AN AIR-TO-AIR HEAT PUMP

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II

TABLE OF CONTENTS

III

INTRODUCTION

The primary purpose of a heat pump is to utilize high-grade energy for pumping low-grade energy to such a temperature that it may be usefully employed. For example, an air-to-air heat pump, usually electically driven, removes heat from atmospheric air during the winter and discharges this heat at a high temperature into the interior air of the building in order to heat the building.

The heat pump has not been accepted by the general public as a standard means of heating. The principal proponents of the heat pump have been the power companies, for electrically-driven units would improve the load factor of these companies and increase their revenue. Electrically-driven heat pumps have proven more efficient than fuel-operated units. Although the power companies have been installing large units in some of their office buildings and devoting much publicity to the advantages of such units the companies have not published sufficient performance data to sell the heat pump to the public at this time.

The first cost of a heat pump is high and the unit cannot economically compete with fuel-fired systems for heating alone but the flexibility of the unit allows it to operate for both heating and cooling and warrants its consideration as a year-round system whose cost must be judged accordingly.

In fairness the cost of the unit should be compared with the combined cost of the fuel-fired heating system and the conventional cooling system.

As a practical heating system the heat pump is in an early stage of development. The value of such a system must be proven in facts and figures before the interest and confidence of the public will be aroused. New equipment must be designed; new refrigerants must be found and tried; older systems must be analyzed and improved; and data must be obtained by research and experimentation. A large amount of general material on heat pumps has been published but specific information on design and operational values is limited. Members of the Mechanical Engineering Department at Virginia Polytechnic Institute felt that it would be desirable to install an air-to-air heat pump in the laboratory in order to secure specific information about this type of unit and also to familiarize the engineering students with such a unit. In addition, such a unit would form the basic system for converting to other heat pump cycles from which additional data could be acquired, provide the necessary range of temperatures for finding the heat-transfer coefficients of coils at low temperatures, furnish a small auxiliary heating system in the winter and provide cooling in the summer for a part of the laboratory, and become a complete air conditioning unit - if additional equipment were used - for demonstration, research, and inclusion in the study program.

IV.

REVIEW OF LITERATURE

The heat pump is not a new discovery: its thermodynamic principles were recognized by Lord Kelvin in 1852 but its development has been slow and the number of applications is small. Essentially, the air-to-air heat pump is a compression-type, household refrigeration system which substitutes the outside air for the water in the ice tray, therefore taking heat out of the outside air instead of from the water in the tray, and discharging heat into the interior of the house in the same manner as the refrigerator. The heat pump attempts to lower the temperature of the atmosphere by expanding the refrigerant to such a low temperature that heat flows from the atmospheric air to the refrigerant in the coil; the refrigerant is then compressed to such a pressure that its temperature is above that of the air in the house, and the heat which the refrigerant picked up from the outside air and from the work of compression flows to the inside air.

The cycle of an air-to-air heat pump has been described. Basically there are four heat pump cycles: the air-to-air, the water-to-air, the water-to-water, and the air-to-liquid cycles. If the heat is absorbed from some water source such as a well or a lake and then discharged into the room air the cycle is a water-to-air cycle. The advantage of taking heat from water sources will be discussed later. The water-to-water cycle derives its heat from a water source and transfers the heat to the refrigerant which, in turn, gives it up to the water in the heating system. Air is used as the heat source in an air-to-liquid cycle. This cycle is more elaborate than the other three, for the air gives up heat to a liquid which transfers it to the refrigerant in the evaporator, and after compression the refrigerant gives up its heat in the condenser to the water which circulates in the heating system.

Why is a heat pump desirable? And if it is desirable, why has it not been more fully developed since it was first suggested nearly a hundred years ago? The heat pump is highly desirable because of its ability to deliver, in an actual installation, as much as five times the heat which is supplied to it as work. Even higher deliveries may be possible. The source of work for a heat pump is usually an electric motor; soot, fumes, fuel storage, and other associated problems of combustion heating systems are automatically eliminated. The reasons for the failure of the heat pump to achieve a higher state of development and use have been ably discussed by Mr. J. Mack Tucker, Associate Professor of Mechanical Engineering at the University of Tennessee.(19) His reasons may be summarized as follows: the existing installations are built from conventional refrigeration equipment although special equipment should be developed; only conventional refrigerants have been tried; none of the demonstration systems have been operated under practical conditions; the cost of a system is not justified unless the purchaser is also willing to pay for summer air conditioning; and the possible use of the system in process work has not been fully investigated.

Of the existing air-to-air installations, perhaps one of the most outstanding is that installed by the Appalachian Electric Power Company in its Roanoke, Virginia, office building.[17] Three separate and independent cycles, each driven by a 75 hp reciprocating Freon-12 compressor, comprise the system. The system was designed to satisfy the heating requirements of the building for outside temperatures as low as 15 degrees and the cooling requirements for temperatures as high as 95 degrees. Two water storage tanks, filled with hot water during the heating cycle and cold water during the cooling cycle, increase the capacity sufficiently to carry peak loads which fall outside of the design temperature limits for short periods of time. Temperate weather permits the heat pump to condition the storage water to the proper temperature while carrying the building load at the same time. The characteristics of this system are still unknown, for no operating data have been correlated and published at the present time.

The partial objective of this investigation is to run performance tests on an air-to-air heat pump; this investigation is primarily concerned with the performance of residential systems, not that of the larger, more expensive commercial installations. Among the residential units which are now available on the market are the "Airtopia" and a unit by York. The "Airtopia", manufactured by Drayer-Hanson, Inc., of Los Angeles, utilizes four coils for heat transfer but only two coils are used in the cycle at any one time. This unit also includes a preheater for the make-up air. The York heat pump has two separate refrigerating systems: the 3 hp system operates on both the heating and the cooling cycles whereas the 2 hp system operates only on the heating cycle. The 2 hp system supplements the other system when the outside temperature drops below 30 degrees. It also acts as a reheater when the 3 hp system is operating on the cooling cycle. Due to the addition of such refinements as dual systems, extra coils, preheaters, etc. it is doubtful if these units can be priced within the budget of the average homeowner. These refinements have been necessary in the past, however, because of the primary weakness of an air-to-air heat pump: less heat is available when the demand is greatest. The operating efficiency of a heat pump, called its coefficient of performance, is equal to the heat output divided by the work required to make this heat available at the upper temperature. Using the Carnot cycle as a basis for the analysis it can easily be shown on the Temperature-Entropy diagram that, for a given discharge temperature (the temperature of the refrigerant vapor in the conditioning coil), the work area is increased as the suction temperature is lowered, thereby decreasing the coefficient of performance for the cycle. Thus, the COP must decrease as the work is increased for a given amount of delivered heat. Emory N. Kemler has plotted the Carnot cycle efficiencies for an air-to-air heat pump;[8] the COP falls from 11.2 at 50 degrees Fahrenheit outside air to 5 at -10 degrees outside air for air delivered at 100 degrees Fahrenheit. Kemler and Oglesby demonstrate in their book[11] that the COP of a commercial compressor will be less than half of the Carnot cycle COP at suction temperatures below 0 degrees for a condensing temperature of 102 degrees. Hence, an actual cycle approaches a COP of 2 or less at low temperatures and the desirability of the heat pump lessens. The maximum COP obtained in the air-to-air installation in the Ohio Power Company building at Portsmouth, Ohio, for an outside air temperature of 32 degrees was 2.4; two 25-hp compressors operated at 160 psi head pressure and 10 psi suction during this test. The COP was increased to 2.9 for 170 psi head pressure and 12 psi suction using a water spray over the coil. When the temperature drops below 20 degrees the outside air is no longer used as the heat source; instead the circulating fan is stopped, the outside air duct is closed off, and city water is sprayed over the evaporating coil. The description of this installation is given in _A Review of Commercial Heat Pump Installations._[12]

The above performance data indicate that it is better to have a water-to-air heat pump because the source of heat remains almost constant the year round. Therefore, it can be seen that when there is the greatest heating demand the COP will not decrease. It is indeed unfortunate that nature has not provided underground water in such abundance as to permit general usage over large areas, for underground water is an almost constant temperature heat source whose temperature is seldom less than 50 degrees. Such high grade energy can be readily "pumped" to a practical condenser temperature with little compressor work, thereby enabling even the simple water-to-air system to maintain high COP's. Kemler[7] considers only three basic sources of heat, the air, underground water, and the earth, although he lists radiant energy from the sun and local bodies of water as sources of heat in nature which man might possibly utilize after further study and research. Kemler states that "practical considerations would appear to limit the air-to-air unit to temperatures of from 10 degrees to 20 degrees Fahrenheit" because of the inherent weakness which has already been discussed. However, this system is particularly advantageous in areas such as California where the average COP may be from 5 to 6 for a mild winter. Kemler cites the advantages of the water-to-air unit as its high temperature heat source, its compactness due to the excellent coefficient of heat transfer, and its ability to deliver at maximum capacity when the demand is greatest. The major problem for this unit is to locate an adequate source of water; a minor problem is the disposal of the water after use. This unit is highly preferred over the air-to-air unit when the water source is available. The third basic heat source is the earth. Probably the two most important methods for removing heat from the earth at the present time are: 1). Drilling of a deep well in which a closed U tube can be placed. Fluid is pumped through the tube, picking up heat from the well water which, in turn, picks up its heat from the earth. The fluid flows through a closed system, and its heat is transferred to the refrigerant in the evaporator. Instead of pumping a fluid through a U tube the well water may be pumped up to the evaporator and then returned to the source but this entails scale and corrosion problems unless the well water is practically pure. 2). Burying tubing at a given depth in the ground. Again, the fluid is pumped through a closed system as it picks up heat from the earth and gives it up in the evaporator. Sometimes the tubing serves as the evaporator but this type of system is conducive to leakage of the refrigerant, trapping of the oil, etc. and according to Kemler and Oglesby,[11] may see little use in the future although G. D. Wetherbee[20] describes such a system, mentions no mechanical troubles, and concludes that the performance data which have been obtained from an actual installation at Mt. Prospect, Illinois, are convincing proof of the system's suitability as "a method of heating buildings in regions with a normal heating season of 6300 degree days". The overall coefficient of performance (this included the input to the blower motor and the automatic controls) was 3.32. The evaporator was composed of 770 feet of one inch tubing and 148 feet of five-eights tubing buried six feet below the surface of the ground. This system gave an overall COP of 3.32 for typical operating conditions (163 psig discharge pressure and 21 psig suction pressure) in the winter of 1949.

