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OPERATING CHARACTERISTICS
OF A DIESEL DRIVEN HEAT PUMP

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A

THESIS

BY

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HISTORICAL REVIEW

INTRODUCTION

The Heat Pump, as a device to provide low grade heat for conditioning purposes, is generally considered to have been originated by Lord Kelvin and its description was contained in papers presented by him in 1852 (75). The original papers described a system capable of transferring heat from a cold temperature to a higher one. Since air was to be the refrigerant, large volumes were required and his cylinder volume was to be 15.63 cubic feet. Power to drive the unit was to be provided by a steam engine and thus was of little practical use. Although this Kelvin cycle was to become the basis of many refrigerating systems, the heat pump itself attracted little attention until 1930, when Haldane reported considerable success with electrically driven units (62). Other installations began to appear quickly and great interest was stimulated in this surprising device, that was capable of heating efficiencies of from 200 to 500 percent.

The possibility of an all-electric dream house awakened America and installations began appearing all over the United States. The work of Marvin Smith and Robert Webber led to the production of the Marvair and the Terra Therm packaged units which are giving excellent results in the field (118, 119, 143).

Although attention was mainly focussed on its air conditioning applications, the industrial possibilities of the heat pump were soon realized and it was pressed into service in evaporation and as an auxiliary heat supply, particularly in fuel-short areas.

A complete bibliography of heat pump development up to 1947 has recently been published by the Southern Research Institute. This paper includes over two hundred references to articles published on the subject (14).

FUNDAMENTAL PRINCIPLES

I THERMODYNAMICS OF CYCLE

The inherent thermodynamic characteristics of the heat pump have attracted the attention of scientists since the time of Kelvin. The apparent simplicity of this device, capable of such unusual heating efficiencies, has long made it a source of considerable conjecture. The limitations of a heat engine, as clearly indicated by simple application of the laws of thermodynamics, is the very principle that leads to the prediction of these heating efficiencies of 200% to 500%.

Basically the heat pump is a device capable of transferring heat from a low temperature "source" to a high temperature "sink". As such, it is identical in performance with a refrigerator and is distinguishable from it only in that, in a refrigerator, attention is focussed on the refrigerating effect, i.e., the heat absorbed from the low temperature source; whereas, with the heat pump, consideration is given to the heating effect, i.e., the heat rejected to the high temperature sink.

The unit consists, essentially, of an evaporator, absorbing heat at a low temperature; a condenser, rejecting heat to the high temperature sink; a compressor, transferring the refrigerant from the low pressure (low temperature) side to the high pressure (high temperature) side; and an orifice, allowing the refrigerant to expand from the high to the low pressure side.

Figure 1 represents the basic cycle which operates by the

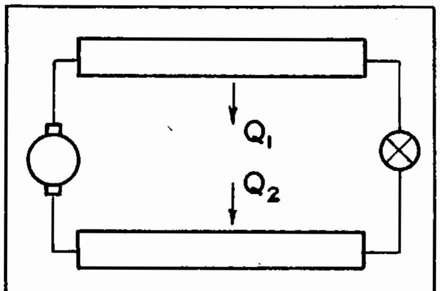


FIGURE I

evaporation, at constant low pressure, of the heat carrying medium, or refrigerant, and compressing the saturated vapour to a higher pressure. The vapour is then condensed at constant pressure giving up its heat and is returned to the evaporator through an expansion valve, which is merely a controllable orifice across which is maintained the difference in pressure between the evaporator and the condenser.

A quantity of heat, Q_2, is absorbed in the evaporator and is rejected, along with the work of compression (W), in the condenser. The heat rejected in the condenser may be termed Q_1 and constitutes the heating effect. The efficiency of the system is measured by means of the coefficient of performance or the COP, which is defined by

$$\text{COP} = \frac{\text{Total heat rejected}}{\text{Work done}} = \frac{Q_1}{W}$$

II THE REVERSED CARNOT CYCLE

The ideal heat engine cycle was first described by Carnot in 1824 (40), and although his mathematical analysis was later proved invalid, the principles he developed were shown to be sound. He established that the maximum thermodynamic efficiency of any heat engine operating between two temperatures was a function of those two temperatures alone and did not depend on the device employed nor on the nature of the refrigerant:

$$\text{Efficiency} = \frac{T_1 - T_2}{T_1}$$

where T_1 and T_2 are the absolute temperatures of the heat source and heat sink, respectively.

The efficiency is the work produced per unit of heat supplied and thus corresponds to

$$\text{Efficiency} = \frac{W}{Q_1} = \frac{T_1 - T_2}{T_1}$$

where W is the work done and Q_1 is the heat absorbed at T_1.

The Carnot analysis became the basis for the thermodynamic study of cyclic processes in general and established the importance of the term Q/T (the heat absorbed, or rejected, divided by the absolute temperature at which the transfer occurs) in the science of thermodynamics. Thus, for the cyclic process proposed by Carnot and consisting of two isothermal and two adiabatic changes, the following relationship exists:

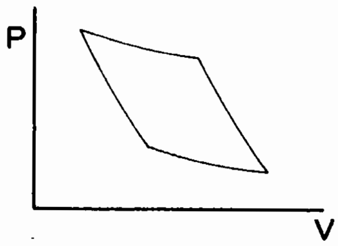


FIGURE 2

$$\frac{Q_1}{T_1} = -\frac{Q_2}{T_2}$$

The negative sign arises from the introduction of the accepted thermodynamic sign convention that heat absorbed will be positive, heat rejected negative, while work done on the system is negative and that done by the system positive.

This relationship when combined with a second

$$Q_1 + Q_2 = W$$

readily leads to the expression of efficiency as proposed by Carnot.

A study of the Carnot efficiency equation immediately reveals the attractiveness of a reversed Carnot cycle, that is the transfer of heat from a low temperature to a higher one by the expenditure of mechanical energy. Defining the efficiency of such a system as the heating effect produced, divided by the work required to produce it, an expression for this efficiency may be derived by the application of these two simple expressions and becomes:

$$\frac{Q_1}{W} = \frac{T_1}{T_1 - T_2}$$

This is the exact reciprocal of the Carnot efficiency equation and it may be seen that the smaller the difference $T_1 - T_2$ the greater the efficiency. Efficiencies of several hundred percent are easily possible. Since efficiency generally applies to a comparison of actual to ideal operation and usually implies a factor less than one, the term "coefficient of performance" was applied to the expression Q_1/W to describe this amplification factor. Although some authors have taken exception to this term for various reasons, their criticism seems hardly justified and it has come into general use. The coefficient of performance adequately describes the factor, whether greater or less than one, and eliminates the confusion of quoting efficiencies of several hundred percent which, indeed, become quite meaningless at this level.

Although the importance of the reversed Carnot cycle was evident to Kelvin, the practical exploitation of this knowledge was not possible at that time. It was not until 1930 that the use of this machine to produce heat attracted new interest. Haldane reported net gains in heat delivery over the coal used to produce the electricity he used, and the era of the heat pump was born (62).

III THE RANKINE CYCLE

Because of the difficulty of building a heat engine operating on the Carnot cycle, the Rankine cycle was proposed as a better means of evaluating the performance of the heat engine. This is the cycle after which steam plants and vapour compression refrigeration systems are

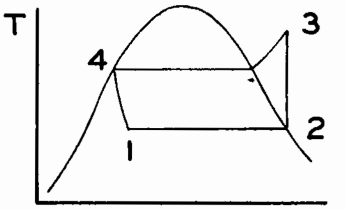


FIGURE 3

patterned. The Rankine refrigeration cycle is represented on a temperature - entropy diagram in Figure 3 and consists of four steps:

(a) Absorption of heat at constant pressure (1 - 2);

(b) Isentropic compression to higher pressure (2 - 3);

(c) Rejection of heat at constant pressure (3 - 4);

(d) Isenthalpic expansion from high to low pressure (4 - 1).

Defining the enthalpy of the refrigerant as H_1, H_2, H_3, H_4 where the subscripts correspond to the points in Figure 3, the heat, Q_2, absorbed at the low temperature, T_2, is given by the enthalpy difference between points 1 and 2, and the heat rejected at high temperature (Q_1) is similarly given by the enthalpy difference between points 3 and 4. Thus

$$Q_2 = \Delta H(1 - 2) = (H_2 - H_1)$$

$$Q_1 = \Delta H(3 - 4) = (H_4 - H_3)$$

$$-W = \Delta H(2 - 3) = (H_3 - H_2)$$

$$COP = \frac{Q_1}{W} = \frac{H_4 - H_3}{H_2 - H_3} = \frac{H_3 - H_4}{H_3 - H_2}$$

The Carnot cycle represents an unattainable ideal system since no machine has yet been devised to operate on this cycle. The Rankine cycle, on the other hand, is the actual method of operation of the vapour compression cycle. It is still an ideal cycle since isentropic compression and isobaric processes are assumed. However, comparison of the actual results with the ideal Rankine cycle represents a measure of the efficiency of the machine. The Carnot efficiency remains as a means of comparison of attainable cycles on the basis of percent of Carnot efficiency and provides an excellent method of assessing the value of any heat engine cycle.

IV THE ACTUAL CYCLE

The Rankine cycle represents ideal operating conditions and can only be approached in practice. Deviations from ideal behaviour will occur and these will all be accompanied by loss of efficiency. The most important deviation from ideal operation occurs in the compression step (2 - 3) in Figure 3 which, though represented as isentropic in the ideal cycle, takes place with an increase in entropy, thus requiring more work to compress the gas from the low to the high pressure. In addition, pressure drops occur in the lines and in the heat exchangers. This means, of course, that a greater pressure differential must be maintained across the compressor than is represented by the evaporating and condensing pressures and thus more work is required to maintain this pressure difference. Considerable attention is being devoted to this problem, and of the various designs which have been proposed for the reduction of frictional resistances, special mention should be made of the longtitudinal fin heat exchanger which is attracting attention elsewhere in industry because of its low pressure drop (31, 36).

Subcooling of the liquid in the condenser results in the reduction of the heat content of the refrigerant entering the evaporator, thus increasing the refrigerating effect and, consequently, the COP of the cycle. Superheating the vapour from the evaporator is necessary in non-flooded evaporators to ensure that the vapour returning to the suction side of the compressor will not be wet. The refrigerating effect is increased but at the expense of additional work. A net loss in COP is the general result. The two effects, subcooling of the liquid and superheating of the vapour, are sometimes combined in a heat interchanger, in which the vapour from the evaporator is used to cool the liquid from the condenser. Although theoretically the increased refrigerating capacity is obtained at a lower cooling capacity to work ratio, in the actual cycle sufficient gains exist to make heat interchangers valuable at certain levels of operation.

In addition to preventing the return of liquid to the compressor, the maintenance of an adequate degree of superheat in the vapour from the evaporator minimizes the possibility of frosting of the suction line. Frosting represents an additional heat pick-up which, since it is not accompanied by a corresponding cooling effect in the refrigerated space, constitutes a loss in the efficiency of operation. This superheat, if obtained in the evaporator alone, requires a large evaporator surface because of the much lower transfer coefficients to vapours. The overall result is a lower evaporating temperature and loss in COP. On the other hand, the reduction of flash gas to the evaporator brought about by subcooling the liquid from the condenser, combined with operation of the evaporator with only one or two degrees of superheat at the outlet, gives rise to a larger cooling effect and a higher evaporator pressure. The superheater can then raise the temperature of the vapour to a temperature sufficient to prevent frosting while at the same time reducing the extraneous heat pick-up. The end result is an increase in the actual COP, increase in compressor capacity, and a reduction in the size of the evaporator (55,56). The advantage of the interchanger is accentuated at lower evaporator temperatures, but disappears above 20°F.

THE HEAT PUMP

Since the pioneering work of Haldane the heat pump has produced considerable interest in the United States and elsewhere. Volumes have been written on the possibilities of this device and much experimental work has been done. Many domestic and commercial installations exist in the United States, while commercial and industrial installations have become extensive in Switzerland.

I TYPES OF INSTALLATIONS

There are four basic heat pump designs, depending on the nature of the heat source from which the heat is extracted, and of the sink to which the heat is rejected (128):

(i) Air to Air;
(ii) Air to Water;
(iii) Water to Air;
(iv) Water to Water.

(i) Air to Air

This system derives its name from the use of air as both source and sink; that is, the heat is withdrawn from outside air and delivered up to air within the building. This heated air is then circulated throughout the conditioned space. There are many advantages to the use of air as a heat sink. The warm air from the condenser is readily treated for complete conditioning (i.e., filtered, washed, purified, deodorized and humidified), before delivery to the conditioned space and the quantity of outside air added is readily controlled. Localized heating effects are reduced and more uniform temperatures maintained by the regular distribution of treated air throughout the conditioned space. The system is easily controlled, quick in response and flexible. Unfortunately the use of outdoor air as a heat source presents many problems and its use is only feasible in mild climates.

When the utilization of outdoor air as a heat source is possible, certain advantages immediately result: the evaporator is free from scale formation and is readily built in a packaged unit, affording greater simplicity of construction (42). However, owing to the low heat transfer coefficient from air, the evaporator requires a very large surface and considerable energy must be expended in moving the air over the evaporator coils. But the greatest objection to air as a heat source is the fluctuation of the coefficient of performance. This fluctuation is an adverse one, the COP being least when the heating load is the greatest. This results in a low performance factor (COP averaged over a period of time) (100). This unfortunate characteristic makes it necessary to employ a larger unit in order to satisfy the peak load requirements. Installation of such a unit costs in the neighbourhood of $600 per ton of cooling capacity (117). In climates where the outside temperature seldom drops below 40° - 45°F, the unit can be economically used and, under these conditions, auxiliary heating is generally supplied for those days when the source temperature falls below this value (118, 124).

(ii) Air to Water

In this design, air is used as the heat source as described above, but the heat is given up to water which is then circulated in a conventional hot water heater system, or is used to heat the conditioned air. The advantage of such a system over the one previously described is that a storage tank may be used. Water is heated when outside air temperatures are favourable and this water circulated when the source temperature is low, thus supplementing the heat pump during periods of high load requirements. This means that a smaller unit may be installed, resulting in saving in initial cost and in running expenses (78, 122).

(iii) Water to Air

Water undoubtedly provides the most satisfactory heat source for heat pump installations. Although still accompanied by many problems, the advantages it offers make it very attractive. Its high heat transfer coefficient and large heat capacity make smaller evaporators possible.

Unfortunately, water is not always available as a heat source and often, though available, it is not usable because of the impurities that it carries.

When using water as a heat source, three choices are available:

(1) Municipal Water;
(2) River or Lake Water;
(3) Underground Water.

City water is, in practically all communities, too expensive for use as a heat source. The demand on the city water plant would be exceedingly large and, although it would be satisfactory from other considerations, it cannot be considered too seriously as a future heat source. In Portland, Oregon, permission was once obtained to return the water to the city mains (60). Special attention to the evaporator lines was, of course, necessary to prevent any contamination of the city water supply. Although this method is permissible in isolated cases, it can hardly be visualized as a common practice with extensive use of heat pump installations.

River or lake water, of course, is limited to communities bordering on such water supplies, and even more particularly to buildings within close proximity to the body of water. Pumping costs must be kept within reasonable limits, else the economic advantage of the whole installation may be lost.

Ground water, when available in sufficient quantities, provides an excellent source of heat for heat pump installations. Ground water temperatures vary roughly from 72°F. in the Southern United States to 37°F. in Southern Canada.

The temperature requirements of ground water are generally considered to be between 40°F. to 60°F. (15, 49, 60, 116, 135, 153). Ground water utilization usually consists of constructing two wells to the water table, with a reasonable distance separating them. Water is then pumped from one well to the evaporator, the waste being returned to the second well. The water in returning through the ground is raised to its original temperature level and is recycled through the system. Unfortunately, if the water requirements are too large, the source temperature will be lowered and may become too low for economic use (129). The cost of sinking the wells is a major item but the advantage of a constant temperature source, added to the advantages of a water source, may justify this expense.

A strong argument in favour of such a system is found in the good results obtained with such a source in the Equitable Building installation in Portland, Oregon. This installation has a capacity of 450 tons and uses two wells as heat source at 64.5°F. and 62.5°F. The system is fully automatic with clockostats partially shutting down overnight. The system provides heating or cooling, or both, as required.

Electrical energy is available at 7 mills per kw-hr and running costs are approximately 46 cents per million B.t.u. Steam costs are about 22% higher, and since the publication of these figures they have increased still further (15, 19, 23, 60, 86, 87). The coefficient of performance of this installation is 3.5 while installation costs were 29 cents per cubic foot, or about $1,250 per rated ton.

(iv) Water to Water

In this type of unit water is again used as a heat source but the heat pump delivers its heat to a closed system in which water is recirculated and used to heat the conditioning air. The advantages and disadvantages of a water source as previously described are still present but the advantage of a heat storage medium can be incorporated.

Although the performance factor (100) in a water source heat pump is generally better than the air source unit (42) it still runs for a small percentage of the time (20% - 30%) (103), since it is designed to supply maximum load requirements which may occur only occasionally. The possibility of storing heat in a large tank permits the use of a smaller unit, running a greater portion of the time. The peak load requirements in excess of actual unit capacity are supplied from the storage tank which is charged during the low demand periods.

II GROUND AS A HEAT SOURCE

Earth provides an unlimited source of heat and in almost all climates this heat is available at the desired temperature not far below the surface. In cold climates operational temperatures may be obtained at greater depth but are not inaccessible. Unfortunately, the transfer of the heat from the soil to the refrigerant presents a problem that, in some districts, appears unsurmountable.

The heat transfer coefficient from soil to a buried coil is very low, with values of 0.5 to 1 being reported by various authors (73, 97). This is of the order that might be expected from an air coil and if the temperature differential across the evaporator is to be kept small, seventy to one hundred square feet of coil would be required for every thousand B.t.u. of heat transferred. If this area is to be supplied by buried coils a very large installation is necessary. Large initial expense is involved and large pressure drops can be expected. A further effect is the dropping off of the coil capacity with time.

Basically, the satisfactory use of ground coils as a heat source depends on the presence of sufficient moisture to provide adequate heat transfer between earth and coil. When the evaporator is used as a heat source, operating on the heating cycle, the temperature gradient set up causes moisture migration toward the coil. On the cooling cycle the effect is opposite. Thus a sandy soil, which allows moisture migration, will give satisfactory results on the heating cycle, while a soil rich in clay, which will inhibit moisture migration, will be more favourable on the cooling cycle (61, 139). The moisture migration also decreases with time (139) and there is undoubtedly a connection between this effect and a similar decrease in the coil capacity.

