A STUDY OF HEAT PUMPS

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ABSTRACT

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This report reviews the thermodynamic principles governing the operation of a heat pump and the various factors which influence its efficiency. The performance characteristics and the operating principles of air-to-air, air-to-water, water-to-water and solar assisted heat pumps are included in this report. A numerical model of a simple air-to air heat pump is derived from equipment manufacturers data and information obtained from published reports and papers. The report concludes that heat pumps principally developed in the U.S.A. have a reduced Seasonal Performance Factor when operated under Canadian climatic conditions. Also the influence of the defrost problem and the inherent inability of heat pumps to maintain their heating capacity as the source temperature falls, have been the main drawbacks for their acceptance as a viable heating system in the residential market. Indications are that the trend to solar assisted heat pumps will continue to grow, because of the unique advantages which they offer over conventional heat pump systems. However this growth will be slow particularly in eastern Canada where the high incidence of cloud cover during the winter months, requires the use of an expensive heat storage system and electric resistance heating systems as a back up.

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1. INTRODUCTION

The objectives of this report are to review the general theory and types of heat pump systems commercially available. To review the performance of the different types of heat pump systems in residential, commercial and industrial environments and the development of a modelling technique with a view of determining the suitability of a specific heat pump system for Canadian climatic conditions.

The heat pump is the name generally applied to a year-round conditioning system, in which the refrigeration equipment is employed to supply useful heat to the space during the heating cycle and to abstract unwanted heat from the space during the cooling cycle. When the heat pump operates as a heating system, the heat is taken from the outdoor air, water or other such low-temperature heat source and delivered together with the heat equivalent of the work of compression to the conditioned space.

Conversely when operating as a cooling cycle, the heat pump abstracts heat from the conditioned space and rejects it together with the heat equivalent of the work of compression to an outside heat sink.

There is no fundamental difference between the well-known vapour-compression refrigeration cycle and the heat pump cycle. Thermodynamically, both systems are heat pumps employing a compressor, condenser, cooling coils or evaporator, throttling valve, controls and piping in order to absorb heat at a low-temperature level and reject it at a higher temperature level. The main difference between the two systems is the primary objective of the application. A refrigeration installation is concerned with the low-temperature effect produced at the evaporator while the heat pump is concerned with both the cooling effect produced at the evaporator and the heating effect produced at the condenser.

Wherever possible in this paper, S.I. units have been employed. In Section 7 (Heat Pump Modelling) British Units are utilized, as the equations employed are based on specific equipment geometry, for which data in S.I. units is not readily available.

1.1. Thermodynamic Principle of the Heat Pump

Commercial refrigeration systems and heat pumps consist of a compressor used to raise the pressure and temperature of the refrigerant vapour, a condenser from which heat is extracted, a storage tank or receiver, an expansion valve used to lower the pressure from the high-pressure or condenser side to the low-pressure or evaporator side of the system, and an evaporator in which heat is absorbed by the refrigerant from some source. An additional major component is the reversing valve, the function of which is to reverse the flow of refrigerant in the system, so that the indoor coil acts as an evaporator on the cooling cycle and as a condenser on the heating cycle.

The defrost and cooling cycle of an air-to-air heat pump is shown in Fig. 1 and a graphical form of the cycle is best illustrated by the Mollier diagram in Fig. 3. Refrigerant is pumped to a high temperature and pressure (isentropic compression) and diverted to the outdoor coil by the reversing valve. A fan forces outdoor air over the coil removing heat and condensing the refrigerant, which then flows through the expansion valve (TXV) and into the low pressure indoor coil (evaporator). Room air forced over the indoor coil is chilled as heat is absorbed by the refrigerant in the evaporator. In absorbing heat the liquid refrigerant boils, becomes a vapour and flows to the compressor suction, whereupon it is re-compressed and the cycle is repeated.

On the heating cycle, the functions of the indoor and outdoor coils are reversed. Since the outdoor coil acts as an evaporator the leaving air is colder than the entering air, as illustrated in Figs. 2 and 4. At certain outdoor temperature conditions, frost will form on the outdoor coil impairing the efficiency and the operation. To remove the frost, the heating cycle is reversed back to the cooling cycle for a few minutes until the hot gas has melted the frost.

In Fig. 3 both cycles are illustrated using the Mollier diagram, beginning at A the compressor takes in the gas which is all vapour and compresses it at constant entropy to B. In the process it puts work WC into the gas and increases its enthalpy. From B to C heat is removed, i.e. rejected to the sink by the condenser. This is the heat rejection (H.R.) process. Since heat is removed the vapoour is condensed and is all liquid at C. From C to D it is throttled at constant enthalpy to the lower pressure in the evaporator. It enters the evaporator as a mixture of 75% liquid and 25% vapour (approx.). From D to A the evaporation process during which heat is added from the outside air. The change in enthalpy is the refrigeration effect (R.E.). As the refrigerant boils and absorbs heat, it vapourizes and enters the compressor as all vapour at point A, and the cycle is repeated.

An index of the performance of the cycle is the coefficient of performance (C.O.P.). By definition the actual coefficient of performance of a heat pump, during the heating cycle, is equal to the total instantaneous heat output (H.R.) divided by the heat equivalent of the net work required to produce the effect (WC). During the cooling cycle the C.O.P. is the ratio of the instantaneous refrigeration effect (R.E.) divided by the heat equivalent of the net work done in producing the effect (WC).

$$\text{C.O.P. (Heating)} = \text{H.R.} = \frac{RE + WC}{WC} = 1 + \frac{RE}{WC}$$

$$\text{C.O.P. (Cooling)} = \frac{RE}{WC}$$

$$\text{C.O.P. (Heating)} = 1 + \text{C.O.P. (Cooling)}$$

Thus the heat pump operating cycle has the inherent advantage of being more efficient than the cooling cycle at the same evaporator/ condenser conditions. The C.O.P. can also be expressed by the use of the absolute temperatures at the various stages of the cycles:

$$\text{C.O.P. (Heating)} = \frac{Th}{Th - Tc}$$

$$\text{C.O.P. (Cooling)} = \frac{Tc}{Th - Tc}$$

Where Th - Condensation Temperature

Tc - Evaporation Temperature

The C.O.P. is useful as a reference to indicate imporant influencing factors, but can never be remotely approached in practice. This is because isothermal compression or expansion cannot be accomplished practically and the low temperature gradient between the refrigerant and the ambient results in prohibitive equipment size and low mechanical efficiency.

1.2. Factors which Influence the Efficiency of the Heat Pump Cycle

1.2.1. Source Temperature

If the source temperature is lowered, the evaporating temperature would also be lowered, which can cause up to a 30% decrease in C.O.P. of the heat pump. Hence the greater the spread between the evaporating and condensing temperatures the poorer will be the performance of the heat pump. Control over the source temperature is limited, as these are fixed by the application.

However, to some degree the elevation of the condensing and evaporating temperatures of the heat pump above the sink and source temperature is determined by the performance of the heat exchangers. The condensing temperature which is determined by the energy balance between the condenser and the room air, which is being heated, can be altered by a modification of the heat exchanger design. If considerably more heat exchange surface is provided, the same amount of heat could be transferred at a lower temperature difference, which would balance out at a lowering of the condenser temperature. Conversely if the efficiency of the heat exchanger is low (too little heat transfer surface) then the condensing temperature would be increased, thus increasing temperature difference and thus reducing C.O.P.

The same argument applies to the evaporator performance, however, this approach will increase the cost of the equipment and would have to be justified on an economic basis. An example of this effect is demonstrated in Fig. 4.

1.2.2. Superheat Vapour, Subcool Liquid

If the refrigerant leaving the evaporator of heat sink surface is in a superheated condition as illustrated by points 1' or 1" in Fig. 5, the refrigeration effect and the work of compression are both increased. The net effect is that the C.O.P. may increase very slightly and may decrease at higher suction temperatures, however, for practical reasons this is necessary so as to avoid damage to the compressor, due to liquid entrainment. Increasing the vapour temperature of the gas entering the compressor increases proportionally the temperature rise in the compressor head. This temperature increase in the compressor heat, caused mainly by the low transfer rate of the superheated vapour, results in an increased discharge pressure and an additional load on the condenser. With hermetically sealed compressor, a larger suction heating effect may result, if the gas is brought through the crankcase or near the motor windings.

The installation of a gas-liquid heat exchanger for the purpose of superheat of the vapour may not be justified on the basis of increased cycle efficiency, however, there are some applications which make the use of such a heat exchanger desirable:

(i) Installations using expansion valves require 8 to 10 degrees of superheat in the evaporator for satisfactory performance. The use of a heat exchanger may make it unnecessary to use the evaporator for superheating the gas, thus increasing the refrigeration capacity of the coil. The expansion valve operation is also improved by maintaining a better control of the hunting of the valve and by reducing the amount of refrigerant carried back to the compressor by entrainment.