When air, used as the source of heat, has a high relative humidity at a temperature of 32 degrees F or below, the outdoor coil has to be defrosted. If periodic defrosting is not provided there will be an accumulation of ice on the surface of the coil, thus lowering the heat transfer between the evaporating refrigerant and the air. This accumulation of ice will cause the compressor to operate at a lower suction pressure with a corresponding reduction in output and coefficient of performance. Philip Sporn and E. R. Ambrose[16] have suggested several systems that could be used for defrosting outside coils. 1). Probably the most convenient method of defrosting the outside coil when it is used as the condenser is to reverse the flow of the refrigerant to it. Louvers which are installed on both sides of the coil are closed during the defrosting cycle to decrease the heat loss to the outside air. One disadvantage of this method is that heat is removed from the conditioned space during the defrosting cycle. 2). Water sprays offer a simple and convenient method for defrosting the outside coil in locations where there is an ample supply of chemically suitable water with a temperature of 50 degrees F or above. The sprays are installed on both sides of the coil if the coil is more than four rows deep. The water is sprayed onto the coil in sufficient quantities and velocities to quickly remove the ice from the coil surface. The air circulating fan should not operate during the defrosting cycle because the heat given up to the outside air by conduction and evaporation would be so great that very little defrosting would be accomplished. Louvers may also be applied to this method to prevent excessive losses to the air. The operation of the compressor during the defrosting cycle may be done provided that the quantity of water is sufficient to maintain the suction temperature above 32 degrees F.; otherwise more ice would accumulate on the coil. Where the cost of water is appreciable it would probably be more economical to stop the compressor and recirculate the water instead of having a once-through system, for the water temperature is reduced very little with just one pass over the coil and, if necessary, a certain percentage of high temperature make up water may be added to maintain the desired water temperature for defrosting. When returning to the heating cycle from the defrosting cycle it is best to start the fan a little before starting the refrigerating compressor so that the water clinging to the coil will be removed when the coil surface temperature is reduced below freezing. 3). Solid adsorbents such as silica gel, activated alumina, activated carbon or a similar material may be employed to dehumidify the air before passing the outdoor coil. The solid agent could be carried on a continuously rotating screen just ahead of the outdoor coil. The screen is rotated at a speed to permit the proper time for the reactivation and dehydration cycle. The water vapor from the outdoor air condenses as it passes over the adsorbent bed, giving up the latent heat of condensation; clean hot air ranging from 300 to 350 degrees F may be passed over the other portion of the rotating screen to reactivate the adsorbent. This system gives a continuous supply of heated air to the conditioned space but the additional initial cost would not justify such a system. 4). By using water sprays as previously described it is possible to bring a liquid absorbent into intimate contact with the air. The vapor pressure of the absorbent can be controlled by the concentration and temperature of the solution. Since the absorbent has a lower vapor pressure the moisture in the air will be absorbed. As the solution continues to absorb moisture from the air the solution becomes diluted,thus losing its power as an absorbent. The diluted absorbent is piped to a solution concentrator where an auxiliary source of heat is used to drive off the excess water of condensation. A water solution of lithium chloride, calcium chloride, and lithium bromide is usually used as the absorbent in this type of system. 5). Electrical energy offers several possibilities for defrosting the outdoor coil. One method is to locate a bank of electric strip heaters adjacent to the coil surface. The outdoor air fan is stopped and the inlet and outlet dampers are closed. The coil is defrosted by the convected and radiant heat given off by the electrical heaters. Another method is to heat the metal surface of the coil instead of the ambient air. This is accomplished by letting the coil act as an electrical resistance. By application of low voltage and high current the coil surface temperature rises quickly and melts the ice. The use of electrical energy for defrosting is very good but the cost of electrical energy in most localities would prevent the use of this type of system. 6). A system using the same principle as described in system number one can be used with two smaller outside coils instead of having one large outside coil. By an arrangement of valves the hot refrigerant gas can be made to pass through two parallel paths. Part of the gas will flow through the conditioner coil and the other part will flow through one of the outside coils. The hot gas flowing through the outside coil would be condensed, giving up its latent heat of condensation to melt the ice. The other outside coil would act as the evaporator, absorbing heat from the outside air to be delivered to the conditioner coil. The main advantage of this system is that one half of the normal amount of heat can be delivered to the conditioned space while one outside coil is being defrosted. 7). The possibilities of storing water during the offpeak periods for heating the conditioned space can also be used to defrost the outdoor coil. During the defrosting cycle the refrigerant flow is reversed from that of the heating cycle, going first to the outdoor coil, then to the liquid receiver, through the expansion valve, the water storage tank coil, and back to the compressor. The coil in the water storage tank acts as the evaporator, absorbing heat from the water which is delivered to the outdoor coil where it is used to defrost the coil. Since there is considerably more heat in the storage tank than that required for defrosting, the circulating pump can still deliver hot water to the conditioner coil. The main advantage of this system is that the refrigerating equipment can be sized to satisfy the average temperatures during both the heating and cooling cycles.

A committee under the chairmanship of R. C. Jordan [6] has published an article on the research needed for the proper utilization of heat pump sources and sinks. A few suggestions where research is needed for air as a heat source will be listed in outline form.

1). Typical climatic data for winter and summer.

2). Recommended winter outdoor design temperatures (dry bulb and wet bulb).

3). Relative capacities for evaporator and condenser service.

4). Air quantities to be handled.

5). Refrigerant quantities required.

6). Effect of wet bulb temperature on coil capacity.

7). Factors affecting frost formation and accumulation.

8). Methods of defrosting.

9). Effect of frost accumulation on heat pump capacity and coefficient of performance.

V

THE INVESTIGATION

A. OBJECT OF THE INVESTIGATION

The object of the investigation was to assemble an air-to-air heat pump from available equipment and to run performance tests on this unit for its heating cycle in Blacksburg, Virginia.

B. DESIGN CONSIDERATIONS

As stated in the object of the investigation the heat pump had to be assembled from available equipment. This meant that the components of the unit had dissimilar capacities and each component had to be considered as a possible limiting-factor when determining the capacity of the unit.

Phillip Sporn and E. R. Ambrose(17) based the design of a heat pump in Roanoke, Virginia, on a dry bulb temperature of 15 degrees F. It would seem reasonable to accept the same design temperature in nearby Blacksburg but a compromise design temperature of 30 degrees F was used since it is impractical to pump a given quantity of heat through a large temperature range with single-stage compression. Table I of this section shows how the volumetric efficiency of the compressor and the theoretical performance of the unit decrease as the temperature range of the compressor increases. The design temperature of 30 degrees created no real hardships since the unit was not to be used or depended upon as the source of heat for an actual residence.

The capacities of the coils were considered first. The coils were unknown quantities, being unmarked both as to identity and performance characteristics, but each was tagged with a card which had the words "5 Tons" written upon it. The value on the card was accepted as the capacity of each coil. This value seemed reasonable when the dimensions of the coils were compared with those of rated coils in manufacturers' guides.

Two identical fans were available for the construction of the unit. The usual practice in the air conditioning industry[3] is to select an air velocity between 400 and 600 feet per minute for the air passing through a cooling coil because higher velocities tend to carry over the moisture which has condensed out of the air. An air velocity of 600 fpm through the cooling coil whose cross-sectional area is 1.67 square feet would give a fan output of 1000 cubic feet of air per minute. This output is far below the maximum output of the fan as given in the Trane Condensed Catalog, Bulletin PB-290, Revised, November, 1946, for the No. 12 single-width, single-inlet, forward-curved fan. Assuming a maximum temperature differential of 18 degrees for the air across the evaporator coil the amount of heat which is given up by the air may be found by

$$w_{ca} = \frac{pV_{ca}}{RT_{ca}} = \frac{13.7 \times 144 \times 1000}{53.3(460 + 30 - 18)} = 78.4 \text{ lb air/min}$$

$$Q_{ca} = w_{ca}c(t_{ca} - t_o) = 78.4 \times 0.24 \times 18 = 339 \text{ Btu/min}$$

where the symbols are defined in the Table of Symbols in the Appendix. Q_{ca} is the minimum amount of heat which will be given up by the air when its temperature drops 18 degrees, for the calculation neglects the latent heat of the moisture which drops out of the air and increases the quantity of heat given up by the air as it passes across the coil. The amount of condensation is dependent upon the relative humidity of the air and the apparatus dew point (mean surface temperature) of the coil, two variables which are difficult to predict accurately for all operating conditions and which will therefore not be considered in the calculations. So for an air velocity of 600 fpm at the design atmospheric-temperature of 30 degrees F, the air will give up at least 339 Btu/min to the evaporator coil if the temperature drop across the coil is 18 degrees. It was decided to run the fan at a speed corresponding to its maximum output of 1884 cfm and to vary the output by means of a damper in the fan inlet. This method would allow the air velocity to be varied without changing the pulley size on the constant-speed motor if the operator desired to test the unit at different air velocities.

The compressor was considered next. Allowing a 10-degree differential between the air leaving the evaporator coil and the Freon within the coil in order to secure proper heat transfer and to compensate for the by-pass factor (the ratio of the unconditioned air across the coil to the total air across the coil) of the coil gives an evaporation temperature for the Freon of 2 degrees F when the outside temperature is at the design condition of 30 degrees. To be more specific, 30 - 18 - 10 gives 2 degrees, the temperature at which the Freon must evaporate for an outside temperature of 30 degrees. Reference to Freon tables gives an evaporation pressure of 24.9 psia for a saturation temperature of 2 degrees F; the compressor must operate at a suction pressure of 24.9 psia for the stated design conditions.

The following analysis will be based upon certain assumptions which will consider the cycle as a perfect or theoretical cycle. The volumetric efficiency of the compressor is the only characteristic of the system for which the calculations will compensate. This method is used by the Trane Air Conditioning Company in designing refrigeration systems and should therefore be acceptable in this work. The cycle analysis will be based upon these assumptions:

(1) Adiabatic compression takes place.

(2) The liquid reaches the expansion valve at the same temperature at which it was condensed.

(3) Vapor with 5 degrees superheat leaves the evaporator.

(4) Vapor with 5 degrees superheat reaches the suction of the compressor.

The first two assumptions were taken from the <u>Trane Air Conditioning Manual.</u>[18]

The Temperature-Entropy Plane of Fig. 1 shows the heating cycle of the heat pump. The liquid refrigerant at 3 is at the discharge pressure of the compressor. The liquid is expanded into a wet mixture by a constant-enthalpy process from 3 to 4 where it enters the evaporator. The wet mixture at 4 evaporates into a superheated vapor and enters the compressor at 1 whereupon it is then compressed adiabatically to 2. From 2 it enters the condenser in a superheated state. The condenser removes the latent heat (plus the superheat) of the refrigerant and condenses it back to its starting point at 3. The suction temperature of the refrigerant must be below the temperature of atmospheric air if heat is to be transferred from the air to the refrigerant and the discharge temperature of the refrigerant must be above the temperature of the room air if heat is to be transferred from the Freon in the condenser to the room air.

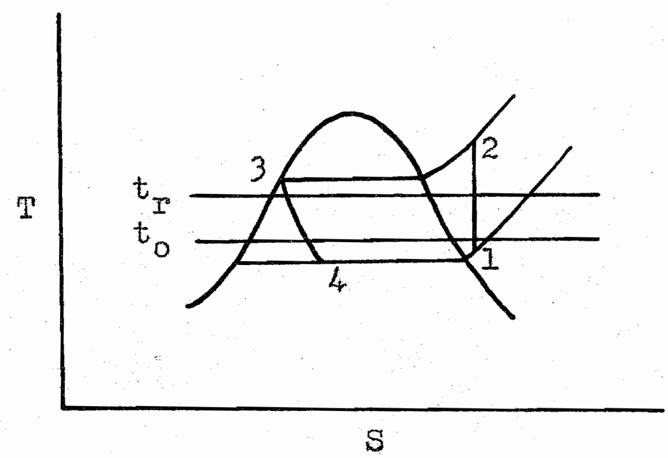


Fig. 1 Heating Cycle

1-2 Adiabatic compression
2-3 Constant-pressure condensation
3-4 Constant-enthalpy expansion
4-1 Constant-pressure evaporation
t_r Room temperature
t_o Outside air temperature

The suction pressure has already been fixed at 24.9 psia for an evaporation temperature t_4 of 2 degrees F. The vapor at 1 is at 24.9 psia and 7 degrees F since it has 5 degrees of superheat. For design purposes the room temperature t_r will be taken as 75 degrees F: therefore the compressor must discharge the superheated vapor at such a pressure that the corresponding saturation temperature is above 75 degrees. The conditions of the refrigerant in the condenser and the conditions of the air leaving this coil will be found by assuming various conditioned-air temperatures and finding the related characteristics of the system for these conditions. For instance, assume that the conditioned air leaves the condenser at a temperature of 80 degrees. Then the saturation temperature of the condensing Freon must be held at 80 + 10 or 90 degrees F if a 10-degree differential is to be maintained between the leaving air and the hotter refrigerant. The saturation pressure at 90 degrees is 114.3 psia.

Known:

$t_1 = 7$ degrees $\qquad h_1 = 79.8$ Btu/lb

$p_1 = 24.9$ psia $\qquad d_1 = 1.58$ cu ft/lb

Assumed:

$t_{da} = 80$ degrees

Found:

$t_3 = t_{da} + 10 = 80 + 10 = 90$ degrees

$p_3 = 114.3$ psia

$h_3 = 28.7$ Btu/lb

$h_2 = 90.8$ Btu/lb

Constant-entropy process to p_3 from 1.

$$r = \frac{\text{absolute head pressure}}{\text{absolute suction pressure}} = \frac{114.3}{24.9} = 4.59$$

The approximate volumetric efficiencies of commercial Freon compressors are listed in Table 6-7 of the <u>Trane Air Conditioning Manual</u> for various compression ratios.

