

DETERMINATION OF FILM COEFFICIENT
FOR BOILING AMMONIA

by

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A THESIS

submitted to

OREGON STATE COLLEGE

in partial fulfillment of
the requirements for the
degree of

MASTER OF SCIENCE

June 1950

ADVANCE BOND

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ACKNOWLEDGMENT

The author is greatly indebted to Professor Wallace H. Martin of Oregon State College for his advice and assistance in construction of apparatus, and critical analysis of subject matter. The writer wishes to thank authors of technical papers from which material has been borrowed, especially to Professor Max Jakob of the Illinois Institute of Technology.

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DETERMINATION OF FILM COEFFICIENT
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INTRODUCTION

In any experiment which determines the physical properties of a substance, it is expedient to discuss the uses of the substance. The reason for this is that the uses demand that the physical properties be determined.

Thermodynamic properties of working fluids have much to do with their suitability for use in a given process. As the boiling temperature of any liquid may be made to change with the pressure exerted upon it, it is quite easy to cause a liquid refrigerant to boil at any desired temperature by placing it in a vessel where the required pressure may be maintained.

Of the many liquids which boil at temperatures suitable for refrigeration, few possess all the requirements for a practical refrigerant. In large refrigeration plants, ammonia is used as a refrigerant almost exclusively. This is because of the high latent heat it will absorb per pound during the change of state from a liquid to a vapor, and also because the operating pressures are desirable. Due to the high toxicity of ammonia, it no longer is used in small units such as household refrigerators.

In the design of ammonia evaporators, the fundamental principles that govern heat transfer must be understood. By correlation of experimental data, much information can be accumulated. This is the reason for running this experiment and many similar experiments on other fluids.

Theory

Knowledge of the thermodynamic properties of fluids has reached a high stage of perfection. Especially where fluids undergo the change of state from a liquid to a vapor and vice versa. This can be seen from the various tables, steam, ammonia, etc, where heat quantities have been determined to tenths of a Btu per pound. In contrast to this, knowledge of the processes connected with the generation and condensation of vapors has been developed but little. The generation of vapors is especially important because of the types of apparatus employed, such as steam boilers and the evaporators used in refrigeration plants and chemical industries.

The process which deserves most consideration is the method by which heat is transferred. When a liquid is heated without boiling, the heat transfer is governed by the laws of convection and the film conductance can be predicted quite accurately. But when boiling occurs, bubbles of vapor are formed and liberated from the surface

in contact with the liquid and turbulence of uncertain magnitude is set up. If vapor is able to exist as a separate phase, a surface of separation is necessary; but the liquid offers resistance to the formation of a curved surface, and it becomes apparent that surface tension has to be considered from the standpoint of a gas to liquid interface.

The heat transfer, then may be divided into two parts. First, the heat that is transferred from the heated solid surface to the boiling fluid, and secondly, the heat that is transferred from the liquid to the bubble. It is known however, that the film coefficient depends upon how great a fraction of the heating surface is in contact with liquid rather than with the vapor bubbles. Since the rate of heat transfer from the heated surface to the liquid is much higher than the rate to the vapor, the film coefficient is increased by anything that either reduces the size of the bubbles formed or reduces the time for the bubbles to break away from the surface. (1, 648-9)*

Film Concept

When a fluid, even though moving, is in contact with a solid surface, there exists an adhering fluid film which

*Numbers in parentheses refer to bibliography.

may be stationary at the surface and moving at low velocities closely adjacent thereto. The fluid film, which on the solid surface side has zero velocity, acts on the flow of heat as if it were an entirely different substance. The fluid on the main stream has considerable freedom of movement, and differences of temperature are not very great especially when the flow is turbulent. The mass velocity flow of the fluid determines the thickness of the film. Let us suppose that water is to be cooled by evaporating refrigerant in a coil immersed in the water. In contact with the surface of the pipe coil there is a film of water that may be thought of as being stationary. This film offers resistance to heat flow through it from the water to be cooled to the pipe surface. The pipe wall offers additional resistance. Inside of the pipe there is a film of refrigerant liquid and vapor that may be thought of as being stationary, which offers additional resistance. If either surface of the pipe has any scale, dirt, or oil on it, the effect will be to offer still more resistance. The total resistance will then be the sum of the above named resistances. Obviously the resistance of any film will depend on the substance of which the film consists and the thickness of the film. It is then quite evident that the resistance of the film would be dependent upon the physical properties of the substance in question. (2, 16-17)

Effects of Various Parameters in Boiling

Analysis of existing data shows that there are a number of factors of importance.

The film coefficients depend not only upon the kind of fluid involved and upon the pressure and temperature of the fluid but also upon the type of retaining tube surface and position.

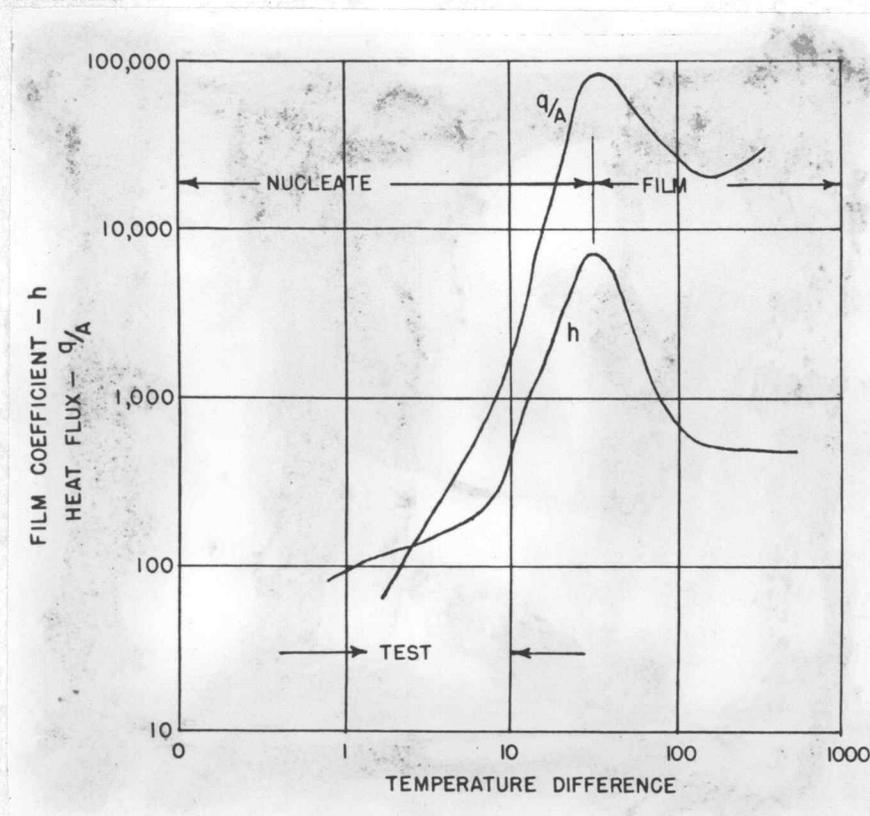


Figure 1. Typical Boiling Curve

Figure 1 shows the general trend of the change of film coefficient and heat flux with the change of mean temperature difference. (3, 469) In this experiment, only the portion of the curve marked which is in the nucleate

boiling region has been determined.

