

**AN INVESTIGATION OF THE EFFECT OF ENTRAINED OIL ON THE
HEAT TRANSFER RATE OF A REFRIGERANT EVAPORATOR**

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**Thesis submitted to the Graduate Faculty of the
Virginia Polytechnic Institute
in candidacy for the degree of**

**MASTER OF SCIENCE
in
MECHANICAL ENGINEERING**

**October, 1964
Blacksburg, Virginia**

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I. INTRODUCTION

It is generally conceded that oil in a direct expansion evaporator will retard heat transfer. In a vapor compression refrigeration system using a reciprocating compressor oil is unavoidably circulated with the refrigerant throughout the system. Very little has been written concerning the effect of this oil on the heat transfer characteristics of an evaporator coil. It was the purpose of this investigation to determine quantitatively the effect of oil on the rate of heat transfer to an evaporator.

Recent investigations have shown that internal surface configuration does affect the heat transfer capacity of a coil by a measurable amount. For their master's thesis projects at the Virginia Polytechnic Institute, D. R. Riffe⁽¹⁴⁾ and R. M. Jesena⁽¹¹⁾ investigated the effects of internal surface configuration on the performance of a direct expansion evaporator. Riffe found that 60 internal longitudinal grooves in a 3/8" diameter evaporator tube increased heat transfer over a similar evaporator with smooth tubes. Jesena found that making the same number of helical grooves of approximately the same dimensions further increased heat transfer 6.62 per cent. Each of the above investigators recommended a study of the effects of oil on the performance of an evaporator.

The 1949 REFRIGERATION DATA BOOK⁽³⁾ contains a table compiled by S. R. Hirsch⁽³⁾ which shows how the evaporator capacity decreased as the oil in the refrigerant-oil mixture increased. Through personal

correspondence with Mr. Hirsch,⁽⁹⁾ the author learned that the results given in the REFRIGERATION DATA BOOK were inconclusive due to the manner in which the investigation was conducted and that they served only to show the relative magnitude of change that might be expected.

C. M. Ashley⁽¹⁾ stated that oil is detrimental to the performance of an evaporator. The oil tends to raise the evaporation temperature and in large enough quantities could retard heat transfer. However, he contended that the amount of oil present cannot be readily predicted. Rather than making an attempt to account for the magnitude of its effect, he recommended that an effort be made to eliminate circulating oil to the evaporator.

In spite of the wide use of direct expansion evaporators using a fluorinated hydrocarbon as the refrigerant, very little has been written concerning the effect of the entrained oil on the evaporator coil heat transfer. It is possible that the oil has a negligible effect on the refrigerant film coefficient since it is so much smaller than the air film coefficient of heat transfer. The purpose of this investigation was to determine quantitatively the effect of an oil admixture on the overall heat transfer coefficient across an evaporator coil, using Refrigerant-12 as the heat absorbing medium and air as the cooling medium.

II. REVIEW OF LITERATURE

General Review of Literature

Vapor compression is the most commonly accepted and widely used method for obtaining refrigeration. The vapor compression refrigeration system utilizes a circulating medium that alternately absorbs heat by evaporation and rejects heat by condensation. The theory of vapor compression refrigeration is discussed in numerous thermodynamics and refrigeration texts. A simple vapor compression refrigeration system includes an evaporator to absorb heat, a compressor to pump the refrigerant from the low side pressure to the high side pressure, a condenser to reject heat and an expansion valve to control the low side pressure.

The ideal prototype of the vapor compression refrigeration system is the reversed Carnot cycle. Between any two given temperatures a system operating on the reversed Carnot cycle is the most efficient refrigeration system obtainable. The reversed Carnot cycle can be represented on the temperature-entropy diagram as shown in Figure I. The processes comprising the cycle are an isothermal expansion 1-2 during which heat is absorbed, an isentropic compression 2-3 during which the pressure is raised from low side pressure to high side pressure, an isothermal compression 3-4 during which heat is rejected and an isentropic expansion 4-1 during which pressure is reduced from high side to low side pressure. The area between 1 and 2 below T_2

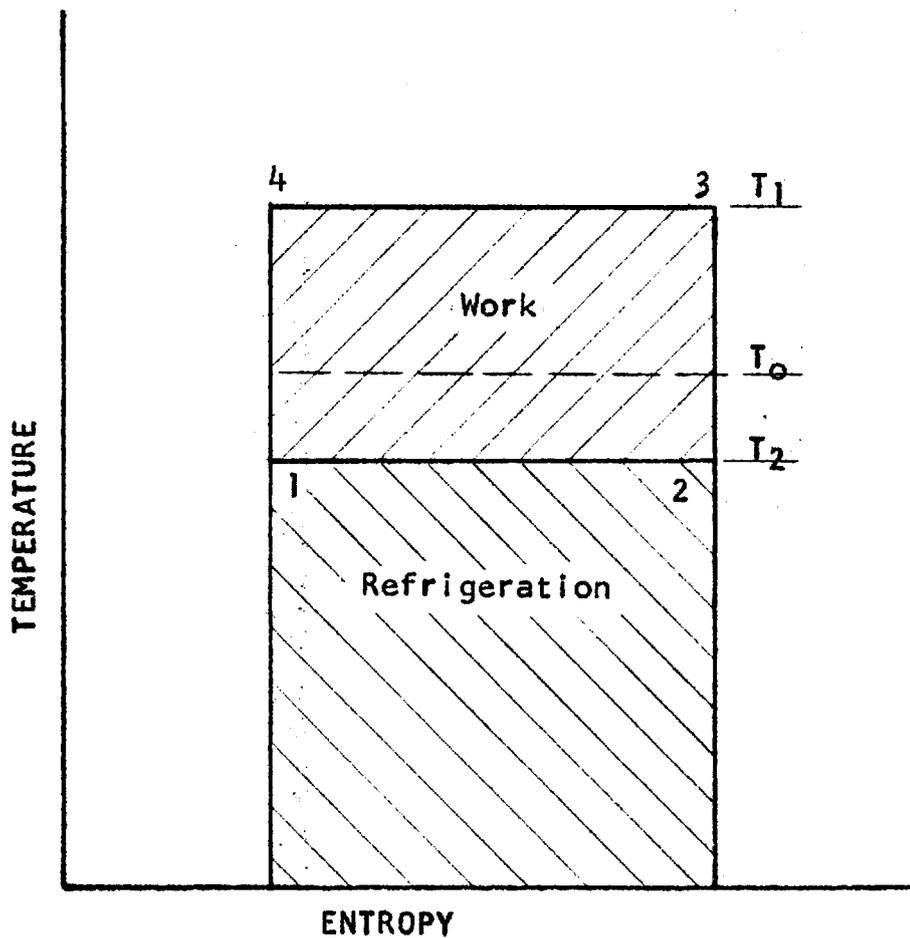


FIG. 1 REVERSED CARNOT CYCLE

represents the heat absorbed by the evaporator and is called the refrigerating effect. The area between 3 and 4 below T_1 represents the heat rejected by the condenser. The difference of these two areas represents the compressor work.

The Coefficient of Performance is a measure of the effectiveness of a refrigeration cycle and it is determined as follows:

$$C O P = \frac{Q_A}{W} \quad (1)$$

where Q_A is the refrigeration effect in Btu/hr and W is the compressor work in Btu/hr. The diagram in Figure I shows that for a given condensing temperature T_1 the Coefficient of Performance for the refrigeration cycle will be a maximum when T_2 is a maximum.

The basic equation for heat transfer to the evaporator is

$$Q = UA(T_0 - T_2) \quad (2)$$

where Q is the transferred heat in Btu/hr, U is the overall transmission coefficient in Btu/hr-ft²-°F, A is the reference area in ft² and $(T_0 - T_2)$ is the temperature difference across the evaporator tube. It is evident from equation 2 that Q is a maximum when the factors to the right are maximum. A maximum for $(T_0 - T_2)$ means a minimum T_2 which contradicts equation 1. It is also desirable that the product of the remaining two factors be a maximum for a minimum size evaporator. The high cost of materials and the demand to limit equipment size and weight requires that A be maintained at a minimum. It is apparent that to have the maximum heat transferred economically U must be a maximum. For the actual refrigeration cycle the quantity $(T_0 - T_2)$ must be replaced by the

logarithmic mean temperature difference ΔT_m which is a close approximation to the true mean temperature difference across the evaporator tube. The resulting equation for heat transfer becomes:

$$Q = UA \Delta T_m \quad (3)$$

The overall transmission coefficient for a clean tube based on the inside tube wall area is given by the equation:

$$1/U_i = 1/h_i + \frac{(D_i)(\ln D_o/D_i) + D_i/D_o h_o}{2k} \quad (4)$$

where U_i is the overall transmission coefficient based on the inside tube area in Btu/hr-ft²-°F, h_i and h_o are the inside and outside film coefficients respectively in Btu/hr-ft²-°F, k is the thermal conductivity of the evaporator tube in Btu-ft/hr-ft²-°F and D_o and D_i are the outside and inside tube diameters respectively in ft. Due to the high thermal conductivity of the evaporator tube material this term is negligible and may be dropped. The wall thickness of the evaporator tube will be small compared to the tube diameter which makes the ratio D_i/D_o very nearly unity. Equation 4 will then be reduced to the following equation:

$$1/U_i = 1/h_i + 1/h_o \quad (5)$$

It is evident from the above equation that for maximum U_i , h_i and h_o must also be a maximum. The purpose of this investigation was to determine to what extent refrigerant lubricating oil affects the overall transmission coefficient. The fact that oil reduces heat transfer is generally conceded, but quantitative data as to the extent of the effect is virtually non-existent.

Specific Review of Literature

Refrigerant Oil. (17), (18) Refrigerant oils are specially refined pure mineral oils referred to as pale oils. Blended oils, compound oils and overrefined pale oils break down and decompose in refrigeration service. A blended oil is a mixture of two fractions of mineral oil. A fraction refers to the temperature at which the various components of a crude oil will condense in a distillation refining process. The lighter fractions condense at the lower temperatures. The lighter fractions in a blended oil will vaporize leaving the heavy fraction which will be too viscous to lubricate the compressor. A compound oil is composed of a mineral oil and a fixed oil. A fixed oil is made from an animal or vegetable fat. The fixed oil will decompose readily in refrigeration service and form sludge. Overrefined pale oils are unstable chemically and also decompose under severe service to form acid conditions and sludge.

The oil for modern refrigeration equipment is selected for long life, especially for hermetically sealed systems where it must last for the life of the system. Some factors considered in the selection of a refrigeration oil are the refrigerant involved, method of lubrication and the operating temperature and pressure range of the system. Some refrigerants, such as, the halogenated hydrocarbons, are miscible in oil while others are not. A refrigerant miscible in oil will require a slightly heavier oil since the refrigerant reduces the viscosity of

the oil. The oil will, in turn, mix with the refrigerant which aids in the oil carryover process. An anti-foaming agent is normally added to help reduce the mixing.

A good refrigerant oil will exhibit such characteristics as maintenance of good body at high temperatures, good fluidity and form an effective oil film at lowest temperatures encountered in the system, resisting carbonization due to hot spots, resisting chemical reaction with the refrigerant and readily separating from the refrigerant. Numerous tests are used to determine the suitability of an oil for refrigerant service. Four tests applied to refrigerant oils are: the dielectric strength test, the cloud and pour point test, the viscosity test and the flash and fire point test. A pure dry mineral oil is a non-conductor of electricity and any impurities or moisture will decrease its dielectric strength rapidly. The dielectric strength is a purity test for the oil. It must withstand 25,000 volts per mm without sparking to pass as a refrigerant oil. The cloud and pour point gives the temperature at which wax crystals begin to form and ceases to flow respectively. Both temperatures must be below the lowest temperature expected in the system in which it is used. This practically limits the source of refrigerant oils to the wax free naphthene base oil. Refrigerant oils should have a high viscosity index. The flash point is the temperature at which an oil vaporizes sufficiently to sustain momentary ignition when exposed to a flame under atmospheric conditions. The fire point is the temperature at which an oil will sustain ignition continually when exposed to a flame under atmospheric conditions.

Refrigerant-Oil Mixture. Numerous investigations have shown that the coefficient of heat transfer for boiling refrigerants will vary widely depending on such physical factors as the temperature difference across the boiling film, character and shape of the surface, viscosity, velocity, quality and impurities, such as, oil. Oil has a tendency to congeal at low temperatures. As a result it will form a coating over the heat transfer surfaces of the evaporator when carried to that area by the refrigerant. This coating will act as an insulating material, thereby reducing heat transfer which in turn reduces system capacity.

The halogenated hydrocarbon refrigerants are to varying degrees miscible in oil. As a result the average refrigeration system using a reciprocating compressor and a halogenated hydrocarbon refrigerant will have oil carryover into the rest of the system. The amount of carryover will depend on the refrigerant used, condition of the compressor and the oil separator used. The following discussion will summarize the results of several significant investigations in which oil in the refrigerant was considered as a factor affecting the heat transfer coefficient of a boiling refrigerant.

While attempting to determine a film coefficient for dichlorodifluoromethane (CCl_2F_2) or Refrigerant-12, C. M. Ashley⁽¹⁾ found that the refrigerant deviated in pressure characteristics from the published data. The temperature variation was $1\text{-}1/2^\circ$ at 40° suction temperature to 4° at 60°F suction temperature. In an effort to obtain published saturation temperatures the system was charged with new

refrigerant, with extra care used to exclude all oil. Ashley stated that oil was detrimental to the evaporator performance, principally due to its tendency to raise the evaporating temperature rather than reducing the film coefficient. He did not attempt to evaluate the effect on heat transfer in an evaporator due to oil in the refrigerant. However, he did list oil as one of the variables which should be investigated.