Research on buried coil design is resulting in basic heat transfer theories and the advancement of mathematical analyses capable of predicting operation and its change over a period of years (26, 34, 44, 61, 65, 66, 76, 107, 139). Other work is being conducted to investigate the effect of hygroscopic fills in increasing the heat transfer and in reducing its decrease with time (99). The lack of a standard earth classification is hampering this work.

Manufactured units using ground coils are in use and are reported to be giving satisfactory results. Their cost is about $1,750 for a 3 hp. unit, plus installation costs (approximately $300). They give performance factors of around 3.5 (143).

III COEFFICIENT OF PERFORMANCE

The coefficient of performance of a heat pump is given by

$$\text{COP} = \frac{\text{Heating effect}}{\text{Work}}$$

that is, the heat supplied to the conditioned space divided by the work supplied to the compressor. The ideal or Carnot COP is found to be a function of the absolute temperatures involved and is given by

$$\text{COP} = \frac{T_1}{T_1 - T_2}$$

where T_2 and T_1 are the source and sink temperatures, respectively. From this relation, it can be seen that the lower the temperature difference between the two temperatures involved the greater is the coefficient of performance.

The COP as defined above actually represents that of the heat pump itself, but certain auxiliary equipment (fans, pumps, etc.) is required for the delivery of the heat to the conditioned space. These auxiliaries consume energy while producing no heating effect and so reduce the Overall Performance Coefficient of the system considerably. In future discussion this term will be used to apply to the ratio of the heating effect produced to the total energy (including auxiliaries) consumed.

The operating conditions vary considerably over any heating season, especially in air source units, and Parkerson's definition of a seasonal performance factor, as mentioned previously on page 11, will be used to apply to the weighted average COP, i.e., Seasonal Performance Factor = $\frac{\text{Total Heating Effect}}{\text{Total Energy Consumed}}$

This actually represents the total heat delivered during the heating season for each unit of energy consumed or paid for.

To make the heat pump a competitive form of domestic heating, the coefficient of performance and the performance factor must be made as high as possible. The first and most obvious means of accomplishing this is to reduce the temperature differential between which the unit operates. Originally, heat pump designs called for a condensing temperature of 150°F. and above, to maintain sufficient temperature potential in the conditioner coil. Modern designs are tending toward lower temperature levels, and 120°F. to 130°F. and lower are being used satisfactorily (48).

The low side temperature is governed by the temperature of the available heat source and here the problem becomes one of heat source availability. Forty to fifty degrees F. is desirable but not easily obtained and becomes more difficult in colder climates.

Increase in the COP can be obtained within the unit itself by the use of a suction line heat exchanger or heat interchanger. This heat exchanger transfers heat from the hot liquid leaving the condenser to the cold vapour from the evaporator. The evaporator efficiency is increased as is also the coefficient of performance. Although there are limits to its effectiveness, the use of a heat interchanger is becoming general, particularly where lower evaporator temperatures are involved (36, 55, 56).

Reduction of the outside power requirements to a minimum and careful attention to pressure drops within the system will all tend to increase the overall performance coefficient.

IV HEAT PUMP EQUIPMENT

The heat pump proper, like the refrigerator, consists of four principal components: a compressor, condenser, evaporator and a pressure reducing valve. These units vary widely in design and application and considerable literature has been devoted to each. Owing to space limitations, only their most fundamental aspects will be considered.

(i) Compressors

The compressor is the heart of the heat pump. There are three basic types of compressors, each with its specific area of application, depending on the size of the installation: rotary compressors (0-1 hp.), reciprocating compressors (0-200 hp.) and centrifugal compressors (150 hp. and above) (82). It can therefore be concluded that, except for very large industrial installations, reciprocating compressors are standard equipment in heat pump installations. Many variations within this general classification exist, depending on number and arrangement of cylinders, splash or forced feed lubrication, design and arrangement of valves, cooling and other features. The combination of these many factors depends sometimes on specific requirements but is often largely dependent on the preference of the manufacturer.

Compressors are generally rated on a volume basis, based on the volume handled at inlet conditions. The volume required will be given by

$$V = \frac{v_2 \quad Q'_2}{(H_2 - H_1)\, e}$$

where V = volumetric capacity of compressor, in c.f.m.;

v_2 = specific volume of refrigerant at suction conditions, in (cu. ft.)/(lb.);

Q'_2 = evaporator heat load, in (B.t.u.)/(min.);

H_2 = enthalpy of refrigerant leaving evaporator, in (B.t.u.)/(lb.);

H_1 = enthalpy of refrigerant entering evaporator, in (B.t.u.)/(lb.);

e = volumetric efficiency, no units.

The volumetric efficiency, e, is given by

$$e = 1 + c \left[1 - \left(\frac{P_1}{P_2} \right)^{\frac{1}{n}} \right]$$

where c = clearance volume, no units;

P_1 = condensing pressure;

P_2 = evaporating pressure;

n = constant lying between 1 for isothermal compression and $C_p / C_v = \gamma$, for adiabatic compression.

The efficiency of a compressor falls off with increasing compression ratio and its capacity will increase with increasing suction pressure. The efficiency decreases with speed. Various mechanical and thermodynamic inefficiencies exist in a compressor due to superheating of entering gas by hot cylinders, expansion through valve ports, etc., and these must be carefully considered in compressor design.

(ii) Condensers

The choice of a condenser for a heat pump will, of course, depend on the type of heating medium involved. The theory of heat transfer and design of condensers is covered in many excellent texts and the only limitation in heat pump condensers is dictated by the restriction that the temperature differential must be kept at a minimum. Increase of the temperature differential across the condenser to economize on surface area must arise at the expense of higher head pressures and a loss in COP. For air circulation systems finned tubes are commonly used, while for water heating systems shell-and-tube or double-pipe arrangements may be employed. An economic balance obviously exists in all cases and specific choice would depend on local conditions. Considerable data on heat transfer coefficients for various refrigerants are available (19, 74, 108, 109, 110).

(iii) Evaporators

The function of the evaporator is to absorb heat from the heat source. Two general methods of operation may be recognized: an evaporator may be operated in a flooded or non-flooded state. Smaller units employ the non-flooded evaporator almost exclusively; this type operates only partly full of refrigerant, the remainder of the evaporator being used to superheat the refrigerant. In a flooded type evaporator, the unit is essentially full of refrigerant and is followed by a separator to prevent liquid entrainment. The vapour leaving the evaporator is then saturated. Since heat transfer to the liquid refrigerant is considerably greater than to the vapour, superheating of the vapour in the non-flooded evaporator is obtained at the expense of capacity and with a lower evaporator pressure. In small units this is not a serious handicap and the cheaper installation counterbalances the slight loss in operating efficiency. In very large installations the converse is generally true. Finned tubes or shell-and-tube exchangers are generally used and in all cases the temperature differential must be maintained as small as possible. Recently, falling film-type evaporators have become popular for water coolers because of the very high U's attainable. Data on heat transfer coefficients for evaporation are available (33, 152).

(iv) Expansion Valves

The pressure differential between the condenser and the evaporator is maintained by a pressure-reducing valve which in most installations combines this function with that of control.

Small units may be satisfactorily controlled by a simple constant pressure expansion valve. This valve contains bellows which control the valve seat. Changes in evaporator pressure actuate the bellows which open or close the valve to counteract the flow change. During the off-cycle, the build-up of evaporator pressure closes the valve and prevents the flow of refrigerant to the low side. An adjusting screw makes possible the selection of any desired evaporator pressure.

The thermostatic expansion valve combines the constant pressure function with that of constant temperature at the outlet of the evaporator, thus maintaining constant superheat. A bulb at the outlet of the evaporator follows the temperature fluctuations at this point and transfers them as pressure fluctuations to a diaphragm within the valve body. This diaphragm actuates the valve to control the flow of the refrigerant to the evaporator. Since the evaporator pressure is impressed on the opposite side of the diaphragm, the resulting action maintains a constant differential across the evaporator or a constant superheat.

In flooded evaporators, various valves may be used such as a float valve, which controls the liquid level, or others. In some cases, injectors are used, but regardless of their form all valves perform a similar function, that of maintaining a pressure difference between condenser and evaporator while controlling refrigerant flow to maintain constant conditions.

V HEAT PUMP AUXILIARIES

The previous section was devoted to the description of the four main parts of any refrigeration or heat pump circuit. The installed heat pump includes various other equipment necessary for automatic, trouble-free operation. These have been classified in the following description as auxiliaries, and the most important of these is the motor to drive the compressor.

(i) Drive Motor

Although some attention has been given to the use of coal, oil or gas as potential basic energy sources, electricity remains as the source of energy commonly used to drive heat pump installations. And why not? An electrical system is the most readily adaptable to automatic operation. Convenient and clean, it is easily controlled and is available at reasonable cost in most localities.

However, the use of an electric motor to drive domestic installations presents many problems. The power available for such installations on this continent is generally 60-cycle, single phase, A.C. power and the operational characteristics of a motor working on such a supply include many objectionable factors. Such a motor has a low power factor at other than its rated load. Starting torque, pull-in torque, and pull-out torque must be carefully matched to the requirements of the compressor (106). This means that, when running, the unit must operate at full load and capacity modulation is not possible with a simple unit, whereas while the load is developing, the motor has unfavourable demand characteristics. The general shortage of three-phase power increases cost of motor and starting equipment (71).

Since to take care of the peak requirements the unit must be overdesigned, it usually operates under low load factors, which are quoted at various values from ten to twenty-five percent (1, 103). These low load factors are objectionable to the utilities since it means that large quantities of equipment are lying idle for extended periods and this, of course, would have the ultimate effect of increasing the unit kilowatt-hour rate.

Whether or not the daily peak load of a heat pump installation will coincide with normal daily load requirements from the utilities, is debatable, although incomplete data seem to indicate that the peak load requirements will be slightly different (1). Regardless of its relationship with normal load requirements, the peak heat pump requirements of various installations in a given area will coincide and these peaks will not be greatly different from the normal load requirements. Although the general opinion is that electricity rates should decrease with extensive use of heat pump heating systems, investigation by utilities indicates that because of the poor load factor it is possible that rates would increase (46, 64).

Use of heat storage has the effect of increasing the load factor (45) and this is favoured by the utilities. With a high load factor special electricity rates (lower than the existing ones) are possible.

To improve the power factor and the load factor, special automatic controls are necessary and these are sufficiently expensive to have a marked effect on the first cost.

(ii) Control

Control of the heat pump is a complex function. The wide variation in load requirements makes control on the capacity essential. The installation must be capable of delivering the required load under extreme conditions and thus, under reduced load requirements, the unit is considerably overdesigned. The delivery of full load heat to the conditioned space under these conditions, without its equal dissipation to the outside, results in an increase in the air temperature until the two are again in balance. Some means of controlling the capacity is therefore obviously required and, as pointed out above, for a simple electrical unit, this can only be accomplished by interrupting the operation. This may be accomplished by temperature or pressure control, thus operating the unit between fixed temperature limits.

In intermediate seasons, fast change-over from the heating to the cooling cycle is required and it is desirable to have an automatic operation. This may be accomplished by four-way valves which can be actuated by the temperature requirements to switch the refrigerant flow from the heating to the cooling cycle.

Where icing is likely to occur on the evaporator or the lines, automatic de-icing is required. This may be accomplished by reversing the cycle but a simpler means is to make the evaporator coil the element of a heating circuit, and when the frost builds up beyond the allowable limit, the unit is cut out and this circuit is cut in. In large installations heating and cooling may be required simultaneously and this must be provided by automatic control.

Thus the control necessary to provide automatic operation of a heat pump installation becomes an extremely complicated network. The latter is further complicated by the use of single phase power. The system required to prevent undesirable reactions on the supply system and to cater to the idiocyncracies of the single phase motor puts considerable strain on the control network and many auxiliary controls are necessary to maintain a fully automatic, trouble-free system (51).

(iii) Piping

Refrigerant piping may be classified under the three headings of liquid lines, high pressure gas lines, and low pressure gas lines. Since pressure drop in the lines impairs operating efficiency, these lines should be as short as possible and so sized that the pressure drop is as small as possible. If the lines are too large, however, more refrigerant is required and this, added to the increased cost of the lines, will produce an unnecessarily expensive installation. In small refrigeration units adequate velocity for oil return limits the line size, but since larger units would include an oil trap for oil separation this ceases to be a factor. In heat pump installations the condenser and evaporator serve double duty, interchanging their function on heating and cooling cycles. The requirements of the installation may make long lines necessary and thus careful attention to line sizes is particularly important. Compromises are necessary, and the design satisfying the worst condition will generally prevail.

(iv) Ducts, Fans, etc.

Where air is used as the heating medium, means must be provided to circulate the air through the conditioned space. Conventional fans used in normal air conditioners are used successfully but the ducts require special design (71). To provide for emergency caused by breakdown or leaks developing in the system, shut-off and blow-off valves are provided at strategic points. If water is used as a heat source, scaling of the exchangers will occur and adequate facilities for descaling must be provided. For complete conditioning, provisions for humidification or dehumidification, filtering and deodorizing the air with their attendant controls, will all form part of the heat pump system.

VI APPLICATIONS OF THE HEAT PUMP

The applications of the heat pump may be classified under three headings:

(i) Domestic Installations;
(ii) Commercial Installations;
(iii) Industrial Installations.

The first two cover the use of the heat pump in domestic homes and commercial enterprises, principally as an air conditioning unit. Industrial applications cover a wide variety of uses where recovered waste heat at low temperatures can be reconverted to usable heat at a higher temperature level.

(i) Domestic Installations

Most of the research projects on heat pumps at present in progress are directed towards domestic use. The goal, of course, is a fully automatic electric home, heated in winter, cooled in summer without requiring care, outside of occasional inspection and repair. The advantages of such a system are many and include clean heat, fully treated air, lack of chimneys, no need for removable double windows or sliding sashes, more responsive system, greater comfort, etc. It is difficult to compare these luxury conditions with convential heating methods on a cost basis, but they are advantages that are an integral part of this device.

The market appears to be great but not so great as would at first be imagined. The heat pump, at present, will not compete with conventional heating systems, due to its high first cost. It is, however, competitive with any other system that combines the advantages of winter heating plus summer cooling (79). The market for the heat pump then appears to be limited to buyers who can afford the luxuries of summer cooling and this eliminates its use in speculative building (71). It has been stated that summer cooling was feasible in houses costing above $6,000 (U.S.A. 1947) (79). This seems to be a very low limit and it seems more likely that this minimum should be raised to from $15,000 to $20,000 for this country.

A number of packaged units are available in the United States including Muncie Gear Works' Marvair (117), Drayer Hansen's Airtopia and General Engineering and Manufacturing Company's "Miracula" (80, 81).

Another domestic use that is receiving attention is in hot water heating and experimental work is being devoted to the study of this problem (126, 148, 149).

(ii) Commercial Installations

Since many commercial establishments with heat pump installations can make use of packaged units, commercial installations are therefore generally interpreted to mean units specifically designed for the particular use, that is, custom built units. A number of different type units are in use in various parts of the world.

The largest installation of its kind was installed in the Equitable Building in Portland, Oregon. This complex installation has already been mentioned and has been described extensively in the literature (15, 19, 23, 60, 86, 87, 95). This installation uses underground water as a heat source and consists of multiple units. It is completely automatic, providing heating or cooling, or any combination of the two, as desired.

The Westinghouse Building in Emeryville, California, uses an air to air system with heat recovery from exhaust gases. Change-over from heating to cooling is obtained by damper control on the air streams. City water is used as an auxiliary source when air temperatures fall below 35°F. The system is completely automatic and provides for filtering and humidification of the conditioning air (27).

The first application of the heat pump to motion-picture theatres was made in Buenos Aires (111). Well water at 64° - 68°F. provides the source of heat. For change-over from heating to cooling the water and air flows are altered while the refrigerant flow is unchanged. The compressor is a four cylinder machine and capacity control is obtained by unloading cylinders, thus operating one, two, three or four cylinders. The unusually high source temperature leads to excellent performance factors.

Many other installations exist throughout the United States and these are of various types (28, 29, 123, 130).

Heat pumps have received considerable impetus in Switzerland and a number of commercial installations are reported in that country (49, 63, 83, 85).

(iii) Industrial Installations

The use of heat pumps in heavy industry in America has not as yet become a widespread practice. The abundance of coal and its resultant low price has been a major deterrent. As long as the price of coal remains low and that of electricity reasonably high it is unlikely that much consideration will be given to the heat pump as a major source of heat supply. Switzerland on the other hand has long suffered from the high price and scarcity of coal, while electricity is abundant and cheap. Even so, direct use of electricity cannot compare with coal burning. The use of the heat pump, however, is an excellent substitute for expensive and scarce coal. The use of heat pumps as auxiliary heat suppliers with existing systems providing peak load requirements has resulted in considerable savings (2, 20, 83, 114, 132).

The use of heat pump in evaporators results in economies unmatched by any other method and their use as such a device can be expected to increase (32, 105, 115, 134, 137). The concentration of milk (88), heating of swimming baths (142), are other uses to which the heat pump has been put.

An interesting application of the heat pump is to be found in Lausanne, Switzerland. The city watermain is tapped and the water passed over the condenser of the refrigeration installation, recovering the heat to raise the temperature of the entire city water supply by 4.5°F., representing a saving of about 350,000 kw-hr in a winter season (22, 98).

A packing plant in Basel, Switzerland has installed a heat pump to recover the heat removed by their refrigeration units. This heat is used to provide hot water for use elsewhere in the plant (30). The silk and paper industries of Switzerland have also turned to the heat pump as a means of recovering low grade waste heat (94).