$e_v = 67.2\%$ for $r = 4.59$

The compressor - Model No. FW-500FH - has a displacement of 1125 cu ft/hr at its rated speed of 435 rpm, according to the specifications issued by the Frick Company.

$$V_d = 1125 \text{ cfh} = 18.75 \text{ cfm}$$

$$w = \frac{e_v \times V_d}{d_1} = \frac{0.672 \times 18.75}{1.58} = 7.98 \text{ lb/min}$$

$$Q_A = w(h_1 - h_3) = 7.98(79.8 - 28.7) = 407 \text{ Btu/min}$$

$$W = w(h_2 - h_1) = 7.98(90.8 - 79.8) = 87.8 \text{ Btu/min}$$

$$Q_R = W + Q_A = 87.8 + 407 = 494.8 \text{ Btu/min}$$

$$Q_{R'} = \frac{Q_R}{200} = \frac{494.8}{200} = 2.47 \text{ tons}$$

$$COP = \frac{Q_R}{W} = \frac{494.8}{87.8} = 5.64$$

$$w_{da} = \frac{Q_R}{c(t_{da} - t_r)} = \frac{494.8}{0.24(80 - 75)} = 412.0 \text{ lb air/min}$$

$$V_{da} = \frac{w_{da}RT_{da}}{p} = \frac{412 \times 53.3(460 + 80)}{13.7 \times 144} = 6000 \text{ cu ft/min}$$

$$v_{da} = \frac{V_{da}}{A} = \frac{6000}{1.67} = 3600 \text{ fpm}$$

By using the known conditions and assuming other temperatures for the conditioned air the above calculations can be used to find the system characteristics which correspond to each conditioned-air temperature. The table on the following page was made up in this manner. No temperatures above 110 degrees were considered for the conditioned air since operation above this point would involve high pressures at risk of damage to the unit.

Table I

System Characteristics

The following conditions are constant:

Compressor suction:

Temperature: 7 degrees F
Pressure: 24.9 psia
Enthalpy: 79.8 Btu/lb
Specific volume: 1.58 cu ft/lb

Saturation temperature (t_3) in condenser:

Conditioned-air temperature (t_{da}) $\neq$ 10

Compressor displacement: 18.75 cfm

	I	II	III	IV
t_{da}	80.0	90.0	100.0	110.0
t_3	90.0	100.0	110.0	120.0
p_3	114.3	131.6	150.7	171.8
h_2	90.8	92.0	93.2	94.4
h_3	28.70	31.16	33.65	36.16
r	4.59	5.28	6.05	6.89
e_v	67.2	64.5	61.8	59.0
w	7.98	7.65	7.34	7.00
Q_A	407.0	372.0	338.0	305.5
W	87.8	93.4	98.3	102.2
Q_R	494.8	465.4	436.3	407.7
Q_R'	2.47	2.32	2.18	2.04
COP	5.64	4.99	4.44	3.99
v_{da}	3600.0	1150.0	659.0	447.0

The air delivery at 80 degrees is impractical for many reasons, some of the more prominent ones being the very high delivery velocity across the coil, the fact that such low temperature air would not be used in a hot-air heating system, and the previous calculation which showed only 339 Btu/min being rejected by atmospheric air to the evaporator. Practically the same limitations apply to the system for air delivery at 90 degrees so the first two sets of conditions are readily discarded. Delivery of conditioned air at either 100 or 110 degrees is practical and the velocities at these temperatures are sufficiently close to the recommended coil velocities to be acceptable. And it is well to remark here that there is no problem of moisture entrainment in a heating coil since the coil reduces the relative humidity of the air: hence the air velocities across heating coils may be higher than those across cooling cools without suffering the consequences of moisture carry-over. The system was sized for the conditions of Column III because the air velocities across the coils are practical, the output and the theoretical COP are higher than for the conditions of Column IV, and the heat picked up in the evaporator is approximately equal to the amount which had previously been calculated to be given up by the outside air to the evaporator.

The tubing was sized according to the methods and tables in the Trane Refrigeration Manual. It was shown in the review of literature how more heat is available at higher temperatures:

Kemler and Oglesby[11] explain that the air-to-air heat pump must furnish the maximum power at the higher outside temperatures because the volumetric efficiency of the compressor increases with the suction temperature for a given discharge pressure. If the compressor motor is operating at rated load when the heat requirement is the greatest, the motor will be severely overloaded when the outside temperature rises and the weight of flow increases with the higher suction temperatures. But less weight of flow is needed to maintain the desired inside temperature when the outside temperature rises and a capacity control is usually installed to reduce the output of, and thereby the input to, the system. The capacity control prevents overheating of the building, overloading of the compressor motor, and waste of electrical energy. Kemler and Oglesby describe the possible methods of capacity control and list such methods as: (1) by-passing a portion of the discharge from the cylinders back into the suction of the same cylinders, (2) placing a stop valve in the suction line of a cylinder or bank of cylinders, (3) throttling the suction, (4) varying the speed of the compressor motor, (5) using a hydraulic speed-control device, and (6) employing cylinder-cutout methods. Two methods were used in this design: a throttle valve was placed in the suction line and a by-pass valve was placed between the suction and discharge lines.

In the discussion of the various methods Kemler and Oglesby indicate that the suction-throttling method is not desirable because of its low efficiency and that friction in the system prevents the by-pass method from reducing the power requirements in proportion to the reduction in capacity. The methods used were the only ones which were applicable to the available compressor unless special equipment were purchased. The flow lines were generously sized to carry more than the design capacity since the future uses of the unit were unknown at the time of its construction. If tests were to be run which required a higher capacity than the design capacity oversized lines would prevent excessive pressure drops in the lines. In any event the capacity would have to be regulated by one of the two controls to hold the compressor input within the rating of the motor. The tubing size was selected from the tables in the Trane Refrigeration Manual according to the capacity and the allowable pressure drop in fifty feet of tubing. All lines were sized for a refrigeration capacity of 2.8 tons and a pressure drop of 3 psi in 50 feet. Because of the differences in the specific volumes of the vapors the suction line to the compressor should be larger than the hot-vapor line which delivers the refrigerant from the discharge to the condenser. But the functions of the two lines are reversed when the unit is switched from the heating to the cooling cycle and both were made equal to the larger size of 1.025" ID and 1-1/8" OD as given in the table for the suction line. The liquid line was sized as 0.430" ID and 1/2" OD. The lines will handle more than 2.8 tons of heating effect because of the fact that they were selected to handle 2.8 tons of refrigeration.

C. PIPING

The piping schematic of the heat pump is shown in Fig. 2. The operation of the heat pump, both the heating and the cooling cycles, and the accessories of the system will be explained.

The heating cycle will be discussed first. Valves 2, 4, 6, 8, 10, and 12 and the accessory valves 14, 15, 18, and 19 are open and all other valves are closed. The hot refrigerant vapor leaving the discharge of the compressor flows to the oil separator where the entrained oil is eliminated from the vapor. From the oil separator the vapor flows to the inside coil and the vapor is condensed, giving up its latent heat of vaporization and its heat of compression to the conditioned air. Then the liquid refrigerant flows to the receiver and through the dehydrator where the moisture, if any, is removed. The refrigerant flows to needle valve 10 where it is expanded into the outside coil. In this coil the refrigerant picks up its latent heat of vaporization from the outside air, changing the liquid refrigerant to a cold vapor. The cold vapor then flows to the suction side of the compressor where the heat of compression is added in the form of work and the cycle is repeated.

When the system operates on the cooling cycle valves 1, 3, 5, 7, 9, and 11 and the accessory valves 14, 15, 18, and 19 are open. All other valves are closed. The hot refrigerant vapor

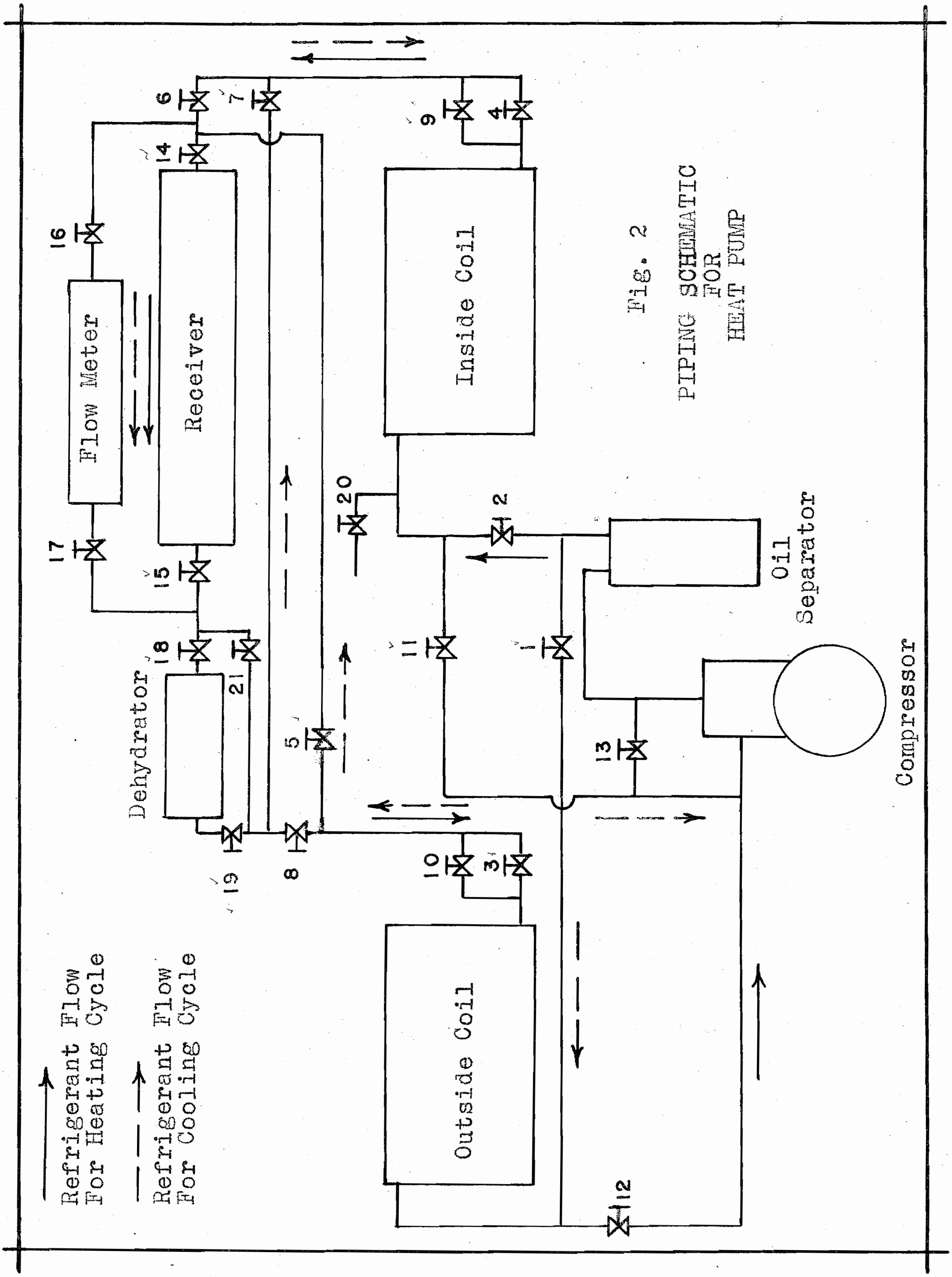


Fig. 2
PIPING SCHEMATIC FOR HEAT PUMP

flows to the outside coil and the vapor is condensed, giving up its latent heat of vaporization and its heat of compression to the outside air. Leaving the outside coil the liquid refrigerant flows through valve 5 to the receiver and on through the dehydrator. The refrigerant is expanded through needle valve 9 into the inside coil where the refrigerant picks up its latent heat of vaporization, cooling the conditioned air. The refrigerant, now a cold vapor, flows to the suction side of the compressor and the cycle is repeated.

Since the flow meter is in parallel with the receiver the refrigerant flow in the system can be measured by the adjustment of valves 14, 15, 16, and 17. With valves 14 and 17 closed and the valves 16 and 15 open the liquid refrigerant, instead of flowing to the receiver, will flow into the flow meter and the system will continue to pump refrigerant from the receiver. When the flow meter is full valves 16 and 15 are closed and valves 14 and 17 are opened. The system is then pumping the refrigerant from the flow meter through the system and back to the receiver. By noting the number of inches of refrigerant drop in the flow meter in a certain interval of time the refrigerant flow can be calculated. The dimensions of the flow meter are 8 inches inside diameter and 20 inches high.

The dehydrator contains silica gel which removes the moisture from the liquid refrigerant as it passes through.

When the system is free of moisture valves 18 and 19 are closed and valve 21 is opened to by-pass the dehydrator. When the silica gel becomes saturated with moisture valves 18 and 19 are closed and the union preceding valve 19 is opened. The silica gel can then be reactivated by applying heat to the dehydrator by means of a torch, thus driving off the excess moisture from the unit.