(ii) Many installations have uninsulated suction lines outside the conditioned space. If the heat gain of such suction lines cannot be avoided, a heat exchanger can be used to increase the refrigeration effect by superheating the gas, which in turn subcools the liquid refrigerant. Thus the necessity for insulating the suction line is then eliminated.

If the liquid leaving the condenser is subcooled below the saturation temperature (from point 3 to 3 as shown in Fig. 5) the enthalpy after the throttling operation would be some point 4. In this case the refrigerating effect and the useful heat output per pound of refrigerant circulated are increased without any change in the work of compression. In contrast to the disadvantages resulting from superheating the suction line, the subcooling of the liquid refrigerant will always improve the heat output and the C.O.P. of the System during the heating cycle. With subcooling the horsepower per ton decreases and the

C.O.P. increases as the liquid refrigerant temperature decreases.

1.2.3. Compressor Performance

The type of compressor and the refrigerant employed to get maximum economcy depend to a large extent on the capacity required and the working pressures needed to satisfy the design conditions. Heat pump compressors are either of the balanced-compression or free-compression type. With balanced compression the necessary positive displacement is generally obtained by either a reciprocating or a rotary mechanism. In this type of compressor not all of the gas leaves the cylinder. There has to be some clearance between the cylinder at top dead center and the valve structure. Hence on the downstroke this trapped high pressure gas expands to the lower pressure occupies some of the cylinder volume, and limits the amount of suction gas which can be taken in. This effect is the volumetric efficiency (V.E.) and is primarily a fucntion of the compression ratio, (C.R.) the ratio of discharge to suction gas pressures. The higher the C.R. the lower the V.E. which results in less gas being pumped. In addition to this effect, as the evaporation temperature is lowered, the specific volume of the suction gas increases. Consequently the mass flow rate of the refrigerant pumped, which determines the heating and cooling capacity of the system is lowered. The positive displacement compressor is best suited for handling smaller volumes of gas with large compression ratios, and are generally selected for all air heat source sink systems and for small and medium sized water heat source sink units. In free compression a trubo-type or a jet-type compressor is used. A turbo compressor whether of the centrifugal or axial flow type is primarily suited for large fluid volumes and low compression ratios. Typical performance curves of a reciprocating compressor at various saturated suction refrigerant temperatures and outdoor temperatures during both the heating and cooling cycles are shown in Fig. 6. This compressor has a 100 ton cooling capacity at a 35°C outdoor temperature and a 5°C saturated scution temperature. It can be seen that during the heating cycle both the refrigerating effect and the corresponding heating output from the condenser reduce proportionally to the drop in outdoor temperature. Similarly during the cooling cycle the cooling capacity and the corresponding condenser output are also reduced as the outdoor temperature increases. The curves also illustrate the importance of liquid sub-cooling on the heating output of the heat pump.

1.2.4. Refrigerants

Some of the most important factors, which influence the choice of the refrigerant are their physical properties such as toxicity, odor, acidity, corrosiveness, lubricating characteristics, reactivity and danger from explosion. The choice of the type and design of compressor will be influenced by the compression ratio and the refrigerant volume factors. It is desirable to have the evaporator pressure as high as is feasible, while keeping the condensing pressure as low as possible. The compression ratio determines the volumetric efficiency of a positive displacement unit and influences the number of stages required in axial flow compressor. Also, the compression ratio in centrifugal compressors is affected by the vapour density of the refrigerant. The density is related to the molecular weight of the gas as well as to the operating temperature and pressures.

Refrigerant characteristics change with operating temperatures. Some refrigerants are advantageous for low refrigeration systems between -40^{0}C and -8^{0}C, whilst others are more favourable in the range of -8^{0}C to 8^{0}C. Common refrigerant oils break down in the 150^{0}C -250^{0}C range. Refrigerant stability at high temperatures is a problem, consequently it is usual practice to desig[illegible]ystems for a maximum compressor discharge temperature of 120^{0}C[illegible] A heat pump must be designed within the constraints described above, thus a system using refrigerant 22 in a reciprocating heat pump, could not operate between a 250^{0}C to 350^{0}C temperature range if oil break-down is to be avoided. This problem can be overcome by using a centrifugal compressor, where oil is not present in the refrigeration circuit. However, this type of compressor is designed for refrigeration tonnages of 200 tons or more.

Thus a refrigerant with a high specific volume is desirable for low tonnage systems, which led to the development of refrigerant 113, which has a specific volume at 5^{0}C saturated evaporating temperature 15 times that of refrigerant 22.

2. HEAT SOURCES AND SINKS

The choice of the heat source and sink is of primary significance because of the heat pump's dual function of providing heating and cooling. Under winter conditions, heat is abstracted from the source and delivered to the conditioned space. Under summer conditions the heat is removed from the conditioned space and discharged to the heat sink. The choice of the most practical medium for a particular application will be influenced by many factors, such as geographic location, climatic conditions, initial cost, availability and suitability. Practically any form of low level heat is readily applicable to the heat pump cycle, but the majority of the systems are outdoor air, or well water as the source and sink. Other possible media include the ground, solar energy and industrial processes.

2.1. Air

One of the universal sources of heat, and perhaps the most common for heat-pump usage is the surrounding air. This medium is used to a large extent on residential installations and is fast becoming the principal selection forthe commercial and industrial applications. The selection or design of the air source heat pump components is affected by the outdoor temperature variation in a given locality. Many localities experience wide fluctuations in outdoor air temperatures and the heating requirements are always greatest when the outdoor temperature is lowest.

Fig. 7 shows the theoretical Carnot-Cycle heat pump efficiency for 35^{0}C delivered air and various air source temperatures. While these C.O.P. valves are much higher than can be realized in practice, it does serve to illustrate that the heat-pump cycle efficiency drops off rapidly as the source temperature increases.

The second curve shows the efficiency calculated on the basis of a 10-degree temperature differential in each of the heat exchangers. This loss greatly reduces the C.O.P. of the heat pump. Since heating requirements increase as the outside temperature drops, the air-to-air unit requires increased size in order to make up for this decrease in C.O.P. as the output requirements increase. Where heating is the controlling load on the head pump and where low outside temperatures are encountered, the air-to-air unit will require a larger horsepower capacity unit than the water-to-air unit.

The air-to-air unit is basically larger in size than the water-to-air unit because the air heat exchanger is larger than the liquid heat exchanger, and provision must be made to bring the outside air to, through and from the unit. Because the temperature drop in the air from which heat is extracted must be kept low to reduce frost formation on the outdoor air coil, which ultimately interferes with heat transfer, large volumes of air must be handled as illustrated in Fig. 8.

This introduces noise problems, and unless the unit is conveniently located to an outside air-source, it will require a considerable amount of power. Practical considerations would seem to limit the air-to-air unit to outside temperatures of from -8°C - -3°C. However, the capacity deficiency of an air source heat pump at the low outdoor temperatures can be neutralized by fully utilizing all methods of conserving heat, proper system design and by using supplemental resistance heating for the short infrequent cold spells during the winter period.

2.2. Water

Water from wells, lakes, rivers and manufacturing processes where the temperature is too low for direct utilization is often a very satisfactory heat source sink. The general advantage of using water as the source of heat is the fact that it is at essentially the same temperature during the entire heating season, which results in a more compact unit. By having a uniform source temperature, it is possible to build a unit designed to operate at capacity. As it will operate at this capacity independent of heating requirements, it is able to deliver at maximum capacity, when this capacity is most needed.

Although water would seem to have some advantages as a source of heat, it also has some distinct disadvantages. The water source must be chemically suitable, e.g. should not be excessively corrosive, and should not require extensive treatment or the use of expensive metals in the fabrication of the heat pump. Well water is often available in dependable quantities at depths of 46 - 180m and at favourable temperatures, usually above 10°C throughout the year. However, it is becoming increasingly difficult to obtain a dependable supply of trouble-free water in many parts of the country.

The disposal of the water after use and the local codes and restrictions do not constitute a difficult problem. In many cases, however, return wells are necessary or storm sewers are subject to a user's tax. Such conditions may make the system cost prohibitive for the heat pump is a once-through system and large quantities of water are required. The uncertainty of finding a sufficient quantity, at a given location, should not detract from its use, however, the cost of developing a well water supply for residential and small commercial installations is not economic, considering the return on investment. In certain industrial applications, use may be made of waste water as a heat source, e.g. warm water discharge from laundries to form large industrial condensers. Fig. 9 shows a plot of the volumetric flowrate of water used for various sizes of heat pumps for a 5, 8 and 11°C temperature difference.

2.3. Earth

The most universal source of heat is the earth, the only problem is in devising the necessary mechanisms to pick up the heat from the earth. Fig. 10 illustrates the large quantities of heat available. This figure shows for example that a cylinder of earth 9m in diameter and 30m in depth will give up 8,207kwh if its temperature is lowered only by 5°C. This amount of heat is sufficient for a five month heating season, with a heat pump delivering 9kw operating 30% of the time.