VII THE ENGINE DRIVEN HEAT PUMP

The electrically driven heat pump has been previously examined. Its potential lies in its ability to contribute to the dream of an all-electric home, but unfortunately there are still a number of disadvantages, inherent in single-phase power principally, that present obstacles in the path of its universal acceptance. An engine-driven heat pump appears to be an attractive means of eliminating some of the difficulties experienced with motor driven units. Such a system would employ a small Diesel engine to drive the compressor and the necessary auxiliaries and the waste heat from the exhaust gases and cooling jacket would be recovered and transferred to the heated space, reducing the requirements of the unit and increasing its efficiency. Such a system has been considered theoretically (3, 59, 93) but little data is available from experimental or actual installations. These systems are described as having a COP of from 1.5 to 1.8. This is actually a misnomer and does not represent a true picture of the capabilities of such a system. What is referred to is the ratio of the heating effect produced to the heat in the fuel supplied to the driving unit, and henceforth this will be referred to as the "Heating Factor". A conventional oil fired furnace is capable of delivering 65 - 75 B.t.u. of useful heat for every 100 B.t.u. supplied as fuel. An engine driven heat pump is then capable of better than twice the efficiency of the conventional heating system and represents considerable saving over an electrically driven heat pump.

Where the installation must be designed for the heating load, a better balance is obtained since heat extracted from the conditioned space coupled with the waste heat from the engine are rejected together through the same coil. This heat is roughly the same in quantity as the source requirements during the heating cycle (93).

The engine is also capable of operating at variable output and throttle control may be used to adjust to the load requirements. Although the compressor efficiency falls off as the demand decreases from the rated capacity, this decrease in efficiency occurs at light demand and therefore its effect is not too serious. Capacity modulation becomes possible, giving a system that readily adjusts its output to satisfy the changing load requirements. Furthermore, design need not be based on the peak load requirements. For example, a normal 5 hp. Diesel engine may be operating from 3.5 to 7.5 hp. under steady conditions and at even higher loading for limited periods.

The factor limiting the operation of motor-driven units to their rated capacity only is the electric motor itself, which must be operated under design load conditions. With the possibility of capacity modulation and the ability of the unit to operate in excess of its rated load, it becomes possible to design on average load requirement and thus a smaller unit may be used.

The cost of such an installation at present exceeds that of the electrically driven unit by a considerable amount (59) but it should be possible to reduce this cost considerably, particularly if such an installation should receive wide application. One means of accomplishing this would be the combination of compressor and engine in one unit, which combination should be cheaper than engine and compressor separately. Recovery of waste heat reduces the evaporator requirement by approximately one-third, making this unit less expensive. The supply and storage of fuel, and the disposal of the exhaust gases are disadvantages that are not encountered in electrical operation. Noise and vibration might be minimized by insulation, or alternatively, the installation may be removed from the house. Maintenance would probably be somewhat higher than with its electrical counterpart. The savings in running cost will be a compensating factor and the Diesel heat pump would still embody all the advantages of a heat pump system. The system can be made fully automatic and a cheaper control mechanism is indicated.

VIII ECONOMICS OF THE HEAT PUMP

It has been proved conclusively that the heat pump is not merely an interesting experimental device, but a practical and highly efficient conditioning unit. For heating, the installation provides the maximum of comfort with the minimum of attention. It must be remembered that the heat pump is a year round conditioning unit and as such can hardly be compared with a simple coal or oil furnace which is capable of providing only winter heat. The actual running costs of the heat pump, when used as a heating unit, are less than the conventional system (124) with the break-even point, according to Penrod, and based on an electricity cost of one cent per kw-hr occurring at a COP of 3.5 (103). Since coal has increased in cost since 1947 (the date of Penrod's article) the break-even point would occur at a lower COP (approximately 2.5 in Canada).

The high initial cost of the heat pump installation, however, is considerably greater and the heavy yearly charges on this capital outlay more than balance the savings on fuel. Davies and Watts (48) by depreciating the heat pump at 7% per annum as against 15% per annum in the case of the conventional heating system, and including costs of tending, arrive at a net saving when using a heat pump. This difference in depreciation rates seems hardly justified.

The true comparison for the heat pump is a system providing winter heating and summer cooling. The cost of the heat pump unit is then not out of line with the combined heating and cooling plant (103, 124). Under these conditions the advantages of the heat pump become apparent. With capital costs approximately balanced, the saving in running costs represents a net saving, its fully automatic operation, lack of waste and continual comfort make the heat pump a very desirable conditioning medium.

Its high capital cost restricts the heat pump to domestic installations where the owner is capable of installing a summer cooling unit, and as previously stated, this would restrict its adoption to houses in the $15,000 class. Although this may represent a small percentage of the total building rate, this still presents a large potential market for the device.

The engine driven heat pump at present involves an even greater investment than the electric unit (59), but its potential fuel saving is greater and it seems feasible that an engine driven unit manufactured in quantity could be made to compare favourably with the electric units. These units are particularly applicable in colder climates and possess the added advantages, as indicated previously, of smaller size and less evaporator capacity. Such a unit is less dependent on local conditions, requiring that only about two-thirds of its heating requirement be supplied from external sources. This advantage would seem to have particular importance in Canada where the problem of finding a suitable heat source becomes of major importance.

EXPERIMENTAL SECTION

The ultimate purpose of this research project was to determine the operating characteristics of a Diesel-driven heat pump, and more particularly to study the effect of evaporator temperature on the performance and capacity, and to explore the feasibility of capacity control by means of engine speed variation.

EQUIPMENT

The apparatus used in this work was designed to approximate as closely as possible the conditions that would be met in a normal air conditioning application. However, the experimental nature of the investigation made it necessary to introduce certain modifications. The major change from normal operation was the substitution of water in place of air as the heat sink medium. The reasons for this modification are obvious in that the measurement of the heat delivered to the water could be more accurately determined and the size and complexity of the equipment was greatly reduced. The equipment consisted essentially of a standard refrigeration circuit with heat source and sink. The refrigeration compressor was driven by a Diesel engine and provision was made to recover heat from the exhaust gases and cooling jacket of the Diesel engine. The complete installation thus consisted of a system of five clearly defined circuits or streams, as shown in Figure 4, and the equipment will be discussed in greater detail on this basis.

I REFRIGERATION CIRCUIT (FREON STREAM)

The refrigeration circuit consisted of a standard Freon-12 vapor compression cycle, utilizing a reciprocating compressor and thermostatic expansion valve. The compressor employed was a two-cylinder Brunner, Model 5000, as supplied with a W-300 condensing unit. This commercial unit is normally driven by a 3 hp. electric motor. The condenser-receiver was mounted beneath the chassis and was of the water cooled, shell-and-tube type. The receiver fed a type 573 Detroit Thermostatic Expansion Valve and then entered a film-type evaporator (Red Diamond 901-1). This evaporator consists of two large concentric tubes sealed at each end to form a hollow cylinder with a narrow annular space between the two tubes. This annulus provides a passage for the evaporating refrigerant, while the water is spread in a thin film over the outer and inner surfaces of the tubes by means of a distributor. The evaporator assembly was 5 feet in height and 10 inches outside diameter. Although designed for flooded operation with injector feed, the unit was successfully adapted to non-flooded operation. The evaporator installation is shown in Figure 7.

The condensing pressure was read with a standard 2-1/2 inch USG pressure gauge with a range of 0-300 p.s.i.g., while the evaporating pressure was measured with a standard 2-1/2 inch compound refrigeration gauge with ranges of 0-30 inches Hg vacuum and 0-60 p.s.i.g.

Copper-constantan thermocouples were installed in appropriate wells in the lines to measure the temperature at the following points:

(i) Inlet to evaporator (T_1),

(ii) Outlet from evaporator (T_5),

(iii) Inlet to compressor (T_6),

(iv) Outlet of compressor (T_3),

(v) Outlet of receiver (T_2),

(vi) Inlet to expansion valve (T_4).

It will be noted that numbers (ii) and (iii) above are in series, as are also (v) and (vi). These points were, however, separated by a length of line and sufficient temperature difference existed between them to justify their separate measurement. This made possible the determination of enthalpy differences through the individual components with greater accuracy.

Standard 5/8 inch O.D. copper refrigeration tubing was used on liquid lines, while 1-3/8 inch and 1-1/8 inch tubing was employed on cold and hot gas lines, respectively. An oil trap was installed in the hot gas line, and a silica gel drier with by-pass connections was provided in the liquid line, which was also equipped with a sight glass.

The normal precautions regarding position of lines were observed, to prevent trapping of liquid and subsequent liquid slugging.

II CONDENSER WATER CIRCUIT

As mentioned previously, water was used as the heat sink. This stream was passed at a constant rate, as measured by a calibrated orifice, through a double-pipe steam heater before entering the condenser of the refrigeration unit. In this way the temperature of the water entering the condenser could be maintained at any desired level. For the purpose of this investigation, the latter was maintained at 64°F., the approximate temperature at which the exhaust air from a conditioned space, when fortified with proper proportions of fresh air from the outside atmosphere, would be returned to the condenser. This temperature would vary, of course, with outside temperature and represents the approximate conditions that would be anticipated in this area. For the same reasons, the water rate throughout the trials was selected to approximate the normal air change rate, on a heat balance basis.

After passing through the condenser, the water was fed through a shell-and-tube heat exchanger to recover heat from the exhaust gases of the Diesel engine, and from there through a double-pipe heat exchanger for the recovery of heat from the Diesel jacket cooling water. The temperature of the condenser water stream was measured by thermocouples at the following points:

(i) Inlet to the condenser (T_9),

(ii) Outlet of the condenser (T_{10}),

(iii) Outlet of exhaust heat recovery exchanger (T_{11}),

(iv) Outlet of jacket heat recovery exchanger (T_{12}).

III EVAPORATOR WATER CIRCUIT

The evaporator water (heat source) was fed at constant rate, as measured by a calibrated orifice, to the distributor of the film-type evaporator. The cooled water was collected in a hold-up tank and then pumped through a heater to be recycled. For evaporator temperatures below 32°F., methanol was added to the water. Direct use of mains water could, of course, have been made, at least for evaporator temperatures above 32°F. The recirculating system, however, made it possible to maintain the water at any desired temperature at the inlet to the evaporator. In addition to the flow rate, the temperature was measured at the inlet and outlet of the evaporator (T_7, T_8).

IV JACKET WATER CIRCUIT

The water for the cooling jacket of the Diesel engine was recirculated in a closed circuit consisting of the jacket heat recovery exchanger, a hold-up tank and a small gear pump in series. The rate of flow was measured by means of a calibrated orifice and the temperature was measured at the following points by means of thermocouples:

(i) Inlet to jacket heat recovery exchanger (T_{14}),
(ii) Outlet of jacket heat recovery exchanger (T_{12}),
(iii) Inlet to Diesel cooling jacket (T_{13}).

V EXHAUST GAS STREAM

The exhaust gases from the Diesel engine were passed through an exhaust gas heat recovery exchanger, of the shell-and-tube type, which discharged to atmosphere. No attempt was made to measure the quantity of these gases but the temperature was taken at the inlet and outlet of the exchanger.

VI POWER, CONTROLS, ETC.

The power to drive the unit was provided by a Ruston Hornsby 1-VTH Diesel engine. This engine was mounted independently of the compressor and on the opposite side to the electric motor. This installation is shown in Figure 6. The unit could thus be run on Diesel or electric power by the simple expedient of changing the belt drive. This Diesel engine is rated at 5 hp., with an operating range of 3.5 to 7.5 hp. The fuel fed to the engine was measured by means of a level gauge calibrated in pounds.

With the exception of valves, all controls were brought to the control panel (Figure 8) and all readings were taken from this point. Mounted on the panel were, the oil level gauge, manometers for the orifices, pressure gauges, motor starting switches, and thermocouple selector switches.

A schematic flow-sheet of the apparatus is given in Figure 4, while Figure 5 is a photograph of the installation. Figures 6, 7 and 8 show the component parts in more detail.

FIGURE 4

EXPERIMENTAL APPARATUS

LEGEND

A	Hold-up tanks
B	Circulating pumps
C	Condenser-receiver
D	Drier
DE	Diesel engine
E	Evaporator installation
F	Steam heaters
FT	Diesel fuel tank
G	Compressor
HE-1	Jacket heat recovery exchanger
HE-2	Exhaust heat recovery exchanger
I	Liquid sight glass
M	Electric motor
O	Orifices
OT	Oil trap
P	Pressure gauges
T	Thermowells
V	Expansion valve

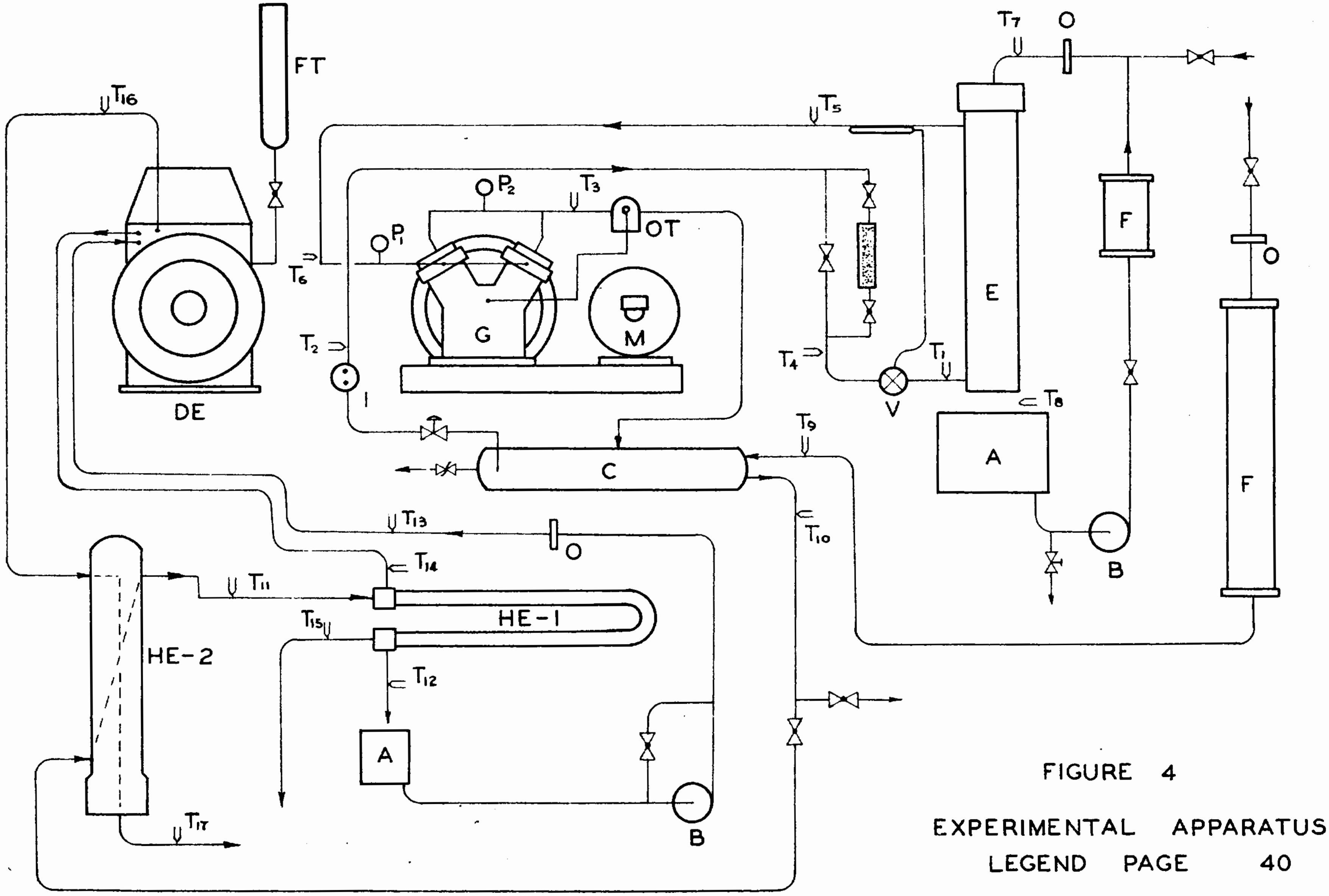


FIGURE 4

EXPERIMENTAL APPARATUS

LEGEND PAGE 40

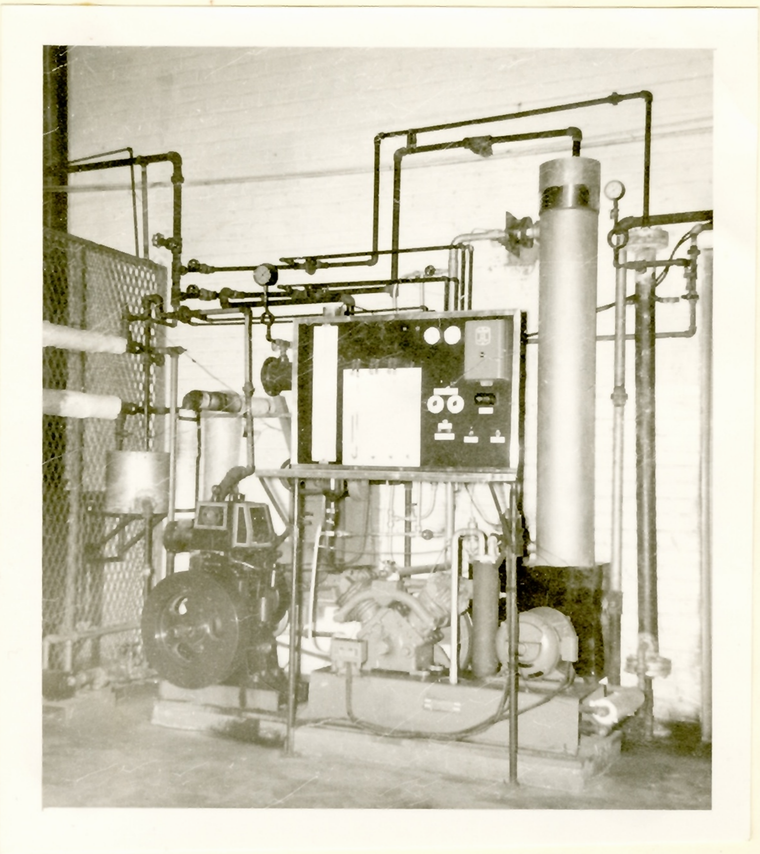


FIGURE 5.