Whitzig, Penny and Cyphers⁽¹⁹⁾ found that 1% by volume of oil in the refrigerant produced a significant reduction in heat transfer. They used a sealed thormal siphon system which made it possible to circulate a predetermined quantity of oil with the refrigerant. This system included a well insulated horizontal tube which served as an evaporator and which was located at the lowest point in the system. A variable output resistance heater wire wrapped around the tube served as the load with a second heater similarly mounted downstream to insure superheat leaving the evaporator. The leaving vapor traveled vertically through a rotometer to a single tube condenser inclined to drain the liquid refrigerant to a vertical receiver. The weight of liquid in the vertical receiver supplied the head on the manual expansion valve. Three separate tests were made regarding oil in R-12; first with no oil, second with 0.1% and third with 1% by volume of oil in the refrigerant. The results of the first two tests were the same. The third test showed a slight decrease in the film coefficient. The results are shown in Figure II. The figure shows, for example, that

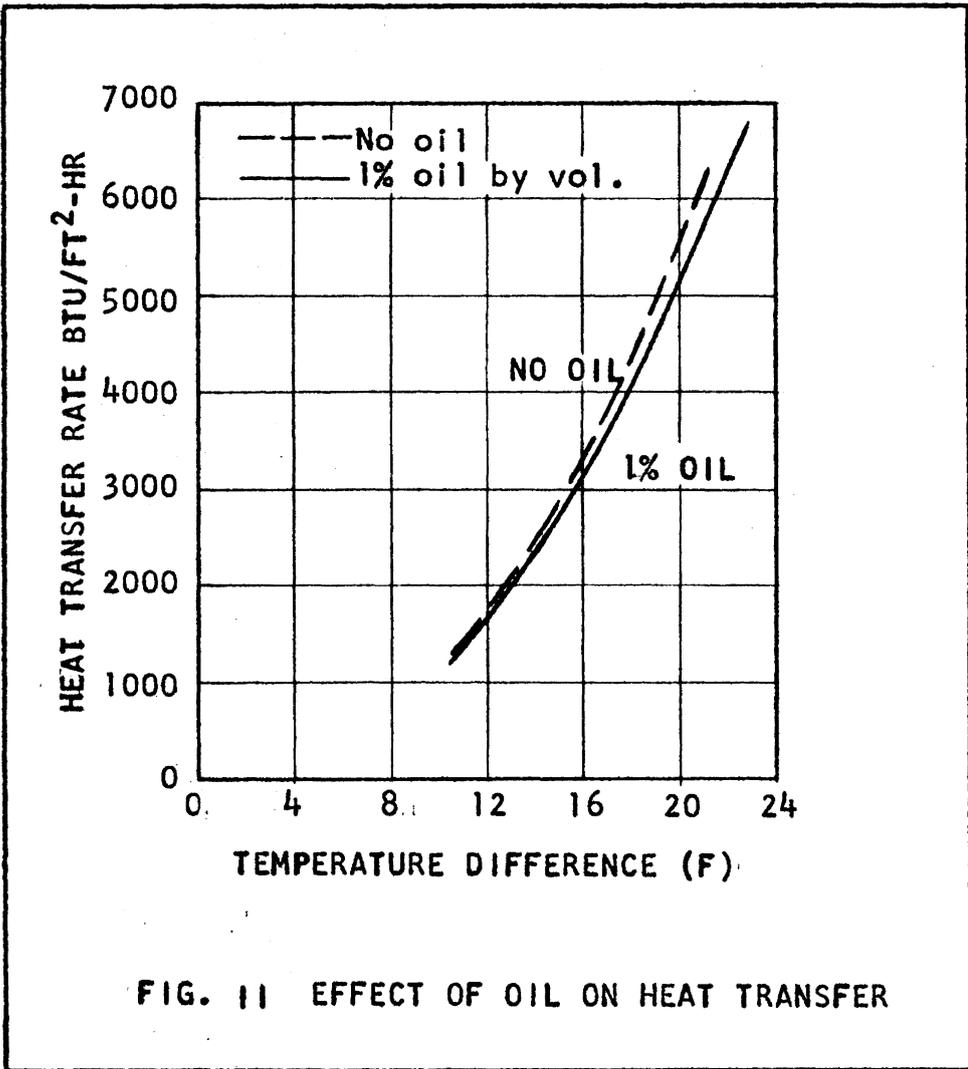


FIG. 11 EFFECT OF OIL ON HEAT TRANSFER

REPRODUCED FROM WHITZIG, PENNY AND CYPHERS, HEAT TRANSFER RATES TO EVAPORATING FREON-12 IN A HORIZONTAL TUBE EVAPORATOR; REFRIGERATION ENGINEERING, VOL 56, PT 2, AUGUST, 1948, P 156.

for a 20°F temperature difference across the refrigerant film there is approximately 350 Btu/hr-ft² difference in heat transfer for a condition of no oil versus 1% by volume of oil. The approximate total heat transferred at this same temperature difference with no oil was 5,700 Btu/hr-ft². The reduction in heat transfer due to oil was 6.1%.

Seigel, Bryan and Huppert⁽¹⁵⁾ made an investigation to examine and correlate the factors which seem to have the greatest effect on heat transfer of a boiling refrigerant in an evaporator tube. They concluded that the presence of oil did not seem detrimental to the heat transfer in the quantities present, but that investigation should be made to determine quantitatively the amount of oil that could be tolerated. The equipment used in their investigation included a variable speed compressor, variable output electric heaters to supply evaporator load and a rotometer in the liquid line. To check the quantity of oil in the refrigerant, the rotometer float was removed and a hydrometer inserted in its place to note any change in specific gravity. The compressor speed was varied in an effort to induce more or less oil to be carried over into the system. The investigators noted a change in color of the mixture prior to the expansion valve and a change in temperature read on a thermometer at exit from the evaporator, however, they reported no change in specific gravity or heat transferred. An oil separator was then installed in the circuit and the refrigerant entering the expansion valve became clear and no further erratic temperature readings were experienced. Seigel et al give the results obtained for four

different evaporator tests on a log-log curve with Q/AW_f versus ΔT the abscissa; where Q/A is the heat flux in Btu/hr-ft², W_f is the flow rate of the R-12 in lb/hr and ΔT is the temperature difference between the tube and the refrigerant in degrees F. The data for all four evaporators gave a single rather well defined curve, however, all data points for the evaporator undergoing the oil test lie to the right and below data points from the other tests in which oil was excluded. The inference being that oil caused a greater ΔT for comparable heat transfer rates. The magnitude of the greater ΔT was from less than 1 to 3°F, however, it is possible this is the increase which should have been expected with the amount of oil that could be induced to circulate with a commercial compressor.

Yoder and Dodge⁽²¹⁾ made a study of boiling R-12 in a vertical tube to obtain heat transfer coefficients in the range of -60 to -100°F. A three stage reciprocating compressor was used. No attempt was made to evaluate the effect of oil, however, periodic checks showed that approximately 5.1% oil by weight was circulated. The method of sampling was not explained. According to the test data given the oil content was checked in each case after several runs. Depending on the manner sampled this could represent an accumulation of oil in the evaporator. Pierre⁽¹³⁾ found the oil to vary from 0 to 18% by volume using a reciprocating compressor.

According to published tables by S. R. Hirsch⁽³⁾ in the 1949, REFRIGERATION DATA BOOK, oil in the refrigerant raises the boiling temperature of the refrigerant and reduces the evaporator capacity. He

shows the rise in boiling temperature due to a given per cent by weight of oil in the refrigerant at various temperatures. A second table gives the per cent reduction in evaporator capacity due to a given per cent by weight of oil in the refrigerant. By personal correspondence, Hirsch states the test conditions by which the data were obtained were not accurately controlled or measured. The results were only to indicate the order of magnitude of the effect for small coils.

Sharpe⁽¹⁶⁾ conducted tests to determine oil flow characteristics throughout a refrigeration system. He found that with as low a concentration as 0.6% by weight of oil, an oil film which flowed with suction gas was established on the tube walls. With a concentration of 1% by weight his photographs show a definite film deposited on transparent test sections.

The general conclusions of the articles researched were that commercial systems invariably circulate some oil with the refrigerant, that the oil will reduce heat transfer if present in large enough quantities and that an investigation to determine its quantitative effect should be made. Since oil is a factor in the overall heat transfer coefficient of a boiling refrigerant it would be appropriate and necessary to have data showing its quantitative effect.

III. INVESTIGATION

Object

The object of this investigation was to determine the effect of oil, circulating with the refrigerant, on the heat transfer characteristics of a direct expansion evaporator coil. The investigation was conducted on a refrigeration system similar to that being used commercially in order that the results might be immediately useful.

Plan

The evaporator on which the test was performed was installed in a duct through which was supplied a constant quantity of air at a predetermined condition of wet and dry bulb temperature. Using a modified refrigeration system, it was possible to change the concentration of oil in the refrigerant passing to the evaporator. A measured sample of refrigerant liquid was collected at entrance to the expansion valve and diverted to an expansion chamber which facilitated volumetric determination of the quantity of oil in the sample.

A multifin, smooth bore evaporator was installed in a conventional refrigeration system similar to that used in commercial applications. The system was modified to accomplish the following: to provide a method to change the oil concentration in the refrigerant, to collect a sample of liquid refrigerant, to separate the refrigerant from the oil in the sample and to observe the events inside of the system at

several locations. An increase in oil concentration was effected by bypassing the oil separator and dryer. The initial tests were made with a minimum of oil in the crankcase and with the separator included in the system. The separator was bypassed and more tests were made with the same oil level in the crankcase. For subsequent tests the oil level in the crankcase was increased to a predetermined operating level and runs were made at each different crankcase oil level with both the separator in the system and the separator bypassed. A refrigerant sampling circuit was installed upstream of the expansion valve to facilitate collecting a sample of liquid refrigerant without disturbing system operation. The sampling circuit is described under equipment and its operation is described under procedure. The collected sample was delivered to the expansion chamber in which a major part of the liquid vaporized. The remaining liquid in the expanded sample aided in washing oil into a calibrated test tube at the bottom of the expansion chamber. The vapor was extracted from the expansion chamber by opening the expansion chamber suction control valve which vented to the system suction line. In order to observe the flow pattern at several selected points inside of the refrigerant system extra strength Pyrex glass sections were installed in the refrigerant lines. A 12" cylindrical glass section 1/2" in diameter was installed in both the liquid sampling section and the return line from the oil separator. A glass tee was installed in the suction line at exit from the evaporator. In addition, Plexiglas safety shields were mounted over the glass sections as a safety precaution.

The air was supplied through an insulated duct which was instrumented to measure the dry bulb air temperature entering and leaving the evaporator and the wet bulb temperature entering the evaporator. Constant entering air conditions were obtained by passing the air in series through a washer and a variable capacity electric heater. The moisture condensed from the air by the evaporator was collected at a drain beneath the evaporator. A water chiller in series with a water-to-water heat exchanger was used to control the washer water temperature. The air entering the evaporator was maintained at 80°F dry bulb and 67°F wet bulb.

Sufficient instrumentation was applied to both the refrigeration system and the conditioned air system to facilitate a heat balance. In order to make a heat balance for the refrigerant system it was necessary to know the refrigerant flow rate, the refrigerant temperature at entrance to the expansion valve and at the exit from the evaporator and for the air conditioning system it was necessary to know the air flow rate, the dry bulb temperature of the air entering and leaving the evaporator, the moisture condensation rate for the air and the average dew point temperature of the air crossing the coil.

There was a possibility that errors were introduced in the heat balances due to heat transfer through the insulated duct and in the thermocouple readings due to radiation interchange and the inherent limitations of thermocouples. Heat transfer could have occurred from the air in the duct to the room before the evaporator and from the room to the air in the duct after the evaporator. The evaporator return bends

and sides were insulated with cork and fiber glass to reduce heat transfer and the thermocouples measuring refrigerant temperatures were located as near the coil entrance and exit as feasible. Since the same range of temperatures prevailed during all tests there was a tendency for errors to be equal for all tests.

The basic equipment constructed and assembled by D. R. Riffe⁽¹⁴⁾ and used by R. M. Jesena⁽¹¹⁾ was modified extensively for this investigation. The modifications are discussed in detail under Equipment.

Equipment

Original Equipment. The original equipment was used to conduct two previous investigations on the effect of internal surface configuration on evaporator performance. The original equipment was fully discussed by Riffe⁽¹⁴⁾ and Jesena.⁽¹¹⁾ Only the modifications and additions necessary for this investigation are discussed here. For information on equipment not herein described references (7) and (10) should be consulted. A view of the original equipment is shown in Figure III.

Modified Equipment. The equipment could be divided into three separate divisions for purposes of discussion: the air conditioning equipment, the refrigeration equipment and the measuring equipment. The modifications and additions are discussed in turn for each division. Figure IV is a schematic diagram for the modified equipment necessary to this investigation.

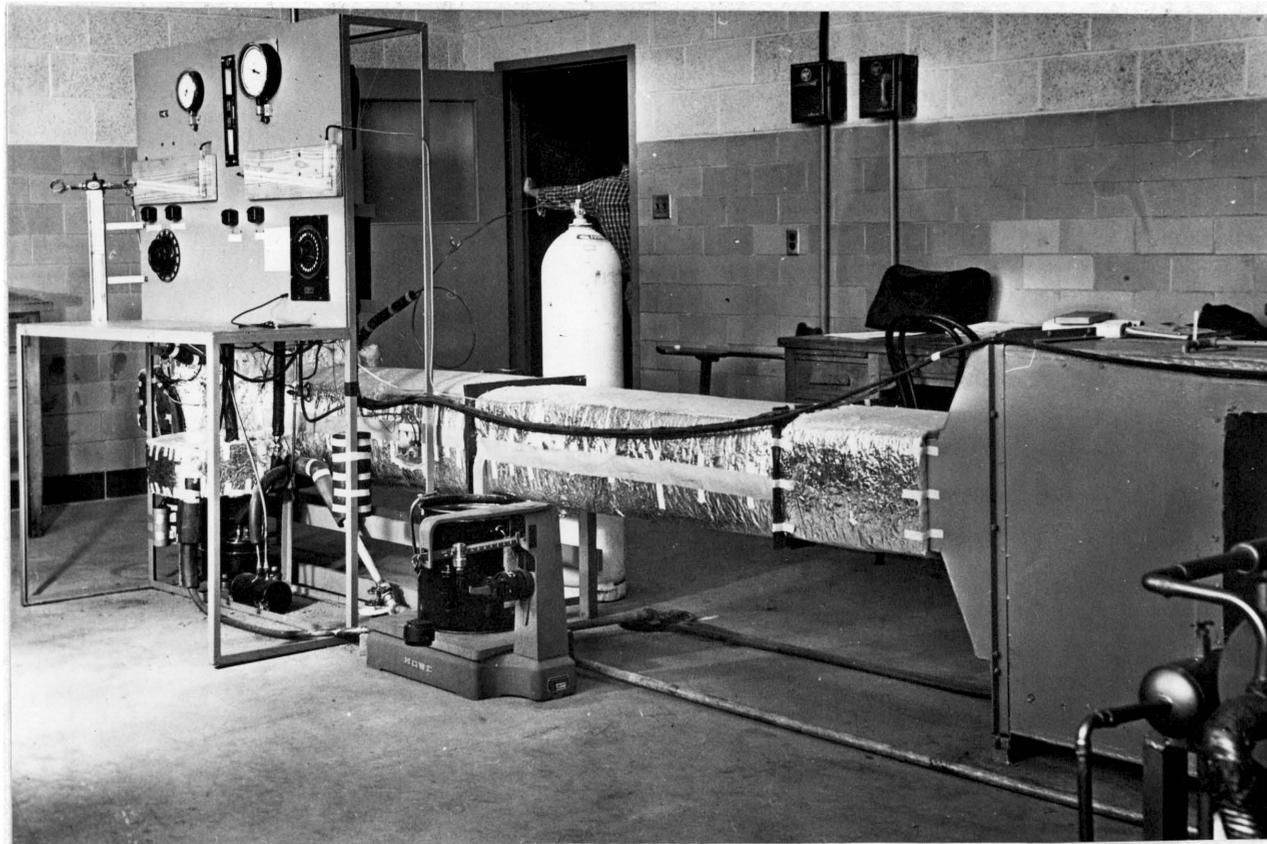


FIGURE III. VIEW OF THE ORIGINAL EQUIPMENT REPRODUCED FROM RIFPE

LEGEND TO CIRCLED LETTERS AND NUMBERS USED IN FIGURE IV

- A - Cold water shut off valve
- B - Hot water shut off valve
- C - Washer water shut off valve
- D - Heat exchanger hot water control valve
- E - Chilled water drain valve
- F - Condenser water control valve
- G - Refrigerant filter bypass control valve
- H - Sampling section bypass control valve
- I - Sampling section inlet valve
- J - Sampling section exit valve
- K - Manual expansion valve
- L - Oil separator bypass valve
- M - Oil separator discharge valve
- N - Expansion chamber delivery line valve
- O - Expansion chamber suction control valve
- P - Evacuating and charging manifold control valve
- Q - Vacuum pump control valve
- R - R-12 charge valve
- S - Drain valve for observation tee in suction line
- T - Oil tank discharge valve
- U - Oil tank fill valve
- V - Damper control knob
- W - Stratification eliminator baffle ahead of evaporator

X - Electric heater control switch

Y - Compressor switch

Z - Fan switch

CIRCLED NUMBERS INDICATE POINTS OF TEMPERATURE MEASUREMENT

1 - Upstream dry bulb thermopile

2 - Upstream wet bulb thermopile

3 - Upstream dry bulb averaging thermocouple

4 - Downstream dry bulb thermopile

5 - Downstream wet bulb thermopile

6 - Downstream dry bulb averaging thermocouple

7 - Refrigerant liquid temperature entering evaporator

8 - Suction gas temperature

Air Conditioning Equipment

Stratification Eliminator Vanes. Four successive rows of flat plate vanes were installed immediately upstream from the air flow measuring nozzle in an effort to eliminate stratification. The vanes were fabricated from USS gage 18 sheet metal in 1-1/2" width strips.