The oil separator employs centrifugal action to eliminate the entrained oil from the hot refrigerant vapor. The vapor enters at the top of the separator and changes direction by 180 degrees around a baffle to reach the outlet port of the separator. The collected oil is removed from the separator by means of a drain valve at the bottom.

The piping schematic of the control board is shown in Fig. 3. Located on the control board are most of the control valves that have to be adjusted when the system is changed from the heating cycle to the cooling cycle. Due to the size of the pipes and the valves of the vapor by-pass section the section is located behind the control board with the valve stems and handles projecting through holes to the front of the board. The entire liquid by-pass section is mounted on the front of the control board. The piping is so arranged that all lines connected to the top of the control board go to the coils and the lines connected to the bottom of the control board go to the receiver, compressor, oil separator,

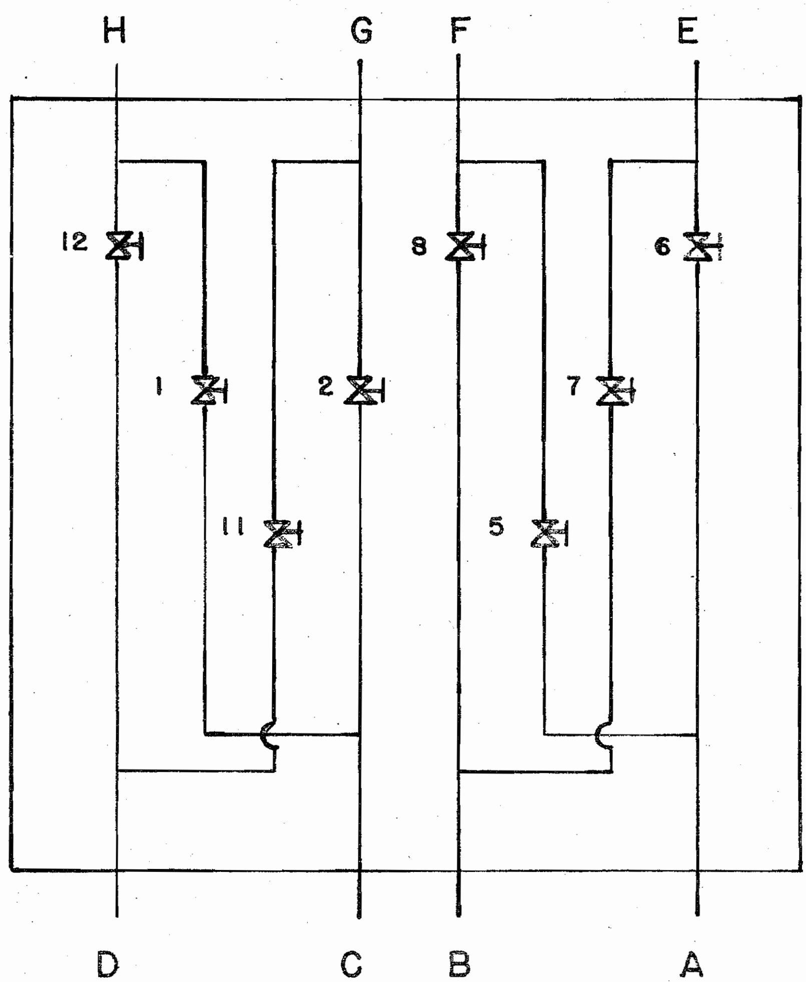


Fig. 3

PIPING SCHEMATIC OF CONTROL

A - To receiver
B - From receiver
C - From compressor discharge
D - To compressor suction
E - Liquid to and from inside coil
F - Liquid to and from outside coil
G - Vapor to and from inside coil
H - Vapor to and from outside coil

dehydrator, and flow meter. Only one vapor and one liquid line service each coil and these lines serve for both cycles. Thus the change of the refrigerant flow, which in turn changes the cycle, is accomplished by adjustment of the valves located on the control board. When the system is operating on the heating cycle valves 6, 8, 2, and 12 are open and valves 7, 5, 11, and 1 are closed. With this arrangement the refrigerant will flow directly to and from the component parts of the system. When the system operates on the cooling cycle valves 7, 5, 11, and 1 are open and valves 6, 8, 2, and 12 are closed. When operating on this cycle the by-passes come into play. The liquid refrigerant flowing from the receiver reaches the control board at point B and flows up through valve 7 to point E and on to the inside coil. After expanding through the inside coil the vapor returns to the control board at point G and flows through valve 11 to point D which is connected to the suction side of the compressor. The vapor after being compressed returns to point C on the control board and flows through valve 1 to point H which is connected to the outside coil. The refrigerant being condensed in the outside coil is returned to the control board at point F and flows through valve 5 to point A and then into the receiver where the cycle is repeated.

Valve 13 is a by-pass valve between the compressor discharge and suction lines. The purpose of this valve was discussed in the design considerations.

Thermometer wells and tees for pressure gages were installed in the lines at the entrance and exit of the compressor and the coils.

An expansion valve and a globe valve comprise the type of by-pass that was installed in the liquid line leading to each coil. The expansion valve remains closed and the globe valve is open when the coil the by-pass precedes is functioning as a condenser. The valves are reversed if the coil operates as an evaporator. A sight glass was installed just ahead of each by-pass to enable the operators to determine whether or not the system is functioning properly. Valve 20 of Fig. 2 is a purging valve which is located on the vapor side of the inside coil at the highest point in the system.

The charging system consists of a 1/4" copper tube connecting a Freon tank to the suction side of the compressor.

D. STANDS, PULLEYS, AND DUCTS

STANDS

The stands that are shown in Fig. 4 and Fig. 5 represent the inside and outside stands respectively. These stands were designed with two thoughts in mind: first, sufficient room for maintenance and operation with as compact a unit as possible; and second, ease in dismantling of the unit when it is moved to a new location. The stands had to be of welded construction because angle iron was used. The frames were

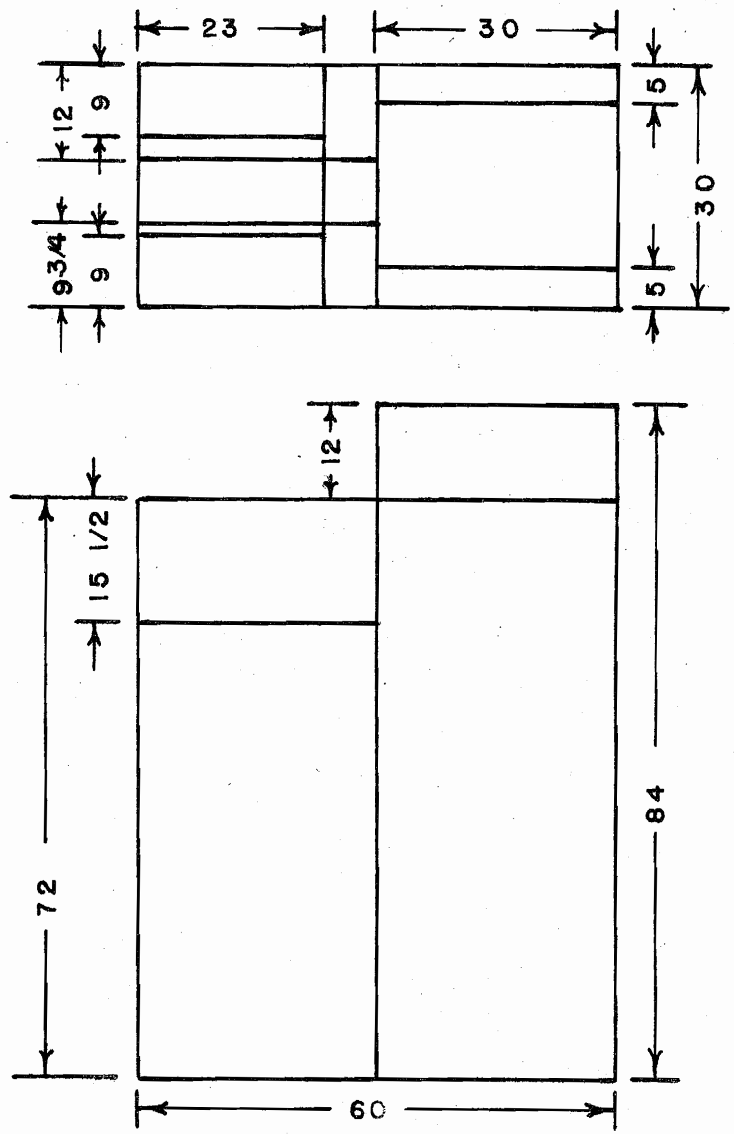


Fig. 4

INSIDE STAND
FOR
HEAT PUMP

Angle iron
1 x 1 x 3/16"
1-1/2 x 1-1/2 x 1/4"

Scale 1/2" = 1'-0"

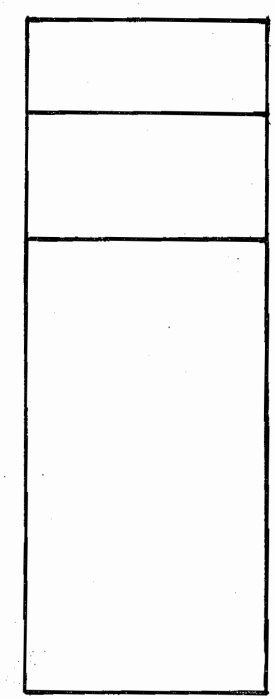
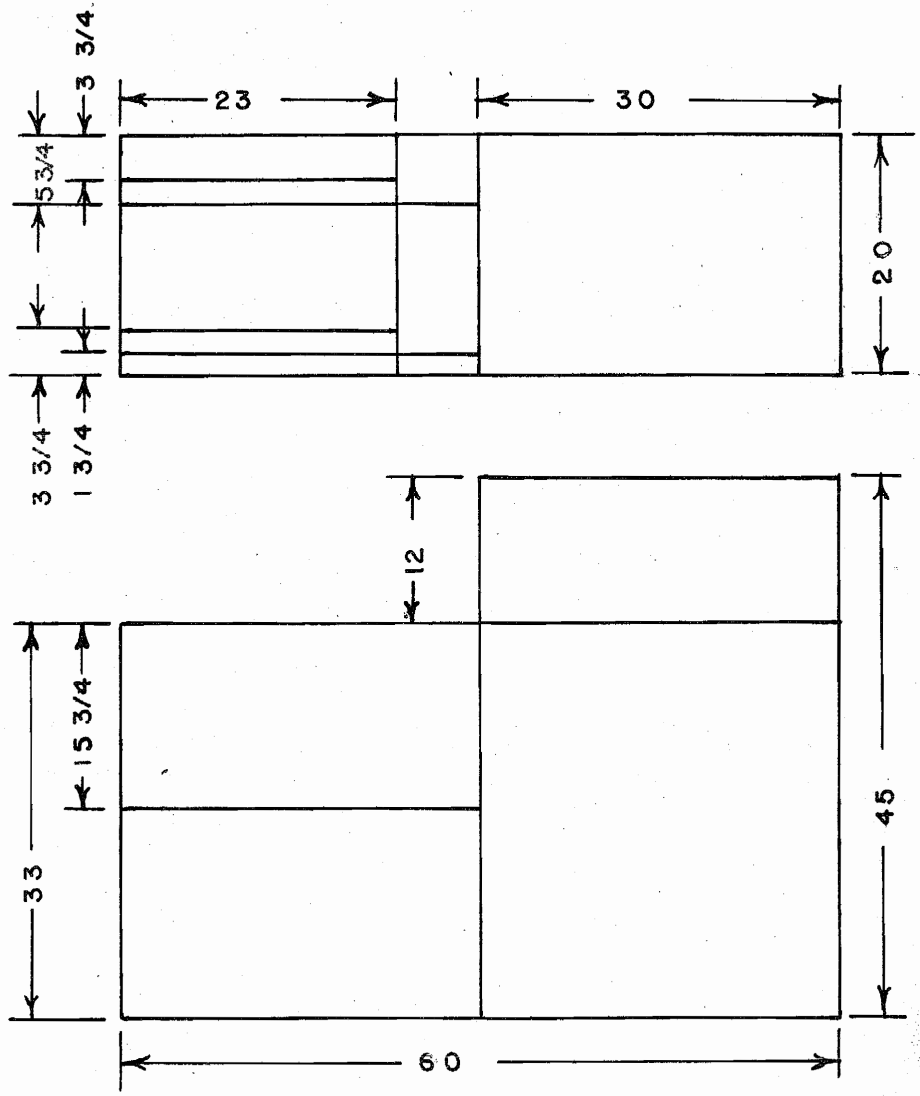


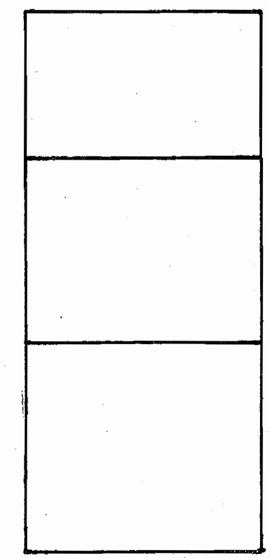
Fig. 5

OUTSIDE STAND
FOR
HEAT PUMP

Angle iron
1-1/4 x 1-1/4 x 1/4"
1-1/2 x 1-1/2 x 1/4"

Scale 3/4" = 1'-0"





made of 1 x 1 x 3/16" and 1-1/4 x 1-1/4 x 1/4" angle iron while 1-1/2 x 1-1/2 x 1/4" angle iron was used for the electric motor mounts. The coils, fans, and motors were bolted to the stands and are readily accessible for disassembly or adjustment.