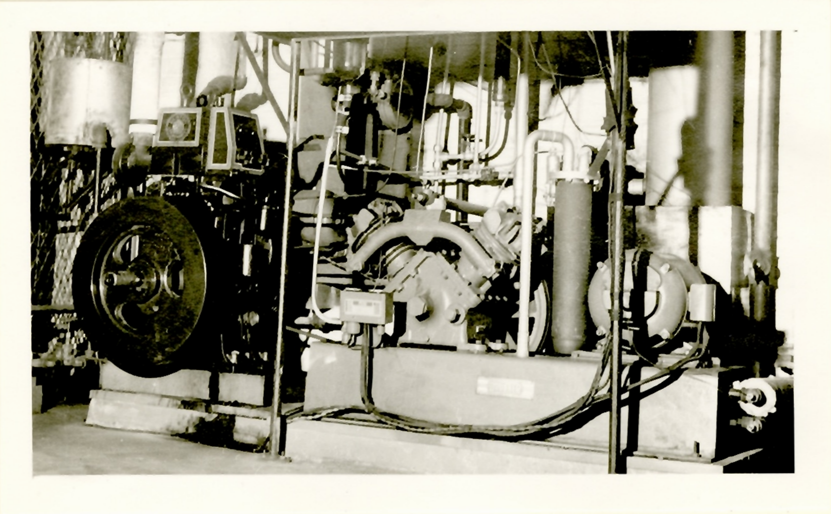


FIGURE 6.

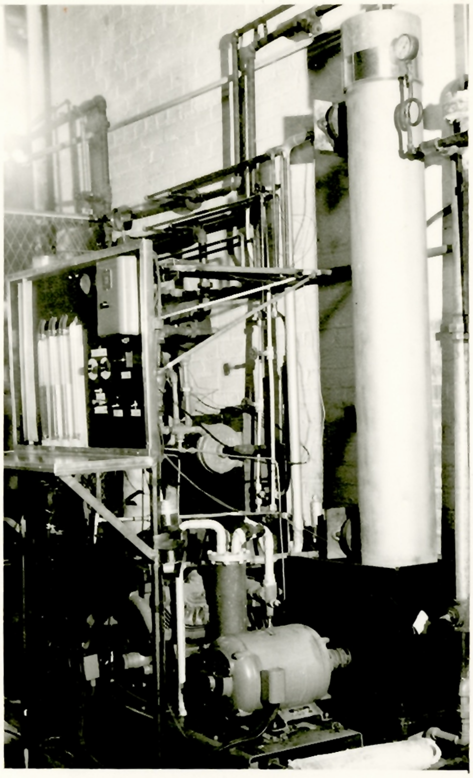


FIGURE 7.

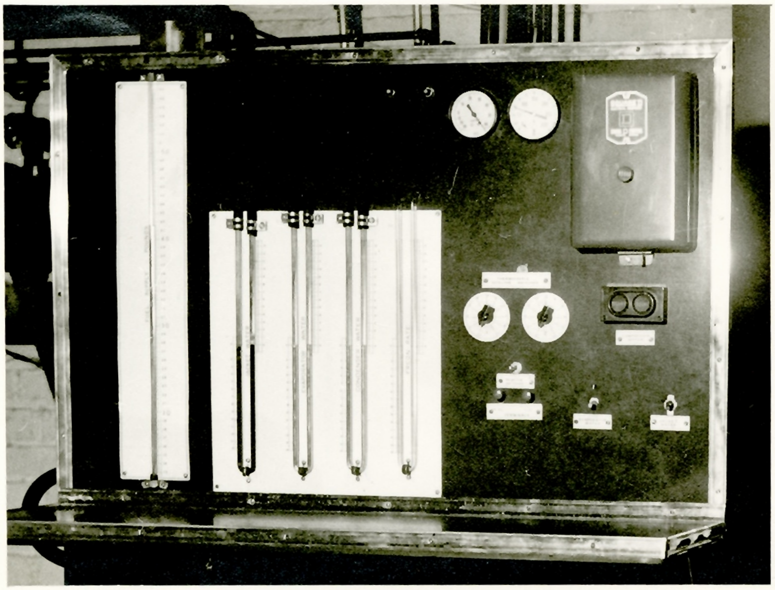


FIGURE 8.

PROCEDURE

I CHARGING AND CALIBRATION

Upon completion of equipment installation, the system was tested and calibrated. The refrigeration circuit required particular attention, and the precautions which were taken to ensure its satisfactory operation will now be described.

The refrigeration installation was first tested at a pressure of 150 p.s.i.g. by filling with nitrogen to that pressure. Leaks appeared at a number of points and these were repaired as required. The system was then allowed to stand under pressure until no loss of pressure was indicated over a period exceeding twenty-four hours. It was then exhausted by means of a small vacuum pump and filled to cylinder pressure with Freon-12. Further tests were made with a halide leak detector.

Some of the fittings and valves used in the refrigeration lines were not standard refrigeration equipment. These were carefully tinned before use, and every precaution was taken to guard against future breakdown.

Upon completion of these tests, final charging of the unit was effected by introducing the required quantity of refrigerant through the suction side of the running compressor. When the refrigeration circuit was charged, the rest of the installation was tested and the orifices calibrated. Particular attention was given to the mounting of the copper-constantan thermocouples in their respective wells, which were then checked for accuracy and found to be satisfactory.

The refrigeration unit was run for a period with the drier in the circuit and then test runs were conducted using the electrical system before the Diesel runs were started.

II EXPERIMENTAL OPERATION

Prior to the initial operation of the refrigeration unit and during all subsequent shut-down periods, the system was "pumped down"; that is, the refrigerant was pumped out of the system and stored as a liquid in the receiver. This was accomplished by shutting the valve at the outlet of the receiver and running the compressor until the evaporator pressure dropped to approximately 10 inches Hg vacuum. This pressure would build up over the shut-down period, due to the release of Freon from the oil, to a positive pressure of 50 to 60 p.s.i.g. The system was not pumped down to a high vacuum because of the necessity of easing the starting load on the Diesel engine, which was not provided with an adequate clutch arrangement. A number of revolutions of the engine were necessary to heat the cylinder sufficiently for starting, and the operation was greatly facilitated by keeping the load at a minimum. At large back pressures, sufficient gas was present to constitute an appreciable load, while at very low back pressures, the pressure difference across the compressor was again sufficient to make start-up difficult. It was soon found that the easiest starting occurred in the range of 15 - 20 p.s.i.g. evaporator pressure.

The starting procedure was as follows: the condenser water was first turned on and allowed to cool the refrigerant and thus lower the head pressure. The jacket water hold-up tank was filled with warm water and this was circulated. The engine was then turned by means of the electric motor until it fired regularly. The drive between the two was then disconnected, and when the engine picked up speed the outlet valve from the receiver was opened slowly to allow the refrigerant to flow into the expansion valve. The evaporator water was then turned on and the system allowed to pull down to the desired evaporator temperature. The steam heaters were then put into operation and the engine speed, condenser and evaporator temperatures were adjusted until all three were at the desired levels. The unit was allowed to run until in balance before commencing a run. A by-pass in the condenser water line prevented cooling of the jacket water or exhaust gases by the cold condenser water during engine start-up, and when the engine was running and the temperature of the cooling jacket water had risen to operating temperatures, the condenser water was diverted through the two heat recovery exchangers.

The general method of experimental evaluation of this heat pump installation was the measurement of heat input and output at constant sink temperature and at various source temperatures. The effects of capacity modulation were investigated by means of varying engine speed at each individual source temperature. Actually, the water temperatures at the inlet to the condenser and evaporator were maintained constant rather than the average source and sink temperatures. It was found that approximately two to three hours were required on start-up to obtain satisfactory running conditions, while one hour between successive runs was generally sufficient when only the engine speed was varied.

Runs were of approximately one hour's duration and the following readings were taken at intervals throughout the run:

General

Condenser pressure,

Evaporator pressure,

Condenser water rate,

Cooling jacket water rate,

Fuel and time (at beginning and end of run only).

Water Circuits (Temperature)

Inlet to condenser,

Outlet of condenser,

Outlet of exhaust heat recovery exchanger,

Outlet of jacket heat recovery exchanger,

Inlet to evaporator,

Outlet to evaporator,

Inlet jacket cooling water to jacket heat recovery exchanger,

Outlet of jacket cooling water from jacket heat recovery exchanger,

Inlet of jacket cooling water to Diesel cooling jacket.

Gas Temperatures

Outlet from Diesel engine,

Outlet from exhaust heat recovery exchanger.

CALCULATIONS AND NOMENCLATURE

From the experimental data taken over a series of trials, heat balances and heating factors were calculated. Standard Freon-12 charts and tables were employed to determine the enthalpy of the refrigerant at the various points in the circuit and the COP of the refrigerant was calculated from the actual enthalpy conditions at the various points in the circuit. The heat delivered to the evaporator was calculated from the flow rate and temperature drop through the evaporator, following correction for extraneous heat pick-up from the surrounding atmosphere, this correction being obtained from blank runs. The sink water heat absorption was determined from flow rates and temperature changes through the individual heat exchangers. A complete heat balance was possible in the jacket water recovery exchanger, but no attempt was made to obtain a balance on the exhaust gas heat recovery exchanger.

The power delivered to the refrigerant was determined by means of the enthalpy conditions at the inlet and outlet of the compressor, while the brake horsepower of the Diesel engine was taken from charts supplied by the manufacturer, assuming full load operation.

The complete experimental data taken during each trial are contained in the Appendix, while the results calculated from these data appear under Experimental Results. For the purpose of physical presentation, all table headings appearing in subsequent pages are presented in the form of symbols only, full nomenclature for which is as follows:

T_1	Freon temperature at inlet to evaporator	°F.
T_2	Freon temperature at outlet of condenser-receiver	°F.
T_3	Freon temperature at outlet of compressor	°F.
T_4	Freon temperature at inlet to expansion valve	°F.
T_5	Freon temperature at outlet of evaporator	°F.
T_6	Freon temperature at inlet to compressor	°F.
T_7	Source temperature at inlet to evaporator	°F.
T_8	Source temperature at outlet of evaporator	°F.
T_9	Sink temperature at inlet to condenser	°F.
T_{10}	Sink temperature at outlet of condenser	°F.
T_{11}	Sink temperature at outlet of exhaust heat recovery exchanger	°F.
T_{12}	Sink temperature at outlet of jacket heat recovery exchanger	°F.
T_{13}	Jacket cooling water temperature at inlet to Diesel	°F.
T_{14}	Jacket cooling water temperature at outlet of Diesel	°F.
T_{15}	Jacket cooling water temperature at outlet of jacket heat recovery exchanger	°F.
T_{16}	Exhaust gas temperature at outlet of Diesel	°F.
T_{17}	Exhaust gas temperature at outlet of exhaust heat recovery exchanger	°F.
P_1	Evaporator pressure	p.s.i.a.
P_2	Condensing pressure	p.s.i.a.

w_1	Source liquid rate	(lb H_2O)/(min.)
w_2	Sink water rate	(lb)/(min.)
w_3	Jacket cooling water rate	(lb)/(min.)
w_4	Freon flow rate	(lb)/(hr.)
H_1	Enthalpy of Freon at inlet to evaporator	(B.t.u.)/(lb)
H_2	Enthalpy of Freon at outlet of condenser-receiver	(B.t.u.)/(lb)
H_3	Enthalpy of Freon at outlet of compressor	(B.t.u.)/(lb)
H_4	Enthalpy of Freon at inlet to expansion valve	(B.t.u.)/(lb)
H_5	Enthalpy of Freon at outlet of evaporator	(B.t.u.)/(lb)
H_6	Enthalpy of Freon at inlet to compressor	(B.t.u.)/(lb)
ΔT_1	Source temperature change through evaporator ($T_7 - T_8$)	°F.
ΔT_2	Sink temperature change through condenser ($T_{10} - T_9$)	°F.
ΔT_3	Sink temperature change through exhaust heat recovery exchanger ($T_{11} - T_{10}$)	°F.
ΔT_4	Sink temperature change through jacket heat recovery exchanger ($T_{12} - T_{11}$)	°F.
ΔT_5	Jacket cooling water temperature change through heat recovery exchanger ($T_{14} - T_{15}$)	°F.
ΔH_1	Enthalpy change through evaporator ($H_5 - H_1$)	(B.t.u.)/(lb)
ΔH_2	Enthalpy change through condenser ($H_3 - H_2$)	(B.t.u.)/(lb)
ΔH_3	Enthalpy change through compressor ($H_3 - H_6$)	(B.t.u.)/(lb)
COP	The coefficient of performance based on the refrigerant ($\Delta H_2/\Delta H_3$)	-

Q_1	The heat removed in the evaporator	(B.t.u.)/(hr.)
Q'_1	The heat picked up in evaporator refrigerant	(B.t.u.)/(hr.)
Q_2	The heat delivered to sink water in condenser	(B.t.u.)/(hr.)
Q_3	The heat recovered in the exhaust heat recovery exchanger	(B.t.u.)/(hr.)
Q_4	The heat recovered in the jacket cooling water recovery exchanger (based on sink water)	(B.t.u.)/(hr.)
Q'_4	The heat given up by jacket water in heat recovery exchanger (based on jacket water)	(B.t.u.)/(hr.)
Q_5	Total heat recovery $(Q_2 + Q_3 + Q_4)$	(B.t.u.)/(hr.)
Q_6	Heat in fuel consumed	(B.t.u.)/(hr.)
W_r	Heat delivered to the refrigerant as work	(B.t.u.)/(hr.)
W_s	Brake horsepower of Diesel	(B.t.u.)/(hr.)
E_w	W_r/W_s - Work efficiency	%
M	Fuel consumption	(lb)/(hr.)
S_e	Diesel engine speed	r.p.m.
k	Correction factor for source water orifice including density correction and specific heat for alcohol solution	(B.t.u.)/(lb ^{o}F.)

The equations employed in the calculation of results are as follows:

1. $Q_1 = (60)(w_1)(k)(\Delta T_1)$
2. $Q_2 = (60)(w_2)(\Delta T_2)$
3. $Q_3 = (60)(w_2)(\Delta T_3)$
4. $Q_4 = (60)(w_2)(\Delta T_4)$
5. $Q_5 = Q_2 + Q_3 + Q_4$

6. $Q'_1 = (w_4)(\Delta H_1)$

7. $Q'_4 = (60)(w_3)(\Delta T_5)$

8. $w_4 = Q_2/\Delta H_1$

9. $Q_6 = 19{,}700\ M$

10. $W_r = (w_4)(\Delta H_3)$

11. $F = Q_5/Q_6$

The symbols employed here correspond to those defined above.

EXPERIMENTAL RESULTS

To determine experimentally the characteristics of the Diesel-driven heat pump just described, a series of trials were conducted at various source temperatures and engine speeds, while maintaining the sink inlet temperature at a constant value. This made possible the study of the effect of source temperature on the capacity and on the Heating Factor, and to study the ability of the installation to respond to load changes. To this end, trials were conducted at source inlet temperatures of 40°F., 34°F., 28°F., 22°F., 16°F., and 10°F., and at five different engine speeds ranging from 500 to 1000 r.p.m. During these runs the source inlet temperature was maintained constant at 64°F.

By keeping the source and sink temperatures constant, the actual condensing and evaporating temperatures varied with load, This procedure was dictated by the necessity of obtaining results which were a function of the source and sink temperatures, the two variables that would normally be fixed in any given installation. This also gave an indication of the flexibility of the installation. The actual condensing temperatures varied from 85°F. at very light loads, to 115°F. at high loads, while under average conditions, a condensing temperature of approximately 100°F. was obtained, consistent with modern design. The evaporator behaved in a similar manner, with the temperature differential between refrigerant and source water increasing with increasing load.

From the experimental data, given in their entirety in the Appendix, the results shown in Tables I to VI were calculated. Each table covers a set of results at one source temperature and at five different engine speeds.

The heat recoveries in the condenser are shown graphically in Figure 9. At a constant source inlet temperature, the heat recovered in the condenser is seen to increase with increasing engine speed, the slope of this line decreasing as the speed is increased. This behaviour can be explained by the fact that the brake horsepower of the engine increases with speed in an approximately linear relationship (for the engine used in this work, the manufacturer gives the full load b.hp. as a linear function of speed), but due to the increasing temperature and pressure differential across the compressor, the COP of the system decreases, producing the "falling off" of the condenser heat recovery curve, when plotted against the engine speed.

At constant engine speed, the condenser heat recovery decreases with decreasing source temperature. The greater pressure difference between which the compressor operates, brought about by the lower evaporator temperatures, results in a decrease in the COP of the system, resulting in a decrease in the quantity of heat delivered for the same expenditure of work energy.

Figure 10 represents the heat picked up in the evaporator plotted as a function of engine speed and source inlet temperature. These curves are similar in shape to the condenser heat recovery curves. Once again the effects of decreasing efficiency and COP are evident.

Figure 11 represents the heat recovered from the exhaust gases plotted as a function of the engine speed. The heat recovered from this source is seen to be approximately linear with engine speed. From Tables A-I to A-VI, in the Appendix, it may be seen that the exhaust gas outlet temperature after cooling (T_{17}), lies generally between 70°

and 80°F. Essentially all the heat in the gases above 70°F. can thus be assumed to be recovered. The heat in the exhaust gases is then directly proportional to the b.hp. since this value, too, is a linear function of engine speed, or the heat recoverable from these gases is constant when expressed in (B.t.u.)/(b.hp.-hr.).

Figure 12 reveals that the heat recovered from the Diesel jacket cooling water increases with speed, while the slope of this line increases also with speed. The values of the heat recovered from the cooling water are those given in Tables I to VI under Q_4; Q'_4, the heat given up by the cooling water in the jacket heat recovery exchanger, is considered less reliable because of difficulties in the measurement of the cooling water flow rate. The flow rate through the Diesel jacket was very low and thus difficult to measure accurately. A carbon tetrachloride manometer was used in conjunction with an orifice, but accurate readings were not possible and therefore the values of the heat recovered from the jacket as calculated from the sink water flow rate have been used throughout as being the more reliable.

The sum of the values Q_2, Q_3 and Q_4 represents the total heat recovery, Q_5, and is plotted as a function of speed and source inlet temperature in Figure 13. These curves show the same general character as the condenser heat recovery curves of Figure 9, that is, increasing with decreasing slope as the engine speed increases. The total heat recovery curves show less curvature however, because of the increasing rate of heat recovery from the Diesel cooling water. The points on this graph show greater scattering than those of the other heat recovery graphs. This is due to the wider variations in the values of heat recovery from the Diesel, which show considerable variation owing principally to the experimental difficulty of accurately measuring small temperature changes through the heat exchangers. However, the trends are clearly indicated and the individual deviations from the norm are not serious.

