocity could be evaluated.

$\rho_v = 4.026 \ lb/ft^3$, (Data Sheet, Air Conditioning, Heating and Ventilating, Oct. and Nov. 1956)

Vapor velocity $= \frac{G}{\rho_v} = 68,970 \ lb/(hr)(ft^2) \times \frac{1}{3600} \frac{hr}{sec} \times \frac{1}{4.026} \frac{ft^3}{lb} = 4.76 \ ft/sec$

Temperature of condensing film by which physical properties were evaluated = $t_f = t_v - \frac{3}{4}(t_v - t_w)$ = 115.3 - 3/4 (115.3 - 73.35) = 83.84°F

Viscosity = μ_f = 0.610 lb/(hr)(ft) (Air conditioning-refrigerating data book)

K_f = 0.0500 Btu/(hr)(ft)(°F) (Heat transmission, McAdams, p. 455)

ρ_f = 80.951 lb/ft^3 (Data Sheet, Air Conditioning, Heating and Ventilating, Oct. and Nov. 1956)

Latent heat of condensation, based on vapor temperature = = 53.399 Btu/lb.

$$\frac{K_f^3 \rho_f^2 \lambda g}{D \ \mu_f \Delta t} = \frac{0.000125 \times 6553 \times 53.399 \times 4.17 \times 10^8}{0.04167 \times 0.610 \times 41.95} = 1711 \times 10^7$$

$$\frac{1}{\Phi} = \left[\frac{\mu_f^2}{k_f^3 \rho_f^2 g}\right]^{\frac{1}{3}} = \left[\frac{0.3721}{0.000125 \times 6553 \times 4.17 \times 10^8}\right]^{\frac{1}{3}} =$$

$$0.001029 \ \frac{(hr)(ft^2)(°F)}{Btu}$$

$$\frac{h_m}{\Phi} = 434.9 \times 0.001029 = 0.4475$$

$$\frac{4\Gamma}{\mu_f} = \frac{4W}{L\mu_f} = \frac{4 \times 93.8}{2.5 \times 0.610} = 246.1$$

APPENDIX B

A Derivation of Equations for Film Condensation with Negligible Vapor Velocity

Consider an inclined plane of unit width, as shown in Fig. 1′. For the viscous incompressible fluid

$$\rho \frac{d\bar{v}}{d\tau} = \rho \bar{F} - \nabla p + \mu \nabla \cdot \nabla \bar{v} \qquad (1')$$

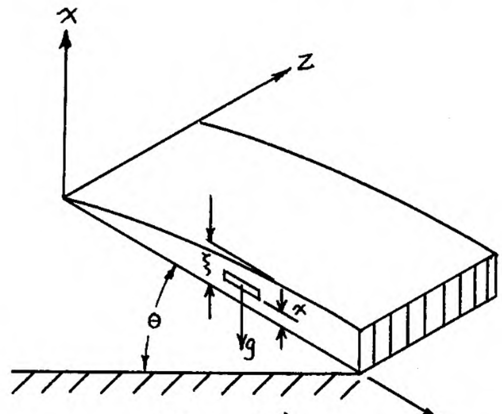


Fig. 1′

or

$$\frac{du}{d\tau} = F_x - \frac{1}{\rho} \frac{\partial p}{\partial x} + \nu \nabla^2 u$$

$$\frac{dv}{d\tau} = F_y - \frac{1}{\rho} \frac{\partial p}{\partial y} + \nu \nabla^2 v \qquad (1a')$$

$$\frac{dw}{d\tau} = F_z - \frac{1}{\rho} \frac{\partial p}{\partial z} + \nu \nabla^2 w$$

where $\nabla^2 = \frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} + \frac{\partial^2}{\partial z^2}$

Assuming no flow in x- and z-direction. Then

$$u = 0, \quad w = 0$$

For a steady flow and no pressure change in y-direction

$$\frac{dv}{d\tau} = 0, \quad \frac{\partial p}{\partial y} = 0$$

Furthermore, assuming that the velocity is the function of x only, $\frac{\partial v}{\partial y} = 0, \quad \frac{\partial v}{\partial z} = 0$

And introducing the external forces which cause the flow of the condensate. Here the external force is the action of gravity per unit mass of the condensate,

$$F_y = g \sin \theta$$

$$0 = g \sin \theta + \nu \frac{d^2 v}{dx^2} \qquad (2')$$

or $\frac{d^2v}{dx^2} = -\frac{g\rho_f}{\mu} \sin\theta$ (2a')

By integrations, with the following conditions:

(1) Zero velocity at the wall

$v = 0$, at $x = 0$

(2) Velocity gradient at the interface of the condensate and the condensing vapor, $\left(\frac{dv}{dx}\right)_{x=\xi} = 0$
where ξ is the thickness of the condensate film.