The effect of changing the saturation pressure shows that the film coefficient is proportional to some power of the reduced pressure. (4, 308)

The effect of change in tube diameter shows that the film coefficient is not changed substantially. (4, 314)

Usually in heat transfer processes there is an attempt at correlating experimental data with the physical properties of the fluid. But in the light of present knowledge of the mechanism of vaporization, mathematical expressions are therefore uncertain and values of the film coefficient must be determined experimentally for most cases. (5, 121)

OBJECT

The purpose of the experiment was to determine the film coefficient of heat transfer from metal surface to boiling ammonia in its capacity as a refrigerant. The operating variables were heat flux and mean temperature difference. Pressure within reasonable limits was held constant.

In actual operating conditions the mean temperature difference between the medium to be cooled and the ammonia must of necessity be kept low for economic reasons. The reason for this is the fact that the suction pressure on the evaporator should be high enough to give sufficient weight of flow to the compressor for a required refrigerating effect.

Since sufficient time was not available for a more extended program, the film coefficient was determined only for a horizontal, single pipe, flooded evaporator.

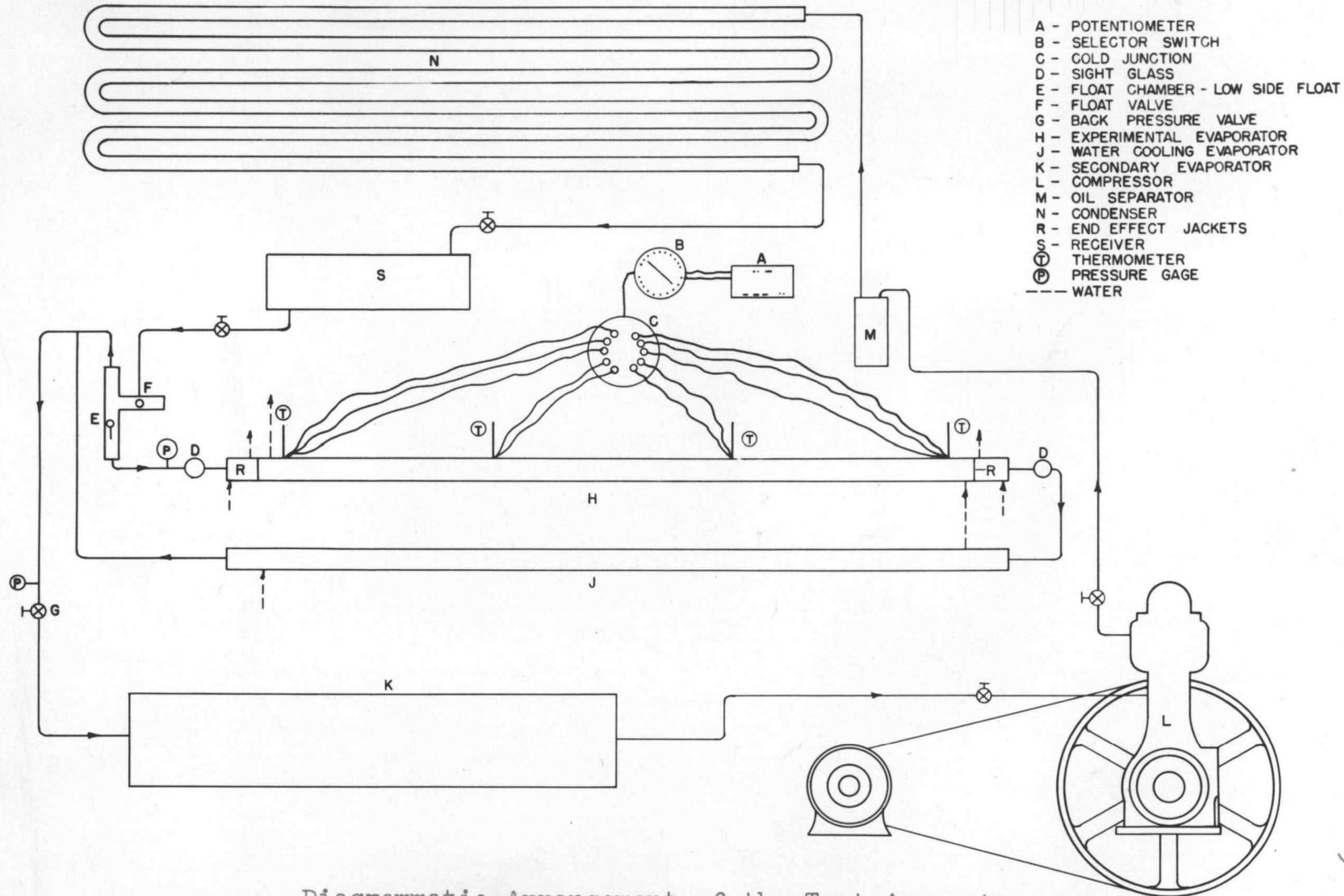
APPARATUS

Ammonia was used as a refrigerant in a horizontal evaporator which for all practical purposes was flooded with saturated liquid. This operation was accomplished by the use of a low side float valve which maintained a constant level of liquid ammonia in an accumulator or separator. The vapor which flashed was taken off at the top of the accumulator and bypassed directly to the suction of the compressor. The liquid from the lower part of the accumulator flowed into the evaporator. Since the rate of flow was much greater than the evaporating capacity of the test unit this unit was thoroughly flooded throughout its length. A sight glass at the inlet and outlet made possible observation of the flow and large quantities of liquid were visible at both points. The desired evaporating pressure was maintained by a back pressure valve through which the flow passed into a secondary evaporator before returning to the compressor.

Figure 2 is a diagrammatic arrangement of the test apparatus. Refrigeration was furnished by a two-cylinder reciprocating compressor. (Figure 3) There was an oil separator and an oil filter in the line before the evaporator, which removed practically all the oil.

Load was applied by means of water circulated through the evaporator as shown. Water circulating through the

FIGURE 2



Diagrammatic Arrangement of the Test Apparatus

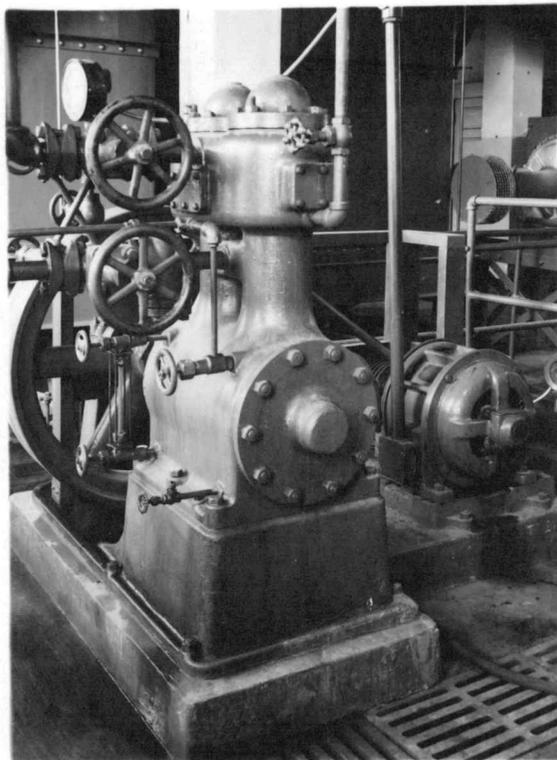


Figure 3. Photograph of two-cylinder, ammonia compressor, with suction valve in piston.

jacket was weighed. Load was determined by the temperature drop and the weight of water flowing per unit time.