Figure 14 shows the variation of the Freon flow rate with engine speed and source temperature. These values have been calculated from the condenser heat recovery values and the enthalpy conditions at the inlet and outlet of the condenser. A condenser balance, rather than an evaporator balance was employed here to obtain the Freon circulation rate, for the reason that enthalpy conditions at the inlet and outlet of the condenser may be accurately measured and thus the enthalpy change through the unit readily determined. In using an evaporator balance, the conditions at the inlet of the evaporator must be determined assuming an isenthalpic expansion through the valve. In addition to this, the evaporator was subject to greater influences from extraneous heat pick-up.

The Freon flow rate is seen to increase with increasing engine speed, the slope of the curve decreasing in the same direction. This behaviour is due to the decreasing volumetric efficiency, brought about by the increasing pressure difference across the compressor as the load increases. The Freon circulation rate also decreases with evaporator temperature due to the decrease in compressor capacity at low back pressures.

The variations in the Heating Factor, obtained by dividing the total heat recovery (Q_5) by the higher heating value of the fuel consumed (Q_6) are shown in Figure 15. When plotted as a function of engine speed, at a given source temperature the Heating Factor is seen to pass through a maximum, while at constant speed, the Heating Factor decreases with decreasing source temperature. This effect is small at first but becomes appreciable below 20°F. In lowering the source temperature from 40°F.

to 20°F. at 600 r.p.m., the Heating Factor is seen to decrease less than 10%, but drops an additional 14% for a further 10° drop in source temperature. In the variation of Heating Factor with engine speed at constant source temperature, a number of effects occur simultaneously. The fuel consumption per b.hp. hour, when plotted against engine speed, goes through a minimum. As the compressor speed decreases the efficiency falls off, while at increasing speeds the increasing pressure differential across the compressor decreases the efficiency and increases the work required to pump a given quantity of heat from low to high temperature. These factors combine to produce a maximum in the Heating Factor curve. The severity of the efficiency decrease at higher speeds is reduced somewhat by the increase in rate of heat recovery per expended horsepower from the Diesel cooling jacket, as indicated by the increasing slope of the curve in Figure 12, but this is not sufficient to offset the other effects. The maximum Heating Factors were found to occur between 575 and 625 r.p.m., shifting towards higher speeds with dropping source inlet temperatures.

Figure 16 represents the compressor efficiency as a function of engine speed. The shaft work was taken from the manufacturer's data assuming full load operation. This assumption is probably not always justified, but deviations are not likely to be serious. Except for the series of runs at 22°F., these curves show good regularity, the maximum efficiency occurring between 800 and 900 r.p.m.

Figure 17, showing variations of condenser heat with source temperature at constant speed, was obtained directly from Figure 9. The condenser heat recovery is seen to increase with approximate linearity with temperature, the slope decreasing with decreasing speed.

TABLE I

EXPERIMENTAL RESULTS

Source Inlet Temperature 40°F.

DESCRIPTION		TRIAL NUMBERS				
		1	2	3	4	5
ΔH_1	(B.t.u.)/(lb)	57.50	57.30	56.75	56.40	56.60
ΔH_2	(B.t.u.)/(lb)	67.55	68.63	69.45	70.85	71.85
ΔH_3	(B.t.u.)/(lb)	10.00	11.33	12.45	14.45	15.15
COP		6.75	6.07	5.57	4.88	4.74
w_1	(lb H_2O)/(min.)	82.0	85.4	87.6	84.6	90.0
k	(B.t.u.)/(lb °F.)	1.00	1.00	1.00	1.00	1.00
ΔT_1	°F.	5.8	6.2	6.6	7.5	7.9
Q_1	(B.t.u.)/(hr)	28,500	31,800	34,700	38,000	42,600
Q'_1	(B.t.u.)/(hr)	28,200	30,900	34,600	36,200	38,100
w_2	(lb)/(min.)	41.5	42.0	46.8	45.1	45.1
ΔT_2	°F.	13.3	14.7	15.1	16.8	17.8
Q_2	(B.t.u.)/(hr)	33,100	37,000	42,300	45,500	48,300
ΔT_3	°F.	1.6	2.2	2.7	3.2	4.4
Q_3	(B.t.u.)/(hr)	3,980	5,550	7,550	8,650	11,920
ΔT_4	°F.	3.2	3.4	3.4	4.2	2.9
Q_4	(B.t.u.)/(hr)	7,960	8,560	9,510	11,330	7,860
w_3	(lb)/(min.)	6.1	6.0	6.5	8.5	8.3
ΔT_5	°F.	17.5	20.4	23.6	16.6	15.2
Q'_4	(B.t.u.)/(hr)	6,410	7,350	9,200	8,460	7,560
Q_5	(B.t.u.)/(hr)	45,040	51,110	59,360	65,480	68,080
M	(lb)/(hr)	1.31	1.59	1.87	2.19	2.46
Q_6	(B.t.u.)/(hr)	25,800	31,300	36,800	43,100	48,500
F		1.745	1.635	1.610	1.518	1.402
w_4	(lb)/(hr)	490	539	609	642	672
S_e	r.p.m.	501	589	685	814	930
W_r	(B.t.u.)/(hr)	4,900	6,110	7,600	9,280	10,200
W_s	(B.t.u.)/(hr)	6,350	7,470	8,700	10,380	11,820
E_w	percent	77.2	81.8	87.5	89.5	86.2

TABLE II

EXPERIMENTAL RESULTS

Source Inlet Temperature 34°F.

DESCRIPTION		TRIAL NUMBERS				
		6	7	8	9	10
ΔH_1	(B.t.u.)/(lb)	57.10	56.80	56.55	56.40	56.10
ΔH_2	(B.t.u.)/(lb)	67.25	68.00	69.95	71.00	70.85
ΔH_3	(B.t.u.)/(lb)	10.00	10.90	13.05	14.35	14.60
COP		6.73	6.24	5.35	4.95	4.82
w_1	(lb H_2O)/(min.)	79.6	86.2	85.2	86.0	84.4
k	(B.t.u.)/(lb °F.)	0.998	0.997	0.999	1.000	1.004
ΔT_1	°F.	3.1	5.1	6.2	6.6	6.9
Q_1	(B.t.u.)/(hr)	24,300	26,350	31,650	34,100	35,100
Q'_1	(B.t.u.)/(hr)	25,600	28,300	31,450	34,000	36,400
w_2	(lb)/(min.)	42.2	42.2	41.7	41.6	41.5
ΔT_2	°F.	11.9	13.4	15.5	17.2	18.5
Q_2	(B.t.u.)/(hr)	30,200	33,900	38,800	42,900	46,100
ΔT_3	°F.	1.6	2.1	2.9	3.6	4.1
Q_3	(B.t.u.)/(hr)	4,050	5,320	7,250	9,000	10,220
ΔT_4	°F.	3.0	3.3	3.8	3.8	4.1
Q_4	(B.t.u.)/(hr)	7,600	8,350	9,510	9,480	10,220
w_3	(lb)/(min.)	6.4	6.0	5.8	6.1	6.3
ΔT_5	°F.	15.6	20.7	22.9	23.7	19.8
Q'_4	(B.t.u.)/(hr)	6,000	7,450	7,970	8,680	7,480
Q_5	(B.t.u.)/(hr)	41,850	47,570	55,560	61,380	66,540
M	(lb)/(hr)	1.32	1.47	1.82	2.13	2.26
Q_6	(B.t.u.)/(hr)	26,000	28,900	35,800	41,950	44,500
F		1.610	1.644	1.555	1.464	1.497
w_4	(lb)/(hr)	449	499	555	603	650
S_e	r.p.m.	508	592	719	835	913
W_r	(B.t.u.)/(hr)	4,490	5,430	7,230	8,650	9,490
W_s	(B.t.u.)/(hr)	6,470	7,550	9,150	10,620	11,600
E_w	percent	69.4	71.9	79.0	81.4	81.5

TABLE III

EXPERIMENTAL RESULTS

Source Inlet Temperature 28°F.

DESCRIPTION		TRIAL NUMBERS			
		12	13	14	15
ΔH_1	(B.t.u.)/(lb)	56.30	55.80	55.90	55.80
ΔH_2	(B.t.u.)/(lb)	69.15	70.30	71.00	72.25
ΔH_3	(B.t.u.)/(lb)	12.20	14.30	15.00	16.25
COP		5.66	4.91	4.73	4.45
w_1	(lb H_2O)/(min.)	82.5	87.5	86.1	86.1
k	(B.t.u.)/(lb °F.)	0.994	1.002	1.004	1.000
ΔT_1	°F.	5.0	5.1	6.0	6.4
Q_1	(B.t.u.)/(hr)	24,600	27,400	31,150	33,050
Q'_1	(B.t.u.)/(hr)	24,600	28,200	31,000	33,050
w_2	(lb)/(min.)	42.2	42.0	41.8	42.2
ΔT_2	°F.	11.9	14.1	15.7	16.9
Q_2	(B.t.u.)/(hr)	30,250	35,500	39,400	42,800
ΔT_3	°F.	2.1	2.7	2.9	4.3
Q_3	(B.t.u.)/(hr)	5,310	6,800	7,270	10,900
ΔT_4	°F.	3.0	3.7	4.0	4.1
Q_4	(B.t.u.)/(hr)	7,590	9,330	10,020	10,390
w_3	(lb)/(min.)	6.3	6.8	6.0	6.2
ΔT_5	°F.	16.8	20.2	25.8	25.2
Q'_4	(B.t.u.)/(hr)	6,350	8,250	9,280	9,370
Q_5	(B.t.u.)/(hr)	43,150	51,630	56,690	64,090
M	(lb)/(hr)	1.39	1.74	1.99	2.25
Q_6	(B.t.u.)/(hr)	27,400	34,300	39,300	44,300
F		1.575	1.502	1.443	1.448
w_4	(lb)/(hr)	437	505	554	592
S_e	r.p.m.	596	708	840	950
W_r	(B.t.u.)/(hr)	5,320	7,220	8,300	9,620
W_s	(B.t.u.)/(hr)	7,580	9,700	10,700	12,100
E_w	percent	70.2	74.5	77.4	79.5

TABLE IV

EXPERIMENTAL RESULTS

Source Inlet Temperature 22°F.

DESCRIPTION		TRIAL NUMBERS				
		16	17	18	19	20
ΔH_1	(B.t.u.)/(lb)	54.75	55.20	54.95	55.30	55.35
ΔH_2	(B.t.u.)/(lb)	69.25	69.53	70.05	71.55	73.45
ΔH_3	(B.t.u.)/(lb)	14.45	14.35	15.15	16.05	17.75
COP		4.79	4.83	4.63	4.46	4.12
w_1	(lb H_2O)/(min.)	78.5	81.5	84.5	85.5	84.5
k	(B.t.u.)/(lb °F.)	0.967	0.967	0.965	0.965	0.965
ΔT_1	°F.	4.1	5.1	5.2	5.8	6.3
Q_1	(B.t.u.)/(hr)	18,700	24,100	25,500	28,700	30,800
Q'_1	(B.t.u.)/(hr)	19,200	23,650	25,450	27,900	29,500
w_2	(lb)/(min.)	41.8	41.7	42.3	42.7	42.2
ΔT_2	°F.	9.7	11.9	12.8	14.1	15.4
Q_2	(B.t.u.)/(hr)	24,300	29,800	32,500	36,100	39,000
ΔT_3	°F.	1.5	2.1	3.0	3.6	4.5
Q_3	(B.t.u.)/(hr)	3,760	5,260	7,610	9,230	11,400
ΔT_4	°F.	2.9	3.0	3.4	4.0	4.6
Q_4	(B.t.u.)/(hr)	7,260	7,510	8,630	10,250	11,680
w_3	(lb)/(min.)	6.4	6.4	6.1	6.6	6.2
ΔT_5	°F.	18.7	20.9	24.0	27.1	31.3
Q'_4	(B.t.u.)/(hr)	7,180	8,030	8,780	10,750	11,650
Q_5	(B.t.u.)/(hr)	35,320	42,570	48,740	55,580	62,080
M	(lb)/(hr)	1.16	1.41	1.67	2.08	2.38
Q_6	(B.t.u.)/(hr)	22,800	27,750	32,900	41,000	46,900
F		1.548	1.538	1.481	1.358	1.325
w_4	(lb)/(hr)	351	428	463	503	533
S_e	r.p.m.	518	630	726	866	995
W_r	(B.t.u.)/(hr)	5,060	6,140	7,000	8,060	9,450
W_s	(B.t.u.)/(hr)	6,590	8,020	9,230	11,000	12,680
E_w	percent	76.8	76.7	75.8	73.3	74.5

TABLE V

EXPERIMENTAL RESULTS

Source Inlet Temperature 16°F.

DESCRIPTION		TRIAL NUMBERS				
		21	22	23	24	25
ΔH_1	(B.t.u.)/(lb)	54.75	55.30	55.25	55.30	55.30
ΔH_2	(B.t.u.)/(lb)	67.70	68.80	70.80	71.70	73.00
ΔH_3	(B.t.u.)/(lb)	12.60	13.40	15.55	16.40	17.50
COP		5.38	5.13	4.55	4.37	4.16
w_1	(lb H_2O)/(min.)	81.0	80.6	87.5	86.4	85.1
k	(B.t.u.)/(lb °F.)	0.970	0.970	0.970	0.936	0.936
ΔT_1	°F.	3.3	4.1	4.5	5.4	5.7
Q_1	(B.t.u.)/(hr)	15,550	19,240	22,900	26,200	27,250
Q'_1	(B.t.u.)/(hr)	16,800	20,500	23,250	24,700	26,000
w_2	(lb)/(min.)	42.2	41.7	41.7	42.0	42.0
ΔT_2	°F.	8.2	10.2	11.9	12.7	13.6
Q_2	(B.t.u.)/(hr)	20,800	25,550	29,800	32,000	34,300
ΔT_3	°F.	1.5	2.2	2.5	3.5	4.7
Q_3	(B.t.u.)/(hr)	3,800	5,550	6,250	8,820	11,850
ΔT_4	°F.	3.2	2.8	3.8	5.1	4.3
Q_4	(B.t.u.)/(hr)	8,100	7,000	9,510	12,850	10,820
w_3	(lb)/(min.)	6.1	6.5	6.5	6.7	6.3
ΔT_5	°F.	20.7	19.1	21.5	27.0	30.1
Q'_4	(B.t.u.)/(hr)	7,580	7,450	8,400	10,850	11,380
Q_5	(B.t.u.)/(hr)	32,700	38,100	45,560	53,650	56,970
M	(lb)/(hr)	1.18	1.34	1.67	2.07	2.38
Q_6	(B.t.u.)/(hr)	23,200	26,400	32,900	40,750	47,000
F		1.408	1.441	1.385	1.315	1.210
w_4	(lb)/(hr)	307	371	421	446	470
S_e	r.p.m.	504	625	751	870	992
W_r	(B.t.u.)/(hr)	3,870	4,960	6,550	7,330	8,220
W_s	(B.t.u.)/(hr)	6,410	7,960	9,570	11,070	12,630
E_w	percent	60.3	62.3	68.4	66.2	65.0

TABLE VI

EXPERIMENTAL RESULTS

Source Inlet Temperature 10°F.

DESCRIPTION		TRIAL NUMBERS				
		26	27	28	29	30
ΔH_1	(B.t.u.)/(lb)	54.90	53.90	55.20	55.00	55.15
ΔH_2	(B.t.u.)/(lb)	68.50	69.35	71.10	72.00	72.00
ΔH_3	(B.t.u.)/(lb)	13.20	14.65	15.90	16.95	16.60
COP		5.19	4.73	4.47	4.25	4.33
w_1	(lb H_2O)/(min.)	75.4	80.3	86.1	80.0	80.0
k	(B.t.u.)/(lb °F.)	0.961	0.961	0.961	0.931	0.931
ΔT_1	°F.	3.2	3.8	4.2	4.3	5.1
Q_1	(B.t.u.)/(hr)	13,900	17,600	20,800	19,200	22,800
Q'_1	(B.t.u.)/(hr)	14,140	17,750	20,100	21,700	22,650
w_2	(lb)/(min.)	42.0	41.5	41.7	42.0	42.0
ΔT_2	°F.	7.0	9.0	10.3	11.3	11.7
Q_2	(B.t.u.)/(hr)	17,650	22,400	25,800	28,450	29,500
ΔT_3	°F.	1.6	2.2	2.8	3.9	4.1
Q_3	(B.t.u.)/(hr)	4,030	5,480	7,010	9,820	10,320
ΔT_4	°F.	3.4	4.0	3.6	4.8	5.4
Q_4	(B.t.u.)/(hr)	8,570	9,950	9,020	12,100	13,600
w_3	(lb)/(min.)	6.6	6.6	6.4	6.6	6.4
ΔT_5	°F.	21.7	24.2	25.4	30.0	33.4
Q'_4	(B.t.u.)/(hr)	8,600	9,570	9,760	11,900	12,800
Q_5	(B.t.u.)/(hr)	30,250	37,830	41,830	50,370	53,420
M	(lb)/(hr)	1.26	1.50	1.75	2.30	2.64
Q_6	(B.t.u.)/(hr)	24,800	29,500	34,400	45,300	52,000
F		1.220	1.282	1.215	1.112	1.028
w_4	(lb)/(hr)	258	323	363	395	410
S_e	r.p.m.	497	630	748	865	963
W_r	(B.t.u.)/(hr)	3,400	4,730	5,760	6,690	6,800
W_s	(B.t.u.)/(hr)	6,305	8,010	9,500	11,000	12,260
E_w	percent	53.8	59.0	60.6	60.6	56.3