Many schemes have been proposed for extracting this heat from the earth. One of the first proposed is shown in Fig. 11. In this system a hole is drilled in the ground to a depth so that there will be a fluid level of about 60m. A closed "U" tube as shown in Fig.12 is inserted into the well. A pump is used to circulate water or other fluids through this pipe to pick up heat from the well, which in turn picks up heat from the earth. This system has many advantages over the circulation of the water itself. The fluid in the "U" tube can be inhibited to eliminate corrosion in the heat exchanger system and power requirements are kept to a minimum, since the only work to be done is that necessary to overcome friction in the system. One objection to this system is the expense involved in digging a well. However, this expense is offset by generally lower operating costs.

Another system adopted was the embedding of heat transfer surfaces, e.g. bundles of horizontal tubes in the earth. Refrigerants such as R12, R22 and brine were pumped through the pipes, which act as the evaporator of the heat pump system. After several years research work on this subject it was concluded that an average safe selection would be 60m to 120m of 22mm to 28mm O.D. pipe per ton of refrigeration effect, buried 1.2m apart at a depth of 1.2m (1). To insure continued contact between the coil and the ground material, the space around the coil was filled with sand. In spite of this precaution, it was found that after several years, the space between the earth and the pipe increased until the surface was no longer satisfactory as a heat sink. This adverse effect did not occur when the surface acts as a heat source, because the moisture in the ground migrates and freezes around the pipe to fill the void.

The heat transfer rates found for this horizontal grid of pipes varied during the heating cycle from 1.68 to 5.15 Btuh (ft) (^{O}F), and during the cooling cycle from 0.64 to 1.3 Btuh (ft) (^{O}F). One of the most important factors to be looked at when the earth is to be considered as a heat source, is the temperature at which heat can be obtained. Fig. 9 shows the approximate temperatures of water from wells at depths of 9 to 18m. At these depths there is in general a seasonal variation of not more than about 1^{O}C. Earth or ground-water temperatures at these depths generally exceed the mean annual air temperature by 1 - 2^{O}C. It can be seen that large sections of the U.S.A. has ground temperatures in excess of 15^{O}C. With this high temperature and allowing a reasonable drop such as 2 to 5^{O}C between the mean earth temperature and the heat exchanger fluid, very efficient heat pumps can be obtained. However, the initial expense in digging deep wells or in laying piping in the earth and the other factors such as corrosion protection of the equipment, etc., ensure that this type of heat pump can only be considered for very specialized applications.

2.4. Solar Power

Considerable interest has been shown in solar energy as a heat source, on a primary basis or in combination with other sources. For direct utilization of solar energy, numerous types of solar collectors have been used, however, most suffer from poor efficiency, as a result of their inability to capture a large percentage of lower intensity radiation, which occurs on cloudy days. The heat pump offers several possibilities for overcoming some of the present handicaps, facing the direct utilisation collector, because of its inability to absorb the solar heat at a relatively low collector temperature. Operating at these low collector temperatures reduces the transmission losses. The reduction in turn materially increases the collector efficiency. In general the principal attraction for using solar energy in this manner is the possibility in providing a higher temperature heat source than the other more common sources.

One drawback of a solar-heat pump system is that an alternate heat source or some means of heat storage must be used to provide heat during periods of insufficient solar radition. The average number of hours of sunshine as well as the percentage of possible sunshine hours, vary widely in the U.S.A., ranging in December from 45 hours and 16% for Erie, Pennsylvania to 254 hours and 82% (14) for Yuma, Arizona. Even under the most favourable conditions, therefore, solar energy can be expected to account for only about 34% of the total heating hours.

A typical example of a solar-heat pump is shown in Fig. 14. During the heating cycle the compressor delivers the high pressure, high-temperature refrigerant to the condenser-cooler, where it is liquidfied by giving up its latent heat of condensation to the water in the storage tank. The refrigerant liquid then changes into a low-temperature, low-pressure gas in the solar collector by obtaining the necessary latent heat of vapourization from solar radiation.

The surface temperature of the solar collector can be maintained at about 15°C or lower and that of the condenser-cooler at a refrigerant condensing temperature of about 50°C or higher. Warm water from the storage tank in turn is circulated to the conditioner coil to provide the necessary heating.

In summation it can be said that as a result of the energy crisis, a renewed interest arose in solar energy as a heat source. Due to the low cost of fossil fuel before the energy crisis, solar energy could not compete with it, if calcualted on the basis of cost per Kw produced. Moreover, since solar energy produces mainly low level energy, through heat conversion, this source of energy became less attractive. A significant amount of research and development is now being done to develop more efficient and less expensive solar-heat pump system components which will allow solar energy as a heat source to reach a high level of utilization in the foreseeable future.

2.5. Other Heat Sources

Heat pumps can also be used to extract low level heat from air, water, fluids, etc., which under normal industrial practices is reflected as "waste heat". Examples such as theuse of the cooling water on a thermal station has been used as the heat source to supply low level heat (25°C) for greenhouses in Russia.

In oil refineries and chemical plants, due to the high cost of energy, low temperature fluid streams, such as cooling tower water and final products, are now being utilized to supply heat to outbuildings on the site. Other uses of the heat pump are in the possible replacement of cooling water condensers for heated products in chemical plants.

Some of the above systems and means of screening such applications from an economical point of view is discussed in some detail in Section 5.

3. AIR-TO-AIR HEAT PUMPS

3.1. Principle of Operation

Air source systems are more common than water source systems due to the availability of air as compared to the expense involved in well drilling and water treatment. The major drawback to air as a source-sink is the inherent characteristic that the heat pump has less heating capability and lower overall efficiency during the coldest weather when the largest quantity of heat is required. When a system is sized for the summer cooling load, it will generally have insufficient heating capacity to match the building load at winter design conditions, particularly in northern climates. To make up this deficit, supplementary electric booster heaters are generally required. Fig. 13 and Fig. 14 illustrates the operation of an air-to-air heat pump.

On the cooling cycle high pressure liquid refrigerant (Fig. 13) high pressure refrigerant is fed to the indoor coil through an expansion device where it evaporates picking up heat from indoor air. Energing low pressure vapour passes through the vapour line, though the reversing valve to the compressor where it is brough to a high pressure vapour and sent back through the reversing valve to the condenser. This vapour gives up heat to the outside air and is condensed back into a liquid. It passes out of the condenser around the expansion device via a check valve and into the liquid line to complete the cycle.

To operate on the heating cycle, the slide in the reversing valve (Fig. 14) moves to the left as shown. The compressor then delivers high pressure vapour to the indoor coil where in condensing it, gives up heat to the indoor air. The liquid refrigerant by-passes the expansion device via the indoor check valve and is conducted to the outdoor expansion device through the liquic line. Here it is fed into the outdoor coil where it is evaporated at a low enough pressure and temperature to absorb heat from the outdoor air. The refrigerant emerges from the outdoor coil as low pressure vapour where it is directed by the reversing valve back to the compressor.

As frost accumulates on the outdoor coil during the heating cycle, it must be periodically removed to permit efficient operation of the heat pump system. To accomplish this the four way refrigerant valve is reversed to the cooling cycle and heat is removed from the indoor air by the compressor and discharged to the outside coil, whose fan is shutdown, until the frost is removed. During the defrost cycle, indoor supplementary heaters are generally energized at least partially to prevent a cooling effect in the conditioned spaces for the short duration of the defrost cycle.

3.2. Controls

The control system to be employed will depend on the operation desired, the performance required, and the type, size and design of the heat pump. The operation can range from manual to completely automatic. A control system can materially effect satisfaction and performance of the installation. It is very desirable to provide a ready means of switching from heating to cooling, automatic defrosting when air is used as a heat source and capacity modulation, particularly in the larger sizes of 20 tons and above.

When designing thecontrols for a heat pump, the following factors should serve as guidelines:

a) It is necessary that conditioned air which is circulated over the indoor coil be maintained at the designed quantity for proper operation on both cooling and heating. This is particularly true of the heating cycle to achieve economical operation and to prevent overload. The importance of keeping air filters clean becomes critical to this operation.

b) Many heat pump installations require supplementary heat to match the heating load at low outdoor temperatures and to temper indoor supply air during defrost cycles. In almost all cases this heat is supplied in the form of electric resistance heating and controls should be designed to minimize the use of the heaters for economy reasons.

A common way od doing this is by controlling strip heaters from the second stage of indoor room thermostats. The strip heater is not turned on until the temperature falls slightly indicating that the heat pump is not handling the load. Many times an outdoor thermostat is also added so that electric heat cannot be turned on until the outdoor temperature falls below the balance point of the system.

Heat pumps using outside air as a heat source must have some means provided to automatically defrost the outdoor coil if it is intended to operate at temperatures below about 10°C. The most universally used method of defrost for air-to-air heat pumps is to stop the outdoor fan and reverse the system to cooling cycle. The defrost period may take from one to six minutes, depending on conditions. During this period it common practice to bring on supplementary electric heat to maintain the supply air temperature at a comfortable level. A device is also necessary to sense the need for initiation of defrost. Some designs use a timer, which calls for a defrost at regular intervals. The interval will vary depending on design, and some units provide a means of adjusting the length of interval to suit climatic conditions in a given geographical area. An equally popular method of control is theuse of an air pressure switch, which senses and increase in static air pressure across the outdoor coil due to frost accumulation between fins (22, 28).