PULLEYS

The pulleys were designed for the maximum volume of air that could be delivered against a two-inch static head. The pulleys were made of aluminum with the exception of the cast-iron motor pulley on the inside stand. Set screws and keys were used to prevent slippage of the pulleys on their shafts. The center distances between the pulleys are 22-1/8" for the inside stand and 22" for the outside stand. The outside diameters for the motor and fan pulleys are 3-1/2" and 4-9/16" for the inside stand and 4-1/2" and 4" for the outside stand. The difference in the speeds of the electric motors necessitated the difference in size for each combination of pulleys. The pulleys are connected with V-belts.

DUCTS

Ducts of 26-weight galvanized iron are used to convey the air to the coils and the conditioned room. A 12-1/4 x 29-1/2" round duct connects the suction side of the outside fan to the outside of the building. Two 12 x 20" to 10 x 12-1/2" reducing ducts, each 16" in length, connect the fans to the coils. The duct connecting the inside coil to the conditioned room is a 12 x 20" rectangular duct 36" long.

The ducts at the entrances of the coils and the duct from the inside coil to the conditioned room were constructed to have the same cross-sectional areas as the coils. For all practical purposes the velocity of the air leaving a coil is equal to the entrance velocity if the temperature change across the coil is neglected. Sturdy duct construction was achieved through the use of riveted seams. The ducts were connected to the fans and the coils by metal screws to insure ease in dismantling.

E. ELECTRICAL CONSIDERATIONS

The switching arrangement and the method of measuring power were selected to: protect the operator and the equipment; eliminate plug-in devices; give reasonable accuracy; permit any one of the three motors to be metered without affecting the operation of the other two; allow the power factor of each motor to be computed from the power readings; and to allow the measurements to be made quickly and easily. The two-wattmeter method of power measurement was selected to fulfill these requirements. This method - which requires two measurements to be made on each motor - dictated the switching arrangement since six readings had to be made with the one available wattmeter.

Precautions for protecting the instruments are few and are fully discussed in the procedure for making power measurements. Each of the motors is separately fused by a fusible disconnect box; the entire unit is then protected by a single fusible disconnect box which is mounted on the unit itself. As far as the safety of the operator is concerned he must be careful to avoid touching the exposed knife switches by other than the insulated handles.

Individual metering of each motor seemed to be an asset to the operation of the unit, for this system allows the operator to check each motor for overload and to determine the effect of each motor on the performance of the unit.

Considering an induction motor as a balanced load the power factor of each motor can easily be computed from the two readings of the wattmeter. The method is explained on pages 244-249 of Alternating-Current Circuits, Second Edition, by Kerchner and Corcoran.

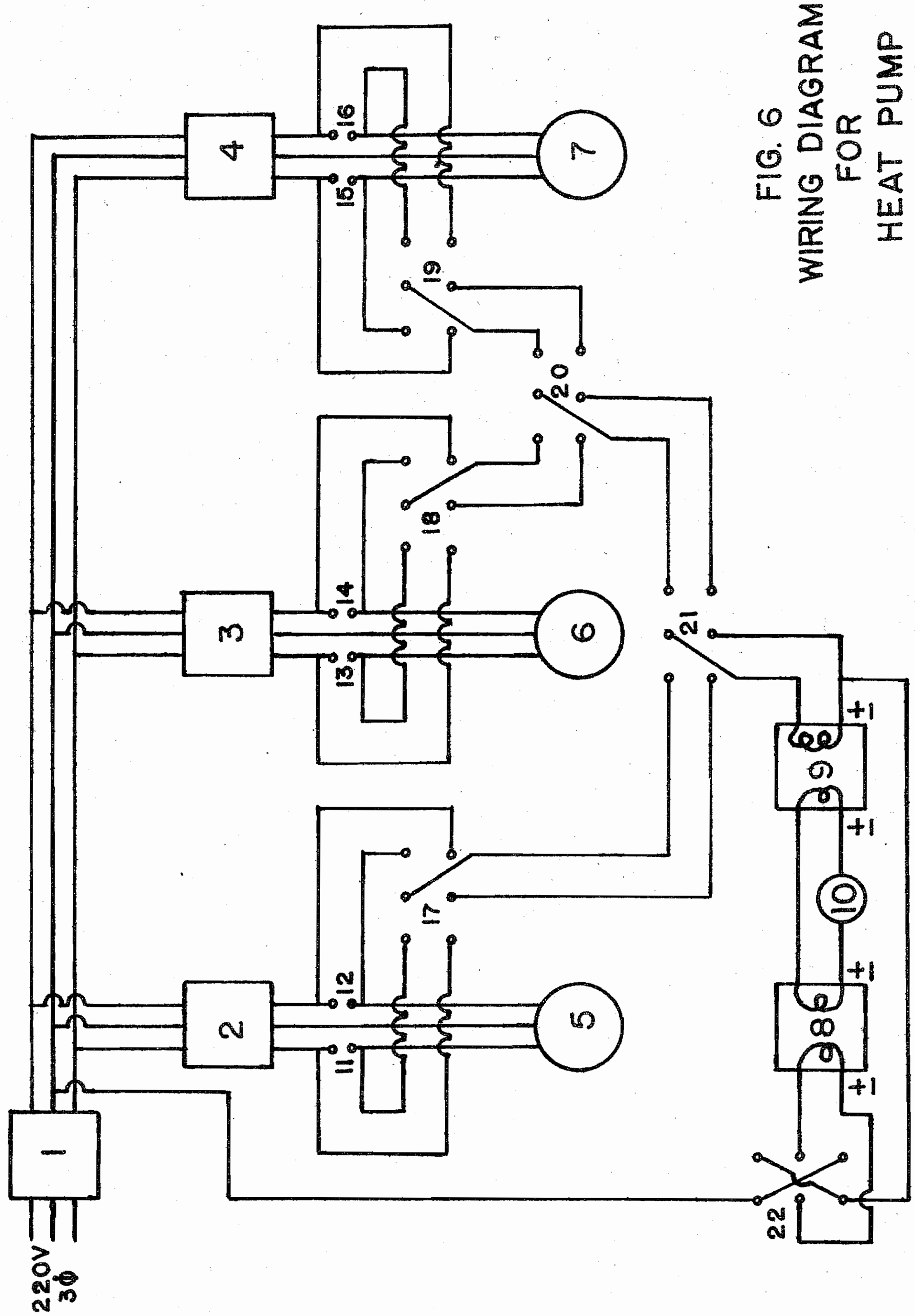


FIG. 6
WIRING DIAGRAM
FOR
HEAT PUMP

Legend for Wiring Diagram

1, 2, 3, & 4 -- Fusible disconnect boxes

5 -- Compressor motor

6 -- Motor for outside fan

7 -- Motor for inside fan

8 -- Wattmeter

9 -- Current transformer

10 -- Ammeter

11, 12, 13, 14, 15, & 16 -- Single-pole, single-throw knife switches

17, 18, 19, 20, 21, & 22 -- Double-pole, double-throw knife switches

F. POWER MEASUREMENT

The SPST switches are closed at all times except when one of them is opened to take a reading on the wattmeter. The DPDT switches may be opened or closed as long as the SPST switches are closed.

The motors are started in the following order: the SPST switches are placed in the closed position; the main switch on 1 is thrown in the "up" position to electrically connect the switchboard to the transformer bank; and the motors are individually started by closing the switches on their fusible disconnect boxes.

Switch 22 is usually left closed in the position which gives upscale readings on the wattmeter when power is being measured. Current transformer 9 is placed in the circuit to prevent damage to the wattmeter when the line current exceeds the five-ampere rating of the current coil in the wattmeter; the selector switch of the current transformer is always placed in the proper position before current is allowed to reach the wattmeter. The selector switch is placed on "10" when measuring power to either of the fan motors and on "25" when measuring power to the compressor motor. The numbers refer to the current flow in the primary winding of the current transformer when the secondary current is five amperes so placing the switch on "10" means that the ratio of primary current to secondary current is 10/5 or 2 and that the readings of the wattmeter and ammeter must be multiplied by this ratio in order to give the actual values of power and line current. Similarly, the "25" position gives a ratio of 25/5 or 5 and the meter readings have to be multiplied by five. The selector switch must be placed in the proper position before opening any of the SPST switches or damage to the wattmeter might result. The meter will not be damaged as long as the ammeter indicates a current of five amperes or less.

Assume that power to compressor motor 5 is to be measured. The currents through switches 11 and 12 must be individually passed through the current transformer, and the two power readings must be added together and their sum multiplied by the primary-to-secondary current-ratio to give the total power to the motor. Reference to the wiring diagram reveals the necessary switching arrangement for making the two power readings. First, the selector switch of the current transformer is placed on "25" since the compressor motor is being considered. Second, switch 21 is closed to the left to connect the current transformer to switch 17. Third, switch 17 is thrown to the left, switch 11 is opened, the wattmeter reading is recorded, and switch 11 is closed; and switch 17 is then thrown to the right, switch 12 is opened, the power is recorded for this configuration, and switch 12 is closed. It does not matter if switch 17 is thrown to the right first or to the left first as long as the corresponding SPST switch is opened for the wattmeter reading. Opening the non-corresponding SPST switch will place the motor on single-phase operation: this will cause no damage and is corrected by closing the switch and opening the other SPST switch for the reading. Fourth, the two power readings are added together and their sum is multiplied by five to give the total power to the motor for the existing load.

The procedure for motors 6 and 7 is practically identical to that of motor 5 except that switch 21 is closed to the right and then switch 20 is thrown to the left for motor 6 and to the right for motor 7. The selector switch is placed on "10" for both of these motors, as explained previously, and the readings are multiplied by two, of course. At light loads one of the two power readings might be downscale when power to either 6 or 7 is being measured; the meter is made to read upscale by throwing reversing switch 22 to the opposite position and considering the reading as negative power which must be substracted from the positive power. Switch 22 should be placed in its usual position after measuring the negative power.

The switching arrangement was color coded on the control board to eliminate the necessity of having to memorize the above operations.

G. APPARATUS

1. Compressor Motor

General Electric Induction Motor
Model 5K 254 B1001
208 volts 14.1 amps
Frame 254 Type K
60-cycle, 3-phase
Speed F. L. 1735 rpm
5 hp continuous 40 degrees C rise
No. ER 12014
Service factor 1.15 at rated voltage and cycle

2. Motor For Inside Fan

Western Electric Induction Motor
Model No. 400014
Type KT936
Form C 3-phase 60-cycle
220 volts 12.7 amps
Speed F. L. 1740 rpm
5 hp continuous 40 degrees C rise
No. 4438545

3. Motor For Outside Fan

Westinghouse Induction Motor
Type CS
220 volts 8.2 amps
Speed F. L. 1160 rpm
60-cycle, 3-phase
3 hp continuous 40 degrees C rise

4. Power Measurement

Wattmeter 0-1.5 kw
Weston Electrical Instrument Corporation
Newark, New Jersey
Model 432 No. 9919
Current Transformer 25 amps
Westinghouse
Type PC-135
No. 3729204
Style 1002149
Ammeter 0-30 amps
Weston Electrical Instrument Corporation
Newark, New Jersey
Model 528
No. 93884
6 Single-pole, single-throw switches
6 Double-pole, double-throw switches

5. Protection for Electrical Equipment

100-ampere fusible disconnect box
2 60-ampere fusible disconnect boxes
30-ampere fusible disconnect box

6. Coils

Dimensions

Length	27 ins.
Height	12 ins.
Width	20 ins.