Refrigeration Equipment

Compressor. In place of the compressor used in the original apparatus a Frigidaire Model CAWO - 100 condensing unit, serial number 5440649 was used for compressing the refrigerant. The unit included a 1 hp, 208/220 volt, 3 ph., 3.8 amp motor, motor starter, condenser and receiver.

Refrigerant Sampling Circuit. The refrigerant sampling circuit shown in Figure V, consisted of a test sample collecting section in parallel with the liquid line. The test sample collector was a 12" section of 1/2" Pyrex glass tube with a packless isolating valve at each end. A 1/4" copper line delivered the sample through a packless control valve to the expansion chamber. The sample collecting section was isolated by valves 6, 7, and 8 in Figure V. The parallel liquid line contained a packless valve which was closed to force refrigerant through the sampling section. The total volume enclosed within the sample collecting section was 74 cc.

Expansion Chamber. An expansion chamber was fabricated from an 18" section of 2-1/2" standard black steel pipe. The chamber was mounted in a vertical position. A steel funnel which emptied into a 5 cc calibrated

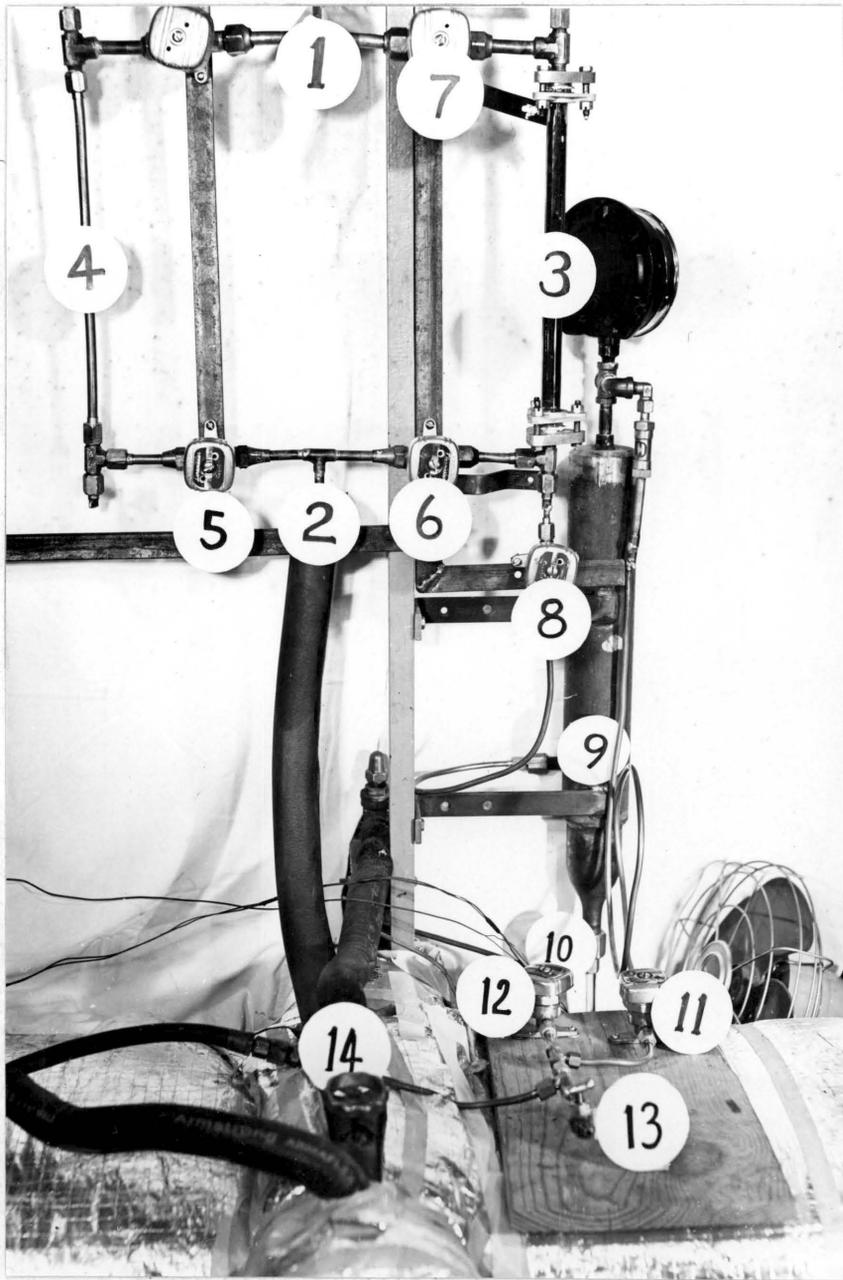


FIGURE V. VIEW OF SAMPLING CIRCUIT

See Next Page for Legend to Circled Numbers

LEGEND FOR CIRCLED NUMBERS ON FIGURE V

- 1 - Entrance to sampling circuit
- 2 - Exit from sampling circuit
- 3 - Glass sampling section
- 4 - Bypass line
- 5 - Bypass shut off valve
- 6 - Sample section exit valve
- 7 - Sample section inlet valve
- 8 - Valve in line to expansion chamber
- 9 - Expansion chamber
- 10 - Calibrated test tube
- 11 - Expansion chamber discharge control valve
- 12 - Control valve between system suction and evacuating and charging manifold
- 13 - Control valve between vacuum pump and evacuating and charging manifold
- 14 - Manual expansion valve

test tube was welded to the pipe to form the lower end. The test tube was held in place by a specially fabricated brass adapter that clamped the test tube lip between two rubber gaskets. A flat steel plate welded in place formed the top of the chamber. A 1/4" copper suction line extended from a tee on top of the chamber to an evacuating and charging manifold. An Ashcroft Bourdon tube dual pressure gage with a range of 150 psi gage graduated in 5 psi increments was mounted at the same tee on top of the expansion chamber. The delivery line from the sampling section entered the side of the expansion chamber tangentially and just above the top of the funnel. The suction line from the expansion chamber contained a packless control valve. The expansion chamber, item 9, and the calibrated test tube, item 10, are shown in Figure VI.

Fan. A 10" Kenmore oscillating desk fan was used to blow room air across the expansion chamber to increase the boiling rate of the refrigerant within.

Evacuating and Charging Manifold. A manifold consisting of the necessary tubing, fittings and packless valves was assembled to facilitate quick and easy evacuation of the expansion chamber, or for replenishing the charge when necessary.

Oil Storage Tank. The oil storage tank was fabricated from a 10" section of 3-1/2" standard black steel pipe by welding flat steel plates on each end. One plate was larger in diameter by 2" than the outside of the tank and contained 3 bolt holes to facilitate mounting on the rear of the control panel. The other end was fitted with a sight



FIGURE VI. VIEW OF GLASS SUCTION LINE TEE AND CALIBRATED TEST TUBE

glass. A 3/8" copper line was installed between the bottom of the tank and the original oil charge line. A 3/8" fill line was attached to the top of the tank. A packless valve was installed near the tank in both lines.

Measuring Equipment

Micromanometer. A twenty-inch micromanometer, serial number B-6361, manufactured by the Trimount Instrument Company of Chicago, Illinois, was used to measure the pressure drop across the air flow nozzle. The manometer used oil with unity density as the manometer fluid and was graduated in increments of 0.0005".

Averaging Thermocouples. The dry bulb temperatures entering and leaving the evaporator were measured with copper-constantan averaging thermocouples consisting of several thermocouples in parallel assembled with twisted and soldered joints. Six thermocouples mounted in a grid at the exit from the air flow nozzle comprised an averaging thermocouple which measured the air dry bulb temperature entering the evaporator. An averaging thermocouple comprised of nine equally spaced thermocouples forming a grid at the end of the duct measured the air dry bulb temperature leaving the evaporator. All thermocouples were fabricated from adjacent sections from the same spool of number 24 copper-constantan wire. The calibration curve for the thermocouples used in this investigation is shown in Appendix V.

Potentiometer. A Rubicon Instruments Potentiometer, model number 2745, case number 121,644, manufactured by the Minneapolis-Honeywell Regulator Company, was used to measure the thermocouple emf. The dial graduations for the potentiometer were in increments of 0.005 mv with enough space between graduations to estimate to the nearest 0.001 mv.

Pressure Gages. A U. S. Gauge Company, number 7037, Bourdon tube dual pressure gage was used to measure suction pressure at the exit from the evaporator. The range of the gage was from 0 to 60 psi in 1 psi increments. The calibration curve for this gage is given in Appendix VI. A Bourdon tube dual pressure gage by Ashcroft was used to give the relative magnitude of the pressure or vacuum in the expansion chamber as an aid in the operation of the evacuating and charging manifold.

Beam Balance. An Ohaus beam balance, manufactured by the Fisher Scientific Company, was used to weigh the condensate. A 400 cc glass beaker was used to collect the condensate.

Stop Watch. A stop watch with a 30-second-per-revolution dial, as manufactured by A. R. and J. E. Meylan, was used to record the time to collect enough condensate to balance the counterweight on the beam balance. The range of the stop watch was from 0 to 15 minutes with graduations to 0.1 second.

Procedure

Preliminary Procedure. A number of preliminary trial runs were made to check the instrumentation and to test the refrigerant-oil

sampling and separating procedures. Several sampling and separating methods were tried and discarded before a suitable method was found. Also, several changes were made in the original instrumentation procedures prior to the investigation. The reasons for these changes are discussed here as an aid to future investigations.

The first method considered for collecting a sample was a removable flask connected to the liquid line through two packless refrigerant valves with a union joint between the valves. The charged flask was removed from the system; weighed; the refrigerant allowed to boil off, leaving the oil; and then the flask was reweighed. The increase in weight over the original dry flask and fittings was the weight of oil collected. The mechanics involved in disconnecting, reconnecting, cleaning and purging or evacuating the flask were laborious and tedious. In addition the weight of oil involved in most runs was less than the possible error involved in weighing the flask and fittings.

A second method considered for collecting a sample was to allow the refrigerant sample to boil off to the atmosphere through a vertical standpipe beneath the sample collecting section shown in Figure V. The oil remaining in the collecting section and the standpipe would eventually drain to a collecting basin beneath the standpipe. The refrigerant boiled violently evaporating any oil with it. In essence the most feasible method to control the boiling refrigerant was by making it a part of the refrigerant system. The expansion chamber and the oil separating procedure used in the investigation evolved as a result of these experiences.

The expansion chamber was designed to be large enough to allow most of the collected sample to vaporize. The chamber was located beneath the sampling section to allow the liquid refrigerant to drain by gravity to the chamber. Due to the tangential entrance, the refrigerant liquid tended to wash any oil to the bottom of the funnel and into the calibrated test tube. The exact size of the expansion chamber was not important so long as a small amount of liquid was available to wash down the oil. Periodically the calibrated test tube had to be emptied of oil and cleaned. As a result it was necessary to evacuate all parts of the system opened to the atmosphere. To facilitate quick and easy evacuation of the expansion chamber without interference to system operation, an evacuating and charging manifold was fabricated. Attached to the manifold with a separate control valve to each were a vacuum pump, a tank of R-12, the expansion chamber and the suction line to the compressor. By proper valve adjustment the compressor could be used to evacuate the expansion chamber. Using this method the boiling rate of the refrigerant was always under close control; in addition the refrigerant was saved.

The effect of the oil on evaporator performance was determined by the rate of heat transfer to the refrigerant in the evaporator. The heat lost by the air served as a check on the amount of heat transferred. The initial heat balance made for the system during the preliminary runs indicated that the air was losing 25% more heat than was being received by the refrigerant based on heat transfer to the refrigerant.

Since the heat to the refrigerant served as the control, the refrigerant system was checked first for errors.

The rotometer was calibrated as described in Appendix III. The resulting calibration curve showed approximately 6% greater flow rate than the curve obtained by Riffe.⁽¹⁴⁾ The heat transfer rate to the refrigerant correlated closely with the manufacturer's published data on the capacity of this unit.

The air conditioning air flow rate was checked with an anemometer, a velometer and a pitot tube. These instruments were first calibrated with an air flow nozzle. The air flow rates for the anemometer, velometer and water filled draft gage were in close agreement. However, it was noticed that reaction of the water column in the slightly inclined draft gage was sluggish and unsteady and for a constant duct pressure did not always give repetitive readings. A micromanometer as described under measuring equipment and calibrated as described in Appendix II was substituted for the draft gage.