We get an expression for the velocity of the condensate at a distance x from the wall

$$V = -\frac{g\rho_f}{2\mu_f}(x^2 - 2x\xi) \qquad (3')$$

The mean velocity of the condensate across any section

$$V_m = \frac{1}{\xi}\int_0^{\xi} V dx = \frac{g\rho_f}{3\mu_f}\xi^2 \sin\theta \qquad (4')$$

Considering a tube section of unit length and an arc element $R d\theta$, as shown in Fig. 2'.

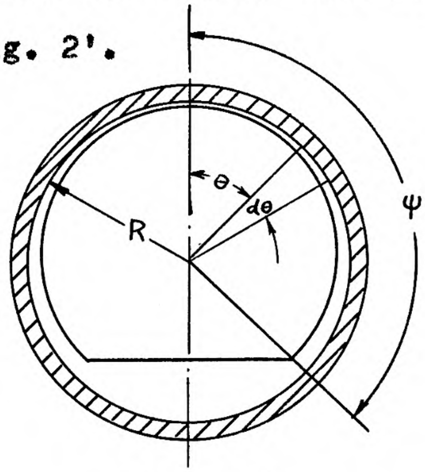


Fig. 2'

The mass rate of flow of the condensate per unit length of the tube is $\Gamma = \rho_f V_m \xi$

Assuming that the total thermal resistance lies in the film of condensate. Then, the energy balance in condensation will be

$$\lambda\, d\Gamma = \frac{K_f}{\xi}\, R\, d\theta\, \Delta t \qquad (5')$$

where λ is the latent heat of condensation.

Hence we have

$$\frac{\lambda g \rho_f^2}{3\mu_f}\left(\xi^3 \cos\theta\, d\theta + 3 \sin\theta\, \xi^2\, d\xi\right) = \frac{K_f}{\xi} R\, \Delta t\, d\theta \qquad (6')$$

or

$$\frac{3K_f R \mu_f\, \Delta t}{\lambda g \rho_f^2}\, d\theta = \xi^4 \cos\theta\, d\theta + 3 \sin\theta\, \xi^3\, d\xi \qquad (6a')$$

Introducing

$$B = \frac{3K_f\, R\mu_f\, \Delta t}{\lambda g\, \rho_f^2} \qquad (7')$$

and

$$S = \frac{\xi^4}{B} \qquad (8')$$

We get

$$d\theta = S \cos\theta\, d\theta + \frac{3}{4} \sin\theta \cdot dS \qquad (9')$$

or

$$\frac{dS}{d\theta} + \left(\frac{4}{3}\cot\theta\right) S = \frac{4}{3}\, \frac{1}{\sin\theta} \qquad (9a')$$

The solution of this first order differential equation is

$$S = \frac{1}{\sin^{(\frac{4}{3})}\theta}\left(\frac{4}{3}\int \sin^{\frac{1}{3}}\theta\, d\theta + C\right) \qquad (10')$$

If no liquid is dropping from the top of the tube, then for $\theta = 0$, the film thickness and therefore S will be finite. This requires

$$C = 0$$

We have

$$S = \frac{4}{3 \sin^{(\frac{4}{3})}\theta} \int \sin^{\frac{1}{3}}\theta\, d\theta \qquad (11')$$

From (8$'$)

$$\xi = S^{\frac{1}{4}} B^{\frac{1}{4}} = S^{\frac{1}{4}} \left[\frac{3 K_f R \mu_f \Delta t}{g \rho_f^2 \lambda} \right]^{\frac{1}{4}} \qquad (12')$$

The local value of the coefficient of heat transfer at the angle Θ is given by

$$h_f = \frac{K_f}{\xi} = \frac{K_f}{S^{\frac{1}{4}} B^{\frac{1}{4}}} \qquad (13')$$