Another control is needed to terminate the defrost cycle. It is customary to stop the outdoor air fan during the defrost cycle, since continued operation would unnecessarily extend the defrost period or even prevent it from terminating entirely. With the outside circulation cut-off and hot gas from the compressor discharging into the coil, the defrost occurs in a matter of minutes. A convenient method for terminating the defrost cycle is to use a pressure control that will switch the system back to the heating cycle when the pressure has reached a pre-detemined level, usually equivalent to a condensing temperature of around 49^{0}C. This temperature can also be instrumental in terminating the defrost period by use of a rapid response thermostat in the coil tubes or line carrying liquid from the outside coil.

Ventilation air requirements and controls used should be very carefully considered in the design of any system using heat pumps. Since a need for cooling must exist before a heat pump can be considered feasible, the possibility of using ventilation air for free cooling can provide worthwhile economies in operating costs. This is especially true in buildings with high internal heat gains, where cooling may be required even at low outdoor temperatures.

An automatic system of ventilation control is shown in Fig. 15. It uses a mixed air controller to operate the damper motor, thereby modulating fresh and return air dampers to maintain a constant temperature input to the heat pump. Controls position the dampers so that only the minimum amount of ventilation air required is admitted during times of cooling demand when outdoor temperature is above 20^{0}C or heating demand. When outdoor temperature is below 20^{0}C and cooling is required, outside air dampers can be opened to admit 100% fresh air for free cooling. These dampers are also (9) controlled to admit minimum ventilation air by the positioning switch shown in Fig. 15 or by an end switch on the damper motor itself.

Many installations require exhaust systems and dampers which may or may not be part of the package for proper control of indoor pressure levels. Simplified ventilation systems are preferred in many cases for economy. One of the simpliest configurations is illustrated in Fig. 16 which employs a spring return motor on the outside air damper. When the fan is started, the damper opens to admit a fixed percentage of outside air. When the fan stops, the damper closes. A more complex system is shown in Fig. 17 where the two position damper motor is replaced with a modulating motor. The amount of outside air is varied by a positioning switch on the control panel. This system allows the outside damper to be opened 100%, but the motor automatically closes it whenever the fan is stopped.

For any installation proper location of the thermostate is an important factor. However, in large spaces with no columns or where shelves cover walls, compromises have to be made. The return aid thermostat is an alternative solution where it is impossible and impractical to mount a room thermostat. A return air thermostat should offer a high sensitivity, fast response, low mass and good repeatability. Electronic controls are now preferred over relay type, as they reduce the bulk of the control panel, offer a higher reliability and an added bonus is remote temperature reset. In the case of rooftop units, resetting return air thermostats can present a problem, but a control panel with remote temperature reset can solve it handily.

3.3. Air-to-Air Heat Pump Systems

3.3.1. Single Package Heat Pumps

Air-to-air heat pumps in this category may be classified by the way in which the indoor and outdoor sections are oriented within the cabinet. Generally speaking, a horizontal unit is considered one in which these two sections are side by side, while in a vertical unit the two sections are stacked one above the other. The cabinet may be designed wholly or partly weather-proof depending on how it is to be installed. For units intended to be installed completely out-of-doors, as on a roof, the entire cabinet must be designed to withstand the weather. Typical arrangements are shown in Fig. 18. Some of the advantages of locating the unit outdoors are:

1) Minimum restriction in the outdoor air circuit, reducing initial and operating cost for the outdoor fan.
2) Ease of removal of melted frost from outdoor coil.
3) Elimination of need for installer to supply outside air ducts and weather louvres.

Among disadvantages to be considered are:

1) Servicing during heating season can be difficult.
2) The indoor section must be well insulated and sealed for efficient operation since it is subjected to the outdoor ambient temperature.

Some units are designed to be installed as in Fig. 19. A unit designed for installation as shown in Fig. 20 requires a blower capable of producing high static pressures in the outdoor section to overcome the extra resistance of outdoor air ducts and weather louvres. If a unit is installed within the building, but outside the conditioned space, special attention must be given to removal of outdoor air condensate. In certain climates it may be necessary to use electric heater cable on condensate lines to insure that the condensate does not re-freeze and block coil draining during the defrost cycle. This problem can be solved by running condensate lines in heated areas of the building.

Vertical cabinet models are most often used for installation entirely within the conditioned space. They are designed to be located near an outside wall for ready access to outside air. They must be used with a supply plenum and grille for free air delivery in certain types of commercial applications.

3.3.2. Split System Heat Pumps

With an air-to-air heat pump the most commonly used arrangement is where the outdoor section (consisting of compressor, controls, outdoor coil and fan) is located remotely from, but coupled directly to the indoor sections, by means of liquid and vapour refrigerant lines. The indoor fan section may be furnished in either a horizontal or vertical cabinet. Vertical cabinet models are often designed so that they can be used for either upflow or downflow of indoor air. Indoor and outdoor sections must be piped in the field with refrigerant grade tubing, evacuated and charged. However, in 5HP sizes and below, some manufacturers are furnishing units designed to be connected with factory charged tubing using refrigerant connectors which may be coupled under pressure. Installation can then be made with no field evacuation and charging of line fittings.

Slab or ground-level installations of split systems are most common for residential small commercial and apartment buildings with only one or two floors as illustrated in Fig. 21. Such installations place major sound producing components outside the conditioned space and give quiet operation for the owner. Since heat pumps must operate year-round in all types of weather, protected locations should be used where possible to insure reliability of operation. Orienting the outside coil so that it does not face the prevailing wind will often aid materially in reducing the length of the defrost cycle in severe winter weather. Snow accumulation can be detrimental to heat pump operation and in climates where this is a factor, the unit should be mounted in a manner, so that the snow will not interfere with operation. On outdoor units, water draining from the coil as a result of defrosting can refreeze and accumulate to prevent complete drainage and partially block the lower portion of the coil. Proper elevation of the unit and provision for water drainage help to alleviate this problem. It should be emphasized that when split systems are installed, that the vapour line connecting indoor and outdoor sections carries compressor discharge gas on the heating cycle as well as suction gas during the cooling cycle. Since there is more likely to be gas pulsations and hence vibration in a discharge line, special precautions should be taken to isolate this line from the building structure. Insulation is also essential on this line and should be capable of withstanding temperatures up to 120°C.

3.4. Performance Characteristics of Air-to-Air Heat Pumps

The winter heat loss and summer heat gain of a home are generally considered to be a form of linear relationship with the outdoor temperature as shown in Fig. 22. (Both the heat loss and the heat gain are based on a 20°C indoor temperature). However, due to internal loads from appliances, people and lighting, heating is not considered to be necessary until the outdoor temperature drops below 10°C. The heating and cooling requirements of a home increase as the outdoor temperature varies from 20°C, the desired indoor temperature.

Capacity and efficiency of the heat pump are also functions of the outdoor temperature. As the temperature difference between the indoor and outdoor air increases, it becomes more difficult to move the heat, thus the capacities decrease and efficiency is reduced as shown in Fig. 23. By comparing Fig. 22 and Fig. 23 as demonstrated in Fig. 24, it is obvious that the heat pump has excess capacity during mild weather, heating and cooling.

The design basis for air-to-air heat pumps are in general as follows:

1) Heat pumps are sized to handle the cooling load at design conditions.
2) Heat pumps require supplemental heating when the outdoor temperature is below the heating balance point.

3) Compressor operation during low temperatures affects seasonal efficiency.

In a study carried out by General Electri in Philadelphia and Minneapolis using a range of heat pump units, a seasonal heating performance factor (S.P.F.) was used to assess the performance of these units.

When calculating a seasonal heating performance factor, the seasonal weather pattern for the particular area under study must be obtained. Fig. 25 shows the weather pattern for the area of Philadelphia in hours at various 2°C intervals of the outdoor temperature. Fig. 25 also plots the electric resistance value (i.e. amount of electric resistance heat required to heat a home at each outdoor temperature) and compares it with the power input required for a heat pump. As the outdoor temperature decreases from 15°C the heat pump power input moves considerably below the electric resistance equivalent showing the efficiency of the heat pump. At approximately 2°C, however, supplementary heat is required and the relative efficiency of the heat pump decreases. The same date is shown in Fig. 26, which shows the heat pump capacity decreasing as the outdoor temperature decreases.

The seasonal performance factor (S.P.F.) is a useful tool in evaluating the performance of a heat pump from an economical viewpoint, and an example of such a calculation is shown in Fig. 27. A diagramatic representation of this particular application is shown in Fig. 28. Several sizes of heat pumps were tested in the same residence and as illustrated in Fig. 29 increasing the unit size above the size required for cooling shows a very poor economic payback. When the same test was carried out in a Minneapolis site, where the peak annual heating/year occurs at -3°C and the balance point occurred at 0°C, the balance point decreased as the size of the unit is increased as shown in Fig. 30.