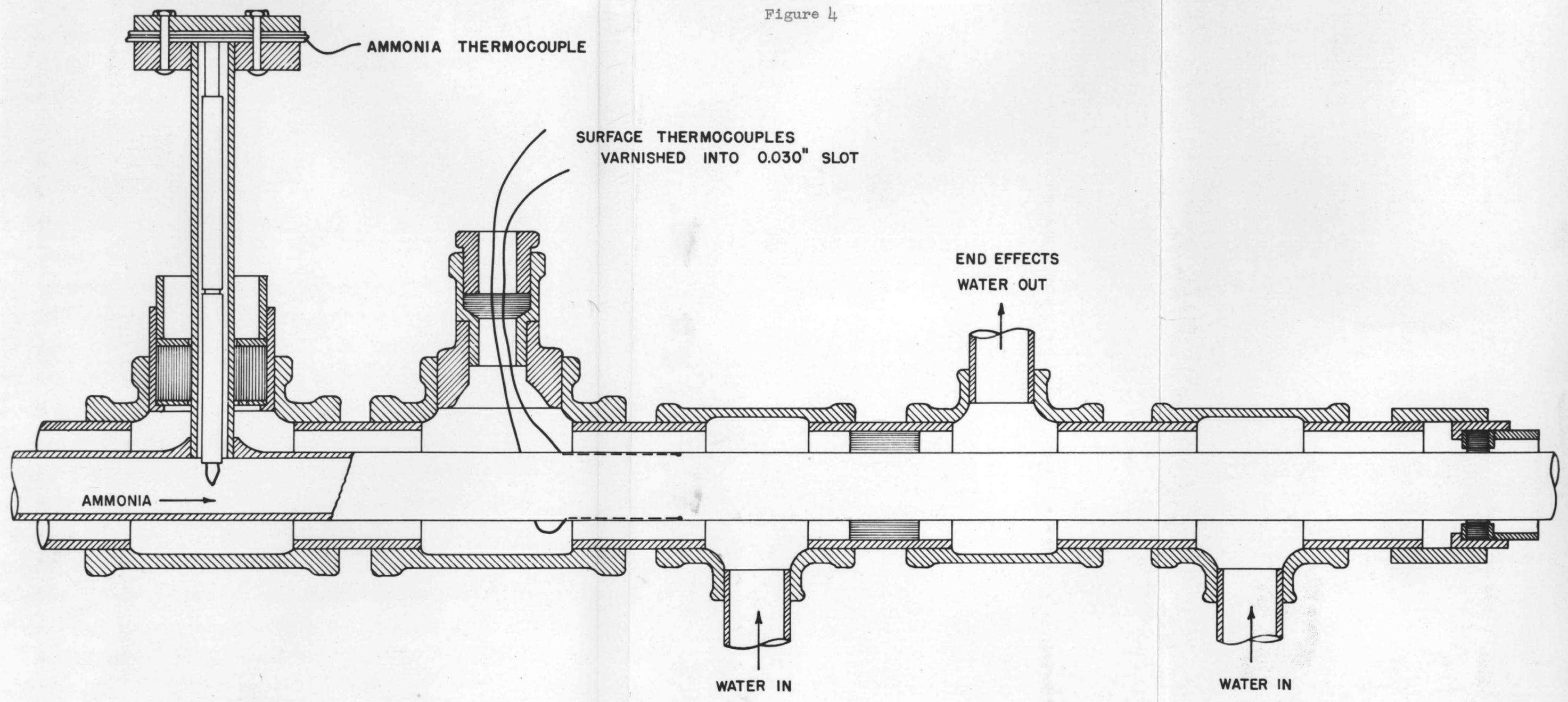
The evaporator was a double pipe type with refrigerant in the inner tube and water flowing in the annular space and consisted of a three-quarter inch extra heavy wrought iron pipe, with seventeen feet of experimental length placed in a horizontal position. Water jacket made up of one and one-half inch pipe was placed over the experimental pipe. The water jacket was covered with strips of three-quarter inch thick Kimsul insulation. The necessary number of tees was installed to allow for the entrance of thermometers, thermocouples, and water. Also water jackets were assembled on both ends of the experimental pipe for the purpose of eliminating end effects. Figure 4 shows a cutaway section of one end of the experimental pipe with the end effects jacket in place.

All of the temperatures were taken by means of No. 26 chromel-alumel thermocouples and calibrated thermometers. To measure outside surface temperatures of the experimental pipe, thermocouple junctions were silver-soldered into the upstream end of a slot milled in the pipe. This slot was two inches long and 0.03 inch deep. The solder was filed down flush with the surface of the pipe to eliminate turbulence at the junction. The trailing wires were varnished with insulating varnish and then varnished into the slot.

DETAIL SECTION
EXPERIMENTAL AMMONIA EVAPORATOR

FULL SCALE

Figure 4



The purpose of this was to keep the temperature of the wire adjacent to the junction the same as the junction so that there would be no heat flow from the moving stream of water to the junction.

Eight thermocouple junctions were installed in the above manner. Four were installed on the top surface and four on the bottom surface of the pipe. Top and bottom pairs were at a definite point along the length of the pipe. The ammonia temperatures were taken by two thermocouples immersed in the ammonia and placed at both ends of the experimental pipe.

Thermocouple leads soldered to copper leads to form the cold junction (Figure 5) were placed in hollow glass containers. These were placed in a cold junction box. The box was so arranged that it could be filled with crushed ice. The box was also equipped with a thermometer so that a check on the cold junction could be made at all times. The copper leads went to a multiple selector switch and from there two leads went to a Leeds and Northrup portable precision potentiometer number 8662 and serial number 706468, which could be read to 0.001 millivolt.

Four thermometers were used to measure water temperatures. They had one degree divisions. Also they were calibrated and used with magnifiers to read to one-tenth of a degree.