The thermopile upstream from the evaporator was checked against the thermopile downstream by turning on the air conditioning system only. The dry bulb thermopiles agreed within 0.1°F. The wet bulb thermopiles disagreed, but never by the same amount over a period of several days. The four thermopiles were simultaneously immersed in a water bath and the temperature of the water changed by adding ice or heat. At each different water temperature the thermopiles gave the same emf. The condensate from the evaporator was collected and weighed. It

was approximately 40% of the amount expected. The enthalpy of the missing condensate was equal to the extra heat attributed to the air. As a result the wet bulb thermopiles were not considered sufficiently accurate for determining enthalpy change of the air across the evaporator. The resulting decision was to weigh the condensate. The wet bulb thermopile upstream continued to be used to give an indication of the wet bulb temperature of the entering air.

For several successive runs following run number 28, shown in Table II, the air to refrigerant heat balance showed a large discrepancy. A horizontal temperature traverse was made by moving a thermocouple across the nozzle throat. The temperature was found to vary more than 2°F. A traverse at exit from the duct showed a similar situation. It was possible that the dry bulb thermopiles were originally located in the same temperature strata and that a change in flow pattern caused a change in stratification pattern. Stratification eliminator vanes were installed in the four foot section of duct between the air flow nozzle and the volume control damper in an effort to thoroughly mix the air. According to D. D. Wile⁽²⁰⁾ a chamber to mix the air sufficiently to require only one temperature measuring point would require more space and fan power than was available in the present equipment. Also, according to Wile, the only other alternative was a large number of thermocouples in parallel giving an average temperature measurement. The vanes did not change the pattern of temperatures at the air flow nozzle or at the exit from the duct. Averaging thermocouples were installed at each location and the dry bulb thermopiles were

discarded. The averaging thermocouples are described under equipment.

Experimental Procedure. The equipment was so arranged that it could be turned on in any order except for the electric heaters, which were turned on after the air conditioner fan. The condenser water for the chiller was adjusted to maintain approximately 225 to 250 lb/in.² gage discharge pressure. The water valve to the washer nozzles was fully opened and the chilled water bleed valve was partially opened. The chilled water bleed valve was adjusted to maintain the required washer temperature and yet bleed enough water to maintain continuous chiller operation. This assured constant temperature water. Additional washer water temperature control was supplied by the hot water heat exchanger when necessary. With the aid of the variac and the washer water temperature controls the air was maintained at 80°F dry bulb and 67°F wet bulb.

The micromanometer was adjusted to indicate the required drop across the air flow nozzle to give 500 cfm flow rate. The micromanometer was adjusted as follows: disconnect the air lines from the duct to the manometer, level the manometer, set the manometer to zero, allow the oil in the manometer reservoir to come to equilibrium, adjust the pointer to touch the surface of the oil, set the manometer scale for the appropriate reading according to the calibration curve in Appendix II and reattach the air lines between the manometer and the duct. The air flow control damper was adjusted until the pointer touched the surface again.

The compressor was turned on, the manual expansion valve opened to approximately 75% of normal operating flow and the condenser water turned on. If the condenser water which was supplied by the chiller was turned on first the compressor tended to slug. With condenser flow established the expansion valve was adjusted to give approximately 10°F superheat. The condenser flow was adjusted to maintain approximately 120 lb/in.² gage discharge pressure. Minute adjustment of the condenser cooling water flow rate was obtained with slight adjustment to the chilled water bleed valve. The slight change in condenser water pressure served as a minute flow adjustment.

At each system start up the refrigeration system was operated a minimum of two hours prior to collecting data. In addition, the system operated a minimum of 45 minutes after final adjustment of refrigerant flow rate and condenser water flow rate prior to collecting data. Due to the coarse adjustment obtainable with the manual expansion valve, a close control of superheat was difficult to maintain without frequent adjustment. Since a change in flow rate changed gas velocity, which, in turn, affected the oil film, the variation in superheat was accepted. During operation the bypass line in the sampling circuit was closed forcing the refrigerant through the glass sampling section, thereby insuring that a representative sample would always be available to the sampling section. The expansion chamber delivery line valve was closed. The chamber was at approximately suction line pressure and ready to receive a sample of refrigerant. The evacuating and charging manifold

control valve was closed. The oil separator discharge valve and bypass valve were appropriately adjusted. To obtain a minimum of oil in the system, the oil separator discharge valve was opened first and then the oil separator bypass valve was closed. During all tests the refrigerant filter was bypassed.

The moisture condensed from the air by the evaporator was collected in a 400 cc beaker resting on the platform of the beam balance. The condensate flow rate was determined by measuring the time required for the collected condensate to balance a 0.4 lb counterweight. Approximately 10 minutes were required to collect 0.4 lb of condensate. The condensate issued from the condensate drain at a steady rate after approximately 45 minutes operation of the refrigeration system.

The procedure used while collecting data was as follows: simultaneously, an empty beaker was placed on the platform of the beam balance under the condensate drain and the stop watch was started. While condensate was being collected other data listed in Table II, run 1, was recorded. If the entering air dry bulb temperature or the suction temperature varied more than 0.2°F while condensate was being collected the data for the run was discarded. Immediately after the condensate had been collected a refrigerant sample was collected. The valve in the sampling section bypass line was opened. The sampling section exit valve was closed first and then the inlet valve. The expansion chamber delivery line valve was opened long enough for the sampling section to completely drain. The expansion chamber suction valve was used to control

the flow rate from the chamber until the suction line pressure was reached. At this point the expansion chamber suction valve was fully opened and the remaining refrigerant was allowed to boil off. During this operation the oscillating fan circulated room air across the exterior of the expansion chamber to increase the rate of heat transfer from the evaporating refrigerant. When the oil in the test tube came to complete rest at room temperature and suction pressure it was considered to be essentially refrigerant free. The quantity of oil collected for the particular refrigerant sample was read directly from the calibrated test tube in cubic centimeters.

An increase in the quantity of oil circulating with the refrigerant was obtained when the oil separator was bypassed. The first 15 runs recorded in Table II were made with the oil level in the crankcase at the bottom of the crankcase sight glass during operation. With the oil at this level several tests were made with the separator included in the system and the rest were made while the separator was bypassed. Runs 16 through 25 were made with the crankcase oil level at the center of the sight glass while for runs 26 through 35 the crankcase oil level was barely visible at the top of the sight glass. For all runs after 35 the oil level was above the sight glass during compressor operation. To raise the compressor crankcase oil level a tank of R-12 was attached to the fill line of the oil storage tank and the oil fill valve opened wide to pressurize the oil tank. The oil tank discharge valve was used to control the quantity of oil flowing to the crankcase. Runs 1, 6, 8,

10, 11, 16, 25, 27, 35, 36, and 37 were made with the separator in the system while the other runs were made with the separator bypassed. Several tests were made each time the equipment was operated, with each test requiring approximately one hour.

Sample Calculations

Thermodynamic Properties of Steam by Keenan and Keyes was used to obtain the enthalpy of water vapor.

Thermodynamic Charts⁽⁵⁾ by Ellenwood and Mackey was used to obtain the enthalpy of superheated Refrigerant-12.

Refrigerant Tables, Charts and Characteristics⁽²⁾ published by ASHRAE was used to obtain the enthalpy of the liquid refrigerant.

Since the wet bulb thermopiles were not sufficiently accurate for this investigation the product of the condensate flow rate and the enthalpy of vaporization for the mean dew point across the evaporator was used.

The values for run number 2 are used in the sample calculations.

1. Absolute suction pressure leaving the evaporator, lb/in.² abs.

$$P_{s2} = P_2 + \text{Bar.}$$

where: P_{s2} = absolute suction pressure, lb/in.² abs.

P_2 = corrected gage pressure leaving evaporator, lb/in.² gage

Bar. = barometric pressure, lb/in.²

$$P_{s2} = 25.5 + 13.97 = 39.47 \text{ lb/in.}^2 \text{ abs.}$$

2. Value of enthalpy for the latent heat of vaporization at the mean dew point temperature across the evaporator.

H_{fg} = the approximate latent heat of vaporization for the condensate collected at the evaporator, Btu/lb

The dew point temperature for air entering the evaporator is 61°F.

The approximate average dew point temperature for air leaving the evaporator is 59°F. Therefore, the mean dew point temperature across the evaporator was 60°F.

$$H_{fg} = 1059.9 \text{ Btu/lb}$$

3. Heat transferred to the refrigerant (Q_T) Btu/min

$$Q_T = W(H_2 - H_1)$$

where: W = rotometer flow rate, lb/min

H_1 = enthalpy of liquid refrigerant entering the evaporator, Btu/lb

H_2 = enthalpy of refrigerant vapor leaving the evaporator, Btu/lb

$$Q_T = 3.58(83.1 - 30.14) = 191 \text{ Btu/min}$$

4. Heat transferred from the air (Q_a) Btu/min

$$Q_a = \frac{A.F. (C_{ps}) (T_3 - T_5) + W_c (H_{fg})}{V_a}$$

where: C_{ps} = the average constant pressure humid specific heat,
Btu/lb (10)

$$= 0.244 \text{ Btu/lb}$$

A.F. = air flow rate, cfm

V_a = specific volume of the air, ft^3/lb

T_3 = temperature of air entering the evaporator, °F, dry bulb

T_5 = temperature of the air leaving the evaporator, °F, dry bulb

W_c = condensate flow rate, lb/min

$$Q_a = \frac{500}{14.6} (0.244) (80 - 59) + 0.0232 (1059.9) = 197.15 \text{ Btu/min}$$

5. The per cent by volume of oil in the refrigerant (% oil)

$$\% \text{ Oil} = \frac{V_o}{V_{sc}} (100)$$

where: V_{sc} = volume of the sampling section, cc

$$= 74 \text{ cc}$$

V_o = volume of oil collected, cc

$$\% \text{ Oil} = \frac{1}{74} (100) = 1.35\%$$

Data and Results

The data measured during the investigation, the calculated results and the significant curves appear in this section. A sample of the calculations used to obtain the results for all runs was shown for run number 2 under Sample Calculations.

Table I is a legend of symbols used in Table II. Table II gives measured data and the calculated results for each run.

Correction curves were applied to all pressure gage readings where necessary. The pressure gage readings recorded in Table II are corrected readings.

The potentiometer emf readings were correlated with temperature tables based on the International Temperature Scale of 1948. All potentiometer readings were taken with the cold junction immersed in an ice bath.

The significant data and results are presented in curve form in Figure VII. The curves in Figure VII show the resulting decrease in heat transfer due to an increase in the quantity of oil in the refrigerant. The curves represent the heat transferred as a function of suction temperature for different quantities of oil mixed with the refrigerant. The method used to correlate the data and the procedure used to obtain the curves is given on the page following the curves.

TABLE I

Legend for Symbols used in Table II

- P_1 - discharge pressure, lb/in.² gage
 P_2 - suction pressure, lb/in.² gage
Bar. - barometric pressure, lb/in.²
 T_1 - refrigerant temperature at evaporator inlet, °F
 T_2 - refrigerant temperature at evaporator outlet, °F
 T_3 - dry bulb temperature of air entering the evaporator, °F
 T_4 - wet bulb temperature of air entering the evaporator, °F
 T_5 - dry bulb temperature of air leaving the evaporator, °F
 T_6 - refrigerant saturation temperature at suction pressure, °F
Rot. - refrigerant flow rate, lb/min
 V_o - volume of oil collected at expansion chamber, cc
 W_c - weight of condensate collected, lb/min
A.F. - air flow rate, cfm

Performance Data

- P_{s2} - suction pressure, lb/in.² abs.
 T_{sh} - suction gas degrees of superheat, °F
 H_2 - enthalpy of the refrigerant leaving the evaporator, Btu/lb
 H_1 - enthalpy of the refrigerant entering the evaporator, Btu/lb
 Q_r - heat transferred to the refrigerant, Btu/min
 V_a - specific volume of the air, ft³/lb
 Q_a - heat transferred from the air, Btu/min
% Oil - per cent by volume of oil in the refrigerant

TABLE II

Data and Results

	Run 1	Run 2	Run 3	Run 4	Run 5	Run 6	Run 7	Run 8	Run 9	Run 10	Run 11	Run 12	Run 13	Run 14	Run 15	Run 16	Run 17
P ₁	119	120	119	120	119	120	120	122	120	123	124	121	119	124	116	120	122
P ₂	25.25	25.5	26.75	27.5	27.5	25.5	26	27.25	27.5	27.25	27	28.8	29.5	29.5	28.75	29.75	28.5
Bar.	13.97	13.97	13.97	13.97	13.92	13.92	13.74	13.74	13.71	13.71	13.71	13.71	13.71	13.71	13.71	13.71	13.71
T ₁	95.5	96	95	95.6	94	95	94.5	94.7	92.5	93.6	95	93.6	93	94	91	93	95.4
T ₂	45	45	45	45	46	45.5	45.5	45	46	45.5	43	46	44	38.5	42	40.5	34
T ₃	79.9	80	80	79.9	80.1	80	80	79.95	80	80	80.1	80	80.1	80	80	80.1	80
T ₄	67	67	67	67.2	67.5	67.2	67	66.5	68	68	68	68	68	67	67	67.2	67
T ₅	59.2	59	59	59.45	59.25	58.5	59.5	59.9	62	62	61.75	61.7	61.25	60.5	59.9	59.6	61
T ₆	24.9	25.2	25.4	25.2	27.9	25.2	25.9	27.3	27.8	27.3	26.9	29.2	30.1	30.1	29.2	29.7	28.8
Rot.	3.58	3.58	3.56	3.6	3.58	3.56	3.6	3.58	3.725	3.71	3.68	3.71	3.725	3.75	3.685	3.75	3.895
V _o	0	1	2.5	0.8	0.5	0	0.4	0	0.24	0	0.08	0.56	0.56	0.8	0.5	0.2	0.4
W _c	0.0277	0.0232	0.0201	0.0256	0.025	0.0275	0.0355	0.0356	0.0467	0.0442	0.0437	0.0478	0.0322	0.031	0.0302	0.0282	0.0356
A.F.	500	500	500	500	500	500	500	500	500	500	500	500	500	500	500	500	500
Performance Data																	
P _{s2}	39.22	39.47	39.69	39.47	41.42	39.42	39.74	40.99	41.21	40.96	40.71	42.51	43.21	43.21	42.46	42.96	42.21
T _{sh}	17	17	17	17	17	17.5	17.5	16	17	16.5	15	16	15	8.5	12	10.5	5
H ₂	83	83.1	83	83	83	84	84	84	84.1	84	83.5	83.3	83.5	82.6	83.1	83	82
H ₁	30.04	30.14	29.9	30.04	29.66	29.9	29.7	29.8	29.3	29.55	29.9	29.55	29.1	29.65	28.95	29.4	30
Q _r	191.5	191	191	192.5	193.5	192.5	195.5	194.1	205.7	202	197.5	199.3	198	198.8	200	201	202.5
V _a	14.6	14.6	14.6	14.6	14.6	14.6	14.8	14.8	14.87	14.87	14.87	14.87	14.87	14.87	14.87	14.87	14.87
Q _a	199.3	197.15	193.8	194.6	195.5	205.7	204	197.9	194.5	191.7	194.2	188.9	185.2	190.1	194	195.4	191.2
% Oil	0	1.35	3.4	1.1	0.7	0	0.54	0	0.32	0	11	0.76	0.76	1.1	0.68	0.26	0.54