It was concluded from the studies that sizing the heat pump for the cooling load is the best economic choice for the consumer most of the time. In northern areas of the U.S.A. such as New York and Minneapolis, it may be more economical to increase the unit size so that the balance point is at or below the temperature of maximum annual heat loss. Due to the growing popularity of heat pumps with some utilities, another important factor which must be looked at carefully is the impact of supplemental heating on the utility peak load demand. The study of the Philadelphia heat pump installation showed that the average power input for the compression cycle actually levels off at 0°C and declines slightly with reduced outdoor temperature (Fig. 31). Thus if supplemental heat can be accomplished by some other means, the utilities winter load could be made very stable.

3.5. Case Studies of Actual Air-to-Air Heat Pump Installations

3.5.1. The Carrier 50DR006 Air-to-Air Heat Pump

This commercially available 5 ton, 3 phase heat pump was developed by the Carrier (17) Air Conditioning Company for the Edison Electric Institute head pump improvement project. The objectives of the project were to develop heat pumps with high performance and giving a high degree of reliability.

The starting point for the development work was the 50DQ006, a standard unit in Carriers heat pump line. In developing an improved version of this heat pump the following hardware changes were made:

Refrigerant Circuit

In Fig. 32 the heating and cooling cycles are shown and several differences from more conventional models are readily apparent. Two liquid lines connect the indoor and outdoor coils. One line is used on heating and the other on cooling, with the refrigerant flow being controlled by the thermostatic expansion valves. This feature eliminates the two refrigerant check valves of conventional cycles. A liquid receiver is included which holds up liquid on the heating cycle and thus reduces the effective refrigerant charge for this cycle. This results in improved heating performance. The circuit also includes a generously sized suction line accumulator, which traps liquid refrigerant that approaches the compressor.

A subcooling section is incorporated in the indoor coil to provide additional heating capacity with no increase in power. An incidental benefit is that when changing from heating to defrost, this subcooler boils of refrigerant and thereby prevents passage of liquid to the accumulator.

Several schemes have been proposed for modifying supplemental heating systems and they are as follows:

1) Controls adapting a new or existing oil or gas furnace with a heat pump are available. The heat pump indoor coil is installed in the position of a normal cooling coil, which means the furnace and heat pump cannot operate simultaneously. The heat pump can be shut down at the heating balance point and all heating below the balance point is accomplished with the fossil fuel furnace. An improvement is to cycle the heat pump off with the second stage of the room thermostat; once the second stage has been satisfied, return to heat pump operation.
2) A more flexible and possibly lower cost scheme for new construction might be to use a hydronic system with either fossil fuel or off-peak electric heat. The stored warm water would be pumped through a water-to-air heat exchanger with the second stage of the room thermostat. The heat pump compressor would operate continuously below the balance point as normal, taking full advantage of the compression cycle.

3) The use of solar collectors to catpure solar radiation and the storage of this energy for supplemental heating could be the first commercially practical application of flat plate solar collectors for space heating.

The refrigerant used was R-500, which gives lower discharge gas temperatures at low outdoor temperatures and has lower system pressures than R22.

Compressor

The compressor was of the hermetic type and was provided with a new 3-phase, 220V, 1750 r.p.m. motor. Use of a 3-phase compressor eliminates the necessity for capacitors and relays, which tend to be a source of trouble in single phase units. Problems in any of the 3-phases are sensed and protected against by two current-sensitive overloads and a winding temperature-sensitive overload. The latter is embedded in the motor windings and will de-energize the motor any time winding temperature begins to exceed safe limits. A low pressure cutout protects against loss of charge and reeze up fo the indoor coil during the cooling cycle and loss of charge during the heating cycle. A discharge gas temperature sensor provides protection against excessive operating temperatures at the compressor discharge, which could result from loss of charge or excessive voltage coincident with low suction pressure. To provide protection against pressures resulting from fan failure and air or refrigerant restrictions of any sort, a high pressure cut-out was used. Finally a crankcase heater was immersed in the crankcase casting, which comes on when the pump is off to warm the compressor crankcase and prevent dilution of the oil by the refrigerant, thereby insuring against flooded compressor starts.

Time Guard

A time guard circuit was incorporated to force a delay of 5 minutes before the compressor can be restarted after any shutdown. This increases the life and reliability of the compressor by preventing short cycling due to such things as loose connections in the wiring or clogged air filters. Since the fan motors start ahead of it, the compressor starts in a more unloaded condition. Also in the event of a momentary power failure, the heat pump is not thrown back onto the line with all the other loads, which helps to protect electrical circuits.

Defrost System

The system incorporated into this design utilizes both time and temperature to determine when defrost will take place and for how long. It automatically adjusts the length of the defrost cycle to changing weather conditions. Normally defrost is complete when the outdoor coil temperature is 18°C. However, since coil temperature might not respond in a high wind, a timer is provided to override the thermostat and end the deforst cycle after 10 minutes.

The performance of this particular heat pump was demonstrated in a residential home in Allentown, Pennsylvania. Figs. 33 and 34 show the seasonal performance of the 50DQ006 and 50DR006 heat pumps respectively. The A.R.I. cooling capacity was increased from 17.5kw to 18kw and the A.R.I. heating capacity rating (10°C outdoor air and 18°C indoor air) was increased from 17.5kw to 19kw. Also the coefficient of performance at these conditions was increased from 3.0 to 3.5.

Greater heating output results in a lower balance point and the higher C.O.P. reduces power input is not only lowered, but also a much smoother input curve results with a reduction in supplementary heating power.

3.5.2. Westinghouse Hi/Re/Li Heat Pump

This 3 ton single phase heat pump was developed by Westinghouse (15) for the Edison heat pump improvement program. In the design of this unit, the emphasis was placed upon reliability and improved operating performance.

In order to eliminate potential field installation problems involved in the charging of split system units, the heat pump was designed as a single package. To minimise outdoor noise level, centrifugal fans were used. A multiple speed fan motor was used indoors for adjusting to the range of duct statics which would be encountered. Plug in electric heaters and filters were located ahead of the indoor coil.

During the heating cycle of the heat pump, the compressor discharges high pressure gas to the condenser, where it condenses to a liquid (Fig. 35). A subcooling control valve then operates to maintain approximately 5^{o}C of subcooling of the liquid leaving the condenser. From there the liquid flows through the manifold check valve and comes in contact with the cool suction gas line. A proximity condition which drops the temperature another 5^{o}C. Finally about 10^{o}C of subcooling occurs when the liquid passes through an accumulator-heat exchanger and just prior to expansion by the subcooling control valve. Due to the increased refrigeration effect given each pound of refrigerant as shown in Fig. 36, there is excess refrigerant introduced into the evaporator beyond the evaporation rate. This is an inherent characteristic of the system, since the refrigerant is highly subcooled and permits the evaporator to function with utilizing surface for superheating.

Liquid and vapour are carried over to the accumulator, where the liquid drops to the bottom and is evaporated by heat exchange with the condenser liquid. The saturated gas in the uppermost section of the accumulator is drawn into the suction U-tube and returned to the compressor. The suction U-tube within the accumulator contains a small oil return bleed which allows a mixture of refrigerant liquid and oil to be drawn into the suction gas. The warm condenser liquid in contact with the suction line boils off the refrigerant, while returning oil to the compressor.

One of the most important aspects of this heat pump is the absence of liquid storage in the coil operating as the condenser. Since subcooling is effectively controller to hold minimum liquid in the condenser coil, the excess refrigerant collects in the accumulator-heat exchanger. This component is sized to handle the excess refrigerant, precluding the possibility of liquid floodback to the compressor during a reversal from heating to cooling. Also the gas formed from the small quantity of liquid in the indoor coil first flows to the accumulator, where it immediately changes state due to contact with the large volume of stored liquid. Therefore, there is no prolonged expansion through the compressor which can wash out the compressor oil.

In the electrical system, the contractors for the compressor and the electric heaters were selected to have ratings approximately twice that required for the application. Heavy duty totally enclosed relays were chosen for protection against mechanical damage, for long life and simplified service. A new control system for the unit was adopted, which included the following requirements:

1) Eliminating or reducing materially the number of relays, interlocking safety switches, starters and other such devices.
2) Provide means for improving excessive short cycling of the compressor.
3) Make the standby resistance heaters automatically energized wherever necessary even though compressor is inoperative.

To achieve these goals, a programmer by the thermostat was used to provide sequential operation of the compressor and electric heaters. Excessive short cycling is eliminated since the programmer does not allow the compressor to recycle until it has been off for 2 minutes. If the compressor becomes inoperative during the heating cycle, the programmer automatically picks up the standby heater.