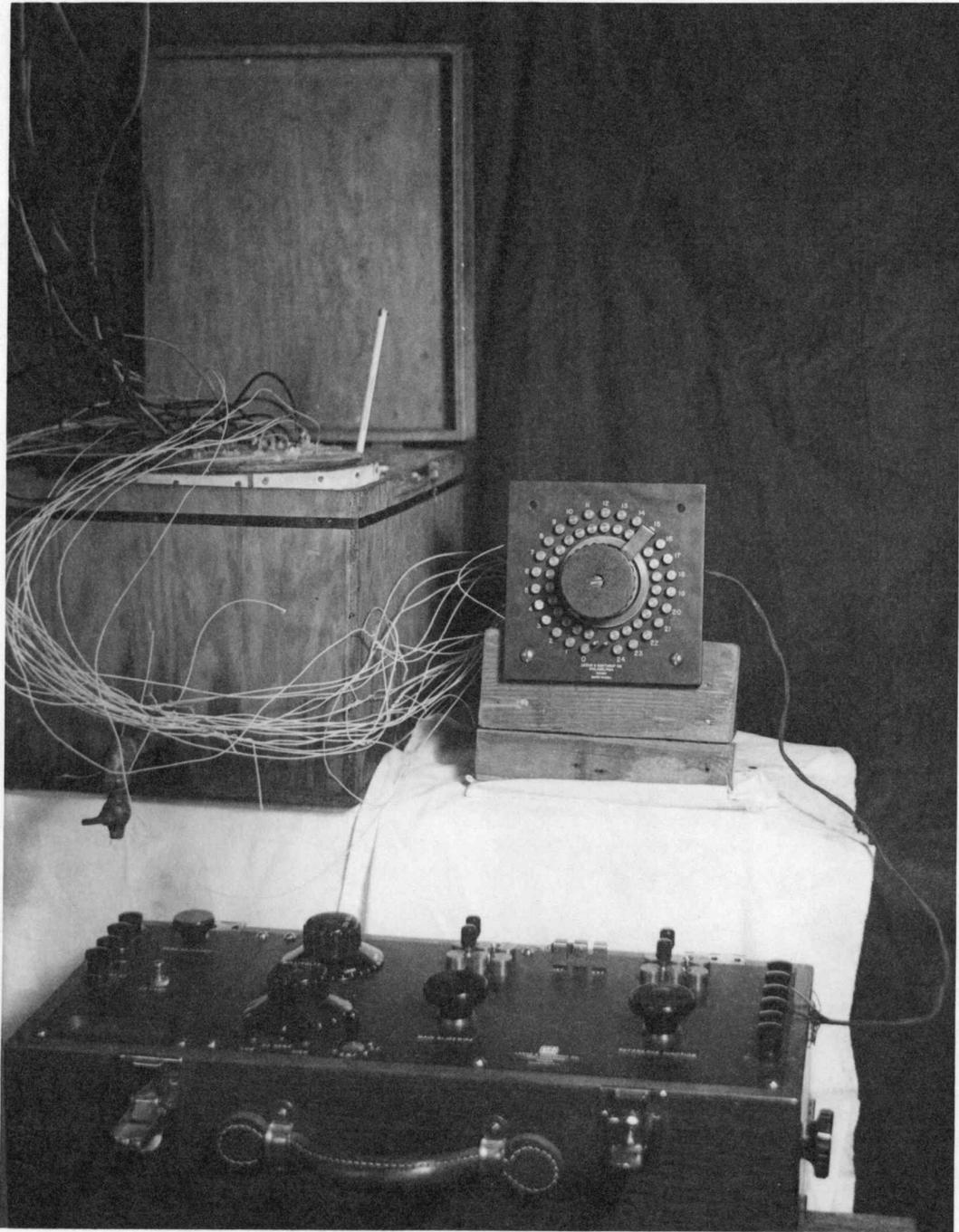


Figure 5. Photograph showing Leeds and Northrup potentiometer, selector switch, and cold junction.

PROCEDURE

In order to calibrate the thermocouples in place, three points were needed to plot a straight line calibration curve. This was done by circulating water through the jackets for at least an hour to be sure of equilibrium conditions. Potentiometer readings of the ten thermocouples were then taken. A separate calibration curve was made for each couple. (Figure 6)

For experiment runs, circulating water was adjusted to a predetermined rate of flow, and the pressure on the evaporator adjusted to about 50-60 pounds per square inch gage. Equilibrium conditions were reached in about two hours, after which readings were taken. Two readings were taken for each run, as a precaution against possible error.

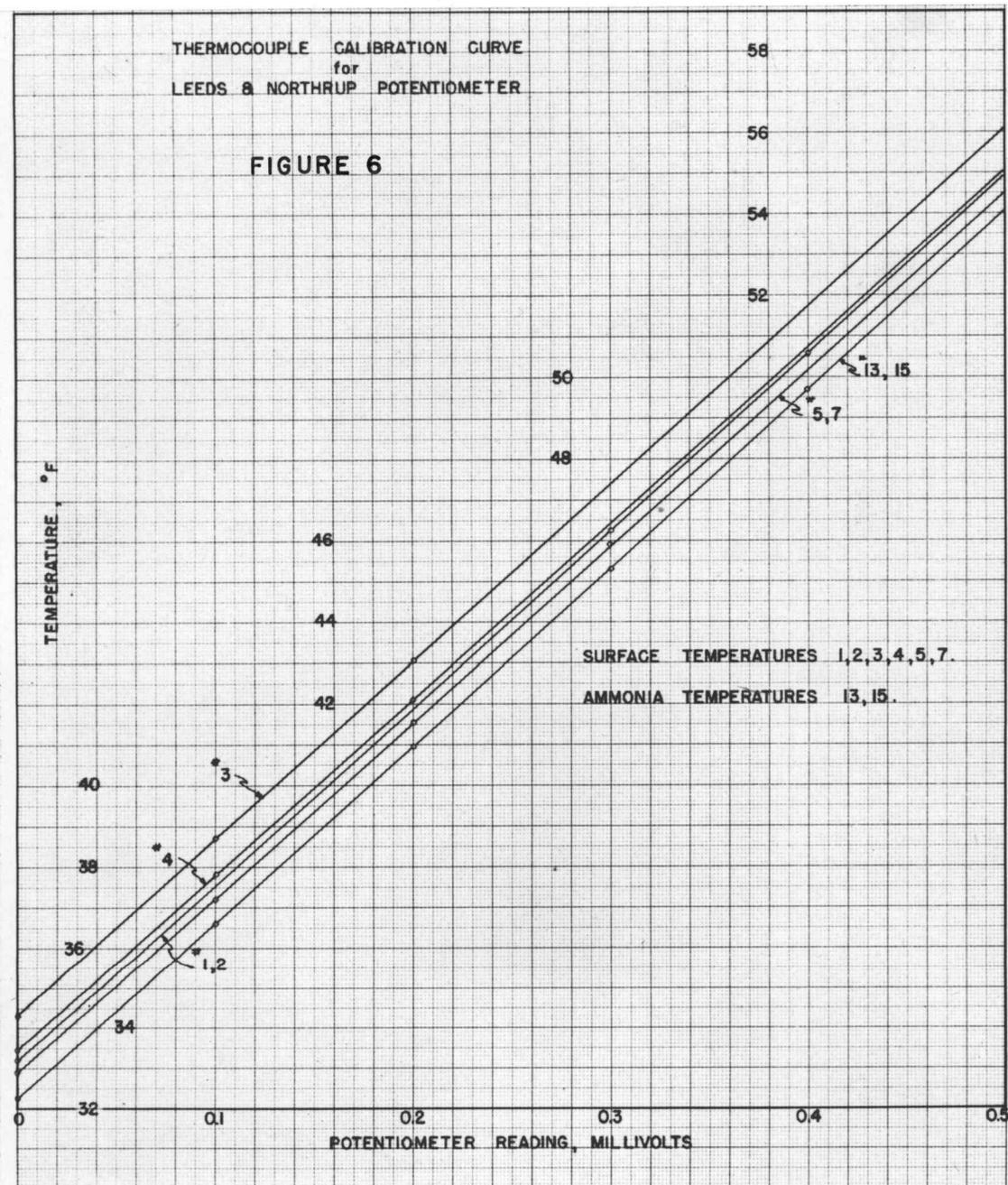
The test results were calculated. To determine the total film coefficient for the ammonia, the average of the top and bottom pipe surface temperature was used. Also the top surface temperatures were used in determining film coefficients.

Derivation of Formulae Used

Laws have been set up which will predict all types of energy flow. The most common law in use is that for the unidimensional flow of electrical energy, Ohm's Law, which

THERMOCOUPLE CALIBRATION CURVE
for
LEEDS & NORTHROP POTENTIOMETER

FIGURE 6



states that the electrical potential is proportional to the electrical flux, where the proportionality constant called resistance is determined experimentally.