Integration from 0 to ψ gives for the average heat transfer coefficient over this area,

$$hm_\psi = \left[\frac{1}{\psi} \int_0^\psi \frac{d\theta}{S^{\frac{1}{4}}} \right] \left[\frac{K_f^3 \rho_f^2 \lambda g}{3 \mu_f R \Delta t} \right]^{\frac{1}{4}} =$$

$$\left[\frac{2}{3} \right]^{\frac{1}{4}} \left[\frac{1}{\psi} \int_0^\psi \frac{d\theta}{S^{\frac{1}{4}}} \right] \left[\frac{K_f^3 \rho_f^2 \lambda g}{\mu_f} \right]^{\frac{1}{4}} \left[\frac{1}{D \Delta t} \right]^{\frac{1}{4}} =$$

$$\beta \Omega \left[\frac{1}{D \Delta t} \right]^{\frac{1}{4}} \qquad (14')$$

where

$$\beta = \left[\frac{2}{3} \right]^{\frac{1}{4}} \left[\frac{1}{\psi} \int_0^\psi \frac{d\theta}{S^{\frac{1}{4}}} \right] \qquad (15')$$

and

$$\Omega = \left[\frac{K_f^3 \rho_f^2 \lambda g}{\mu_f} \right]^{\frac{1}{4}} \qquad (16')$$

The heat transfer per unit length of the tube is given by

$$hm_\psi \psi D \Delta t = 2 \Gamma \lambda \qquad (17')$$

From (14$'$) and (17$'$), eliminating $D \Delta t$

We have

$$hm_\psi = \beta \Omega \left[\frac{hm_\psi \psi}{2 \Gamma \lambda} \right]^{\frac{1}{4}}$$

or
$$hm_\psi^{\frac{3}{4}} = \beta\left[\frac{K_f^3\ \rho_f^2 \psi g}{2\ \mu_f\ \Gamma}\right]^{\frac{1}{4}}$$

rearranging
$$hm_\psi = \beta^{\frac{4}{3}}\left[\frac{K_f^3\ \rho_f^2 \psi g}{2\mu_f\ \Gamma}\right]^{\frac{1}{3}} =$$

$$(2\psi)^{\frac{1}{3}}\ \beta^{\frac{4}{3}}\left[\frac{K_f^3\ \rho_f^2\ g}{\mu_f^2}\right]^{\frac{1}{3}}\left[\frac{4\Gamma}{\mu}\right]^{-\frac{1}{3}} \tag{18'}$$

By designating $\Phi = \left[\dfrac{K_f^3\ \rho_f^2\ g}{\mu_f^2}\right]^{\frac{1}{3}}$ (19')

This is a property of the refrigerant. We get

$$hm_\psi = (2\psi)^{\frac{1}{3}}\ \beta^{\frac{4}{3}}\ \Phi\left[\frac{4\Gamma}{\mu_f}\right]^{-\frac{1}{3}} = \beta'\ \Phi\left[N_{R_e}\right]^{-\frac{1}{3}} \tag{20'}$$

where $\beta' = (2\psi)^{\frac{1}{3}}\ (\beta)^{\frac{4}{3}}$ (21')

and $N_{R_e} = \dfrac{4\Gamma}{\mu_f}$ (22')

The term $\frac{4\Gamma}{\mu_f}$ appearing in equation (18') is the Reynolds number of the condensing film. Since the diameter of the tube, D, is inherently small, the condensate flow for the horizontal tube is considered to be always laminar.

If one inserts value of $\psi = 2\pi$ into equation (14') and (20') the resulting expressions are the same as those for condensation on the outside of a horizontal tube, namely

$$hm = 0.725\left[\frac{K_f^3 \rho_f^2 \lambda g}{D\mu_f\ \Delta t}\right]^{\frac{1}{4}} = 0.725\ \frac{\Omega}{(D\ \Delta t)^{\frac{1}{4}}} \tag{23'}$$

and
$$hm = 1.2\left[\frac{K_f^3\ \rho_f^2 g}{\mu_f^2}\right]^{\frac{1}{3}}\left[\frac{4\Gamma}{\mu_f}\right]^{-\frac{1}{3}} = 1.2\ \frac{\Phi}{Re^{\frac{1}{3}}} \tag{24'}$$

APPENDIX C

Table 4. Experimental data of condensing Freon-12 inside a horizontal tube.