This system controlled well from a temperature standpoint, but it could not prevent unnecessary operation of booster heat because the stepping function could not be made to differentiate between compressor and booster heater operation. This resulted in heating kwh to be excessive. To overcome this problem, all the wall thermostats were replaced with ones having two, rather than one, heating stages which was a return to conventional practice.

The primary thrust of this heat pump study was directed towards obtaining performance characteristics of geographically scattered heat pump installations in a form, which would facilitate comparison of the energy requirements for the various regions and to some extent between the heat pump energy use levels and those of other systems. Results of this study were based on 53 heat pump installations (40 residential and 13 commercial) some of which afforded two years of acceptable data. The ultimate comparison factor used for the heating function was that of "kilowatt hours - per kilowatt of design loss per degree day" (Kwh/HD/DD) and for the cooling function "kilowatt hours per ton of design heat" (Kwh/ton). Fig. 37 shows the heating results obtained for both the residential and commercial installations.

Commercial applications show no trend towards uniformity with respect to the heating function, which may be due to the variation in such things as occupancy, internal heat gains and resultant requirements of supplementary heating. Residential applications show a definite percentile average at the level of 0.17 Kwh/HD/DD.

Fifty five percent of the installations were under 0.20 Kwh/HD/DD and these were regarded as satisfactory and acceptable energy use levels for heat pump performance.

The input requirements of a heat pump system are non-linear with respect to outdoor temperature. In Fig. 38 which shows the heat inputs to several typical installations, it may be noted that the slopes of input lines for both the cycle and supplemental heating are about the same up to approximately $3^{0}C$. Beyond that point the supplmental heat input increases at a rapid rate whereas that for the cycle trends downwards. Thus the rate of usage of both the heat pump and supplemental heat will vary as a function of the temperature at each of the several points experienced during a given interval as shown in Fig. 38. The conclusion reached was that the supplemental heat was being brought into use before the full capability of the heat pump cycle is exhausted. This facet may offer an area for further investigation with potential improvement in economy of energy use.

Fig. 39 and 40 present the weekly energy use of the several load components making up the total use of an installation in New Jersey and one in Ohio. The similarity of inputs to the heat pump cycle is of particular interest especially when the marked difference in supplementary heat is noted. The New Jersey installation had a reported loss of 16.1 Kw compared to 13.6 for the one in Ohio. However, the New Jersey degree days 4828 compared to over 6000 for Ohio. Fig. 41 presents similar data for a commercial installation. The effect of the much greater base load in reducing the heating requirements and increasing the cooling requirements is apparent. In spite of the reported design heat loss of 21.8 Kw the heating use, particularly that of supplemental heat is of dramatic proportion when referenced to the foregoing residential data.

Fig. 42 shows the same installation as shown in Fig. 39 and attempts to reconcile the actual use of both the heat pump and supplemental heating on the coldest day with an estimate of requirements derived from the reported heat loss. The outdoor temperature ranged between 0° and -10°C and the output from the cycle was stable and varied as a function of its input and C.O.P. however, the total heat requirement is great than the calculated requirements. At first it was suspected that too high a C.O.P. was used in determing the output of the heat pump cycle, however, since the excess heat supplied is about equal to the full heat pump output, this was ruled out. The most probable explanation is that the calculated design loss was lower than actually experienced. If then the heat supplied at -18°C is used as a base and the hourly requirements recalculated, good compatibility with the heat supplied is obtained.

3.5.3. Heat Pump Performance in North Western United States

This report was based on actual data obtained over a period of five years from an air-to-air heat pump installed in a residential home in (14) Pullman, Washington. The residence was a six-room frame of 106m^2. All the walls and ceilings were insulated and storm windows and doors were used during the heating season. A basement of 56m^2 located under the central part of the residence was used for laundry, hot water equipment and heat pump equipment as well as for general storage. Basement temperature was maintained above 15°C and the calculated heat load of the house at 20°C temperature difference was 11.3 Kw.

The heat load was obtained by shutting off the heat pump during several periods of the first heating season and supplyign all the heat by electrical resistance heaters. Four calibration periods totalling 28 days were used for heating during which 2912 Kw/hr were used for heating. The accumulated degree days for these periods was 756 or an average of 27. A convenient constant was derived from these results - $\frac{335 \text{ Kw-hr}}{\text{day}}$ per degree day.

An analysis of the weather data as shown in Table I and II show that January, the coldest month, has an average temperature of -2°C. The minimum temperature is -35°C, but -22°C is normally used as the design temperature. Rather than providing a large heat pump to handle the entire heating load which would be expensive due to the fact that its full capacity would only be used a few hours every year, a smaller heat pump was chosen supplemented with electrical resistance heaters for use in abnormally low temperature conditions.

A 2 Kw heat pump was used which meets the heating requirements of 7 Kw/hr when the outdoor temperature is -2^{0}C. The supplemental heat was supplied by three resistance heaters of capacity 2.4 Kw (7.2 Kw total) which were controlled by a two-step thermostat.

Provision was made to connect two of the units on the first step during extremely cold weather. The indoor section of the heat pump consisted of the motor, compressor, condenser, indoor air fan, supplemental heaters, controls and defrosting system. An evaporator and a fan comprised the outdoor section. The evaporators were equipped with spreader type trays for a wet defrost system. Although the use of water flooding was considered to be second best to a hot gas method, it was adopted on a trial basis. During heat pump operation, a tank of water is warmed by subcooling the refrigerant on its way to the expansion valve. At defrost time, the water is pumped to flood the evaporator coils, melting the frost. Defrosting is controlled by an automatic switch, timed to cycle twice daily.

A balance point of -3^{0}C was obtained with the system, which is just below the normal January average temperature. This was determined during a period of days each year when the outdoor temperature remained steady and no wind was present, which normally occurred under conditions of thick cloud cover. Fig. 43

shows the heat output of the heat pump plotted against outdoor temperature and the energy demands of the heat pump is shown in Table III.

One of the principal drawbacks to the use of heat pumps in this region of the U.S.A. is that the supplemental heaters impose a relatively high load on the power supplies which during abnormally cold weather already faces the maximum load of the year. In Fig. 44 the outdoor temperature and total heating system demands are shown plotted against the three coldest days of the five year study period. It should be noted that the third step of the supplemental heating operated only intermittently during the system peak hours of 4 to 6 p.m. each day.

In Table IV it can be seen that the supplementary heaters increases the power demands of the residence. Comparison of the several years, shows that the supplemental heaters are responsible for the major part of the increase in demand. Comparison of the load factors of the heat pump with that of the other residential uses, shows that this heating plant, including the supplement heaters is of the same order as the domestic use without the heat pump.

During the five year operating period, the following problems arose with the heat pump system:

1) Fatigue Cracks in Pipe

Cracks occurred in the compressor suction line, as a result of insufficient flexibility in the pipe to handle compressor vibration. The pipe was re-routed and fitted with a flexible metal hose.

2) Defrosting Cycle

When large snow flakes were carried by the air entering the evaporator, the snow quickly formed a blanket over the screen and stopped air flow in the evaporator. A hood was built over the evaporator air inlet which prevents wet snow and rain from reaching the screen surfaces, thereby eliminating the snow removal problem.

Due to the presence of dust and other air-borne materials, which in this machine were washed off the evaporator tubes and returned to a tank with the defrost water. On subsequent cycles this material was pumped with the water and collected on the water spreading trays above the separator, which in time clogged the trays. The installation of a strainer in the defrost drain line and smaller mesh screens across the evaporator air inlets eliminated this problem.

3.5.4. Comparison of Heat Pump Installations in Canada and the U.S.A.

Since most of the data available in journals on heat pumps is mostly American in origin, no direct corellation should be drawn for the performance of a Canadian installation due to the differences in climatic conditions between the two countries. In order to determine the effect the Canadian climate would have on the Seasonal Performance Factor (S.P.F.) of a heat pump presently installed in a Philadelphia residence, a S.P.F. calculation using Montréal weather was carried out as shown in Fig. 75.

The seasonal heating hours (32) together with the dry bulb design temperature (31) for Montréal were inserted into the S.P.F. calculation detailed in Figs. 75 and 76 lists the design conditions for the two installations and manufacturers data on the heat pump.

Due to the larger number of degree hours experienced in the Montréal location as compared to Philadelphia, the seasonal heat input of the resistance heaters is increased quite substantially. This has the effect of reducing the "Overall Seasonal Performance Factor" of the heat pump by 18%.

Thus in order to lower further the balancepoint and to duplicate the Philadelphia performance of the heat pump system, a larger capacity unit would be required which has the effect of reducing the economic savings generated by this type of installation.

4. AIR-WATER/WATER-AIR/WATER-WATER HEAT PUMP SYSTEMS

4.1. Principles of Operation

4.1.1. Air-Water System

The air-to-air system is commonly used in the tonnage range up through 30 tons. This type of system is limited to a single indoor coil restricting its operation to only one zone. Multiple indoor coils are not practical because of problems of liquid draining and controlling capacity of condensers in parallel in different locations. The main difference between the air-to-water system and the air-to-air system is that the indoor coil now becomes a refrigerant to water heat exchanger. This serves as a water chiller during the cooling cycle and chilled water is then circulated to multiple fan coil units, which are individually controlled to maintain comfort conditions in each zone.