TABLE II (Continued)

Data and Results

	Run 18	Run 19	Run 20	Run 21	Run 22	Run 23	Run 24	Run 25	Run 26	Run 27	Run 28	Run 29	Run 30	Run 31	Run 32	Run 33	Run 34
P ₁	121	120	124	125	124	125	124	124	123	124	125	119	119	123	122	117	119
P ₂	27.75	28.75	28.25	30.25	30.75	31	31	29	29	29.25	30.5	26	26.5	27	27	27	27.4
Bar.	13.71	13.71	13.55	13.67	13.67	13.67	13.67	13.67	13.67	13.67	13.67	13.8	13.8	13.82	13.82	13.82	13.82
T ₁	95.4	94.8	94.3	97.7	97.7	98.1	97.2	96.5	96.3	96.5	96.8	90	89.5	92	94.4	90	91
T ₂	44	40	41.4	39.7	39.5	40.5	41	36.5	41	38.6	42	36	39	35.5	37.5	38.3	39.8
T ₃	80	80	80	80	80	80	80	80	80	80	79.9	79	80.2	80	80.2	80.1	80
T ₄	67	66.9	67	67	67	66.9	66.9	66.9	67.1	67.1	66.9	67	68	67.6	68	68.4	68.8
T ₅	61	60.6	61.4	61.7	61.7	61.5	61.3	61	61.1	61.3	61.1	58.5	57.5	57.5	58	58.2	58.5
T ₆	27.8	29.2	28.3	31	31.6	31.9	31.9	29.3	29.3	29.7	30	25.8	26.4	27	27	37	27.7
Rot.	3.81	3.83	3.85	3.89	3.95	3.975	3.89	3.935	3.935	3.935	3.935	3.58	3.54	3.62	3.54	3.54	3.58
V _o	0.3	0.3	0.16	0.88	0.64	0.64	0.7	0.08	0.08	0.08	0.72	1.22	1.3	0.9	0.5	0.96	0.56
W _c	0.0358	0.0321	0.0585	0.065	0.0685	0.0664	0.067	0.07225	0.073	0.072	0.0683	0.01625	0.0098	0.01175	0.0143	0.0143	0.0213
A. F.	500	500	500	500	500	500	500	500	500	500	500	500	500	500	500	500	500
<u>Performance Data</u>																	
P _{s2}	41.46	42.46	41.8	43.92	44.42	44.67	44.67	42.67	42.67	42.92	44.17	39.8	40.3	40.8	40.8	40.8	41.2
T _{sh}	15	10.4	11.5	7.7	7.0	7.5	8	6.5	11	8.6	10	10	11	6.5	8.5	10.3	10.8
H ₂	83.5	83	83.2	82.7	82.7	82.8	82.9	82.3	83	82.5	83	82.2	82.8	82.2	82.6	82.7	83
H ₁	30	29.5	29.73	30.55	30.55	30.62	30.41	30.26	30.2	30.26	30.33	28.7	28.6	29.2	29.71	28.71	28.95
Q _r	204	205	205.5	203	206	207	200.5	205	208	206	207.5	192	188.5	191	187.5	191	193.5
V _a	14.87	14.87	15.08	14.78	14.78	14.78	14.78	14.78	14.78	14.78	14.78	14.8	14.8	14.72	14.72	14.72	14.72
Q _a	191.3	191	210	217.8	222	220.4	223	230	230	229.35	234.3	183.2	195.5	196	197.1	193.6	197.5
% Oil	0.4	0.4	0.21	1.78	0.87	0.87	0.96	0.11	0.11	0.11	1.0	1.65	1.76	1.2	0.68	1.3	0.76

TABLE II (Continued)

Data and Results

	Run 35	Run 36	Run 37	Run 38	Run 39	Run 40	Run 41	Run 42	Run 43	Run 44	Run 45	Run 46	Run 47	Run 48	Run 49	Run 50	Run 51
P ₁	116	116	115	121	119	118	118	119	118	120	120	120	120	120	120	122	121
P ₂	26.5	26.25	26	26	25.25	25	26	26	26.25	27	27	27	28	28	28	28.75	28.75
Bar.	13.82	13.82	13.82	13.6	13.6	13.6	13.6	13.6	13.8	13.8	13.8	13.8	13.8	13.8	13.8	13.77	13.77
T ₁	89.5	89.5	89.6	90.5	89	88.5	88.7	89	91	91.3	92	91.3	91.3	92	92	94	94
T ₂	34	34	30.5	35	33	36	37.2	34	39	36.5	40.3	40.2	33	36	36	33	38
T ₃	80	80	80	80	79.8	79.8	79.8	80	80	80	80.1	80	80	80	80	80	80
T ₄	68.7	68.6	68.7	68	67.7	67.6	67.7	67.2	68	68	68.2	68.2	68.3	68	68.5	68	68
T ₅	58	58	58	56.7	56	57	57	57.7	57.5	58.5	57.8	57.5	57.5	58	58	60.1	60.2
T ₆	26.4	26	25.8	25.2	24.4	24	25.2	25.2	26	27	27	27	28.3	28.3	28.3	29.3	29.3
Rot.	3.835	3.835	3.79	3.54	3.51	3.51	3.51	3.51	3.54	3.62	3.58	3.58	3.68	3.68	3.68	3.87	3.85
V _o	0.08	0	0	0.58	0.48	0.32	0.48	0.4	0.82	0.42	0.9	0.64	0.32	1.26	0.6	0.8	0.8
W _c	0.0265	0.0279	0.0289	0.0161	0.0156	0.0147	0.0131	0.00965	0.0186	0.020	0.0186	0.0201	0.0195	0.02	0.0308	0.045	0.048
A.F.	500	500	500	500	500	500	500	500	500	500	500	500	500	500	500	500	500
<u>Performance Data</u>																	
P _{s2}	40.3	40.05	39.8	39.6	38.9	38.6	39.6	39.6	40.05	40.8	40.8	40.8	41.8	41.8	41.8	42.62	42.62
T _{sh}	7	7	3	7	6	9.5	10.2	7	11.5	8.5	12.3	12.2	4	7	7	4	9
H ₂	81.4	81.4	81.5	82.2	82	82.3	82.6	82.1	82.8	82.2	82.9	82.9	81.9	82.2	82.2	81.8	82.6
H ₁	28.6	28.6	28.7	28.83	28.5	28.4	28.43	28.5	28.95	29	29.2	29	29	29.2	29.2	29.66	29.66
Q _r	204.4	204.4	200	189.3	188	189.5	190.5	188	191	193	192.5	193.2	195	195	195	202	203.5
V _a	14.72	14.72	14.72	14.97	14.97	14.97	14.97	14.78	14.75	14.75	14.75	14.75	14.75	14.75	14.75	14.77	14.77
Q _a	207.4	208.8	209.9	194.1	203.5	198.6	197	193.2	202.7	196.1	200.7	204.3	203.7	200.4	211.85	209.6	211.8
% Oil	0.11	0	0	0.785	0.65	0.43	0.65	0.54	1.1	0.57	1.215	0.865	0.43	1.7	0.81	1.1	1.1

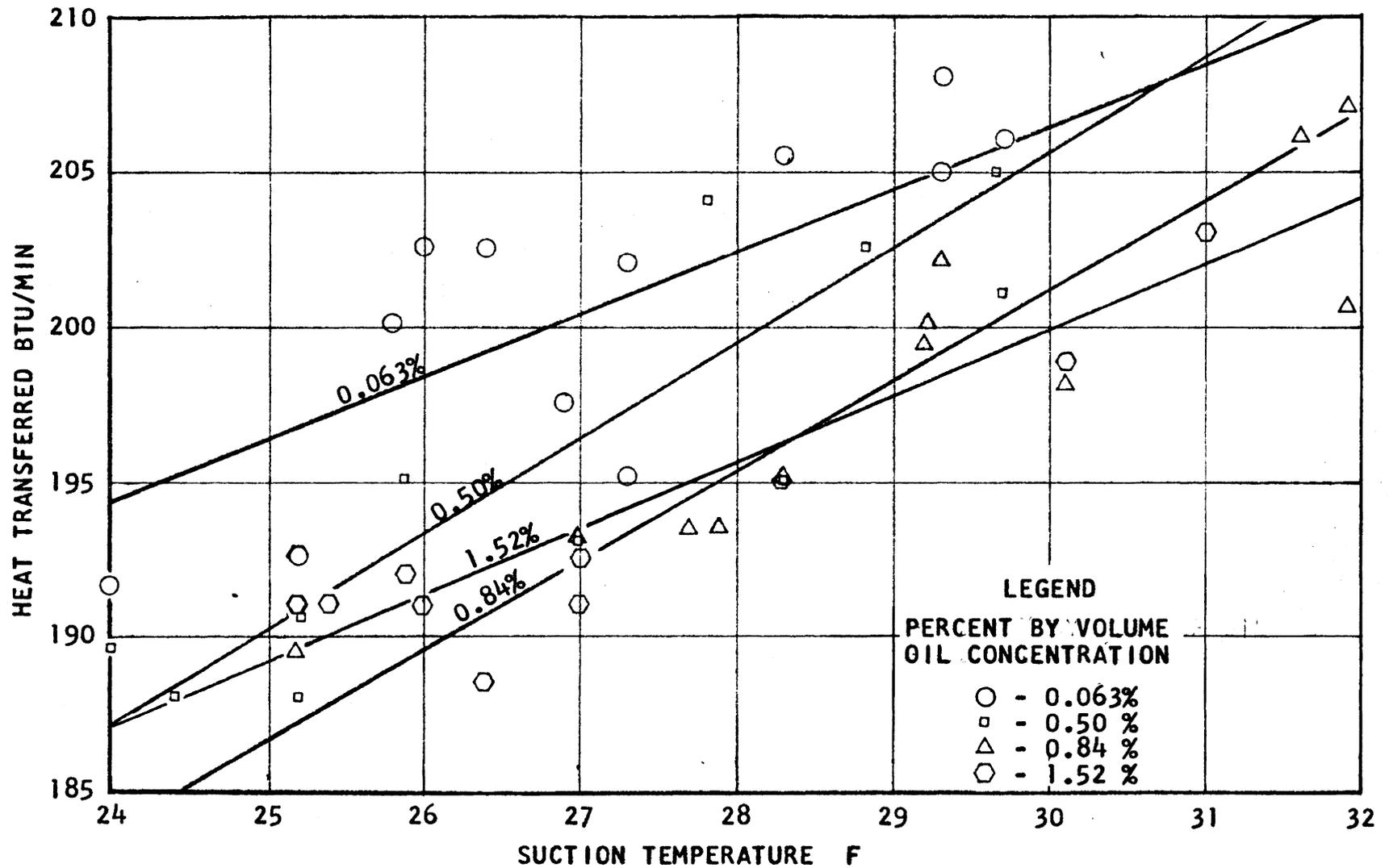


FIG. VII EFFECT OF OIL ON EVAPORATOR HEAT TRANSFER

Procedure for Obtaining the Curves in Figure VII

The data for all test runs was arranged in ascending order according to the per cent of oil obtained for each run. If two or more runs had the same per cent of oil these runs were further arranged in ascending order of suction temperature. If two or more runs had the same per cent of oil and suction temperature these runs were further arranged according to the ascending quantity of heat transferred. In order to have several curves showing the relative decrease in heat transferred as the quantity of oil in the refrigerant increased, the arranged data was divided into four equal parts. The first group of data points represented a curve for heat transferred as a function of suction temperature for a minimum of oil. Each succeeding group of data represented a curve for the corresponding average quantity of oil in the refrigerant.

The method of averages as given by Davis⁽⁴⁾ was used to obtain the equation representing the curves. All data points which showed a deviation of more than 3% were adjudged to be non-characteristic. The equations which are represented by the curves in Figure VII are as follows:

$$(0.063\% \text{ oil}) \quad Y = 145.85 + 2.02X$$

$$(0.50\% \text{ oil}) \quad Y = 112 + 3.09X$$

$$(0.84\% \text{ oil}) \quad Y = 114.5 + 2.89X$$

$$(1.52\% \text{ oil}) \quad Y = 135.4 + 2.15X$$

IV. DISCUSSION

Due to the amount and variety of equipment required for the investigation it was considered necessary to discuss its operation separately from the results obtained in the investigation. Therefore, the discussion is divided into two sections. The first section concerns the equipment performance and limitations. The second section concerns the results.

Equipment

Since much of the equipment had been used for two previous investigations, many limitations and sources of error were discussed previously. The items discussed by Riffe⁽¹⁴⁾ and Jesena⁽¹¹⁾ are referred to here but not discussed except where additional information can be added.

The sampling circuit performed as desired. The glass section made it possible to observe the rapidity with which the refrigerant drained to the expansion chamber and also the thoroughness with which the refrigerant washed the sampling section. In a matter of seconds the interior of the glass appeared clean and dry, thus indicating that any liquid left on the surface had vaporized.

The expansion chamber performed exceptionally well. The rate of boiling of the refrigerant was completely under control. Violent boiling caused the oil to be thrown from the test tube to the side of the test chamber, from which it returned very slowly to the test tube. Best

results were obtained when the ebullient action of the refrigerant was held to a minimum. As the refrigerant vaporized the liquid surface, which was in effect an oil surface, descended in the chamber. The surface tension of the slowly descending oil surface stripped the oil from the chamber wall. The pressure in the expansion chamber was equal to the suction pressure when the volume of oil in the test tube was recorded. It is possible that a small amount of refrigerant remained in the oil which would have vaporized if the oil had been reduced to atmospheric pressure. This procedure would have extended each test to a period of several hours instead of the approximately one hour which was required in this investigation. To expedite such a test procedure three expansion chambers could be used simultaneously as follows: one open to the compressor suction reclaiming refrigerant from a test sample, another being evacuated with a vacuum pump after having been vented to the atmosphere and a third ready to receive a sample.

Due to the coarse adjustment available with the manual expansion valve it was difficult to maintain a constant superheat of 10°F at the evaporator outlet without frequent adjustment. However, since refrigerant flow rate affected the oil film in the evaporator, frequent expansion valve adjustment was considered undesirable. Therefore, superheat readings from 3°F to 17.5°F were accepted. A needle valve with linear characteristics might have been a better choice.

A possible source of error in determining the heat transferred to the refrigerant due to recording incorrect temperature for the refrigerant entering and leaving the evaporator was fully discussed by Riffe⁽¹⁴⁾ on

page 63 of his thesis. He calculated the possible temperature difference between the refrigerant and that indicated by the thermocouple, due to the evaporator tube wall thickness, to be approximately 0.02°F. He concluded that this was less than experimental error.