Run No.	Vapor Temp. of Freon T_v °F	Tube Wall Temp. Tw °F	Cooling Water Temp. Rise °F	Boiler Pressure psig	Heat Input KW	Freon Flow G.P.M.	Water Flow lb/hr	Liquid at Entrance
1	107.25	70.84	4.5	137	1.014	0.230	867.5	Yes
2	95.15	66.29	3.6	114	0.70	0.085	957.5	Yes
3	97.95	65.19	3.2	120	0.78	0.10	991.7	Yes
4	107.55	69.23	4.4	137	1.034	0.18	944.9	Yes
5	99.7	67.67	4.2	119	1.40	0.09	1049.6	No
6	103.5	69.13	4.6	125	1.53	0.095	1025.6	No
7	107.4	71.03	5.3	132	1.676	0.11	1005.6	No
8	106.0	71.44	5.1	127	1.594	0.10	991.7	No
9	108.1	72.15	5.1	131	1.720	0.111	942.4	No
10	108.5	69.98	4.7	132	1.82	0.120	1040.5	No
11	108.1	70.46	4.8	132	1.77	0.115	1002.8	No
12	111.5	71.69	5.5	140	1.93	0.13	997.2	No
13	112.8	73.14	5.5	141	1.93	0.132	1022.7	No
14	114.8	73.22	6.2	147	2.02	0.142	947.4	No
15	115.3	73.35	5.8	148	2.06	0.145	1028.6	No
16	119.5	74.32	6.5	156	2.24	0.163	1039.0	No
17	117.7	73.3	6.1	153	2.14	0.155	954.9	No
18	115.0	74.29	5.7	147	1.996	0.14	967.7	No
19	118.5	72.37	4.8	154	2.21	0.16	1313.9	No
20	121.2	73.99	5.5	160	2.26	0.165	1210.5	No
21	122.9	73.90	5.6	166	2.37	0.175	1194.8	No
22	102.2	67.07	3.7	122	1.54	0.095	1469.4	No
23	103.9	68.39	3.8	127	1.64	0.105	1445.8	No
24	112.4	72.21	3.8	141	1.942	0.135	1460.4	No
25	96.5	68.78	3.1	109	1.22	0.065	1011.2	No

Table 4. Experimental data of condensing Freon-12 inside a horizontal tube. (concl.).

Run No.	Vapor Temp. of Freon T_v °F	Tube Wall Temp. Tw °F	Cooling Water Temp. Rise °F	Boiler Pressure psig	Heat Input KW	Freon Flow G.P.M.	Water Flow lb/hr	Liquid at Entrance
26	96.5	67.95	3.6	111	1.26	0.070	1011.2	No
27	99.4	67.49	3.7	118	1.42	0.085	1081.1	No
28	106.6	69.74	4.1	129	1.71	0.11	1162.8	No
29	96.7	66.85	2.9	112	1.356	0.08	1254.4	No
30	115.3	71.21	4.5	150	2.10	0.15	1232.9	No
31	85.8	63.76	2.3	97	0.84	0.31	1208.1	Yes
32	92.1	67.18	3.0	107	1.074	0.39	1139.2	Yes
33	95.4	66.32	3.1	111	1.18	0.40	1142.9	Yes
34	79.65	62.54	1.8	85	0.55	0.11	1034.5	Yes
35	84.15	63.52	2.3	93	0.744	0.13	1028.6	Yes
36	88.35	65.15	2.9	98	0.849	0.16	952.4	Yes
37	96.4	67.38	3.5	113	1.157	0.25	975.6	Yes
38	105.65	69.92	4.3	131	1.50	0.37	962.6	Yes
39	114.2	74.66	4.5	145	1.738	0.33	1008.4	Yes
40	118.4	73.22	4.9	155	2.046	0.21	1022.7	Yes
41	123.5	74.93	5.3	166	2.34	0.175	1061.9	No
42	129.3	76.62	5.9	181	2.54	0.20	995.9	No

APPENDIX D

A Cross-sectionalarea of the inner tube of the horizontal condenser, ft^2.

A_1 Area of inside surface of the inner tube, ft^2.

A_2 Area of the cylindrical surface where thermocouple junctions are located, ft^2.

A_m Logarithmic mean area, ft^2. $Am = \frac{A_2 - A_1}{\ln \frac{A_2}{A_1}}$

C_{pf} Specific heat of condensate at constant pressure, Btu/lb-°F.

C_{pw} Specific heat of cooling water at constant pressure, Btu/lb-°F.

d Prefix, indicating denvativl, dimensionless.

D Inside diameter of the inner tube, ft.

f Friction factor, dimensionless

F Force per unit mass - lb_f/lb_m; also total force or drug per unit surface of laminar layer, $(lb_m/ft^2)(ft/hr^2)$

g Gravitational acceleration, ft/hr^2

G Mass velocity of Freon-12 condensate, $lb/hr\text{-}ft^2$

h_f Local value of condensing film coefficient $Btu/hr\text{-}ft^2\text{-}°F$.

h_m Average value of h with respect to height of condensing surface, $Btu/hr - ft^2 - °F$.