Header Diameter

O. D.	$2\frac{1}{2}$ ins.

Rows of tubes

Deep	17
High	8

7. Compressor

Frick Company
Model No. FW-500FH
High Temperature Range, Freon-12

Condenser Cooling Medium	Water
Size of Motor - H. P.	5
Compressor Speed - R. P. M.	435
Displacement - Cu. Ft./Hr.	1125
No. of Compressor Cylinders	3
Bore - ins.	$3\frac{1}{4}$
Stroke - ins.	3
Flywheel - Outside Diameter - ins.	15.7

Pulley - Outside Diameter - ins.	4.2
Motor Shaft Diameter - ins.	1 1/8
No. of V Belts	4
Part Number	43702
Outside Circumference - ins.	65
Refrigerant Charge - lbs.	3
Receiver Pump-down Capacity - lbs.	66
Receiver Liquid Outlet Valve - ins.	7/8 O.D.
Compressor Suction Line Valve - ins.	1 3/8 O.D.
Oil Charge in Compressor - pts.	8

8. Fans (2)

The Trane Company
Single Width, Single Inlet Fans
Type FC
Model No. 12

Circumference - ft.	3.14
Wheel Diameter - ins.	12
Outlet Area - sq. ft.	0.785

9. Motor and Fan V Belts

Gates Truflex
Model No. 2580

10. Performance Measurement

4 Wet and Dry Bulb Mercury Thermometers
Taylor
Range 25- 120 degrees F at 2 degree intervals
6 Mercury Thermometers
Southern Scientic Company
Range 0 - 220 degrees F at 2 degree intervals
2 Pressure Gages
Frigidaire
Freon-12 Gage
Range
0 - 30 in. Hg. at 1 in. intervals
0 - 60 psi at 5 psi intervals
Pressure Gage
The Ashcroft Company
New York, N. Y.
Range 0 - 160 psi at 5 psi intervals
Pressure Gage
Marsh Company
Freon-12 Gage
Range
0 - 30 in. Hg. at 10 in. intervals
0 - 300 psi at 5 psi intervals

Pressure Gage
  Lonergan
  Philadelphia, Pennsylvania
  Range 0 - 200 psi at 5 psi intervals
Pressure Gage
  Frigidaire
  Range 0 - 300 psi at 5 psi intervals
Stop Watch
  A. R. and J. E. Meylan
Anemometer
  Keuffel and Esser Company
  New York, N. Y.
  Serial No. 5963

H. HEAT PUMP OPERATING PROCEDURE

The procedure of the starting, operation, and shut down of the heat pump on the heating cycle will be explained.

STARTING

1. Check the oil level sight glass on the compressor and be sure that the oil level is at least in the center of the glass. Lubricate the fan and motor bearings and drain the oil separator.

2. Open the necessary valves in the order listed, beginning at the receiver discharge.

   a. The receiver valve 15.
   b. The by-pass around the dehydrator, valve 21.
   c. Expansion valve 10 preceding the outside coil.
   d. Vapor valves 2 and 12 on the control board.
   e. Liquid valve 4 leaving the inside coil.
   f. The receiver valve 14.

Then check the by-pass valve 13 and all other valves to be sure that they are closed.

3. Remove the covers from the sight glasses.

4. The compressor and fan motors are started in the following order: the SPST meter switches are placed in the closed

position; the main switch is thrown in the "up" position to energize the switchboard; and the motors are individually started by closing the switches on their fusible disconnect boxes. Immediately valves 6 and 8 on the control board are opened and the system is in operation.

OPERATION

As the system goes into operation an overall picture should be acquired by the operators to determine if the system is operating properly. Notice the discharge pressure and do not let it exceed 150 psig. Also, do not let the suction pressure fall below 10 inches of mercury vacuum. These conditions - if they exist - can be corrected. The discharge pressure can be lowered by increasing the quantity of air flowing across the condenser and the suction pressure can be raised by opening the expansion valve further.

The oil level of the compressor should be checked frequently because of the carry over of oil by the refrigerant vapors. If the oil level cannot be seen in the sight glass, shut down the unit immediately. Usually there is sufficient foaming of the oil to clearly indicate the oil level.

No trouble has been encountered with motor overload or overheating although the power should be observed when the load on the unit is changing. Also, notice the amount of superheat the vapor has when leaving the evaporator. The vapor should have approximately 5 to 10 degrees of superheat to prevent a liquid slug from reaching the compressor. If the superheat is excessive decrease the quantity of air flow or increase the weight of refrigerant flow.

SHUT DOWN

1. Receiver valve 15 is closed.

2. The compressor by-pass valve 13 is cracked to prevent the suction pressure from falling below 10 inches mercury vacuum. This point cannot be stressed too greatly because at low suction pressures the oil from the crankcase is pumped up through the suction valve into the compressor.

3. When all the refrigerant has been pumped to the receiver, valve 14 is closed, sealing the refrigerant in the receiver.

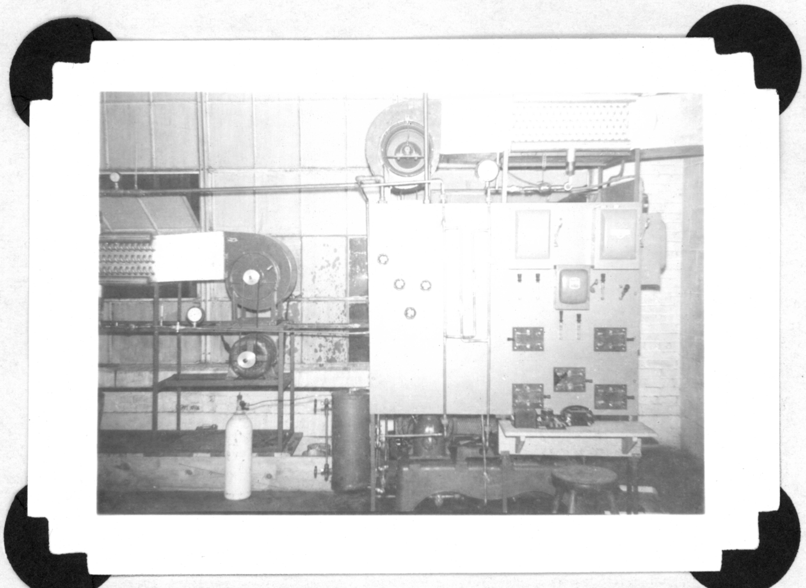
4. Disengage the fans and compressor at their switch boxes and then disconnect the switchboard from the transformer bank. Close all valves.

TESTING OF THE UNIT

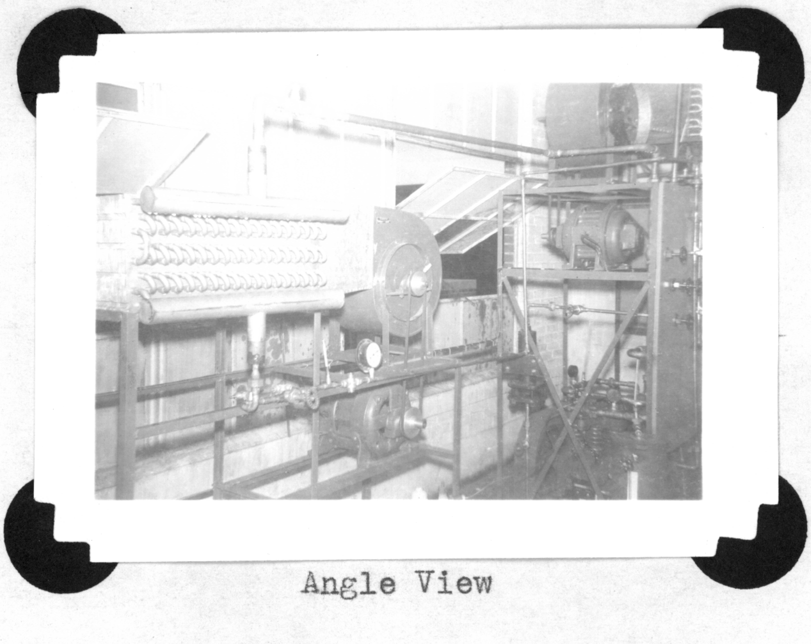
Two tests were run on the unit but the first test was discontinued because the control of the unit was impossible. Dampers were installed on the suction side of the fans and the second test was conducted. By adjusting the dampers complete control of the unit was obtained. The data and results of this test are presented later. The air velocity across the outside coil was held as near as possible to 500 fpm and the superheat was held at approximately 5 degrees. The volume of air flow across the inside coil was adjusted so that 100 degree F conditioned air was obtained. Readings were taken at 30 minute intervals for three hours after the system had reached equilibrium.

I. RESULTS

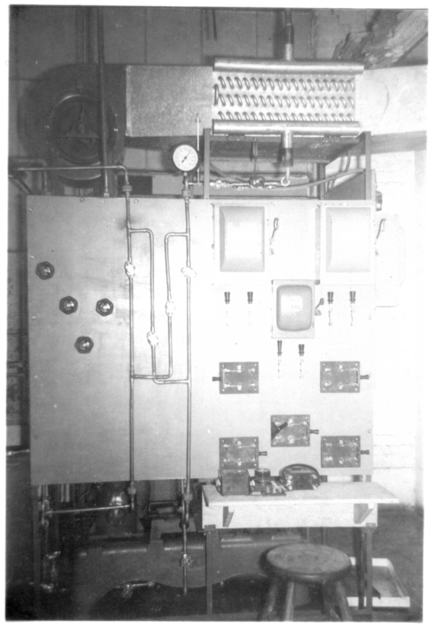
1. THE AIR-TO-AIR HEAT PUMP



Front View



Angle View



Inside Stand

2. THE PERFORMANCE OF THE AIR-TO-AIR HEAT PUMP

The performance of the air-to-air heat pump will be presented in three parts: operating data, method of calculation, and table of results.

Table II

Operating Data For Heat Pump Heating Cycle

Read.* No.	Conditioned Air			Room Air		Outside Air		Cold Air		
	Vel. Ft/Min	DBT F	WBT F	DBT F	WBT F	DBT F	WBT F	Vel. Ft/Min	DBT F	WBT F
1	910	101	69	74	60	69	56	550	51	50
2	884	99	68	73	60	69	55	535	46	44
3	894	99	68	73	60	68	55	545	46	44
4	765	100	68	72	60	66	54	548	45	42
5	878	98	68	70	58	66	54	551	44	42
6	871	100	69	73	60	66	54	545	44	42
7	814	99	69	71	59	65	53	547	44	41
Average	859	99.4	68.4	72.3	59.6	67.0	54.4	546	45.7	43.6

Read.* No.	Outside Fan		Inside Fan		Compressor	
	KW_1	KW_2	KW_1	KW_2	KW_1	KW_2
1	-0.11	0.50	0.53	0.01	0.68	0.31
2	-0.12	0.50	0.54	0.01	0.67	0.31
3	-0.12	0.51	0.54	0.01	0.67	0.30
4	-0.12	0.52	0.54	-0.02	0.67	0.29
5	-0.13	0.53	0.56	-0.02	0.665	0.28
6	-0.12	0.52	0.55	0.00	0.67	0.29
7	-0.12	0.52	0.54	0.00	0.66	0.29
Average	-0.12	0.514	0.543	0.00	0.669	0.296

*Readings every 30 min.

Barometric pressure: 13.6 psia

Table III

Operating Data For Heat Pump Heating Cycle

Read.* No.	Compressor Suction		Compressor Discharge		Entering Condenser		Leaving Condenser	
	F	PSIG	F	PSIG	F	PSIG	F	PSIG
1	44	27	169	140	155	136	108	139
2	44	28	168	140	154	134	106	135
3	35	28	161	137	148	134	106	136
4	43	26	169	140	154	137	108	141
5	44	26	170	135	155	133	105	134
6	44	26	170	142	155	142	108	138
7	43	25	170	135	155	134	105	135
Avg.	42.4	26.6	168	136.2	154	135.7	106.5	136.8

Read.* No.	Entering Evaporator		Leaving Evaporator		Superht. Leaving Evap. F	Press. Drop For Cond "H_2O
	F	PSIG	F	PSIG		
1	102	127	37	32.0	3	1.70
2	100	123	38	31.0	5	1.80
3	101	123	36	31.5	3	1.85
4	104	130	36	29.5	4	1.60
5	101	125	36	29.0	5	1.60
6	103	128	38	29.0	7	1.65
7	102	127	36	29.0	6	1.65
Avg.	101.8	126	36.7	31.5	4.72	1.69

*Readings every 30 min

Barometric pressure: 13.6 psia

METHOD OF CALCULATION

The following calculations were based on an average of the data obtained.