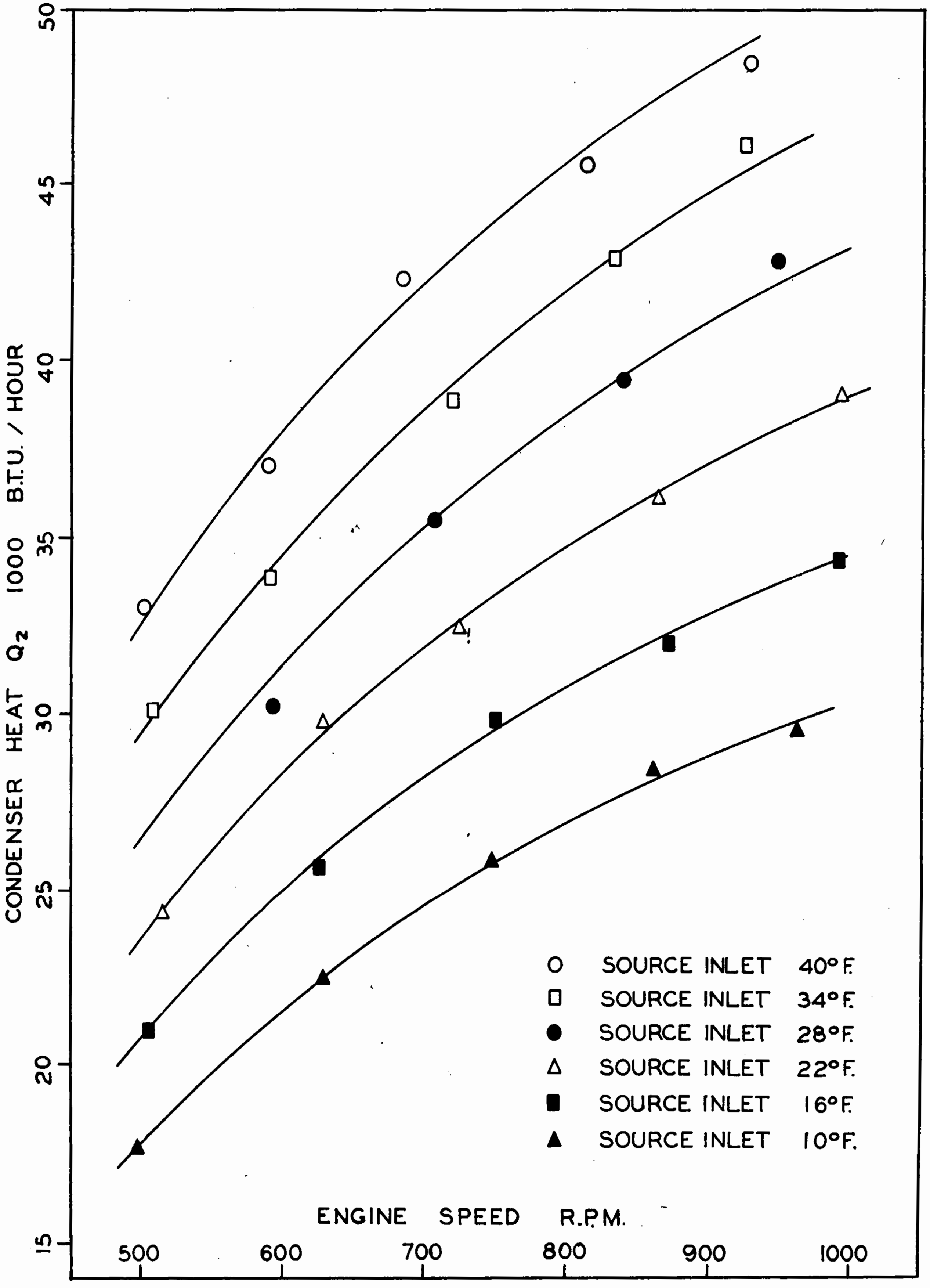


FIGURE 9 CONDENSER HEAT

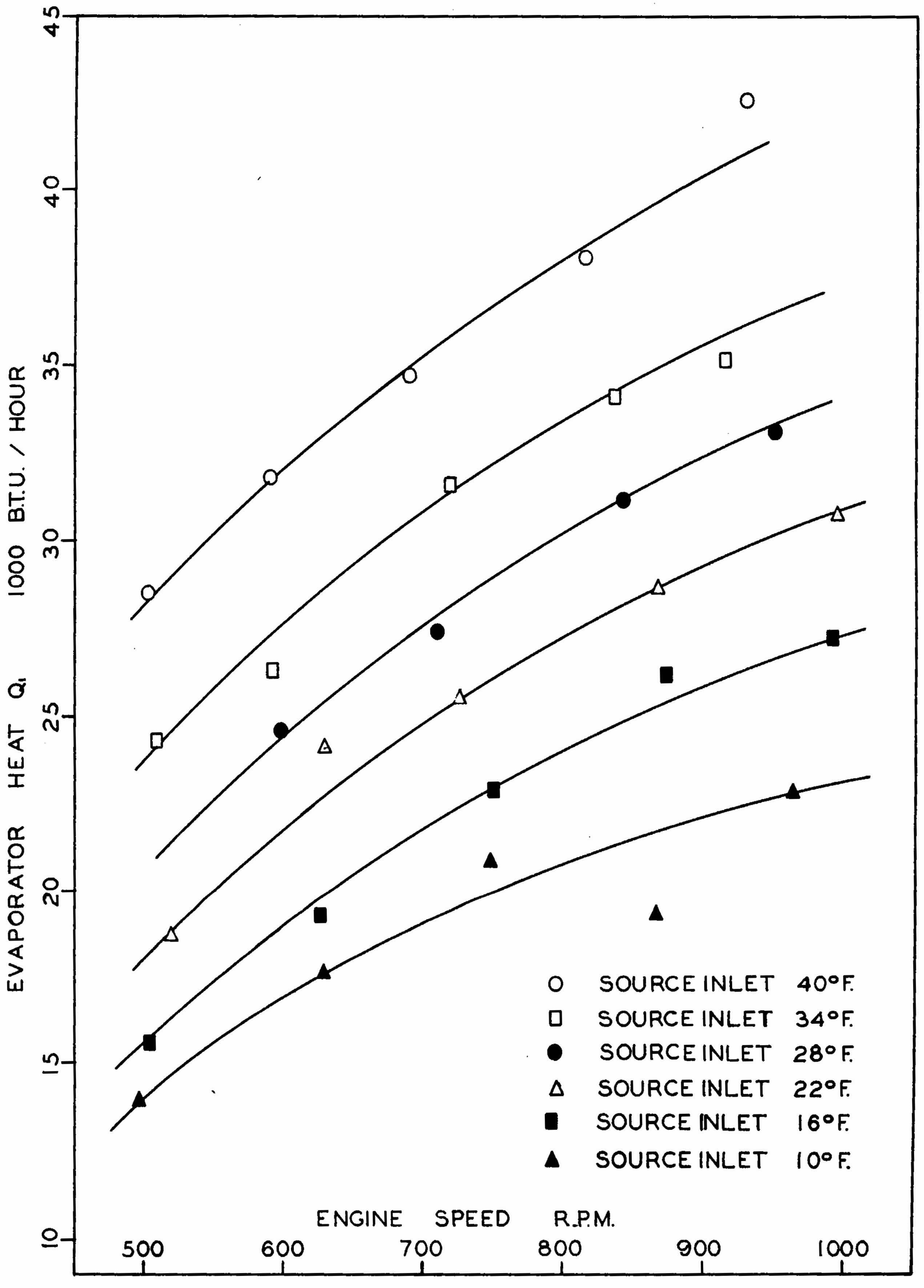


FIGURE 10 EVAPORATOR HEAT

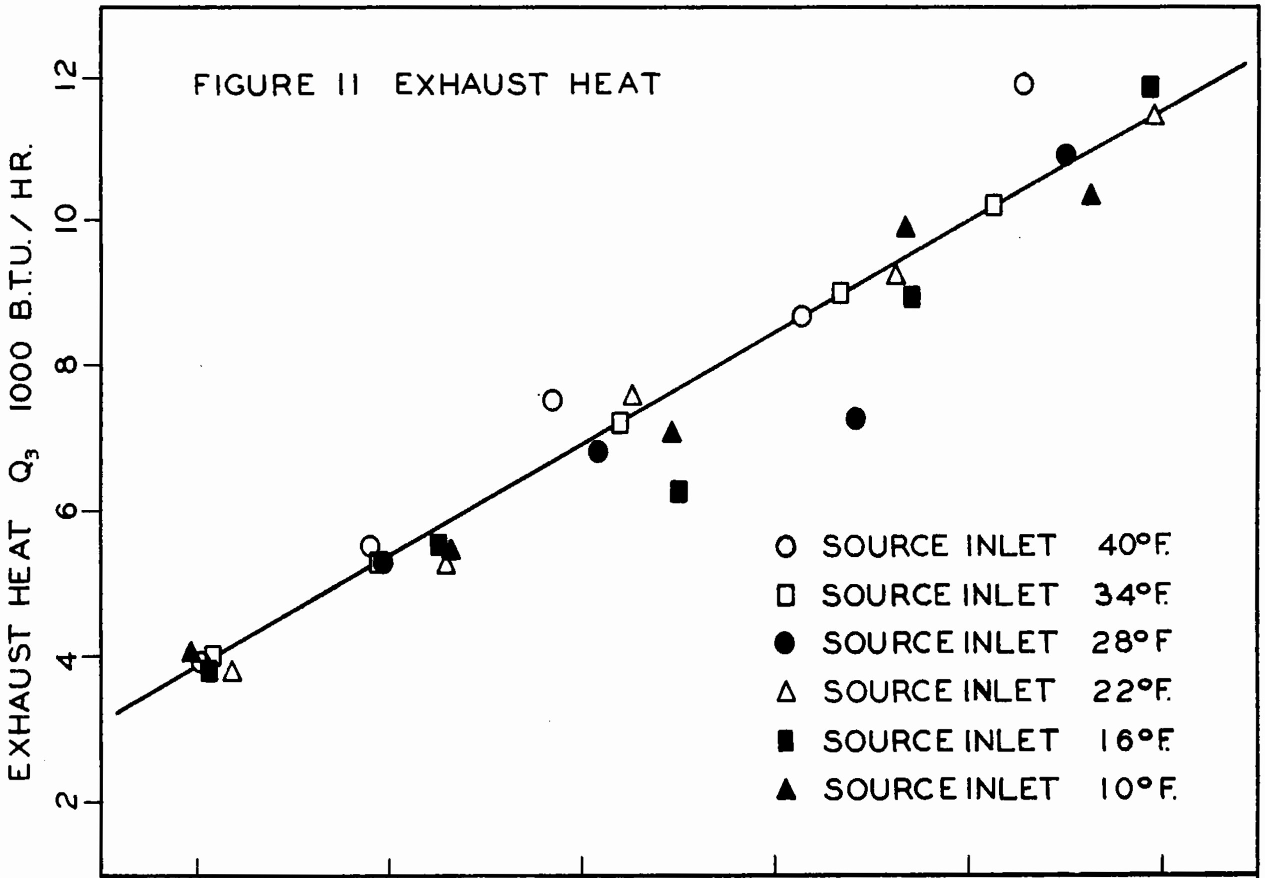


FIGURE 11 EXHAUST HEAT

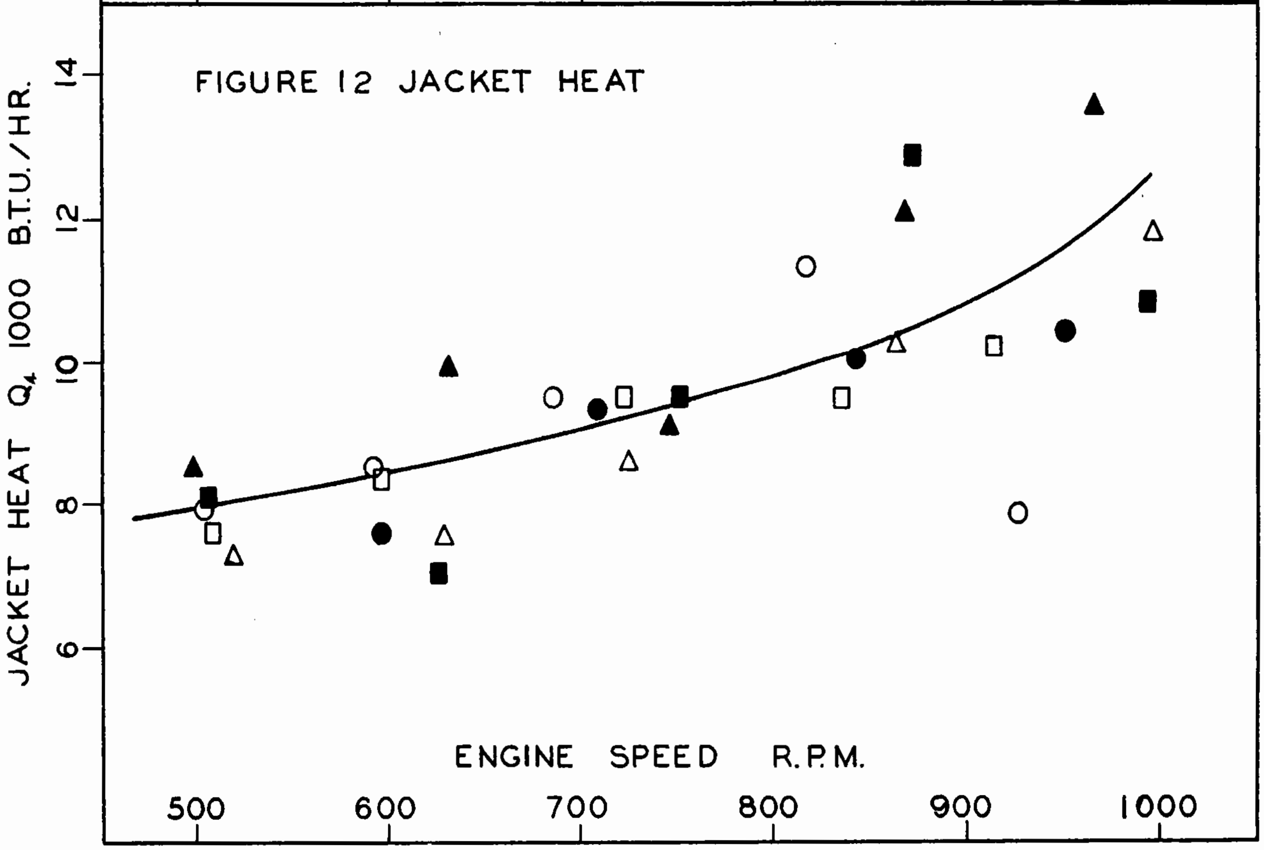


FIGURE 12 JACKET HEAT

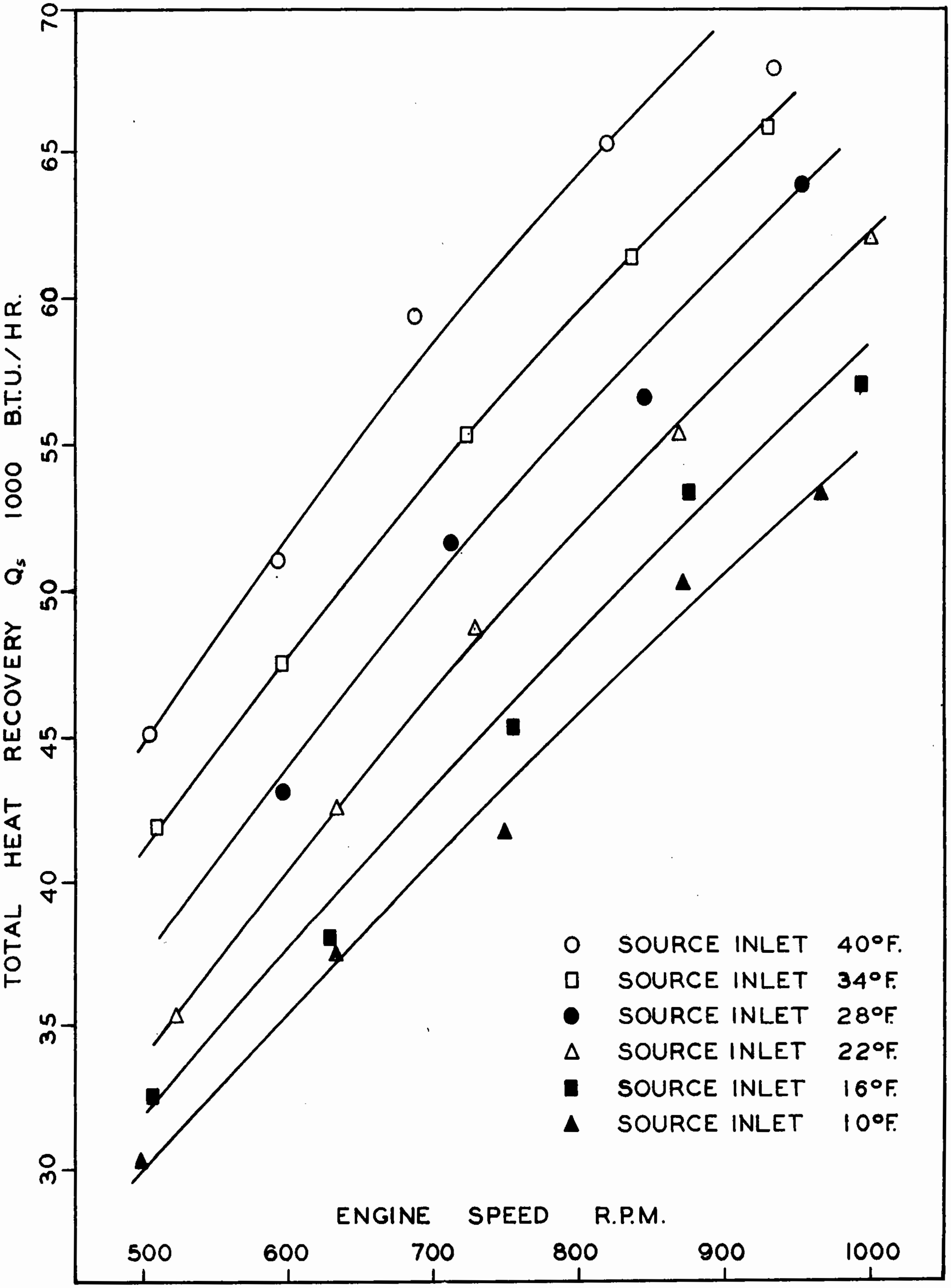


FIGURE 13 TOTAL HEAT RECOVERY

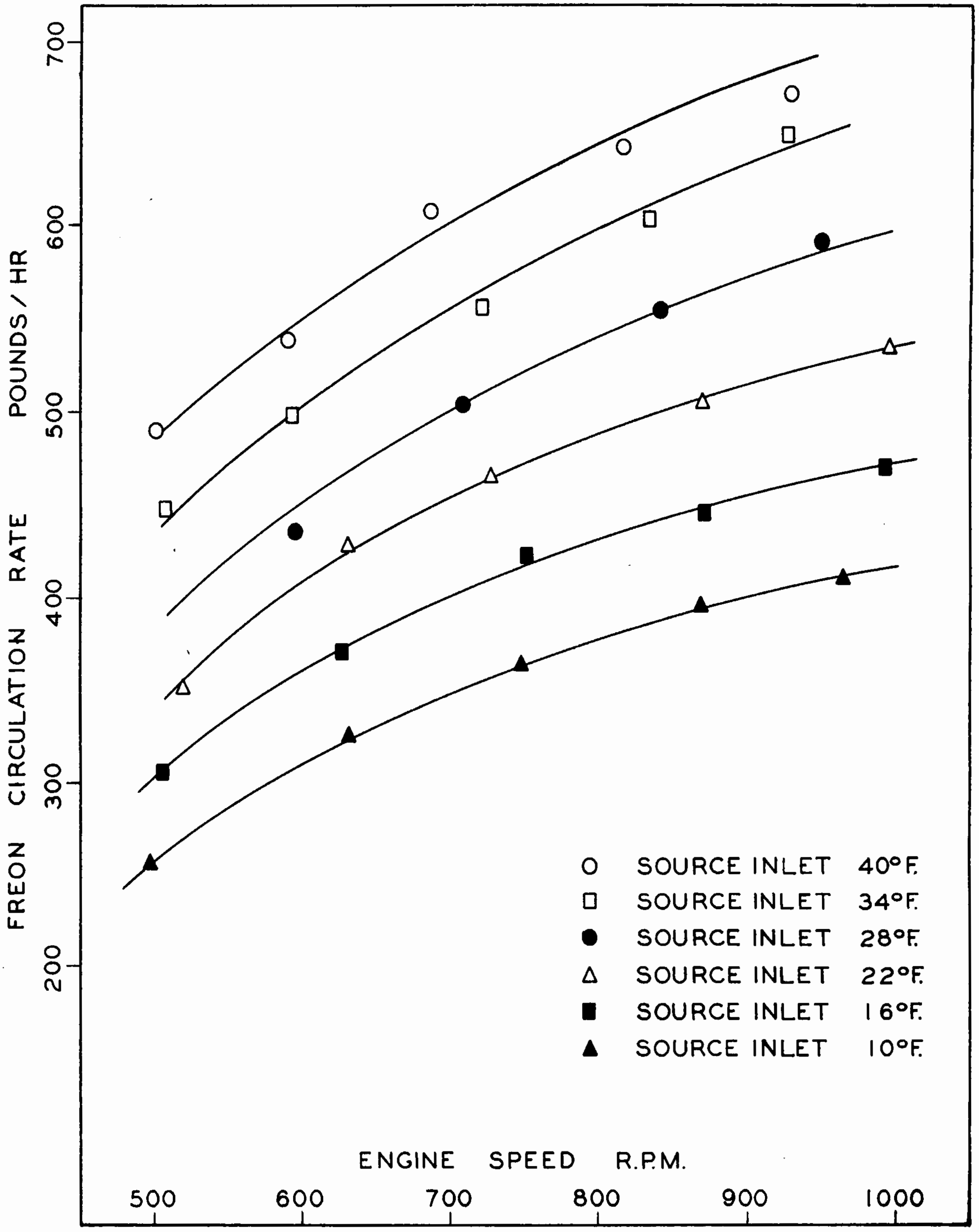


FIGURE 14 FREON CIRCULATION

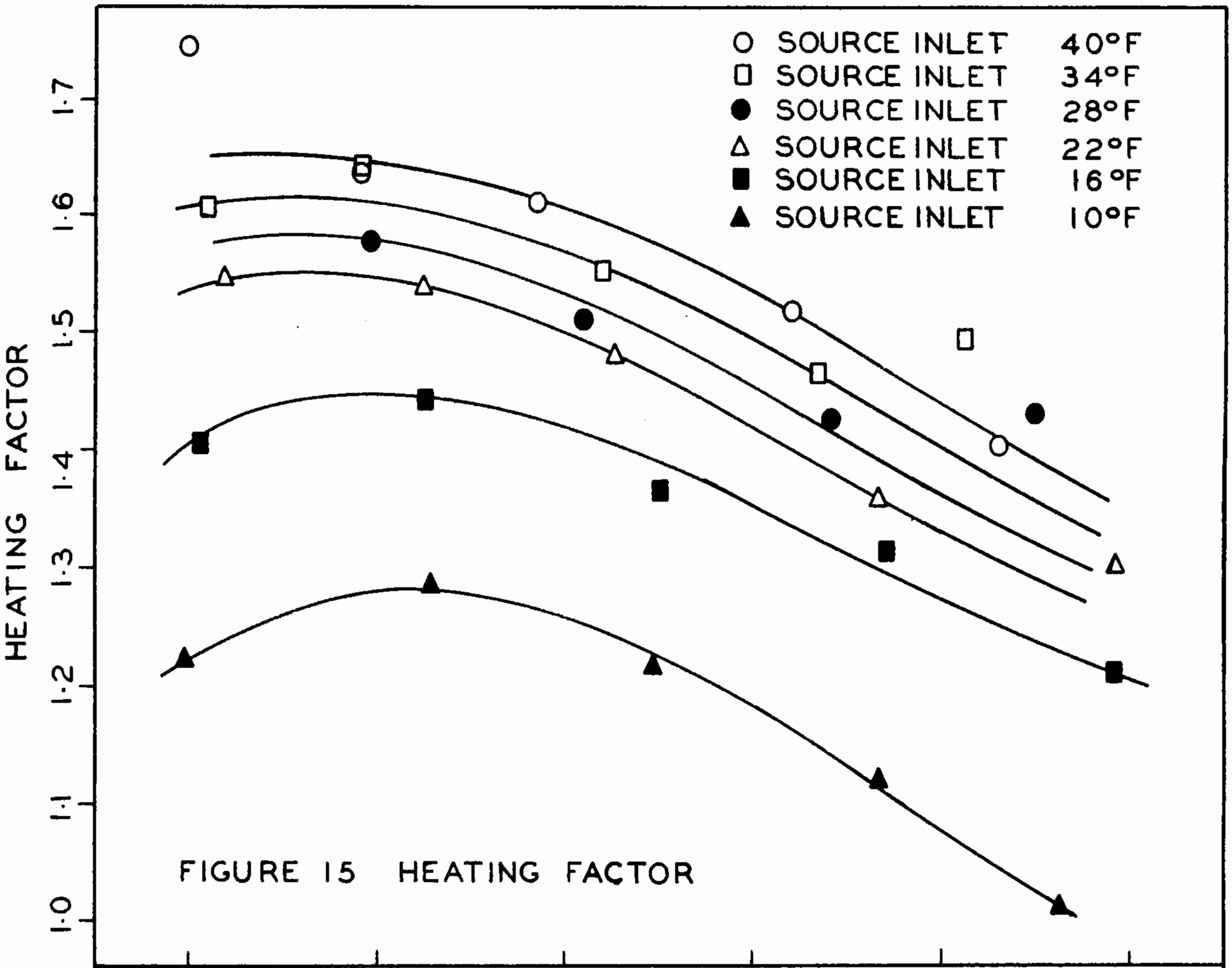


FIGURE 15 HEATING FACTOR

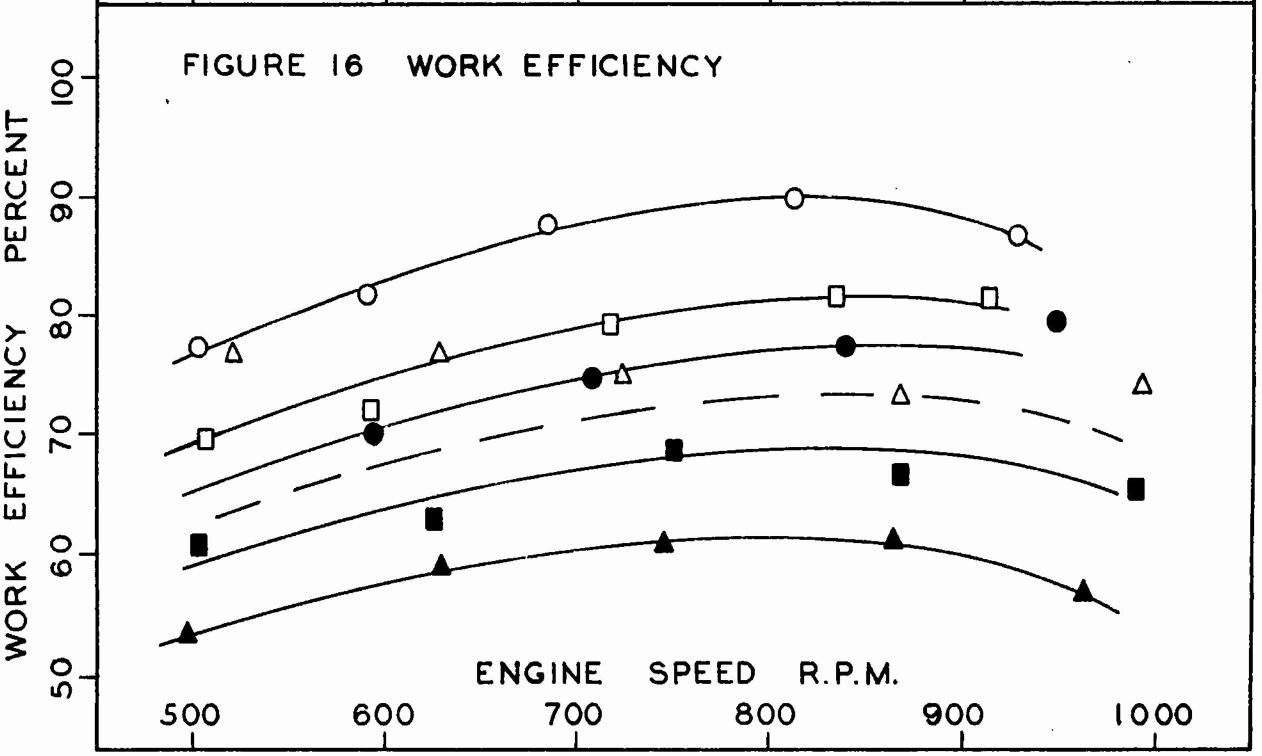
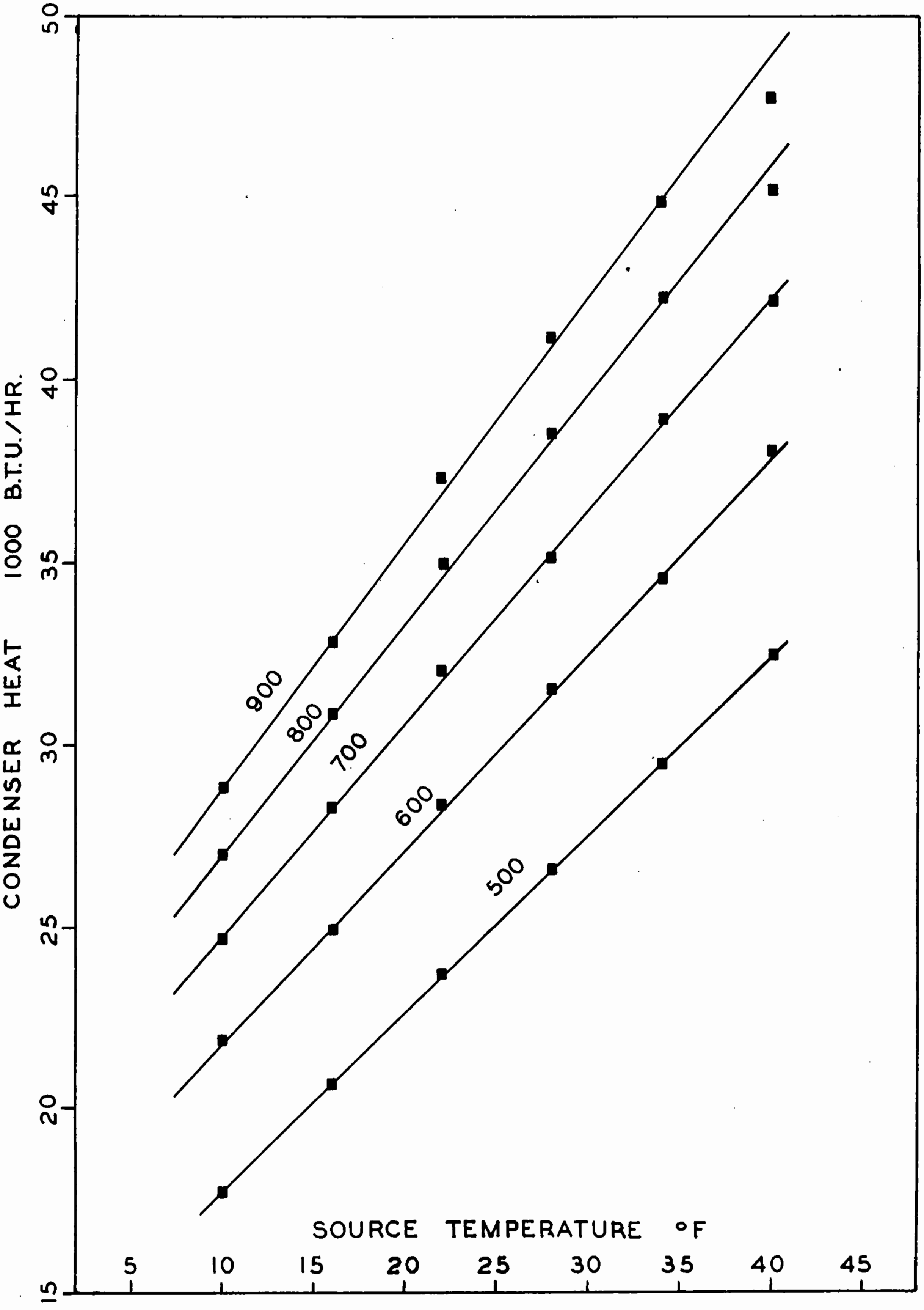


FIGURE 16 WORK EFFICIENCY



CONDENSER HEAT WITH SOURCE TEMPERATURE

FIGURE 17

CONCLUSIONS

The interpretation and evaluation of the operating characteristics of heat pump installations are frequently made very difficult because of the narrow range of operating conditions investigated or of the specific nature of the heat source used. In the present study, every effort was made to make the installation independent of the nature of the heat source, and emphasis was placed only on its temperature. Source temperatures from 10°F. to 40°F. were selected as representative of the range likely to be encountered in this area. The effect of source temperature was studied in six degree steps over the complete interval giving a good coverage of the available temperatures. In actual operation, the unit was found to have good flexibility and excellent response and, with the exception of a few minor difficulties previously mentioned, the controls and measurements proved to be quite adequate.

The experimental results indicate that although the major portion of the heat delivered is provided by the condenser, the direct recovery of waste heat from the Diesel engine is quite appreciable. This contribution becomes more significant as the source temperature falls. A better appreciation of this important conclusion can be obtained from a quantitative comparison of the results. Taking trial seven as a representative example, obtained at a source temperature of 34°F., and 592 r.p.m., it is seen that of the total recovery of 47,500 (B.t.u.)/(hr), 28.6% is provided by waste Diesel heat. Trial ten at the same source temperature but at 913 r.p.m. shows an increase in the waste heat recovery to 30.8% of the total. At a source temperature of 10°F., the corresponding values of the Diesel heat recovery are 41% and 49.5% in runs 27 and 30, respectively. To realize the significance of these figures, a brief glance at the economics of a Diesel heat pump is required. The heat introduced into the Diesel engine is provided in the form of Diesel fuel at a cost comparable with domestic fuel oil. Based on the distribution obtained with this experimental unit, this can be accounted for as work (25 - 30%), heat to the cooling water (25%), heat to the