The oil storage tank provided a convenient method with which to introduce oil into the compressor crankcase. The hot gas line of the refrigeration system would have been a more convenient source for pressure than was the spare tank of R-12 which was used. A connection from the hot gas line to the top of the oil storage tank, controlled by a packless valve, was needed to pressurize the oil tank.

Apparently, the only control over the quantity of oil circulated with the refrigerant was that obtained with the oil separator. Overloading the crankcase had no noticeable effect on the quantity of oil mixed with the circulating refrigerant. For a minimum of oil in the refrigerant system the separator was included in the refrigeration system. For a maximum of oil the separator was bypassed. The quantity of oil circulating was unpredictable.

One of the effects of bypassing the separator was that both the suction and discharge pressures increased immediately from 1 to 2.5 psi. This phenomenon could have been the result of an oil film on the tube walls of the condenser which caused increased resistance to heat flow. Therefore, the condensing temperature and corresponding saturation pressure increased. According to the theory of gas compression this will reduce the volumetric efficiency of the compressor, which, in turn, will reduce the refrigerant flow rate through the system. Since the same

quantity of air was supplied to the evaporator at the same conditions of wet and dry bulb temperature for all tests, a small reduction in flow was reflected in a small rise in suction pressure. As a result, the temperature difference between the air and the refrigerant was reduced, which resulted in a reduced heat transfer rate.

The drip leg of the glass tee at exit from the evaporator remained approximately 2/3 full of oil while the separator was bypassed. Upon closing down the system, after operating with the separator bypassed, oil filled the glass tee to the top of the side inlet. With the separator in operation the drip leg of the tee was virtually dry during system operation. Upon closing down the system, after operating with the separator engaged, the amount of oil draining from the evaporator was negligible. With the separator in the circuit, the separator oil return line appeared to be continuously full of foam. The foam was a result of the mixture of oil and refrigerant immediately expanding upon leaving the separator. The foam was similar in appearance to that obtained as a result of draining the refrigerant sample to the atmosphere in an attempt to separate the refrigerant from the oil.

In the preliminary procedure, the reasons for discarding the wet bulb thermopile readings as a means for determining the heat balance for the air conditioning system were discussed. Following this, the method that was used to determine this heat balance was given. However, it is possible that dew point control of the air leaving the washer might have been more accurate than the wet bulb thermopiles and more

convenient than weighing condensate. This might be achieved in the following manner. An averaging thermocouple grid in the exit from the washer would indicate a dry bulb temperature very near the washer dew point, depending on the washer efficiency. The actual dew point leaving the washer could be determined, and with the dry bulb temperature at the washer exit, the washer efficiency also could be determined. With reference to a psychrometric chart, an error of several per cent in washer efficiency would be an error of less than 1°F in dew point. For a final dry bulb temperature at entrance to the evaporator, 1°F error in dew point leaving the washer is less than 1°F error in the final wet bulb temperature. Therefore, control of the washer by the dry bulb temperature of the leaving moist air might be more accurate than the wet bulb thermopile. The success of this method depends upon the thermocouples in the thermocouple grid in the exit from the washer remaining dry.

Radiation interchange at both averaging thermocouples could introduce error in temperature measurement of the air. It is possible a direct line of sight existed between the averaging thermocouple at exit from the air flow nozzle and the electric heater, introducing some error due to radiation exchange. Also, it is possible that the downstream averaging thermocouple experienced radiation interchange with the duct, evaporator and wall of the room. However, the conditions in the duct and in the room were relatively constant for all runs. Therefore, any errors would have been of the same magnitude for all runs, and consequently,

their effect would have been canceled. Therefore, these possible errors would not have interferred with the purpose of the investigation, which was to measure the change in the quantity of heat transferred across the evaporator tube wall.

Results

The results of this investigation are tabulated in Table II and shown graphically in Figure VII. Figure VII is a presentation of the heat transferred to the evaporator as a function of suction temperature. The four curves in Figure VII represent different quantities of oil mixed with the refrigerant passing through the evaporator. It is significant to note that as the quantity of oil increases the heat transferred decreases. It is apparent that the oil entering the evaporator remained a liquid while the refrigerant evaporated, and that the oil wetted the tube surface as it was pulled through the evaporator due to the velocity of the refrigerant vapor. The oil on the tube wall interposed an additional film that increased the total inside film resistance.

Each curve rerpresents the average of the data from approximately the same number of runs. The uppermost curve, for example, represents the average of the data for the runs in which the minimum of oil was found. The reduction in heat transfer shown by the curves is from 5% at 25°F suction to 3% at 30°F suction temperatures. The variation in the quantity of oil in the refrigerant was from 0 to 3.4% by volume.

Inspection of Table II reveals that the results were limited to a very small change in oil quantity due to the extremely low percentage of oil induced into the refrigerant by the compressor. Also, inspection of the data and results shows why the curves for the 0.84% and the 1.52% line cross. A number of values for heat transferred used to obtain the 0.84% line are less than the lowest value of heat transferred for the 1.52% line. The curves obtained indicate a definite trend in that oil does reduce the heat transfer rate to a boiling refrigerant, but that much more data is required over a wider range of refrigerant-oil mixtures to establish the fact with predictable quantitative results.

C. M. Ashley⁽¹⁾ has stated that the average refrigeration system in good operating condition would not contain over 3% by volume of oil in the refrigerant. Through personal correspondence, Mr. Ray M. Hawkins,⁽⁸⁾ an engineer with Aerofin Corporation, Syracuse, New York, stated that good quality compressors would circulate from 3/4 to 1-1/2% by volume of oil in the refrigerant. A maximum value of 3.4% was obtained for one run only during this investigation. There were several runs with approximately 1.75% while all other runs contained less oil in the refrigerant.

Whitzig, Penny and Cypher,⁽¹⁹⁾ using a thermal siphon system in which the volume of oil mixed with the refrigerant could be closely controlled, found that 1% by volume of oil caused a 6.1% reduction in heat transfer. This is a 1% to 2% greater reduction than found by this author. In this thesis the upper curve represents a minimum oil

concentration of 0.063%, whereas in the report of Whitzig et al the upper curve represents refrigerant free of oil.

Seigel, Bryan and Huppert,⁽¹⁵⁾ using slightly different parameters, found that refrigerant containing oil required a greater temperature differential across the evaporator tube for a given rate of heat transfer. In order to conform to the discussion, this statement could be changed to read that at a particular suction temperature less heat was transferred as oil concentration increased. No curves were given to distinguish oil-free runs versus runs with oil; however, the trend of the plotted data points shown by Seigel et al support the trend found in this thesis.

The quantity of oil dealt with in this thesis was extremely small and any surface imperfections in the expansion chamber could cause a relatively large error in the percentage of oil. A perfected expansion chamber operating technique is essential to prevent errors in measuring oil concentration. A chamber with a smooth wall and a minimum of joints and crevices should be used.

The glass sections of the refrigerant circuit proved interesting and revealing. The phenomena of returning oil and refrigerant gas could be observed at the evaporator exit through the glass tee. The oil was observed as it was being pulled vertically up the side of the tee by the high velocity suction gas. Also, the evaporator was slightly flooded on occasions to observe the violent boiling occurring in the tee. An oil-rich mixture in the tee exhibited foam as opposed to an oil-free mixture in which drops or slugs of refrigerant appeared with the refrigerant vapor until they suddenly evaporated.

V. CONCLUSIONS

The following conclusions were drawn from the investigation of the effect of oil on the performance of a direct expansion evaporator:

1. The oil circulating with the refrigerant has a tendency to reduce the rate of heat transfer. At 25°F suction temperature the reduction in heat transferred between the runs with a minimum of oil and the runs averaging 0.84% by volume of oil in the refrigerant was 5%. At 30°F suction temperature the reduction in heat transferred between the runs with a minimum of oil and the runs averaging 1.52% by volume of oil in the refrigerant was 3%. Since production line compressors would circulate from 3/4 to 1-1/2% by volume of oil these systems could expect from 3 to 5% reduction in capacity unless an effective oil separator was used in the system.
2. The oil circulating with the refrigerant causes an elevation in the refrigerant boiling temperature. Within minutes after the separator had been bypassed the suction and discharge pressure rose as much as 2.5 psi, representing an increase in suction temperature of as much as 3.3°F.

VI. RECOMMENDATIONS

Recommendations for modifications to the equipment used in this investigation were included in the discussion of the equipment. In addition it is recommended that the following related test be conducted:

1. Additional tests be made with the same equipment covering wider ranges in quantity of oil in the refrigerant and in suction temperatures. This could be attained by using a variable speed compressor, a compressor in poor mechanical condition or a parallel suction gas return line through the bottom of the crankcase to induce more oil into the refrigerant. The increased compressor speed or suction return through the oil would increase the agitation of the crankcase oil causing foaming, which, in turn, would make it easy for the refrigerant to entrain more oil. A unit in poor mechanical condition would introduce oil into the refrigerant due to excess oil in the cylinder.
2. Tests on operating commercial systems to determine the actual quantity of oil in the refrigerant. A sampling circuit and expansion chamber similar to that used in this investigation could be adapted temporarily to some commercial systems for this purpose.

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VIII. ACKNOWLEDGMENTS

The author is grateful to his wonderful wife and children for their patience, understanding and encouragement during the years spent on this endeavor.

In addition the author extends sincere appreciation to those staff members of the Mechanical Engineering Department of Virginia Polytechnic Institute who gave their time and advice and especially to:

The members of the thesis committee, Professors R. K. Will, J. B. Jones and C. H. Long for their advice, criticisms and encouragement in preparing this thesis.

Professors D. S. Cassell, M. Crawford, R. S. Gay, S. T. Hsu and H. S. Miles for their suggestions and constructive criticisms.

The personnel of the machine shop Mr. F. B. Fisher and Mr. H. M. Smith for their cheerful cooperation and help in fabricating the test equipment.

Sincere appreciation is extended to Mr. S. R. Hirsch of the Worthington Corporation and Mr. Ray M. Hawkins of the Aerofin Corporation for their helpful correspondence.

Special appreciation is extended to D. R. Riffe and R. M. Jesena who designed and constructed the apparatus used in this investigation.

Sincere appreciation is extended to Mrs. R. L. Price who typed this thesis in its final form.

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X. APPENDICES

1. Specific Volume of Air for Changing Barometric Pressure
2. Micromanometer Calibration Curve
3. Rotometer Calibration Curve
4. Thermopile Calibration Curve
5. Thermocouple Calibration Curve
6. Suction Pressure Gage Calibration Curve

APPENDIX I

The following relations were used to determine the specific volume of moist air (V_a), ft³/lb of dry air for 80°F dry bulb and 67°F wet bulb temperature, for a changing barometric pressure. For convenience the derivation was made in terms of dry air since the difference in specific volume for the moist air and the dry air involved in this investigation is negligible. (6)

Thermodynamic Properties of Steam by Keenan and Keyes was used to obtain the saturation pressures and enthalpy for water and steam.

The characteristic gas equation was used to determine the volume of dry air.

$$\begin{aligned} 1. V_a &= \frac{M_{da} R_{da} T}{P_{da}} \\ &= \frac{M_{da} R_{da} T}{P_m - P_v} \end{aligned}$$

where: M_{da} = mass of dry air, lb

R_{da} = universal gas constant for air, ft/lb °R

P_{da} = partial pressure of dry air, lb/ft²

P_m = barometric pressure, lb/ft²

P_v = partial vapor pressure of moisture in the air, lb/ft²

T = dry bulb temperature, °R

The partial pressure of the vapor was determined from

$$W_v = \frac{0.622 P_v}{P_m - P_v}$$

from which

$$2. P_v = \frac{W_v P_m}{W_v + 0.622}$$

where: W_v = specific humidity of moist air, lb/lb dry air

Substitute equation 2 into equation 1 and let M_{da} equal 1 lb.

$$V_a = \frac{(1) (53.3) T}{P_m - \left[\frac{W_v P_m}{W_v + 0.622} \right]}$$

$$3. V_a = \frac{53.3 T}{P_m} \left[1 + \frac{W_v}{0.622} \right]$$

Let P_B equal barometric pressure, in Hg and substitute into equation 3.

$$V_a = \left[\frac{53.3 T}{P_B (0.491) (144)} \right] \left[1 + \frac{W_v}{0.622} \right]$$

$$4. V_a = \left[\frac{0.754 T}{P_B} \right] \left[1 + \frac{W_v}{0.622} \right]$$

where: 0.491 = pressure in inches Hg equivalent to 1 psi

144 = number of square inches per square foot

The specific humidity of the moist air was also obtained from

$$5. W_v = \left[\frac{W_s h_{gs} + c_p (T_4 - T_3)}{h_{gv} - h_{fs}} \right]^{10}$$

where: W_s = specific humidity of the moist air at saturated conditions,
lb/lb dry air.

h_{gs} = enthalpy of saturated vapor at the wet bulb temperature,
Btu/lb

h_{gv} = enthalpy of saturated vapor at the dry bulb temperature,
Btu/lb

h_{fs} = enthalpy of saturated water at the wet bulb temperature,
Btu/lb

T_3 = dry bulb temperature of air entering evaporator, °F

T_4 = wet bulb temperature of air entering evaporator, °F

The saturated specific humidity (W_s) at 67°F was determined from

$$W_s = \frac{0.622 P_s}{P_B - P_s}$$

where: P_s = the saturated vapor pressure at 67°F bulb temperature
= 0.6669 in. Hg

$$6. W_s = \frac{(0.622) (0.6669)}{P_B - 0.6669}$$

$$= \frac{0.4148}{P_B - 0.6669}$$

Substitute equation 6 into equation 5 and simplify

$$W_v = \frac{\left[\frac{0.4148}{P_B - 0.6669} \right] \left[1056 + .24 (67 - 80) \right]}{1096.6 - 35.05}$$

$$7. N_V = \frac{436 + 3.12 P_B}{1061.55 P_B - 707.95}$$

Substitute 7 into equation 4 and let T = 540°R

$$\begin{aligned} V_a &= \frac{(0.754) (540)}{P_B} \left[1 + \left(\frac{436 + 3.12 P_B}{1061.55 P_B - 707.95} \right) \left(\frac{1}{0.622} \right) \right] \\ &= \frac{267,570 P_B - 98}{660.28 P_B^2 - 440.34 P_B} \end{aligned}$$