1. Input work to motors,

A. Heat to motors,

$$KW = (MR)(MF)$$

Where,
KW Power input (KW)
MR Meter reading, $KW_1 + KW_2$ (KW)
MF Meter Factor

Inside fan motor,

$$KW = 0.543 \times 2 = 1.086 \text{ KW}$$

Outside fan motor,

$$KW = 0.394 \times 2 = 0.788 \text{ KW}$$

Compressor motor

$$KW = 0.965 \times 5 = 4.825 \text{ KW}$$

B. Btu equivalent of total KW input,

$$W_I = (\Sigma KW)(CF)$$

Where,
W_I Total input work to motors (Btu/min)
ΣKW Total kilowatts to motors (KW)
CF Conversion factor to change KW to Btu/min.(56.9 Btu/kw min)

$$W_I = 6.699 \times 56.9$$

$$W_I = \underline{\underline{380}} \text{ Btu/min}$$

2. Heat lost by air in outside coil,

A. Density of dry air leaving outside coil,

$$\frac{W_a}{V_a} = \frac{(P_a)(144)}{(R)(T_{ca})}$$

Where,

$\frac{W_a}{V_a}$	Density of dry air leaving outside coil (lb dry air/cu ft)
P_a	Pressure of dry air leaving outside coil (psia)
R	Gas constant for dry air (ft-lb/lb degree Rankine)
T_{ca}	Dry bulb temperature of air leaving outside coil (degrees R)

$$\frac{W_a}{V_a} = \frac{13.476 \times 144}{53.3 \times 505.7}$$

$$\frac{W_a}{V_a} = \underline{\underline{0.072}} \text{ lb dry air/cu ft}$$

B. Volume of air leaving outside coil.

$$V_{ca} = (v_{ca})(A)$$

Where,

V_{ca}	Volume of air leaving outside coil (cu ft/min)
v_{ca}	Velocity of air leaving outside coil (ft/min)
A	Cross-sectional area of air duct (sq ft)

$$V_{ca} = 546 \times 1.67$$

$$V_{ca} = \underline{\underline{914}} \text{ cu ft/min}$$

2. (Continued)

C. Weight of dry air leaving outside coil,

$$w_a = \frac{(W_a)(V_{ca})}{(\overline{V_a})}$$

Where,

w_a	Weight of dry air leaving outside coil (lb dry air/min)
$\frac{W_a}{V_a}$	Density of dry air leaving outside coil (lb dry air/cu ft)
V_{ca}	Volume of air leaving outside coil (cu ft/min)

$w_a = 0.072 \times 914$

$w_a = \underline{\underline{65.8}}$ lb dry air/min

D. Enthalpy of air leaving outside coil and outside air.

$h_{ca} = 17.49$ Btu/lb dry air

$h_o = 23.47$ Btu/lb dry air

Where,

h_{ca}	Enthalpy of air leaving outside coil (Btu/lb dry air)
h_o	Enthalpy of outside air (Btu/lb dry air)

These enthalpies were obtained from a Carrier psychrometric chart and have been corrected for atmospheric pressure.

2. (Continued)

E. Heat lost by air in outside coil,

$$Q_A = w_a (h_o - h_{ca})$$

Where,

Q_A	Heat lost by air in outside coil (Btu/min)
w_a	Weight of dry air leaving outside coil (lb dry air/min)
h_{ca}	Enthalpy of air leaving outside coil (Btu/lb dry air)
h_o	Enthalpy of outside air (Btu/lb dry air)

$$Q_A = 65.8\ (23.47 - 17.49)$$

$$Q_A = \underline{\underline{394}} \text{ Btu/min}$$

3. Heat delivered to room,

A. Density of conditioned dry air.

$$\frac{W_1}{V_1} = \frac{(P_{a1})(144)}{(R)(T_{da})}$$

Where,

$\frac{W_1}{V_1}$	Density of dry conditioned air (lb dry air/cu ft)
P_{a1}	Pressure of dry conditioned air (psia)
R	Gas constant for dry air (ft-lb/lb degree Rankine)
T_{da}	Dry bulb temperature of conditioned air (degrees R)

$$\frac{W_1}{V_1} = \frac{13.421 \times 144}{53.3 \times 559.4}$$

$$\frac{W_1}{V_1} = \underline{\underline{0.065}} \text{ lb dry air/cu ft}$$

3. (Continued)

B. Volume of conditioned air,

$$V_{da} = (v_{da})(A)$$

Where,

V_{da}	Volume of conditioned air (cu ft/min)
v_{da}	Velocity of conditioned air (ft/min)
A	Cross-sectional area of air duct (sq ft)

$$V_{da} = 859 \times 1.67$$

$$V_{da} = \underline{\underline{1432}} \text{ cu ft/min}$$

C. Weight of dry conditioned air,

$$w_1 = \frac{(W_1)}{(V_1)}(V_{da})$$

Where,

w_1	Weight of dry conditioned air (lb dry air/min)
$\frac{W_1}{V_1}$	Density of dry conditioned air (lb dry air/cu ft)
V_{da}	Volume of conditioned air (cu ft/min)

$$w_1 = 0.065 \times 1432$$

$$w_1 = \underline{\underline{93.2}} \text{ lb/min}$$

D. Enthalpy of room and conditioned air,

$$h_r = 26.895 \text{ Btu/lb dry air}$$

$$h_{da} = 33.63 \text{ Btu/lb dry air}$$

Where,

h_r	Enthalpy of room air (Btu/lb dry air)
h_{da}	Enthalpy of conditioned air (Btu/lb dry air)

These enthalpies were obtained from a Carrier psychrometric chart and have been corrected for atmospheric pressure.

3. (Continued)

E. Heat gained by air and delivered to room,

$$Q_R = w_{da}(h_{da} - h_r)$$

Where,

Q_R	Heat gained by air and delivered to room (Btu/min)
w_{da}	Weight of dry conditioned air (lb dry air/min)
h_r	Enthalpy of room air (Btu/lb dry air)
h_{da}	Enthalpy of conditioned air (Btu/lb dry air)

$$Q_R = 93.2(33.63 - 26.895)$$

$$Q_R = \underline{\underline{628}} \text{ Btu/min}$$

4. Actual coefficient of performance,

$$COP' = \frac{Q_R}{W_I}$$

Where,

COP'	Actual coefficient of performance
Q_R	Heat gained by air and delivered to room (Btu/min)
W_I	Total input work to motors (Btu/min)

$$COP' = \frac{628}{380}$$

$$COP' = \underline{\underline{1.65}}$$

TABLE OF RESULTS

Conditioner coil:	
Air flow, cfm	1432
Entering air, degree F	72.3
Leaving air, degree F	99.4
Outdoor coil:	
Air flow, cfm	914
Entering air, degree F	67
Leaving air, degree F	45.7
Compressor:	
Suction pressure, psig	26.6
Evaporating temperature, degree F	26.
Discharge pressure, psig	136.2
Condensing temperature, degree F	109.5
Electric consumption (watts):	
Compressor	4825
Inside fan	1086
Outside fan	788
Capacity:	
Refrigerating effect, Btu/hr	23,600
Heating output, Btu/hr	37,700
Coefficient of performance	1.65

VI

DISCUSSION OF RESULTS

Construction of the air-to-air heat pump was not completed until springtime because of the difficulty in obtaining parts and the length of time involved in the construction itself. An eight-hour test was run on the unit at an outside temperature of 45 degrees F but due to the lack of dampers on the fans, control of the unit was impossible and the varying air deliveries prevented the system from attaining a state of equilibrium from which typical performance data could be recorded. Although the results of this first test were discarded the test demonstrated the effects of air delivery on the COP of the unit. These effects will be fully discussed later.

Dampers were placed on the fan inlets to control the outputs of the fans and then another test was run. The weather had turned warm by this time however so the test was conducted during the hours from 8:00 to 11:00 P.M. in order to take advantage of the lower temperatures which occur after sundown. Even then the outside temperature averaged 67.0 degrees F for the three-hour test and subsequent operation has been at even higher temperatures so this discussion will be based upon the performance of the heat pump at an average outside temperature of 67.0 degrees F. It was not the object of the thesis to test the unit over the entire range of temperatures encountered in this area. The object of the investigation was to assemble an air-to-air heat pump from available equipment and to run performance tests on this unit for its heating cycle in Blacksburg, Virginia.

The COP of the heat pump was calculated to be 1.65 for a heating effect of 628 Btu/min or 3.14 tons at an outside temperature of 67.0 degrees F. According to E. R. Ambrose,[2] a packaged heat pump built by the York Corporation, York, Pennsylvania, operated with a COP of 3.1 at this same outdoor temperature and rejected almost 50,000 Btu/hr or 834 Btu/min. The York heat pump is a dual unit which contains two entirely separate Freon-22 refrigerant circuits; this unit was described in the Review of Literature as having a 3 hp system which operates alone for temperatures above 30 degrees F and a 2 hp system which supplements the other whenever lower temperatures are present. Evidently only the 3 hp system was operating when the above data were recorded. According to the test results the York unit provided 32.8% more heating effect than the laboratory unit and at an 87.9% greater COP when both units operated at the same outdoor temperature. These differences are great and should therefore be explained in detail.

The York unit is a factory-built heat pump whose components have been carefully selected for optimum performance. The laboratory heat pump had to be built from odd pieces of available equipment, none of which could be selected for proper size or optimum performance. The fan motors are too large for their loads and cause the COP of the unit to be penalized accordingly. Fig. 41 on page 247 of Alternating-Current Circuits, Second Edition, Kerchner and Corcoran, shows that the power factor of a balanced load is 0.5 when one wattmeter reads zero in the two-wattmeter method. The inside fan-motor had an average power factor of 0.5 during the test, as is shown by the zero reading of one of its two power readings, and this was due primarily to the fact that the 5 hp motor was carrying a load of less than 1.5 hp. Fig. 175 on page 271 of Alternating-Current Machines, Second Edition, Puchstein and Lloyd, gives the characteristics of a 5-hp, 220-volt, 3-phase, 60-cycle, squirrel cage induction motor (similar to the motor on the inside fan) and demonstrates how both the power factor and the efficiency fall off at low loads. The efficiency also falls off at overloads. The characteristics of a particular motor depend upon its design and construction but all motors of a given classification will have similar characteristics since the National Electrical Manufacturers Association has set up minimum standards for general-purpose motors. From the preceding discussion it is apparent that the 5 hp motor on the inside fan and the 3 hp motor on the outside fan should both be replaced with 1-1/2 or 2 hp motors in order to operate the fan drives at maximum efficiency and reduce the total work to the heat pump for a given output. An improved power factor would also result from the use of smaller motors and would thereby reduce the current to the unit. It is doubtful if the power readings at low power factors are accurate enough for other than general performance tests because such low power factors introduce phase-angle errors into the instruments and no means are available for calibrating the wattmeter and current transformer at these power factors without going through a process which is too involved to be worthwhile. For this reason a watthour meter should be installed for future tests. Watthour meters can be made to have an accuracy of 99.5% under almost any kind of condition. The wattmeter is sufficiently accurate at large power factors.

The small cross-sectional areas of the coils are probably the chief causes of the small COP for the laboratory heat pump. The authors feel that articles on heat pumps have failed to emphasize the importance of coil construction on the performance of the unit. And the articles have neglected to indicate a method of selecting the proper coils for use on a heat pump. The York unit has two evaporator coils in series: the first coil - for the 3 hp system - is twelve rows deep and has a frontal area of six square feet while the second coil - for the 2 hp system - is four rows deep and has the same frontal area as the first coil. In relationship to the fan the two coils represent a single coil which is sixteen rows deep with a frontal area of six square feet. The outdoor fan pushed 2500 cfm of air through these coils at an average velocity of 417 fpm and a temperature drop of 8.9 degrees F. The evaporator coil on the laboratory unit is seventeen rows deep and has a frontal area of one and two-thirds square feet.