exhaust gases (20%) and as other losses (25 - 30%). By using the work developed to drive a heat pump, an additional heat premium is obtained at no cost, which is delivered to the condenser along with a portion of the heat supplied as shaft work. Assuming for the moment an operating COP of 4, the 25 work units are capable of delivering 100 heat units to the condenser. When added to the 45 units recovered from the Diesel waste heat, this leads to 145 units of heat delivered for each 100 units that have initially been purchased. A Heating Factor of 1.45 is thus obtained. Although the heat recovered from the Diesel engine represents a considerable portion of the delivered heat, it should not be overlooked that the total direct heat recovery (work plus waste heat) accounts for only about 70 of the 145 delivered heat units, the remainder, or 75 units, being supplied by the heat pump system proper.

The actual Heating Factors obtained varied from 1.28 to 1.64 in the range of peak efficiency, which figures represent operating Heating Factors exclusive of auxiliaries. A comparison of Diesel and electrical heat pump costs at this point would be instructive. Again considering trial seven, the Diesel unit operated with a Heating Factor of 1.64, which is equivalent to an operating cost of approximately 60 cents per million B.t.u. with Diesel fuel at 17.5 cents per gallon. Based on the results of this run, an electrical unit, in order to provide the same output at like temperatures, would require a condensing temperature of 115°F. with subcooling to 97°F. possible. Under these conditions a COP of 4.77, based on the refrigerant, is estimated. With electricity at one cent per kw-hr and 75% overall efficiency, the comparable cost of an electrically driven unit becomes 82 cents per million B.t.u.

Apart from the operating economies involved, the Diesel unit presents the advantage of continuous capacity control over a considerable range. In the series of trials at 34°F. source temperature, the delivered heat varied from 42,000 (B.t.u.)/(hr) at 500 r.p.m. to 72,000 (B.t.u.)/(hr) at 1,000 r.p.m., representing considerable flexibility. This flexibility is not without some disadvantage for as the capacity is increased to higher levels a decrease in the Heating Factor and economy results. Heavy loads would, however, be of short duration. Similarly the unit can be run at reduced capacity, but this, too, is at the expense of reduced Heating Factor, and though of longer duration, the total cost would be small compared to average load conditions.

Since only 60 - 70% of the total heating requirements are supplied by the refrigeration unit, an immediate saving in equipment size is apparent. A comparison of a Diesel and electrical unit based on the results of trial seven will serve as an example.