By assuming the second term of the numerator is insignificant the equation becomes

$$V_a = \frac{267,570}{660.3 P_B - 440.3}$$

APPENDIX I

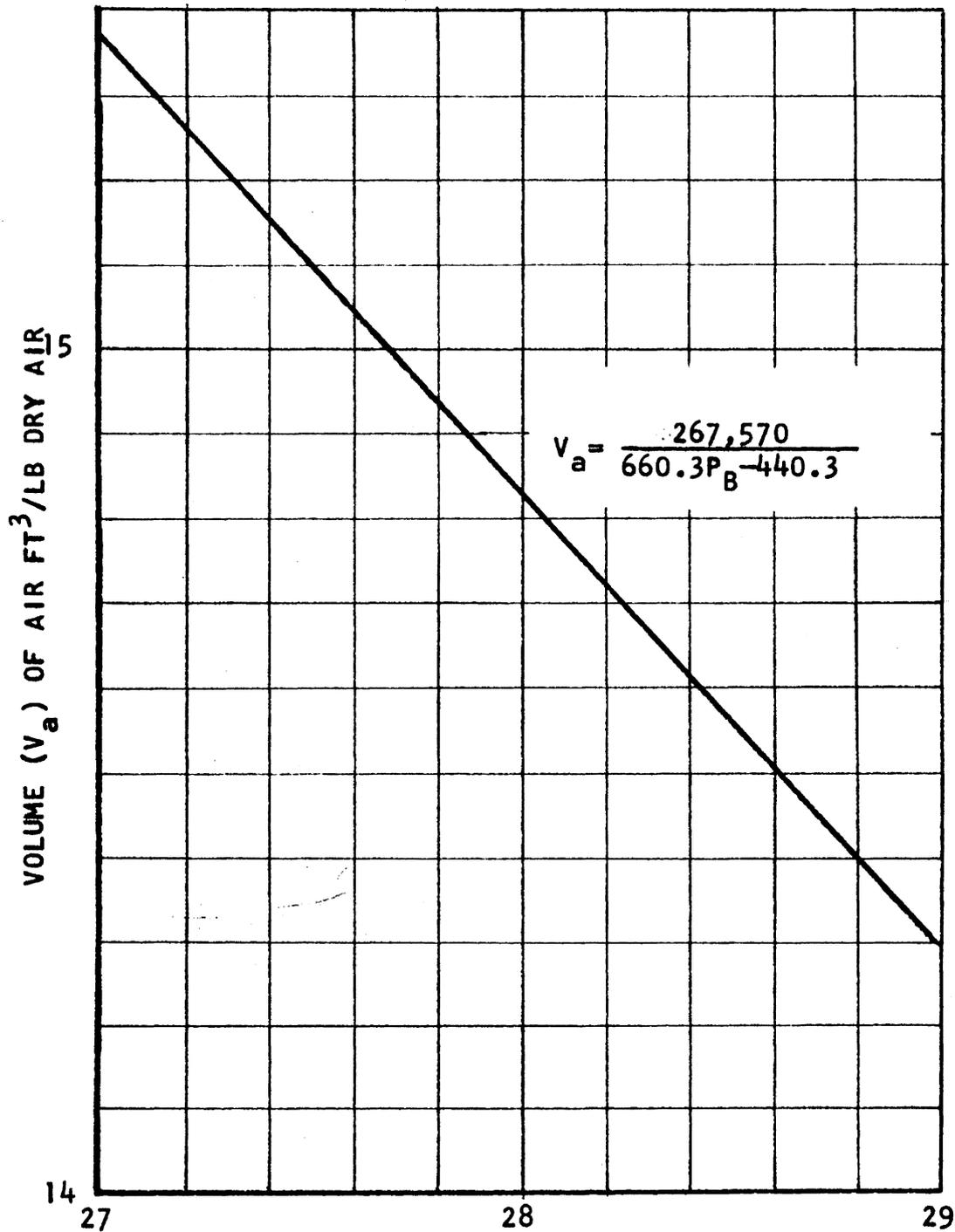


FIG. VIII SPECIFIC VOLUME OF AIR AT 80 F DRY BULB AND 67 F WET BULB

APPENDIX II

Derivation of relations used to obtain the micromanometer calibration curve for a changing barometric pressure.

The general equation for the velocity of a flowing fluid modified to account for a coefficient of discharge and approach velocity correction factor is

$$1. \bar{V} = \frac{C \sqrt{2 g H}}{\sqrt{1 - \left(\frac{A_2}{A_1}\right)^2}}$$

where: \bar{V} = velocity at the throat, ft/sec

C = coefficient of discharge, dimensionless

g = standard acceleration of gravity, ft/sec²

H = Head on the flowing fluid, ft

A₂ = throat area, ft²

A₁ = duct area, ft²

For high Reynolds numbers the coefficient of discharge (C) is 0.99. (12) The value of the approach velocity correction was also 0.99.

If the head (H) is changed to be the pressure in inches of water (ΔP) the equation becomes

$$2. \bar{V} = \sqrt{2 g V_a 5.19 \Delta P}$$

where: V_a = volume of moist air, ft³/lb dry air from Appendix I

Equation 2 was solved for ΔP in terms of a fixed velocity through the nozzle and for the volume of moist air (V_a) flowing in the duct. The diameter of the nozzle throat was 4.5 in. The resulting equation was

$$3. \Delta P = \frac{16.8}{V_a}$$

where V_a may be taken from the curve in Appendix I for different barometric pressures.

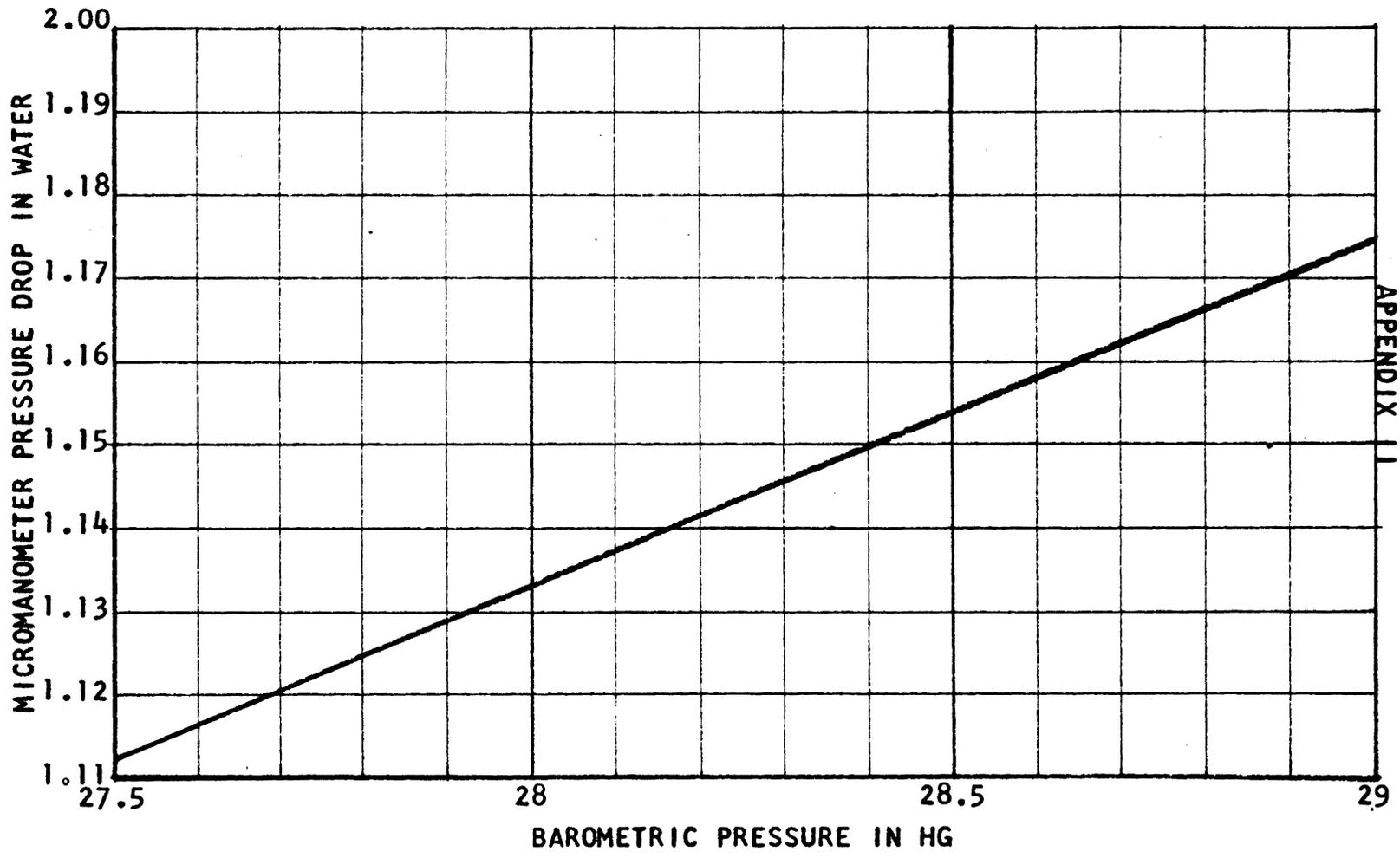


FIG. IX MICROMANOMETER CALIBRATION CURVE

APPENDIX II

APPENDIX III

The Schuttle and Koerting rotometer was calibrated by comparing the heat given up by the refrigerant to the heat absorbed by the cooling water in the condenser. Before calibrating the rotometer the condenser, refrigerant piping and water piping for at least 12 inches beyond the condenser were insulated with a minimum of 2 inches of fiber glass insulation. Copper-constantan thermocouples were brazed to the refrigerant and water lines near the condenser with the insulation extending a minimum of 6 inches beyond the thermocouple.

The heat absorbed by the cooling water is equivalent to the product of the condenser water flow rate and the change of enthalpy for the water flowing through the condenser. The condenser water flow rate was determined by the time required to collect 20 pounds of water. The water was weighed with platform scales which have a range of 0 to 300 pounds in 2 ounce increments and a sensitivity of 1 ounce. The time was measured by a stop watch with a 30 seconds per revolution dial. The enthalpy change for the water was taken as the difference of enthalpy for saturated liquid at the temperatures entering and leaving the condenser.

The heat given up by the refrigerant is equivalent to the product of the refrigerant flow rate and the enthalpy change through the condenser. The compressor discharge pressure and the temperature at entrance to the condenser were used to obtain the enthalpy of the superheated refrigerant entering the condenser. The enthalpy of the refrigerant leaving the

condenser was equivalent to the enthalpy of saturated liquid at the discharge pressure. The heat added to the water divided by the enthalpy change of the refrigerant through the condenser gave the refrigerant flow rate.

The rotometer manufacturer supplied a correction curve to be applied to the rotometer used in this investigation. The curve shows that for a change in temperature of 12.5°F beyond the temperature for which the rotometer was calibrated, a correction in specific gravity of 0.01 should be applied. While the rotometer was being calibrated the liquid refrigerant leaving the receiver varied between 85 and 95°F. This was very close to the temperatures experienced during the investigation. Since the rotometer was calibrated at the same temperature at which it was used further correction seemed unnecessary.