An analogy to Ohm's Law is the unidimensional flow of heat energy. Here the potential is measured by temperature difference. Flux is the time rate of heat flow "q", per unit area "A", normal to the flow of heat. The proportionality constant is determined experimentally. It may be written:

$$\Delta t = m q/A \quad (1)$$

When heat is flowing through a solid of given thickness, the proportionality constant "m" becomes L/k , where L is the thickness parallel to the flow of heat, and "k" is called the thermal conductivity. When heat is flowing from a solid to a liquid in motion, there is no method of measuring the film thickness; therefore, "m" becomes $1/h$ or the film resistance to heat transfer, where "h" is called the film coefficient of heat transfer.

It will be assumed that the experimental pipe is a hollow cylindrical body of circular cross section, and the direction of heat flow is at all points radial and perpendicular to the axis of the cylinder. Therefore, for heat flow through the solid portion let the inner radius be " r_1 " and the outer radius be " r_0 ", and the corresponding temperatures be " t_1 " and " t_0 ". Heat flow per unit

time, "q" flows from the outer to the inner surface of the cylinder of length "l". This heat flows across progressively smaller areas of the cylinder. Therefore equation (1) becomes an integral:

$$dt = \frac{dr}{k} \frac{q}{A}$$

$$\int_{t_1}^{t_0} dt = \frac{q}{2\pi l k} \int_{r_1}^{r_0} \frac{dr}{r}$$

$$t_0 - t_1 = \frac{q}{2\pi l k} \ln \frac{r_0}{r_1} \quad (2)$$

Also the same amount of heat must flow through the film on the inner surface of the pipe. Therefore equation (1) becomes:

$$t_1 - t = \frac{l}{h} \frac{q}{2\pi r_1 l} \quad (3)$$

where "t" is the temperature of the fluid flowing in the cylinder. Solving for the temperature differences in the above two equations and adding, the equation becomes:

$$t_0 - t = \frac{q}{2\pi l} \frac{\ln \frac{r_0}{r_1}}{k} + \frac{l}{h r_1} \quad (4)$$

From this, solve for "h":

$$h = \frac{q \frac{k}{r_1}}{2\pi l k (t_0 - t) - q \ln \frac{r_0}{r_1}} \quad (5)$$

The experimental constants are r_1 , r_0 , k , and l . The values to be determined are q , and $(t_0 - t)$. It must be remembered that the flow of liquids is in counterflow and that $(t_0 - t)$ must be represented by the logarithmic mean

temperature difference. The nomenclature and units used are:

h = film coefficient of heat transfer, Btu/hr- $^{\circ}$ F-ft²

q = heat transferred, Btu/hr

k = thermal conductivity, Btu/hr- $^{\circ}$ F-ft²/ft

l = length of pipe, feet

$t_o - t$ = logarithmic mean temperature difference, between outside pipe surface and fluid in pipe, $^{\circ}$ F

r_o = outside radius of pipe, feet

r_i = inside radius of pipe, feet

Experimental constants are:

r_o = 0.0437 feet

r_i = 0.0309 feet

k = 34.9 Btu/hr- $^{\circ}$ F-ft²/ft wrought iron, (4, 381)

l = 11.5 feet

Substituting experimental constants, equation (5) becomes:

$$h = \frac{1130 q}{2520 (t_o - t) - 0.348 q} \quad (6)$$

Heat transferred is determined from the weight and change in temperature of water:

$$q = w (t_1 - t_3) \quad (7)$$

w = weight of water flow, lb/hr

t_1 = temperature of water entering evaporator, $^{\circ}$ F

t_3 = temperature of water leaving evaporator, $^{\circ}$ F

Substituting constants in equation (3),

$$\Delta t = t_1 - t = \frac{q}{2.22} \times \frac{1}{h} \quad (8)$$

where:

$$\frac{q}{2.22} = \text{heat flux, Btu/hr-ft}^2 \text{ inside surface}$$

Δt = log-mean temperature difference between inside surface and ammonia, F

Sample Calculations

Data:

$$w = 1230 \text{ lb/hr}$$

$$t_1 = 52.5 \text{ F}$$

$$t_3 = 49.4 \text{ F}$$

$$t_o = 48.4, 40.3 \text{ F}$$

$$t = 35.7 \text{ }^\circ\text{F}$$

$$q = 1230 (52.5 - 49.4) \quad (7)$$

$$= 3813 \text{ Btu/hr}$$

$$t_o - t = \frac{48.4 - 40.3}{\ln \frac{48.4 - 35.7}{40.3 - 35.7}} = \frac{8.1}{\ln \frac{12.7}{4.6}}$$

$$= 8.1 \text{ F}$$

$$h = \frac{1130 \times 3813}{2520 \times 8.1 - 0.348 \times 3813} \quad (6)$$

$$\begin{aligned} &= 220 \text{ Btu/hr-F-ft}^2 \\ \Delta t &= \frac{3813}{2.22} \times \frac{1}{220} && (8) \\ &= 7.83 \text{ F} \end{aligned}$$

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DISCUSSION OF RESULTS

Trend of Curves

The following set of curves represents the significant data obtained as described in the procedure. Figures 7 and 8 were obtained at an evaporator pressure of from 50 to 56 psig. The temperature difference between the inside metal surface and the ammonia varied from two to ten F. In this range the film coefficient varied from 160 to 370 Btu per hr-F-ft², while the heat flux varied from 420 to 3300 Btu per hr-ft² of inside surface.

The trend of these two curves follows that outlined in Figure 1, for the portion of the curve which is of particular interest for this experiment. Log-log plot was used because it has been employed by previous investigators who have studied a greater range of mean temperature differences. For the range of mean temperatures covered in this experiment a semi-logarithmic plot gives a more comprehensive curve. The plot of heat flux against temperature difference gives a straight line while the plot of film coefficient gives a curve with a slightly increasing slope.

The equation determined for heat flux in terms of mean temperature difference is:

$$\ln (q/A) = 0.28(t) + 5.31$$

FIGURE 7

AMMONIA BOILING AT 50-56 PSIG. IN A FLOODED HORIZONTAL SINGLE TUBE EVAPORATOR. PLOT OF FILM COEFFICIENT, h , AGAINST INSIDE MEAN TEMPERATURE DIFFERENCE, Δt .

- Values for average temperature of pipe surface.
- Values for top temperature of pipe surface.