During the heating cycle, the indoor heat exchanger becomes a water cooled condenser with warm water circulated to the same fan coil units. The system's cycle of operation is controlled by an outdoor thermostat which may be manually overriden. Interlocks to space thermostats are provided so that they are always controlling for the same cycle of operation as the heat pump. For defrosting the cycle is reversed to cooling, but the outdoor fan is turned off. Heat is removed from the warm circulated water and supplemental heaters, if used.

Air to water systems range in size from 30 to 800 tons. Their big advantage over air-to-air systems is that multiple zones can be provided on the same system. Also, where factory packaged equipment is used, less refrigerant field piping is required. A typical example of an air-to-water heat pump is shown in Fig. 45.

4.1.2. Water-Air-System

A water-to-air system uses water as a heat source and air to transmit heat to the conditioned space. Water used as a heat source can be supplied from a well, river, lake, the earth or other sources that have fairly constant temperatures. This type of system operates with a relatively constant C.O.P. regardless of outside air temperature, hence it is used in climates where extreme temperature variations are experienced. This constitutes its greatest advantage over a system using outside air as the heat source. Heat pumps of this type are also smaller in size, since requiring only one air heat exchanger.

Its disadvantages lie in the fact that abundant sources of suitable water are becoming increasingly scarce and as a result the application of this system is rather limited. Frequently sufficient water may be available from wells, but the condition of the water often will either cause corrosion in heat exchangers or it may induce scale formation. Other considerations to be made are costs of drilling, piping and pumping and means for disposing of used water.

A typical water-to-air system is shown in Fig. 46. As with air source types, the refrigerant cycle must be reversed to alternately provide heating and cooling. This cycle is basically the same as the air-to-air system, except that a water chiller-condenser is substituted for the outdoor air unit. Water temperatures underground vary from about 10 to 15°C depending upon the location. In Fig. 46, 12°C well water is circulated to the chiller where it is cooled to 8°C and returned to another well or more suitable means of disposal. Heat extracted from the water is rejected in the indoor heat exchanger to the mixture of outside and return air in order to satisfy space conditions. When the cycle is reversed, the indoor coil becomes an evaporator and the water chiller-condenser rejects building heat to the well water supply.

As with air-to-air systems, this type is limited to operation on a single zone. Capacities range up to 30 tons and occasionally as high as 50 tons.

4.1.3. Water-to-Water Systems

Since water is the heat source, it has the same advantages as the water-air type. It is used where an adequate water supply is available for a heat source and where hot and cold water for an industrial process or for a hot-water heating system is the final product.

In a water-to-water system the water flow is reversed rather than the refrigerant flow for economic reasons. In other words both chiller and condenser are non-reversible and the chiller will always provide chilled water, whereas the condenser provides warm water, regardless of operating cycle.

During the cooling cycle source-sink water is circulated through reversing water valves to the condenser where heat is rejected and the resulting warm water is drained to a sewer or another well. For the heating cycle, well water is circulated directly to the chiller where heat is removed from it and rejected to the condenser which now has its water circulated to heating and cooling coils.

An example of a typical water-to-water system is shown in Fig. 47. The size range of these systems varies from 20 up to 1000 tons. Its main advantages over the water-to-air system is that it can be used with multiple zones, each individually thermostatically controlled. Also, a standard packaged water chiller can be used as the heat pump with modifications made in the control circuits.

On large installations where the water supply is limited, it is sometimes advantageous to combine an air and a water heat source. This takes advantage of the higher C.O.P. when the outside air temperature is higher than the water temperature and also gives a higher output than an air-to-air system at low outside air temperatures. Such a system is illustrated in Fig. 48. When the outside air temperature reaches the valve set on the thermostat, the outside fan is shut down and the water pump circulates water through the water coil which can be built into the air coil.

4.2. Water Source/Sink Heat Pump Systems

4.2.1. Simultaneous Heating and Cooling

In larger commercial and industrial buildings, it is often necessary to provide heating in one area and cooling in another. Heat pumps which utilize water to heat and cool spaces within a building at the same time accomplish this task in two ways. One way is to make provision for excess fresh air to each of the air handling units. The system is then operated to provide heating anytime the outside air temperature is below a pre-determined valve for the most critical zone. Other zones which may require cooling obtain it by using large quantities of outside air which, for this cycle, should be at 15°C or below. At higher outside air temperatures, the system provides cooling, and no heating should be required except where close humidity control is desired.

A better but more expensive arrangement is to apply a heat pump that can simultaneously provide heating and cooling to the spaces, regardless of season. Types of systems applicable to this heat pump are multi-zone, dual duct, induction unit, three-pipe and four-pipe systems. Fig. 48 shows a typical building load graph as a function of outside air temperature for an application where simultaneous heating and cooling would be desirable. The building cooling requirements are shown to be 350 Kw at 38°C and 88 Kw at -18°C. Heating requirements are 800 MBH at -18°C and zero at 25°C. Interior zones requiring mechanical cooling, however, do not make the excess heat available to the perimeter of the building, where it is required, unless the heat is transferred through a heat pump system.

It can be seen that at 10°C outside air temperature, the cooling requirement is substantially greater than the heating requirement so all heat removed from the interior cannot be distributed to the perimeter. Where provision is made for up to 100% ventilation air at air handlers, the heat pump system can be unloaded to the point where it removes only enough heat from interior spaces to satisfy the perimeter. Compressor capacity is regulated to match the heating load, which is supplemented with heat from the interior area, plus the heat of compression. At these intermediate temperature levels, this system does not provide sufficient cooling, so the fresh air dampers are modulated to increase the quantity of outside air used for cooling purposes.

As the outside air temperature falls to -3°C, heat removed from interior spaces plus the heat pump motor gain is in balance, theoretically, with perimter heat loss. At this point, fresh air dampers would be reset to provide air for ventilation purposes only. At lower outside air temperatures, heat removed from interior spaces would be supplemented by the heat pump with heat removed from outside air well water, storage or supplemental heaters. Such a system is illustrated in Fig. 49.

The chiller and condenser shown in this system are both non-reversible, as they respectively circulate chilled water and warm water to cold and hot decks with individual three way valves, each controlling desired temperature conditions. The outdoor air unit becomes an evaporator when the heat available from the cold air circuit plus the heat of compression are not great enough to satisfy the warm air circuit. When excess heat is available from inside, the outdoor coil becomes the condenser. When running on the heating cycle, the outdoor coil operates in parellel with the water chiller. An evaporator pressure regulator in the suction line leaving the chiller, prevents water freeze-up. On the cycle where excess heat is available internally, condenser and the outdoor coil operate in parallel as condensers.

For the arrangement shown separate fans on hot and cold air circuits permit extreme flexibity in operation. Excess ventilation air is introduced to the cold air duct only, to provide economizer cooling in areas with high internal loads without affecting the heating load in the warm air circuit. This permits the system to operate at optimum efficiency and follow the pattern illustrated in Fig. 48.

4.2.2. Exhaust Heat Recovery

On most larger commercial and industrial buildings, it is necessary to provide positive exhaust to induce sufficient low internal static pressures throughout and obtain uniform ventilation. Since exhaust air temperature is frequently well above outside air temperatures, it is desirable to remove heat from exhaust air to permit more efficient operation of the heat pump. A system for recovering heat in this fashion is shown in Fig. 50.

An evaporator coil is located in the exhaust plenum and functions in parallel with the outdoor air coil to simultaneously remove heat from both. During mild weather operation, the outdoor coil becomes inoperative and all the heat is removed from exhaust air resulting in very efficient operation. Where physically practical, exhaust air may be ducted into the outdoor coil directly, thereby raising the temperature of air entering the outdoor coil to increase the heat pump capacity and efficiency. During cooling cycle, the evaporator coil in the exhaust plenum is inoperative, but the 23^{0}C leaving air, mixing with warmer outside air entering the outdoor coil, will further help the capacity and efficiency of the system, causing lower condensing temperatures.

The liquid refrigerant subcooling coil shown in Fig. 50 subcools the liquid refrigerant leaving the condenser on heating cycle, while preheating ventilation air. When practical to apply this principle, capacity and efficiency of the heat pump system is further improved and preheating helps to offset possible freeze-up hazards of water heating coils.

4.3. Case Studies of Heat Pump Installations using Water as the Sink or Source or Both

4.3.1. Wichita School Water-to-Water Heat Pump

This heat pump system was installed in the Sedgwick County, Kansas, whose campus style layout with air-conditioning and heating was accomplished with a 700 hp water-to-water heat pump. The school has accommodation for 1,200 students on an 80 acre site and all the educational facilities were housed in nine separate buildings, the details of which are shown in Fig. 51.