During the test the outdoor fan pushed 914 cfm of air through the evaporator coil with an average velocity of 546 fpm and a temperature drop of 21.3 degrees F. The York fan motor drew 712 watts and the laboratory fan motor drew 788 watts. It has already been stated that a smaller fan motor on the laboratory unit would be more efficient and draw less power than the large lightly-loaded motor so part of the power to this large motor can be charged to its light loading. But even if the motor were replaced with a smaller motor which drew 15% less power for the same air delivery the outdoor fan on the laboratory unit would still be drawing 670 watts or 94% of the power to the York fan while delivering only 36.5% as much air. The smaller input to the York fan is chiefly due to the low air velocity of 417 fpm as compared to 546 fpm for the laboratory unit. The large flow area (six square feet) of the York coil permitted the fan to push a large volume of air across the coil at a low velocity. Fig. 13.4 on page 286 of <u>Modern Air Conditioning, Heating, and Ventilating</u>, Second Edition, Carrier, Cherne, and Grant, shows that the air friction of a coil increases at a much faster rate than the air velocity when the coil is more than ten rows deep. It may be concluded that a large coil frontal area allows a large quantity of air to be shoved through the coil at low velocity and small static pressure loss, causing the fan to draw less power than if it had to pump the same quantity of air through a smaller flow area at a higher velocity. Actually the laboratory fan drew more power than the York fan when delivering only 36.5% as much air.

At first thought the larger air temperature differential of 21.3 degrees F for the laboratory evaporator coil seems more desirable than the differential of 8.9 degrees F for the York coil since the same amount of air would give up more heat to the first coil. But a large differential increases the work area of the heat pump and lowers the COP. Looking at Fig. 1 it can be seen that the work area is increased if the evaporation temperature t_4 is decreased. If the suction pressure drops while the discharge pressure is held constant the compression ratio of the compressor is increased, the volumetric efficiency decreases, and the output of the unit drops off. So for a given outside temperature t_o and a constant discharge pressure the compressor will do less work on each pound of refrigerant as t_4 approaches t_o and the COP will be improved. This situation can best be explained by using actual operating data. The temperature drop of the air moving across the evaporator coil of the laboratory unit was 21.3 degrees F so the air left the coil at a temperature of 45.7 degrees F for an outside temperature of 67.0 degrees F. At this time the evaporation pressure of the refrigerant was 30.1 psig or 43.7 psia and the corresponding saturation temperature was 30.7 degrees F or 15.0 degrees below the temperature of the leaving air. The refrigerant was colder than the outside air by 15 + 21.3 or 36.3 degrees F. When tested at an outdoor temperature of 35 degrees F the evaporating temperature of the York unit was 25 degrees F or only 10 degrees below the air temperature as compared to 36.3 degrees for the laboratory unit. This means that if the laboratory unit had held the temperature difference between the outside air and the evaporating refrigerant to 10 degrees instead of 36.3 degrees the suction pressure would have been 68 psia instead of 43.7 psia and the COP would have shown a significant increase. Thus it becomes advisable to hold the saturation temperature fairly close to the outside temperature in order to keep the suction pressure as high as possible. The low velocity of the air will provide sufficient time for the necessary heat transfer between the air and the coil even when the two temperatures are only 10 degrees apart. The air temperature across the York coil dropped from 35 to 27.7 degrees F for the evaporation temperature of 25 degrees F when its velocity was 410 fpm. Although its temperature differential was small the quantity of air was sufficiently large to provide the required amount of heat for the evaporation of the refrigerant.

The condenser should also have a large cross-sectional area to permit large fan capacities with low fan input but the temperature differential of the air across the coil is maintained large since it is usually desirable to heat the room air from 70 or 75 degrees to approximately 100 degrees F.

The large temperature differential of the air is secured by maintaining a large temperature differential between the room air and the condensing refrigerant. The conditioner coil on the York unit has two-thirds of the frontal area and three-fifths of the air flow of the evaporator coil.

The performance of the unit could be determined more accurately if a rotameter were installed to measure the weight of refrigerant flow per unit length of time. The calculations could then be based upon the conditions of the refrigerant at critical points throughout the system instead of upon the conditions of the air. The measurement of air conditions in a duct depends upon making traverses which yield an average temperature or velocity but this method is subject to greater error than is the method of measuring refrigerant temperatures and pressures.

The absence of such refinements as 4-way valves and automatic controls created the necessity of using a larger number of fittings in the construction of the heat pump. Pressure losses due to friction in the system are small however and the operation of the unit is more flexible than if the fittings had been held to a bare minimum.

The authors do not believe that the system has been operated under optimum conditions. They do believe that the COP of the unit can be improved by individually varying the components of the system until each is operating at its optimum point but that the COP will always be less than that of a commercial unit for the same outside temperature because of the inherent faults which have been discussed in this section. The unit should prove to be a valuable asset to the school since it will fulfill the many uses which were described in the Introduction as being the motivating reasons for its construction.

VII

CONCLUSIONS

A. The COP of the heat pump will always be less than that of a commercial unit for the same outside temperature because it was built from odd pieces of equipment which do not meet the requirements of heat pump operation.

B. The fan motors are too large and have to operate at low efficiencies. Smaller motors would provide more efficient performance and improve the COP.

C. Each coil should have a large cross-sectional area in order to have sufficient air flow at low velocities for maximum COP at rated loads.

D. The saturation temperature of the refrigerant in the evaporator should be held fairly close to the outside air temperature in order to keep the suction pressure as high as possible. This means that the temperature differential of the air across the coil will be small.

E. More accurate power measurement could be achieved by installing a watthour meter.

F. More accurate performance data could be obtained by installing a flowmeter and basing the calculations upon the conditions of the refrigerant.

G. The unit fulfills the objectives for which it was constructed.

VIII

SUMMARY

All available information on heat pumps was read and the articles pertaining directly to design, operation, and the overall performance of air-to-air heat pumps were included in the Review of Literature. The advantages and disadvantages of the basic types of systems and their heat sources were explained. A few of the outstanding units as well as small commercial units were discussed, but the data published on these units were very limited as to design and operating values. One of the major problems encountered with air-to-air heat pumps is the defrosting of the outside coil. Several methods were outlined but the best solution for defrosting the coil would depend upon the conditions pertaining to each unit. Several suggestions were outlined where research is needed for the proper utilization of air as a heat source and sink.

After several trial and error solutions the system was designed to give a conditioned air temperature of 100 degrees F for an outside temperature of 30 degrees F and a heat delivery of 436.3 Btu/min. The assumed room temperature was 75 degrees F which gave a 25 degree differential across the inside coil. With an 18 degree differential across the outside coil a refrigerating effect of 338.0 Btu/min could be picked up from the outside air. These values appeared to give the best operating conditions. The tubing was sized accordingly and considerations were given to the fact that the vapor lines would be used for both cycles.

The stands were constructed and the component parts of the system were mounted on their respected stands. The piping, duct, and electrical work were completed. The design and construction of each of these phases was done with emphasis on clarity, simplicity, and ease in operation and maintenance.

The valves numbered 1 through 12 are the control valves. The system is operated on the heating cycle with the even-numbered valves opened and the odd-numbered valves closed. All the control valves, except the expansion valves and their by-passes, are located on the control board. A dehydrator and oil separator were installed in the piping circuits.

A two-wattmeter method of power measurement was installed on the unit. This method enables the operators to determine the power supplied to each motor and instantaneous power when the load on the unit is changing. Each of the motors is protected by fusible disconnect boxes.

The procedure for starting, operating, and shut down was explained with emphasis on the factors that should be watched for safe and satisfactory operation.

The calculations were based upon the conditions of the air, the heat picked up from the outside air, the work supplied to the motors, and the heat gained by the room air.

IX

RECOMMENDATIONS

In order to prolong the life of the equipment, simplify the operation of the unit, and secure additional operating data the following recommendations are made.

(1) A rotameter should be acquired and installed for the purpose of measuring the flow of refrigerant.

(2) The inexpensive valves on the unit should be replaced with standard refrigeration valves whenever such items become available.

(3) A calibrated watthour meter should be used to measure the total energy input to the unit during future tests. This meter (accurate within 0.5%) will provide greater accuracy than the wattmeter and simplify the power measurement over a period of time.

(4) The wattmeter should be retained for observing the instantaneous variation in motor input when the load changes.

(5) The fan motors should be replaced with 1-1/2 or 2 hp motors in order to increase the efficiency of these drives and raise the COP of the heat pump.

(6) Drain pans should be placed beneath the coils.

(7) The COP of the unit for the range of temperatures encountered in the Blacksburg area should be found by running tests throughout the cold months of the year.

(8) The optimum COP for various outside temperatures should be found by varying the system components individually until the maximum COP is obtained.

X

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XI

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XII

VITA

Mr. Wilson E. Ringler was born in Portsmouth, Virginia, on May 7, 1925. Upon his graduation from Woodrow Wilson High School in 1942 Mr. Ringler worked on various construction jobs for a year before entering the Navy where he served as an aviation radioman until honorably discharged in 1946. He entered Virginia Polytechnic Institute in the fall of 1946, received his degree of Bachelor of Science in Mechanical Engineering in June of 1950, and remained in school to do graduate work in his department.

Mr. Evander H. Rogers was born in Kingstree, South Carolina, November 25, 1922. He was graduated from Woodrow Wilson High School of Portsmouth, Virginia, in June of 1940. He had completed approximately two thirds of a machinist's apprenticeship at the Norfolk Naval Shipyard before entering the Army Air Force in October, 1943. There he served for two and one half years as a radio and radar technician and was honorably discharged from the Army in March of 1946. He enrolled in the St. Helena Extension of William and Mary College that fall and transferred to Virginia Polytechnic Institute at the beginning of his sophomore year. He completed the requirements for a Bachelor of Science Degree in Mechanical Engineering in June of 1950 and remained after graduation to do graduate work in the same department.

XIII

APPENDIX

TABLE OF SYMBOLS

A	Cross-sectional area of air duct (sq ft).
c	Specific heat of air at constant pressure (Btu/lb-degree F).
COP	Theoretical coefficient of performance for heat pump.
COP'	Actual coefficient of performance for heat pump.
d_1	Specific volume of vapor entering compressor (cu ft/lb).
e_v	Volumetric efficiency of compressor.
h_1	Enthalpy of superheated vapor entering compressor (Btu/lb).
h_2	Enthalpy of superheated vapor leaving compressor (Btu/lb).
h_3	Enthalpy of liquid refrigerant in receiver (Btu/lb).
h_4	Enthalpy of wet mixture leaving expansion valve (Btu/lb).
p	Atmospheric pressure (lb/sq ft).
p_1	Evaporator pressure (psia).
p_3	Condenser pressure (psia).
Q_A	Heat added to refrigerant in evaporator (Btu/min).
Q_{ca}	Heat given up by air to evaporator (Btu/min).
Q_R	Heat rejected by refrigerant in condenser (Btu/min).
$Q_{R'}$	Q_R in tons.
r	Ratio of compression for compressor.
R	Gas constant for air (ft-lb/lb-degree Rankine).
t_1	Temperature of vapor entering compressor (degrees F).
t_3	Saturation temperature of condensing refrigerant (degrees F).
t_{ca}	Dry bulb temperature of cold air leaving evaporator (degrees F).
$t_{ca'}$	Wet bulb temperature of cold air leaving evaporator (degrees F).
T_{ca}	Dry bulb temperature of cold air leaving evaporator (degrees R).
t_{da}	Dry bulb temperature of conditioned air (degrees F).
$t_{da'}$	Wet bulb temperature of conditioned air (degrees F).
T_{da}	Dry bulb temperature of conditioned air (degrees R).
t_o	Dry bulb temperature of atmospheric air (degrees F).
$t_{o'}$	Wet bulb temperature of atmospheric air (degrees F).
t_r	Dry bulb temperature of room air (degrees F).
$t_{r'}$	Wet bulb temperature of room air (degrees F).
v_{ca}	Velocity of cold air leaving evaporator (fpm).
V_{ca}	Volume of cold air leaving evaporator (cu ft/min).
V_d	Compressor displacement (cu ft/min).
v_{da}	Velocity of conditioned air (fpm).
V_{da}	Volume of conditioned air (cu ft/min).
w	Refrigerant flow (lb/min).
W	Work of compression (Btu/min).
W_I	Input to motors (kw or Btu/min).
w_{ca}	Weight of cold air leaving evaporator (lb/min).
w_{da}	Weight of conditioned air (lb/min).