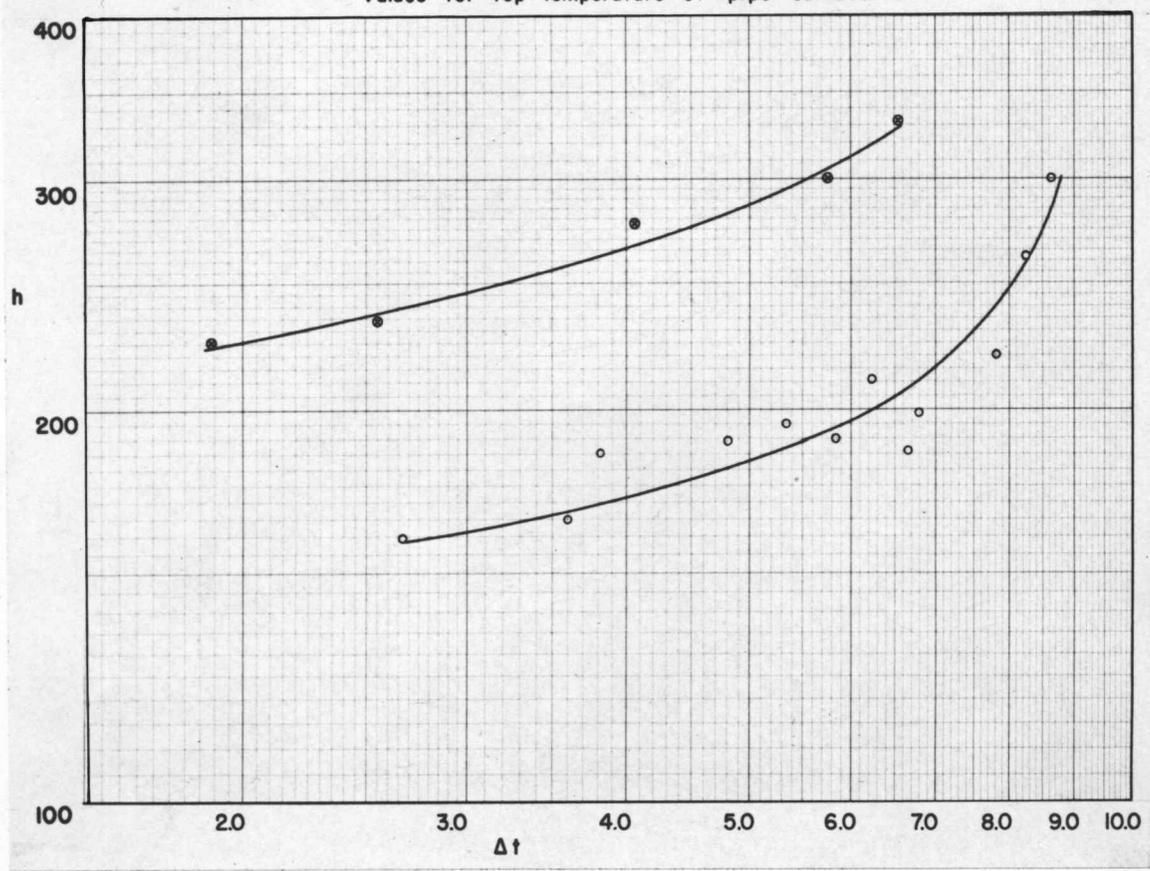
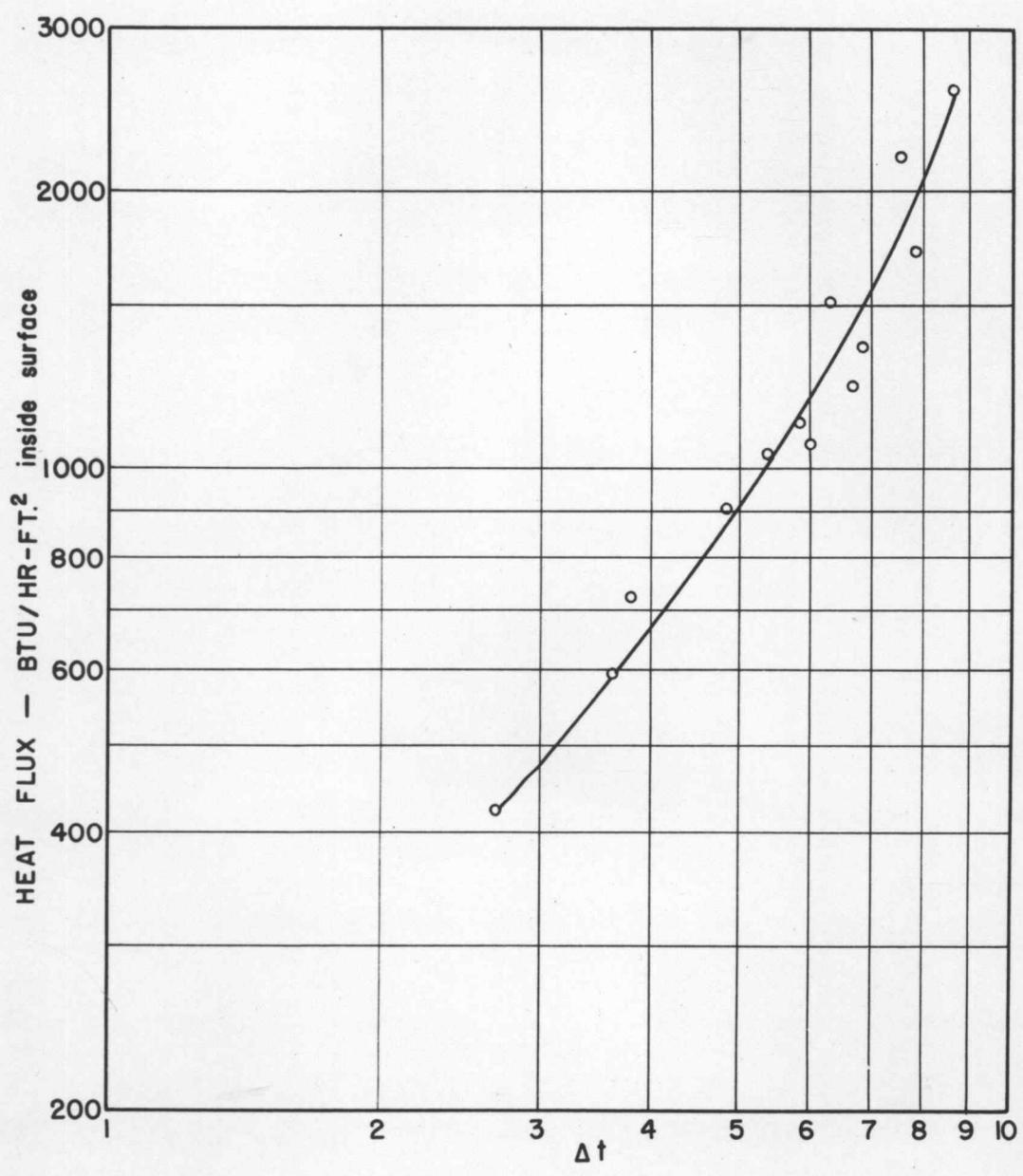


FIGURE 8

AMMONIA BOILING AT 50-56 PSIG.
IN A FLOODED HORIZONTAL SINGLE TUBE
EVAPORATOR. PLOT OF HEAT FLUX AGAINST
INSIDE MEAN TEMPERATURE DIFFERENCE (Δt).



This equation is of course limited to the operating pressure which for this experiment was 53 psig. For other pressures the above equation would probably take the form:

$$\ln (q/A) = 0.28 (t) + 5.31 + \#(p)$$

where $\#(p)$ is some function of the pressure, which would give a family of straight lines on semi-logarithmic paper.

In Figure 7, the top surface temperatures were used in determining the film coefficient for the upper curve. It is believed any values between the limits of these two curves will give reasonable design values for the film coefficient, although it would be considered that the values of film coefficient found by using the average surface temperature will give the worst possible conditions encountered. This of course is the value to be used in the design of evaporators.

Effect of Oil in the Evaporator

Contrary to what would be expected by laws of convection, the temperature of the bottom surface was higher than the top surface of the pipe. This would seem to indicate that there was more resistance to flow of heat through the bottom surface than through the top surface. This can be attributed to lubricating oil which flows through the system despite all attempts to remove it.

As stated in the procedure, the average of top and

bottom surface temperatures was used to determine the film coefficient for ammonia under actual operating conditions. This is not a truly correct method due to the fact that the oil film probably does not offer the same resistance at all points of the surface. This condition affects the temperature distribution. The fact that the temperature difference along the bottom of the pipe was greater than along the top would indicate that the oil film on the bottom of the pipe is thicker than along the top of the pipe. This can be substantiated by the fact that the oil is heavier than the ammonia. Also oil was conclusively known to be in the system because of the presence of oil in the sight glass.