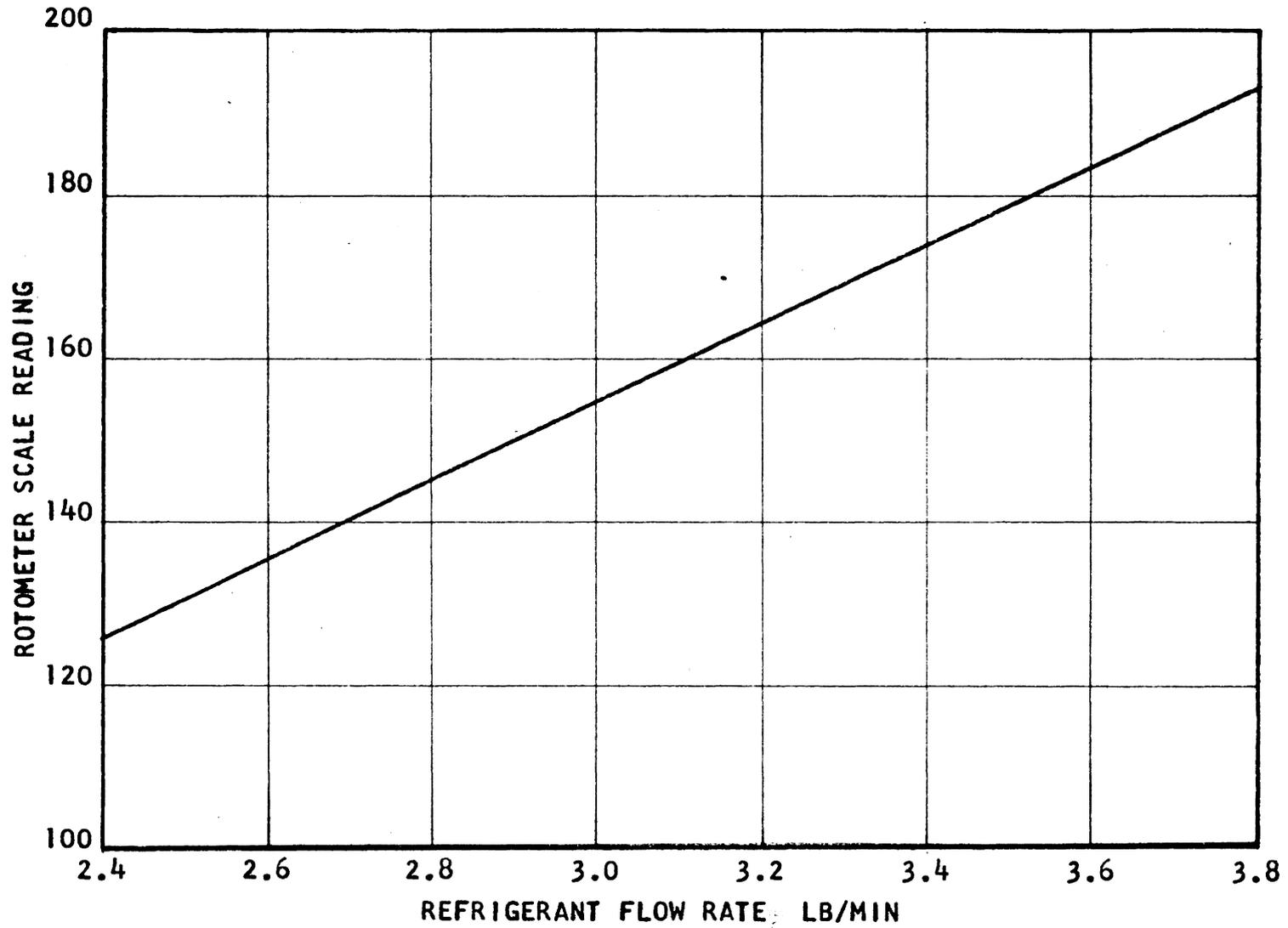


FIG. X ROTOMETER CALIBRATION CURVE

APPENDIX IV

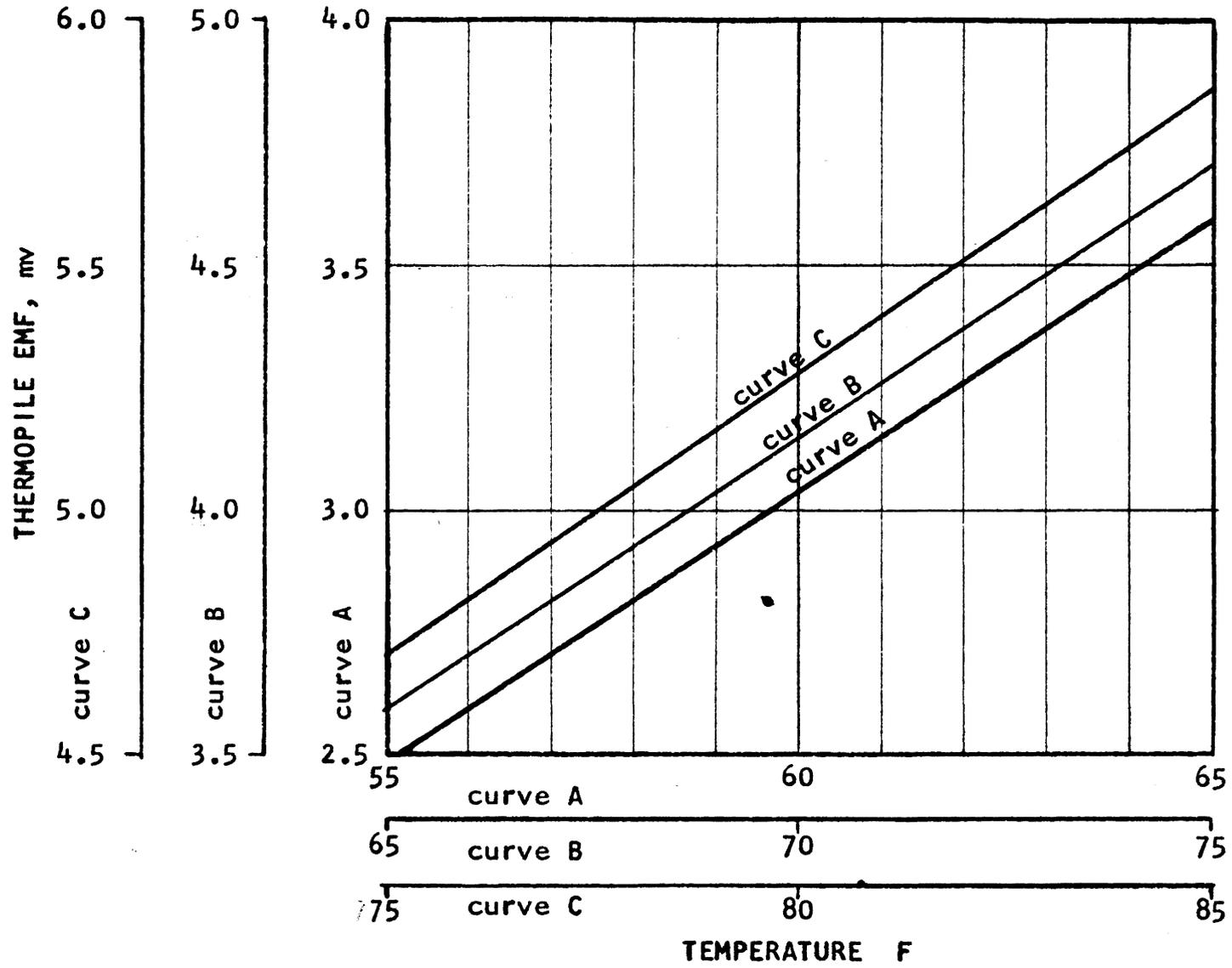


FIG. XI THERMOPILE CALIBRATION CURVE REPRODUCED FROM RIFFE¹⁴

APPENDIX V

Thermocouple Calibration. A section of copper-constantan wire adjacent to that used to fabricate thermocouples for this investigation was calibrated as described below. It was assumed all wire in a particular spool would have the same characteristics, therefore it was considered necessary to calibrate only one thermocouple. The potentiometer described under equipment was used for the calibration. The calibration curve is shown in this section of the Appendix.

Three fixed points as given by the Handbook of Chemistry and Physics* was selected as calibration points. They are: the freezing point of mercury -38.87°C at 760 mm of Hg, the freezing point of water 0°C at 760 mm of Hg and the boiling point of water 100°C at 760 mm of Hg. The freezing point of mercury was obtained by placing a test tube containing approximately 25 cc of technical grade of mercury into a flask of dry ice and gasoline mixture. The emf for the thermocouple was -1.419 mv. The freezing point of water was obtained by inserting both junctions in the ice bath. The emf read for the ice point was 0.00 mv. A hypsometer was used to obtain the steam point. The emf read for the steam point was 4.191 mv. An ice bath was used as a reference junction for all calibration points. The brass scale correction for the barometer as given by the Handbook of Chemistry and Physics** was

*Handbook of Chemistry and Physics, Chemical Rubber Publishing Company, 42nd Edition, 1960, p. 2322-2323.

**ibid., p. 2485.

applied for the prevailing atmospheric conditions of 77°F and 28.08 in. Hg. The corrected boiling point was obtained from Thermodynamic Properties of Steam by Keenan and Keyes and was found to be 208.54°F.

The calibration curve obtained showed the copper-constantan wire to agree with standard emf temperature correlation tables. Therefore, the tables were used in lieu of the curve obtained.

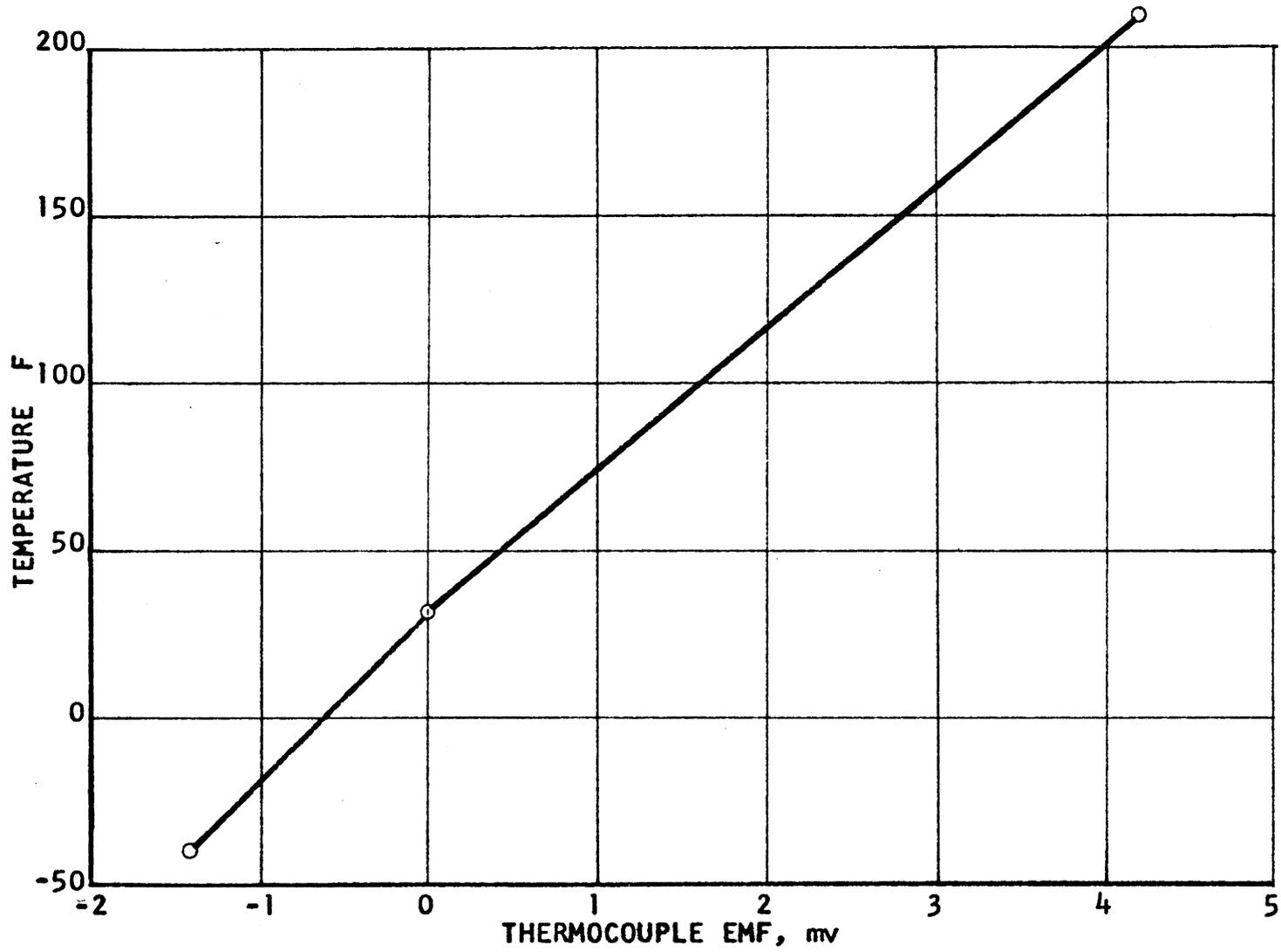


FIG. XII COPPER-CONSTANTAN THERMOCOUPLE CALIBRATION CURVE

APPENDIX VI

The suction pressure gage, number 7037 U. S. Gauge Company, was calibrated with a dead weight gage tester. The gage was tested in 5 lb increments throughout its range by placing a known weight on the platform of the dead weight gage tester and recording the actual reading of the gage. The true pressure readings versus the gage readings were plotted on the following calibration curve.

APPENDIX VI

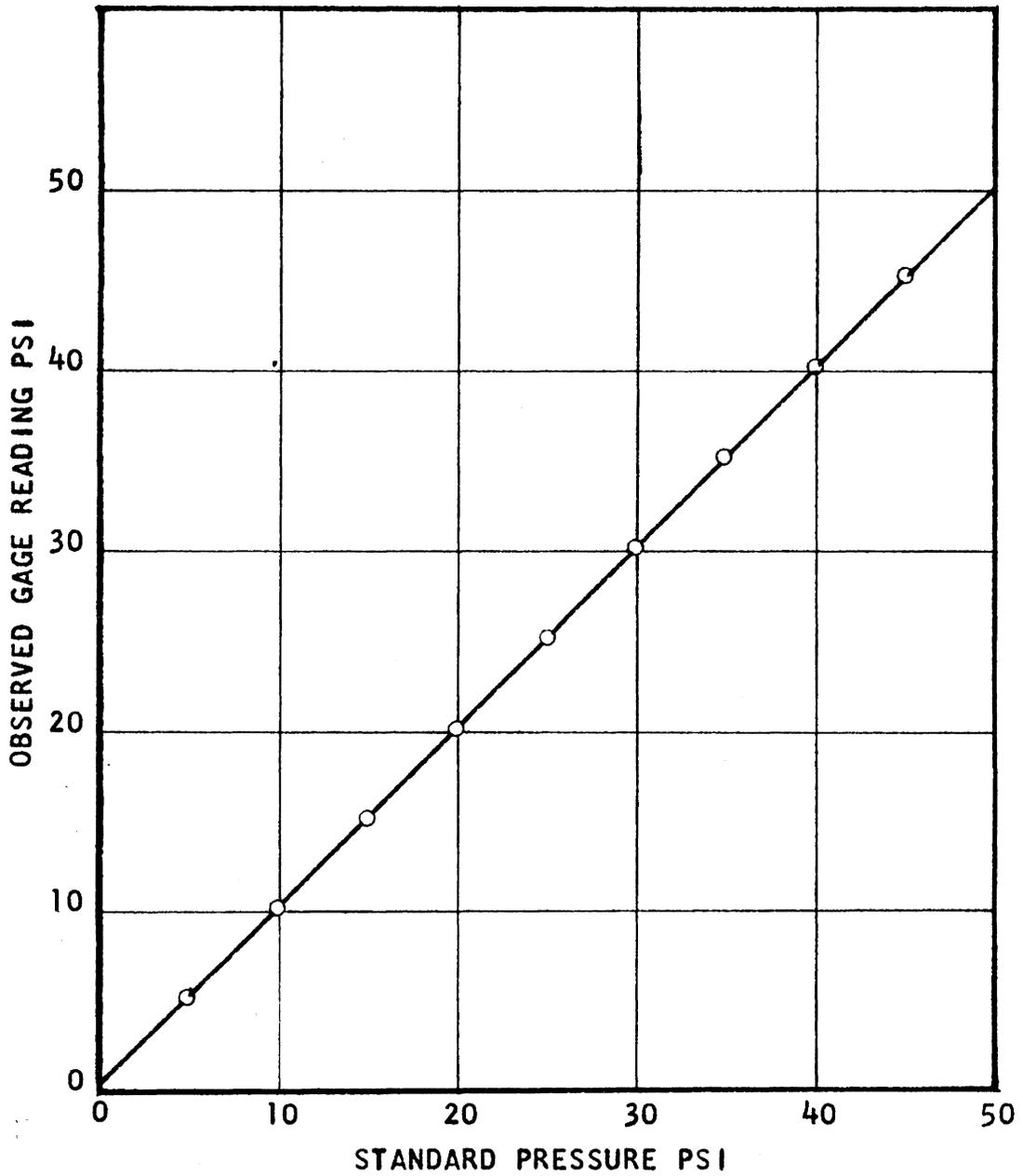


FIG. XIII SUCTION PRESSURE GAGE CALIBRATION CURVE
(US Gauge No. 7037)

ABSTRACT

The investigation is concerned with the effect of entrained oil on the heat transfer characteristics of a direct expansion evaporator coil. A modified refrigeration system, in which the oil content of the refrigerant could be varied, was used in conjunction with an air conditioning system, which supplied the evaporator load. The heat transferred to the refrigerant was measured for different concentrations of oil in the refrigerant.

The load to the evaporator was supplied by 500 cfm of air maintained at 80°F dry bulb and 67°F wet bulb by a washer and an electric heater. The heat transferred to the refrigerant was determined by measuring the temperature of the liquid refrigerant entering the expansion valve and the suction temperature and pressure of the superheated vapor at exit from the evaporator. A sample of the liquid refrigerant flowing to the evaporator during the test was trapped in a sample collector and channeled to an expansion chamber, where the oil and refrigerant were separated and the oil concentration was determined.

The results of the investigation showed that oil tends to reduce heat transfer across the evaporator tube wall. At 25°F suction temperature the reduction in the heat transferred was 5% with an oil concentration of 0.84% by volume compared to the refrigerant with an oil concentration of 0.063%. At 30°F suction temperature the reduction in the heat transferred was 3% with an oil concentration of 1.52% by volume. The range of suction temperatures for the investigation was from 24°F to 31.9°F. The maximum average oil concentration obtained during the investigation was approximately 1.75% by volume of oil in the refrigerant.