$h_{m\psi}$ Mean value of h over the film angle ψ, $Btu/hr\text{-}ft^2\text{-}°F$.

K_f Thermal conductivity of the Freon-12 condensate, Btu/hr-ft-°F.

Km Thermal conductivity of the material of the inner tube, mean value, Btu/hr-ft-°F.

L Length of a straight tube, ft.

ln Prefix, indicating natural logarithm.

M Weight rate of cooling water, lb/hr.

Q Rate of heat transfer to cooling water in test section, Btu/hr.

$\frac{Q}{A_1}$ Heat flux, $B/hr\text{-}ft^2$.

R Radius of tube, ft.

Re Reynolds number, dimensionless

S A function of the film angle ψ .

t_f Temperature of condensing film by which physical properties are evaluated, ^oF.

tv Temperature of the saturated vapor of Freon-12, ^oF.

tw Inside surface temperature of inner tube F.

Δt Temperature drop across the film condensate ^oF.

Δt_b Temperature difference between inside surface of the inner tube and the points where the thermocouples for measurement of the tube-wall temperature are located ^oF.

V Velocity, ft/sec.

W_v Flow rate of Freon-12 condensate, lb/hr.

x Linear distance, ft; also rectangular coordinate.

Δx Thickness of the inner condenser tube, ft.

X Non-dimentionalized coordinate.

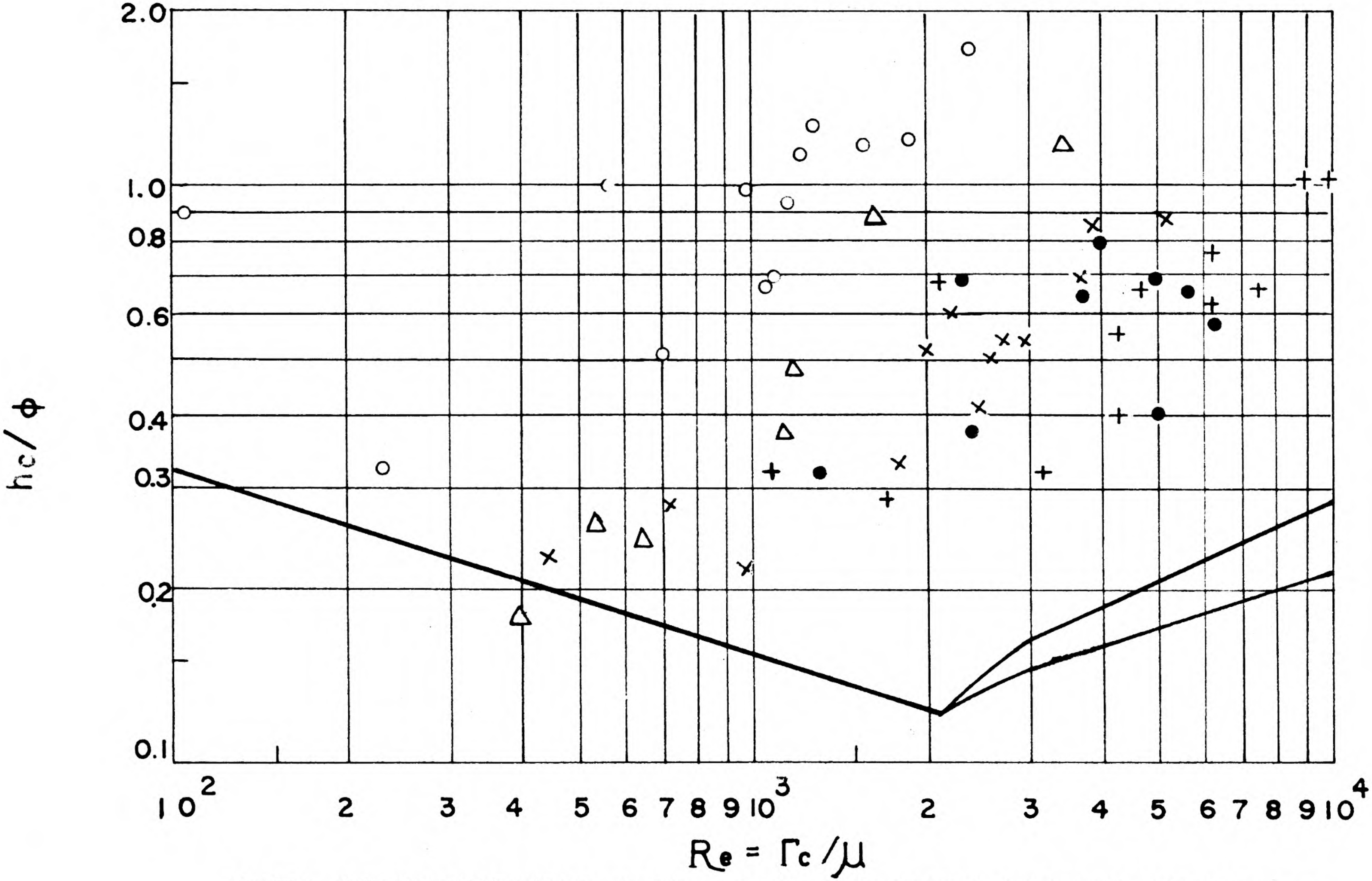
y Rectangular coordinate.