Because the thickness of the oil film was not known, no attempt was made to determine the film coefficient for ammonia alone. The objective was to simulate actual operating conditions.

Effect of Change in Saturation Pressure

Because the only heat exchange medium available was water, saturation pressures corresponding to temperatures above the freezing point of water had to be used. In actual practice, temperatures much below this are used. To simulate these conditions, a relation recommended by M. Jakob (6), will be used. The equation is:

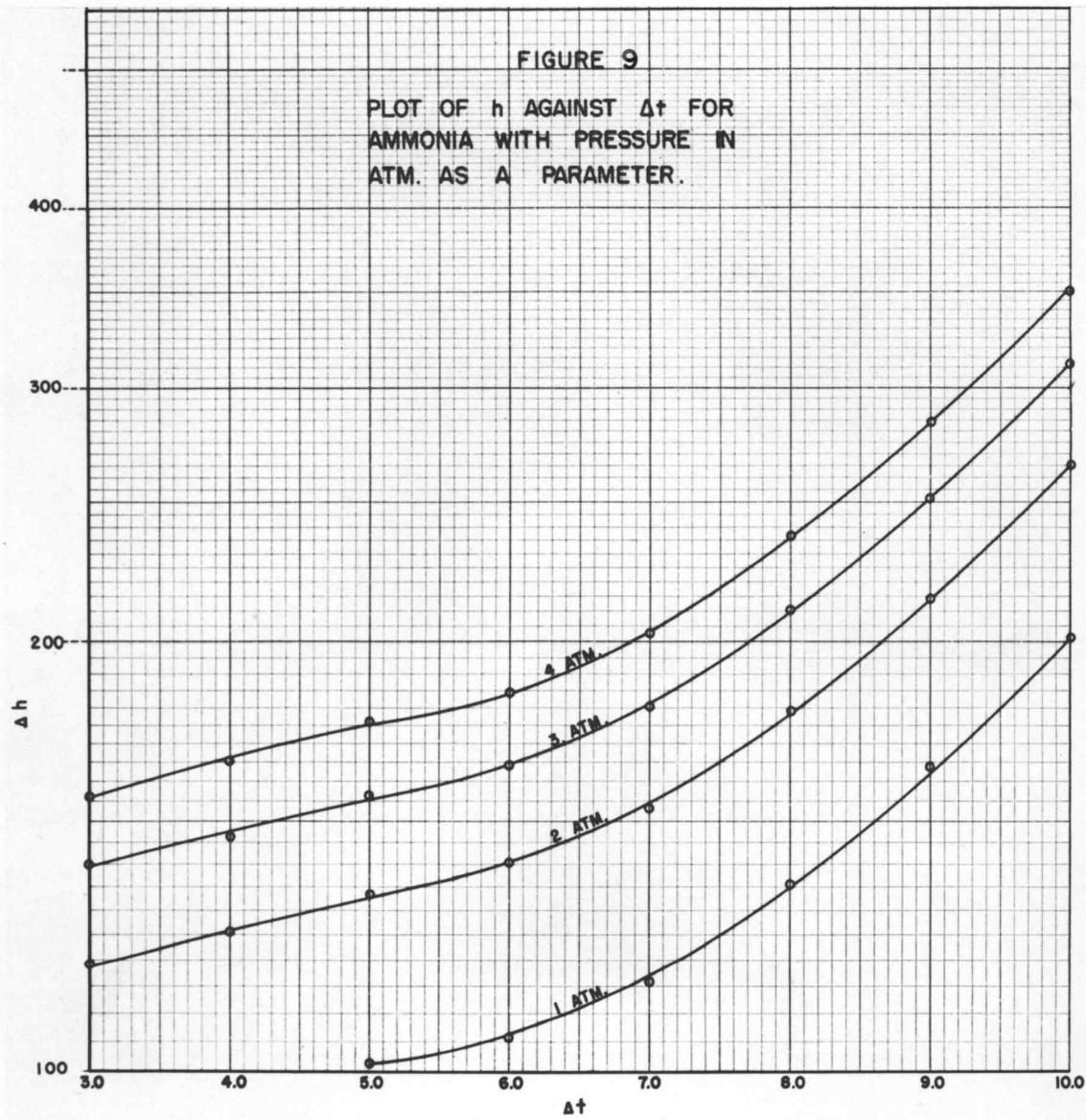
$$h = h_s (p/p_s)^{0.4}$$

wherein the subscript "s" relates to standard atmospheric conditions. This relation has been used in plotting Figure 9.

Comparison to Other Experimental Methods

The determination of the film coefficient for boiling liquids has been carried out by many investigators in the past. Their apparatus and procedure have differed from that used in this experiment. Most of the investigations have included the film boiling region as well as the nucleate boiling region (Figure 1). Also the evaporators take various and sundry shapes and positions, e.g., boiling inside and outside of tubes, vertical, horizontal, and inclined evaporators, horizontal plates, rough and polished heat exchange surfaces, etc. Therefore any comparison to other experimental work would be added only for operation similar to that employed in this experiment.

Ashley (7) has conducted a similar experiment on Freon-12, but the procedure was somewhat different. He used the total resistance method, that is, the mass velocity of the water had to be determined to calculate the film coefficient for the water, where this experiment completely eliminated the water film by the use of surface thermocouples. Also Mr. Ashley's evaporator was not



operated flooded, but was used as a direct expansion system where flash gas entered with the liquid. The results of Mr. Ashley's experiment seem comparable, although the units used in plotting graphs do not conform to those used in standard practice and seem meaningless.

Error

As has been stated in the description of apparatus painstaking precautions have been taken to make sure that the measurement of temperatures would be as accurate as possible. Surface temperatures which are difficult to measure would probably offer a sizeable error if thermocouples are not properly installed. This was illustrated by thermocouples nine and eleven (Figure 2) which were not filed down flush with the outside surface of the pipe and gave obvious error. Readings from these thermocouples were not used.

Error introduced by the use of mercury thermometers would be mostly due to not having the average water temperature. This can be neglected because the bulb of the thermometer would register an average temperature as its length would cover the cross section of flow.

Heat gain to the water from ambient outside conditions has been estimated to be about two per cent of the heat transferred and therefore can be neglected.