	Evaporator Duty (B.t.u.)/(hr)	Condenser Duty (B.t.u.)/(hr)	Waste Heat (B.t.u.)/(hr)	Total (B.t.u.)/(hr)
Diesel Unit	28,000	34,000	14,000	48,000
Electrical Unit	38,000	48,000	-	48,000

This 29% reduction in condenser load would result in considerable saving in the entire cost of the refrigeration unit, however two additional heat exchangers for waste heat recovery would be required. Predicated on a comparison such as this, it has been stated in the literature and previously quoted herein, that the installed cost of a Diesel unit would still be in excess of an electrically driven unit. Based on comparable design loads this is probably quite true but the possibility of continuous capacity control over a wide range of load requirements introduces additional economies that may ultimately favour Diesel-operated units.

An electrically driven heat pump must be operated at rated capacity, and in order to satisfy the probable peak loading it must be designed for the maximum load requirements. For the series of runs at a source temperature of 34°F., a 9 hp. installation would be required to satisfy the peak load of 72,000 (B.t.u.)/(hr), while evaporator and condenser would be designed to handle a duty of 72,000 (B.t.u.)/(hr) and 52,000 (B.t.u.)/(hr) respectively, and under all load requirements these conditions would apply. For a Diesel unit, a compressor rated normally at 3 hp. and an engine with an operating range of 2 to 5 hp., rated probably at 3.5 hp., would be sufficient. This unit would operate under average or reduced load for a large portion of the time but would be capable of supplying heavy loads of short duration when called upon to do so. This represents a considerably increased economic advantage not previously considered seriously, and should be appreciably in excess of the cost of the additional equipment required. In areas where the heating requirements vary over an extremely wide range, the full capacity on-off cycling operation of an electrical unit would produce undesirable fluctuations in the temperature of the conditioned space under light load conditions. Control of this temperature between narrow limits would result in short-cycling of the unit with its unfavourable demand characteristics. To avoid these conditions some capacity control would probably be required. This would necessitate expensive multiple units or multiple cylinder units with automatic cylinder unloading. Under these conditions the ability of the Diesel unit to provide continuous capacity control over a wide range, which is an inherent property of this type of unit, may be of considerable advantage.

Actual comparison of the first cost of Diesel and electrical installations is difficult but in general terms it can be stated that economies favouring the Diesel installation can be realized in the entire refrigeration installation including compressor, lines, condenser and evaporator, while two additional heat exchangers are required. Auxiliaries for the transfer of source and sink mediums would be comparable. A saving in moving the source medium is possible but an additional pump for transfer of cooling jacket water would probably be necessary. For comparable outputs very little saving in motor cost could be credited to the electrical unit.

For normal capacity control, throttle control on the Diesel is sufficient, but for in-between seasons on-off control would be required. In change-over from the heating to the cooling cycle, disposal of Diesel heat is necessary, and while the exhaust gases may merely be by-passed, the jacket heat must be removed by other means.

No economic comparison of the control systems can be made at this time, for the complexity of complete Diesel control has yet to be determined.

Noise and vibration are two serious disadvantages in a Diesel installation but it is possible that these can be satisfactorily overcome. At present it would seem that application of Diesel driven heat pumps would be restricted to larger installations but future developments may overcome this; however, greater care and maintenance would be required.

In summary; a Diesel driven heat pump can be operated economically, producing heat at a cost of approximately 60 cents per million B.t.u., exclusive of auxiliaries, compared with a cost of 82 cents per million B.t.u. for an electrical unit. Continuous capacity modulation is possible by throttle control. Twenty-five to forty percent of the total heat delivered is obtained from waste heat recovery. Major equipment economies are possible and these may be sufficient to result in a net saving in a Diesel installation. Considerable flexibility is an inherent characteristic of the unit but noise and vibration and the lack of complete push-button control may be serious deterrents to its use.

This study has presented a preliminary evaluation of the operating characteristics of a Diesel driven heat pump and the results obtained indicate that this unit possesses many attractive features. This study is by no means complete and many questions remain to be answered before unqualified recommendations can be made; however, it is hoped that the advantages revealed in this work will stimulate interest in this device and lead to the development of new fields of heat pump engineering.

APPENDIX

EXPERIMENTALLY OBSERVED DATA

AND ENTHALPY OF REFRIGERANT

TABLE A-I

EXPERIMENTAL DATA

Source Inlet Temperature 40^{o}F.

DESCRIPTION			TRIAL NUMBERS				
			1	2	3	4	5
Refrigerant:							
	T_1	^{o}F.	31.6	30.3	27.8	26.1	24.7
	T_2	^{o}F.	75.0	75.8	77.1	79.1	78.1
	T_3	^{o}F.	121.4	131.2	141.4	153.7	159.7
	T_4	^{o}F.	74.4	75.5	76.8	78.5	77.6
	T_5	^{o}F.	35.7	35.1	32.9	33.0	32.2
	T_6	^{o}F.	36.6	35.7	35.1	34.1	34.2
Source Water:							
	T_7	^{o}F.	39.8	39.9	39.9	39.9	39.7
	T_8	^{o}F.	34.3	34.0	33.5	32.5	32.1
Sink Water:							
	T_9	^{o}F.	63.9	63.8	64.2	64.1	64.0
	T_{10}	^{o}F.	77.2	78.5	79.2	80.9	81.8
	T_{11}	^{o}F.	78.8	80.7	81.9	84.1	86.2
	T_{12}	^{o}F.	82.0	84.1	85.3	88.3	89.1
Jacket Water:							
	T_{13}	^{o}F.	93.4	95.9	98.6	101.9	103.2
	T_{14}	^{o}F.	111.5	117.5	123.5	119.8	120.8
	T_{15}	^{o}F.	94.0	97.1	99.9	103.2	105.6
Exhaust Gas:							
	T_{16}	^{o}F.	412	480	547	602	608
	T_{17}	^{o}F.	77.0	77.8	79.5	82.5	83.7
General:							
	P_1	p.s.i.a.	45.4	43.7	41.3	38.8	37.2
	P_2	p.s.i.a.	122.9	134.8	142.0	153.3	161.9
	w_1	(lb)/(min.)	82.0	85.4	87.6	84.6	90.0
	w_2	(lb)/(min.)	41.5	42.0	46.8	45.1	45.1
	w_3	(lb)/(min.)	6.1	6.0	6.5	8.5	8.3
	S_e	r.p.m.	501	589	685	814	930

TABLE A-II

EXPERIMENTAL DATA

Source Inlet Temperature 34°F.

DESCRIPTION			TRIAL NUMBERS				
			6	7	8	9	10
Refrigerant:							
	T_1	°F.	26.0	25.1	22.9	21.3	21.2
	T_2	°F.	75.5	76.7	76.1	77.3	78.7
	T_3	°F.	117.2	126.0	140.1	150.5	154.9
	T_4	°F.	74.5	75.7	76.0	77.0	77.8
	T_5	°F.	32.0	30.7	27.3	27.6	28.6
	T_6	°F.	34.4	33.3	30.4	30.1	31.4
Source Water:							
	T_7	°F.	34.4	34.4	34.3	34.3	34.5
	T_8	°F.	29.7	29.7	28.5	28.1	28.0
Sink Water:							
	T_9	°F.	64.1	64.3	63.9	64.3	64.1
	T_{10}	°F.	76.0	77.7	79.4	81.5	82.6
	T_{11}	°F.	77.6	79.8	82.3	85.1	86.7
	T_{12}	°F.	80.6	83.1	86.1	88.9	90.8
Jacket Water:							
	T_{13}	°F.	91.9	96.0	99.0	101.9	103.1
	T_{14}	°F.	107.9	116.7	123.4	126.0	124.2
	T_{15}	°F.	92.3	96.0	100.5	102.3	104.4
Exhaust Gas:							
	T_{16}	°F.	383	447	530	582	608
	T_{17}	°F.	75.2	76.9	79.9	83.2	83.6
General:							
	P_1	p.s.i.a.	40.8	39.9	37.5	36.3	35.3
	P_2	p.s.i.a.	114.7	123.1	140.7	148.1	156.5
	w_1	(lb)/(min.)	79.6	86.2	85.2	86.0	84.4
	w_2	(lb)/(min.)	42.2	42.2	41.7	41.6	41.5
	w_3	(lb)/(min.)	6.4	6.0	5.8	6.1	6.3
	S_e	r.p.m.	510	592	718	835	913

TABLE A-III

EXPERIMENTAL DATA

Source Inlet Temperature 28°F.

DESCRIPTION			TRIAL NUMBERS			
			12	13	14	15
Refrigerant:						
	T_1	°F.	18.7	17.4	17.3	15.1
	T_2	°F.	73.4	75.3	75.2	75.6
	T_3	°F.	128.4	141.6	147.5	156.7
	T_4	°F.	73.6	74.9	74.7	75.0
	T_5	°F.	23.6	21.1	21.8	20.4
	T_6	°F.	27.4	23.5	23.9	22.8
Source Water:						
	T_7	°F.	27.8	28.0	28.2	27.8
	T_8	°F.	23.3	23.3	22.7	21.9
Sink Water:						
	T_9	°F.	63.8	64.7	63.8	64.1
	T_{10}	°F.	75.7	78.8	79.5	81.0
	T_{11}	°F.	77.8	81.5	82.4	85.3
	T_{12}	°F.	80.8	85.2	86.4	89.4
Jacket Water:						
	T_{13}	°F.	90.5	95.8	98.4	100.6
	T_{14}	°F.	107.8	117.0	124.5	126.2
	T_{15}	°F.	91.0	96.8	98.7	101.0
Exhaust Gas:						
	T_{16}	°F.	436	517	564	592
	T_{17}	°F.	73.0	76.9	78.6	82.9
General:						
	P_1	p.s.i.a.	35.2	33.5	33.9	31.4
	P_2	p.s.i.a.	118.7	133.2	142.0	149.8
	w_1	(lb)/(min.)	82.5	87.5	86.1	86.1
	w_2	(lb)/(min.)	42.3	42.0	41.8	42.2
	w_3	(lb)/(min.)	6.3	6.8	6.0	6.2
	S_e	r.p.m.	596	708	840	950

TABLE A-IV

EXPERIMENTAL DATA

Source Inlet Temperature 22°F.

DESCRIPTION			TRIAL NUMBERS				
			16	17	18	19	20
Refrigerant:							
	T_1	°F.	14.5	12.5	12.5	10.6	9.0
	T_2	°F.	80.1	76.5	75.0	75.5	75.3
	T_3	°F.	121.8	131.2	138.1	149.1	160.4
	T_4	°F.	79.1	75.9	74.2	74.6	74.7
	T_5	°F.	20.3	17.4	13.6	14.7	16.4
	T_6	°F.	22.2	23.0	13.9	18.3	17.7
Source Water:							
	T_7	°F.	21.4	21.2	21.6	21.0	21.5
	T_8	°F.	17.9	16.6	16.9	15.7	15.7
Sink Water:							
	T_9	°F.	64.4	64.4	64.6	64.6	64.6
	T_{10}	°F.	74.1	76.3	77.4	78.7	80.0
	T_{11}	°F.	75.6	78.4	80.4	82.3	84.5
	T_{12}	°F.	78.5	81.4	83.8	86.3	89.1
Jacket Water:							
	T_{13}	°F.	90.3	92.8	94.9	98.3	101.7
	T_{14}	°F.	109.0	114.2	119.6	126.3	134.6
	T_{15}	°F.	90.3	93.3	95.6	99.1	103.3
Exhaust Gas:							
	T_{16}	°F.	394	457	517	593	637
	T_{17}	°F.	72.9	75.2	76.7	76.5	81.7
General:							
	P_1	p.s.i.a.	31.9	30.8	30.8	28.9	28.5
	P_2	p.s.i.a.	100.2	109.5	126.2	134.7	144.7
	w_1	(lb)/(min.)	78.5	81.5	84.5	85.5	84.5
	w_2	(lb)/(min.)	41.8	41.7	42.3	42.7	42.2
	w_3	(lb)/(min.)	6.4	6.4	6.1	6.6	6.2
	S_e	r.p.m.	518	630	726	866	995

TABLE A-V

EXPERIMENTAL DATA

Source Inlet Temperature 16°F.

DESCRIPTION			TRIAL NUMBERS				
			21	22	23	24	25
Refrigerant:							
	T_1	$^{\circ}$F.	9.4	8.2	7.4	6.7	4.9
	T_2	$^{\circ}$F.	77.6	76.5	73.1	73.6	73.3
	T_3	$^{\circ}$F.	120.7	126.1	138.5	145.3	154.2
	T_4	$^{\circ}$F.	76.0	75.6	72.4	72.7	72.5
	T_5	$^{\circ}$F.	12.9	16.8	11.9	9.5	10.0
	T_6	$^{\circ}$F.	20.2	19.4	13.3	13.5	14.4
Source Water:							
	T_7	$^{\circ}$F.	15.5	16.0	15.8	16.3	15.7
	T_8	$^{\circ}$F.	12.9	12.5	11.8	11.5	10.6
Sink Water:							
	T_9	$^{\circ}$F.	63.8	63.8	63.9	63.9	64.1
	T_{10}	$^{\circ}$F.	72.0	74.0	75.8	76.6	77.7
	T_{11}	$^{\circ}$F.	73.5	76.2	78.3	80.1	82.4
	T_{12}	$^{\circ}$F.	76.7	79.0	82.2	85.2	86.7
Jacket Water:							
	T_{13}	$^{\circ}$F.	87.8	90.6	93.9	96.9	98.9
	T_{14}	$^{\circ}$F.	108.5	110.0	116.0	124.4	130.0
	T_{15}	$^{\circ}$F.	87.8	90.9	94.5	97.4	99.9
Exhaust Gas:							
	T_{16}	$^{\circ}$F.	383	432	492	597	665
	T_{17}	$^{\circ}$F.	70.0	71.8	73.1	78.9	79.8
General:							
	P_1	p.s.i.a.	28.7	28.1	27.9	26.8	24.9
	P_2	p.s.i.a.	95.7	102.7	120.2	127.7	132.7
	w_1	(lb)/(min.)	81.0	80.6	87.5	86.4	85.1
	w_2	(lb)/(min.)	42.2	41.7	41.7	42.0	42.0
	w_3	(lb)/(min.)	6.1	6.5	6.5	6.7	6.3
	S_e	r.p.m.	504	625	751	870	992

TABLE A-VI

EXPERIMENTAL DATA

Source Inlet Temperature 10°F.

DESCRIPTION			TRIAL NUMBERS				
			26	27	28	29	30
Refrigerant:							
	T_1	°F.	4.2	2.7	1.2	0.3	-0.8
	T_2	°F.	75.2	77.2	71.1	72.4	71.5
	T_3	°F.	120.6	129.6	135.5	143.6	142.5
	T_4	°F.	73.6	76.3	70.3	71.7	71.2
	T_5	°F.	9.0	6.4	6.1	5.4	5.1
	T_6	°F.	16.8	14.0	8.1	9.1	10.0
Source Water:							
	T_7	°F.	9.3	9.3	9.5	9.2	9.3
	T_8	°F.	6.8	6.1	6.0	5.5	4.9
Sink Water:							
	T_9	°F.	63.8	63.8	63.9	63.9	64.0
	T_{10}	°F.	70.8	72.8	74.2	75.2	75.7
	T_{11}	°F.	72.4	75.0	77.0	79.1	79.8
	T_{12}	°F.	75.8	79.0	80.6	83.9	85.2
Jacket Water:							
	T_{13}	°F.	88.1	91.1	92.4	98.2	99.6
	T_{14}	°F.	110.0	115.1	117.5	129.1	133.8
	T_{15}	°F.	88.3	90.9	92.1	99.1	100.4
Exhaust Gas:							
	T_{16}	°F.	412	475	518	649	671
	T_{17}	°F.	70.1	73.3	74.2	78.4	77.5
General:							
	P_1	p.s.i.a.	25.4	24.5	24.4	23.1	21.9
	P_2	p.s.i.a.	91.7	97.7	113.0	116.7	120.7
	w_1	(lb)/(min.)	75.4	80.3	86.1	80.0	80.0
	w_2	(lb)/(min.)	42.0	41.5	41.7	42.0	42.0
	w_3	(lb)/(min.)	6.6	6.6	6.4	6.6	6.4
	S_e	r.p.m.	497	630	748	865	963

TABLE A-VII

ENTHALPY OF REFRIGERANT

TRIAL NO.	ENTHALPY					
	H_1	H_2	H_3	H_4	H_5	H_6
Source Inlet 40°F.						
1	24.75	24.90	92.45	24.75	82.25	82.45
2	24.98	25.00	93.65	24.98	82.28	82.32
3	25.25	25.35	94.80	25.25	82.00	82.35
4	25.75	25.90	96.75	25.75	82.15	82.30
5	25.50	25.60	97.45	25.50	82.10	82.30
Source Inlet 34°F.						
6	24.00	24.10	92.90	24.00	82.00	82.30
7	25.00	25.20	93.20	25.00	81.80	82.30
8	25.00	25.00	94.95	25.00	81.55	81.90
9	25.15	25.35	96.35	25.15	81.55	82.00
10	25.40	25.80	96.35	25.50	81.65	82.05
Source Inlet 28°F.						
12	24.60	24.60	93.75	24.60	80.90	81.55
13	24.80	25.00	95.30	24.80	80.60	81.00
14	24.80	25.00	96.00	24.80	80.70	81.00
15	24.90	25.00	97.25	24.90	80.70	81.00
Source Inlet 22°F.						
16	25.90	26.10	95.35	25.90	80.65	80.90
17	25.00	25.20	94.55	25.00	80.20	80.20
18	24.70	24.95	95.00	24.70	79.65	79.85
19	24.60	25.00	96.55	24.60	79.90	80.50
20	24.80	25.00	98.15	24.80	80.15	80.40
Source Inlet 16°F.						
21	25.00	25.60	93.30	25.00	79.75	80.70
22	25.00	25.20	94.00	25.00	80.30	80.60
23	24.30	24.50	95.30	24.30	79.55	79.75
24	24.10	24.50	96.20	24.10	79.40	79.80
25	24.20	24.50	97.50	24.20	79.50	80.00
Source Inlet 10°F.						
26	24.40	25.00	93.50	24.40	79.40	80.30
27	25.10	25.30	94.65	25.10	79.00	80.00
28	23.80	24.00	95.10	23.80	79.00	79.20
29	24.00	24.30	96.30	24.00	79.00	79.35
30	23.90	24.00	96.00	23.90	79.05	79.40

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