Y Non-dimentionalized coordinate.

z Rectangular coordinate

β A function of the film angle, defined by equation (8)

Γ	Mass rate of flow of condensate from the lowest point in the condensing surface divided by the breadth, lb/hr-ft.
Δ	Prefix, indicating finite difference.
θ	Angle
λ	Latent heat of vaporization Btu/lb.
μ_f	Absolute viscosity of Freon-12 condensate, lb/hr-ft.
ν	Kinematic viscosity, ft^2/hr.
ξ	Thickness of condensate film, ft.
π	3.1416
ρ_f	Density of Freon-12 condensate, lb/ft^3.
ρ_v	Density of Freon-12 vapor, lb/ft^3.
Φ	$\left[\frac{K_f^3 \rho_f^2 g}{\mu_f^2}\right]^{\frac{1}{3}}$ Btu/hr-ft^2-F
Ω	$\left[\frac{K_f^3 \rho_f^2 \lambda g}{\mu_f}\right]^{\frac{1}{4}}$ Btu/hr-ft$^{\frac{7}{4}}$-F$^{\frac{3}{4}}$
ψ	Film angle, angle between a radius of a horizontal tube, and a vertical plane at the axis.
τ	Time, hours.



DATA OF CARPENTER FOR CONDENSATION IN THE PRESENCE OF HIGH VAPOUR VELOCITIES

o ETHANOL. ● TOLUENE. x WATER. △ METHANOL. + TRICHLORETHYLENE.

Fig. 17.

HEAT TRANSFER OF CONDENSING FREON-12
INSIDE A HORIZONTAL TUBE

by

CHENG-CHIEH HWANG

B.S., National Taiwan University, China, 1953

AN ABSTRACT OF
A THESIS

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KANSAS STATE COLLEGE
OF AGRICULTURE AND APPLIED SCIENCE

1957

An experiment was made to investigate the mechanism of condensing Freon-12 inside a horizontal tube in the presence of vapor velocity.

A double-pipe test condenser with an inner tube of 0.500 I. D., common to refrigeration equipments, was designed and constructed. The temperatures of the Freon-12 vapor and the cooling water were measured at the inlet and at the outlet of the condenser. Tube wall temperatures were measured by fifteen thermocouples. The average coefficient of heat transfer, h_m, was evaluated from the rate of heat transfer to the cooling water, Q/A_1, the temperature difference between the Freon-vapor and the tube wall, $\triangle t$, using the equation,

$$Q/A_1 = h_m \triangle t$$

A visual study was made with sight-glasses installed on both ends of the test condenser.

A total of 42 runs was made with h_m ranging from 300 to 500 Btu/(hr)(ft^2)(f), Q/A_1 from 5700 to 21000 Btu/(hr)(ft^2), and $\triangle t$ from 17 to 53^{o}F. Freon-12 entered the condenser in some cases totally in the form of vapor and in others partly in the form of vapor and partly in the form of liquid. In the latter case, the entering liquid formed a stream, maximum depth of about 1/8-inches, at the bottom of the test condenser tube. The depth of the condensate stream at the exit was in excess of 1/4-inches, and varied with entrance conditions and the rate of vapor condensation. Wave motions in the flow of the condensate in the bottom of the tube were also observed.

The theoretical equations for condensation on horizontal tubes were found inapplicable, because the experimental data showed 0 to 70 percent higher values of h_m than that predicted by these equations. A similarity was observed between data published on condensation in a vertical tube with a turbulent flow of condensate and the data obtained from the horizontal tube investigated in this research. This similarity was regarded as being due to the effect of interfacial vapor friction on the flow of the condensate.