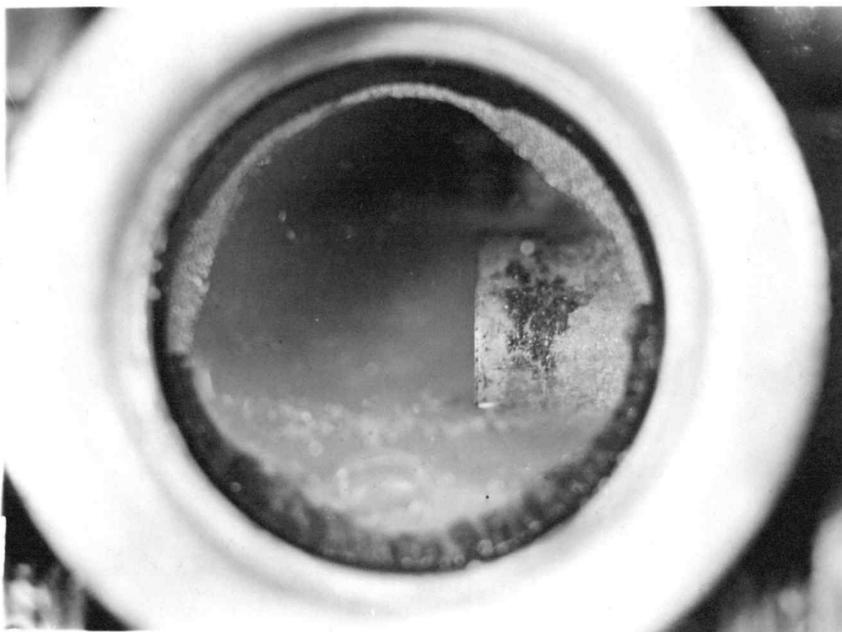
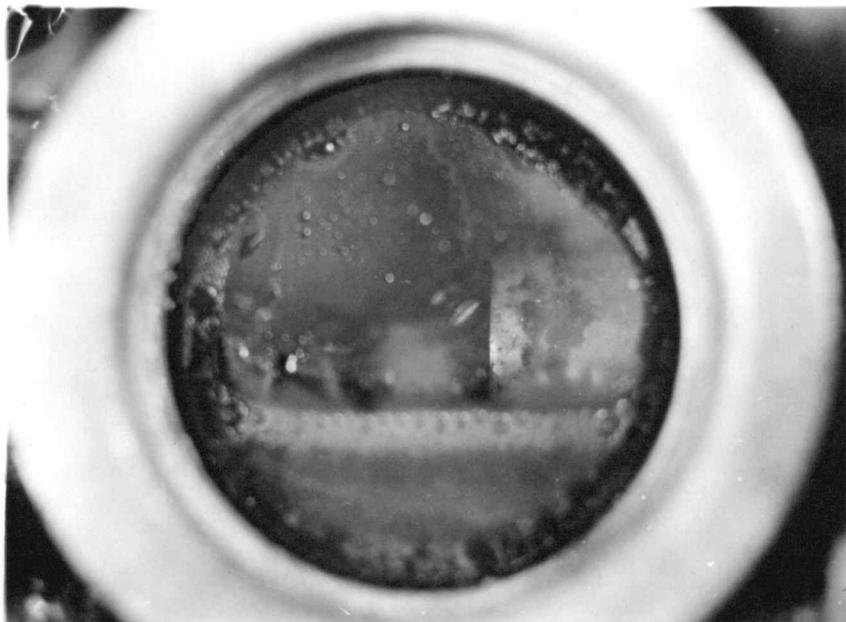


Figure 10. Photographs of ammonia boiling in an inclined sight glass. Photographs taken looking down into glass at an angle of 45 degrees.

Regulation by the low side float valve caused surges which could be seen in the sight glass and recorded by the pressure gage. The variation of static pressure due to this was not more than one pound per square inch for the low heat flux used in this experiment. This alone would account for appreciable error because the change in pressure also changes the saturation temperature of ammonia. A change in one pound per square inch changes the temperature about one degree F.

Pressure drop due to pipe friction was negligible. It can be seen from Figure 2 that the pressure on the entrance and exit to the evaporator was the same.

CONCLUSION

In trying to assess the value of experimental work presented in this paper one of the most important questions is: How far can the experimental results be extended with good accuracy? To some extent this question can be answered by a comparison of the operation of the experimental evaporator with operation in practical applications. Field applications are seen to be very comparable by the following description.

Recently flooded evaporator systems have come into use especially when used with blower units (8); Figure 11 shows how a flooded ammonia system operates. An accumulator, which is simply a large drum is connected to the discharge of the refrigeration coil. This accumulator then connects to the bottom of the coil by means of a tube. A definite liquid level is maintained in the accumulator by means of a float switch or a float valve, whichever may be desired. In operation liquid ammonia flows into the bottom connection of the coil and as it absorbs heat, gas bubbles are formed which tend to rise to the top and thus create a thermosyphon which causes liquid and vapor to circulate through the coil and back into the accumulator.

Except for conformation of the evaporators used in practical application it can be readily seen that results

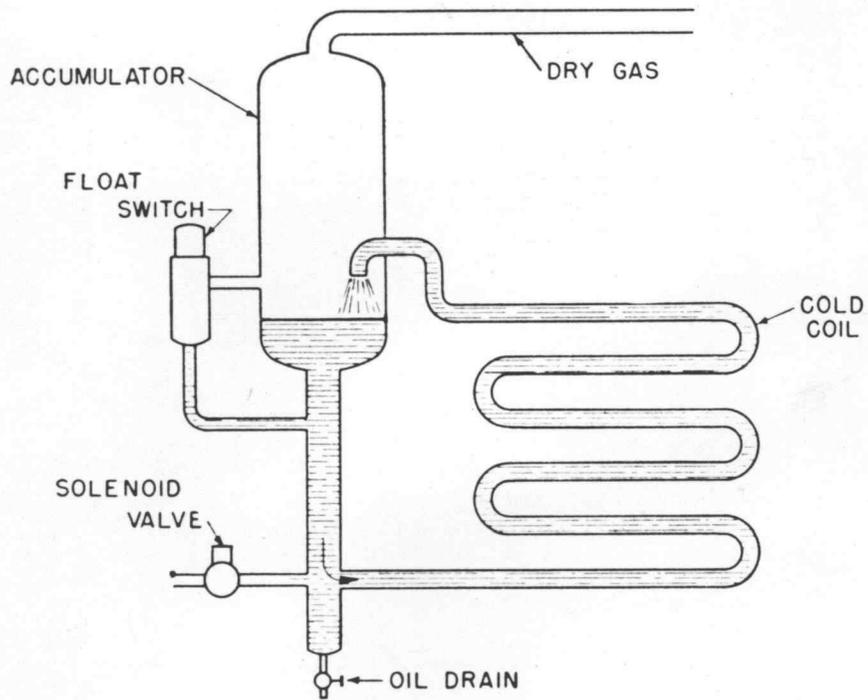


Figure 11. Schematic diagram of a flooded ammonia evaporator. Cold coil can be used with an air blower unit.

of this experiment can be extended to design of evaporators.

It is well known that the effect of oil in a refrigerant is very detrimental to the performance of an evaporator. This effect showed conclusively that there was an actual decrease in the film coefficient.

The oil which is used as a lubricant for the compressor in a refrigeration system is carried over into the condenser with hot ammonia gases. From the condenser the oil enters the evaporator. Even in high grade ammonia systems the carryover of oil into the evaporator cannot be wholly prevented even with separators and purgers. By this line of reasoning it would be believed that the accumulation of oil in the evaporator would give unpredictable performance over a period of time. This was not found to be true as reproducible results were obtained for an extended length of time.

In Figure 11 is shown an oil drain to purge oil from the system. The oil flows to this drain by gravity. This would work satisfactorily if static conditions existed, but intermittent operation would cause surges where oil would be carried over into the coils.

For the reasons given in the above paragraphs it is believed that the film coefficient found by experiment is sound and applicable. The range of mean temperature

differences used gives the lowest values of the film coefficient. By extrapolation to higher values of mean temperature difference, the film coefficient has an increasing slope up to a maximum. It is expected then that the worst possible conditions to be encountered in a flooded system are contained in the range determined in this experiment.

Further investigations in the determination of film coefficient for boiling ammonia would include operation of an inclined, and vertical evaporator. Present data on the various positions of an evaporator show that different results are obtained.

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