Total cost of construction was 2.25 million dollars (1960) and the cost of mechanical equipment such as air-conditioning heat pump, water wells, etc., worked out to $2.75 per square foot. During preliminary stages of design an examination was made of the available fuels for heating purposes and their related costs. However, it was decided that a water-to-water heat pump would be economical, due to the fact that the school was situated over an unlimited supply of nearly constant temperature water (16°C) supplied from the Arkansas River. A comparison of the annual heating bills for the various fuels considered is shown in Fig. 52. Factors which influenced the decision to use a heat pump were as follows:

1) A study of fuel costs indicated that the cost of natural gas would rise by 55%.
2) The cost of electricity would remain fairly constant.
3) Elimination of boiler installation costs.

4) A smaller equipment room as a result of eliminating boilers.
5) Lower insurance rates as a result of elimination of boilers.
6) Elimination of cooling tower installation.
7) Lower operating costs for air-conditioning.

The heat pump system consisted of a 700 hp open type refrigerant 12, centifugal compressor with condenser capable of heating 1,775 g.p.m. of water (112 litres/sec) from 114 to 123°F (45 - 50°C) plus resistance heaters to boost the water from 123 to 125°F while 800 g.p.m. (50 litres/sec) of 60°F (15°C) well water is cooled to 45°F (8°C) in the evaporator. During the cooling cycle 1775 g.p.m. of water for the building loop is cooled from 53 to 45°F (12 - 8°C) while 800 g.p.m. of 15°C water is heated to 24°C through the condenser. Well water was supplied through a 30 H.P. (22 Kw) turbine type pump from a 40 feet deep (12 m) well.

A heat pump with supplemental electric resistance heater was chosen over a straight heat pump installation, however, this cost was justified by the relatively few hours during the year the heaters were used. Total heat loss of the building at -10F (-25°C) was 10.6 million Btuh. At night this load reduces to 8.9 million Btuh by eliminating much of the ventilating air. The heat pump produces 7.75 million Btuh leaving only 1.15 million Btuh to be produced by the resistance heaters at -10°F (-25°C) outside air temperature.

During the day the heat load required was 10.6 million Btuh, however, this was reduced to 9.5 million Btuh as a result of internal heat gains. It was found that the outside air temperatures below above which the supplemental heaters are not required for an indoor design temperature of 70°F (21°C) was 0°F (-18°C) at night and 5°F (-12°C) during the day. Since only 60 hours of temperatures in the range of 5 to -4°F (-15°C - 20°C) were experienced during the heating season, the resistance heaters were rarely in operation.

Classroom heating and cooling units were of the standard year-round unit ventilator type with coils for either hot or chilled water, drippan and damper controls, capable of handling 100% outside air. On some days when the central system was operating on the heating cycle, certain rooms required cooling due to solar heat gains. This was solved by using unit ventilators with pneumatic damper control, which automatically positioned the outside air damper to bring in enough outside air to maintain the desired room temperature.

4.3.2. Water-to-Water Heat Pump in the United Illuminating Company Building, New Haven, Connecticut

This application was particularly interesting since the heat pump was located in a region where most winters are severe. The installation was a water-to-water design using an unlimited supply of underground water at 58°F (15°C) as the source of heat. It is located in six wells in the basement of the building and is pumped through a sand-settling tank which has a capacity of 400 g.p.m. (25 litres/sec) under maximum heating conditions. The total capacity of the wells are 600 g.p.m. (38 litres/sec) so that only four wells are in operation during maximum conditions. This arrangement allows the water from the evaporators to be returned to the two wells not in use and gives a higher pumping level for the wells that are in operation. All the well pumps are controlled by electric float switches located in the settling tanks, which allows any of the pumps to be controlled by any of the float switches.

A simplified diagram of the system as connected during the heating seasons is shown in Fig. 53. Water from the sand settling tank is pumped through the heat pump evaporator where heat is removed from it. This heat is transferred to water circulating around the condenser coil, which is used to heat the building. Fig. 54 shows the piping arrangement for summer operation, in which water from the settling tank is pumped through the evaporator of the heat pump where it is chilled. It is then pumped to the building for cooling and is returned to the condenser of the heat pump where it is used to cool the refrigerant and is then returned to the wells.

The air-conditioning system is divided into three circuits as shown in Fig. 55. One circuit contains both heating and cooling units, another heating units only and the third cooling units only.

A central ventilation system, which operates only on outside air, supplies all the air-conditioned rooms through a duct system. Heating cooling and control of humidity are accomplished by water-to-air heat exchangers located in the duct system. In addition to the central ventilating system, local heating and cooling units were placed in each small office. These local units consisted of a fin tube heat exchanger and a fan assembly to recirculate the room air and provide the individual heating and cooling requirements of the office. In addition to the local units, convector heating units were installed in the offices on the north side of the building, which have a high heat loss and a low heat gain, and in hallways, stairways, small toilets and service offices not air-conditioned. The garage was heated with ventilators and convectors, which operate with secondary water from the convector heaters and central heaters. This arrangement decreased the temperature of the water fed to the heat pump condenser and thus allows maximum performance of the heat pump.

The heat pump consisted of eight condensing units. Each unit is composed of a 40 H.P. (30 Kw) Westinghouse vertical eight cylinder compressor, a two-pass horizontal condenser and a vertical shell and coil water chilling unit. All the condensing units were mounted on Korfund adjustable spring supported based, and piping to the building was connected using flexible vibration eliminators to prevent noise transmission in the building. All eight evaporators of the heat pump were connected in parallel in the water circuit, with a capacity sufficient to cool 400 g.p.m. (25 litres/

sec) of water from 55 to 40^{0}F (13 to 5^{0}C). The eight condensers are arranged in series so that each is required to raise the temperature of the water only a small amount. This gives a higher C.O.P. than if the condensers were connected in parallel and each operated at the final temperature.

Temperature control is maintained by an outside thermostat and program switch, which controls the operation of the condensing units, and thermostats located on the local unit heaters. The thermostats located in the condenser exit controls the operation during the heating cycle and starts and stops the compressor as needed. The controls are connected so that the water to the evaporator flows only when the compressor is in operation. Temperature control for the local heating and cooling units is maintained by a thermostate which by-passes the water around the heat exchanger, when room temperature reaches the valve set on the thermostat.

The heat pump was designed to supply heat for -5^{0}C minimum temperature, if the temperature falls below this valve, additional heat is supplied by a booster heater. These heaters consist of a 480 kw heater and a 7,000 gal. storage tank. The heaters were connected through a time clock switch so that they operated from off-peak electric power. At night the storage tank is charged to 350^{0}F (177^{0}C) and discharges to 150^0 (66^{0}C) under maximum requirements. An auxiliary electric heating system which is a duplicate of the booster system was installed to be used as a spare for the booster or as an auxiliary for the compressors.

The change from winter to summer operation is accomplished by changing the position of several shut-off valves and changing the position of the control switch, which changes the action of the heating and cooling thermostats located on the local unit heaters and changes the control of the compressors from a thermostat located in the condenser exit to a thermostat located in the evaporator exit.

Figs. 56 and 57 show the electrical energy consumption for 1941 - 1943 and the operating costs over the same period. With electric energy costs at that time 1 cent per kilowatt hour, the heat pump compared favourably with other heating systems under consideration for the same installation.

5. LARGE SCALE INDUSTRIAL HEAT PUMPS

5.1. Factors Governing the Use of Industrial Heat Pumps

The heat pump is a special kind of heat generator, for it allows us to use the heat content of environmental air, water or soil and also allows the recovery of low level waste heat produced by technical industries, which would otherwise be rejected. Heat pumps have been used for up to 50 years in commercial buildings, swimming pools, etc., however, their use in industry has been curtailed, due to some limitations of Rankine Cycle machines.

The most important of those limits at present is the temperature which can be attained by a heat pump process. Domestic space heating requires a technique that guarantees maintenance-free and safe operation. Under these circumstances, however, temperatures of more than $60^{0}C$ ($140^{0}F$) cannot at present be exceeded while on the other hand, conventional hot water heating systems require a maximum water temperature of $90^{0}C$ ($194^{0}F$). To replace an oil-fired boiler for instance, by a heat pump, it is necessary to enlarge the radiators to correspond to the lower maximum temperature. This is not a problem of heat pump technique, but a problem that limits heat pump application.

This example shows that only low temperature heat demand can be covered by heat pumps, but it means that the lower the temperature needed, the more economical the heat pump can be, especially when there is at the same time a small difference between the temperature needed and the temperature of the heat source. In general, one cannot simply replace any other heat generator by a heat pump without at the same time, modifying the energy consumption. This means, for instance, the installation of ceiling or floor heating or of convectors with an additional fan, because these heating systems need a lower temperature than radiators.

During recent years, combined systems of heat pumps and boilers have been developed and parially operated. Space heating is done completely by the heat pump with outdoor temperatures down to $0^{O}C$ ($32^{O}F$), where the water temperature attainable by the heat pump is still sufficient to meet the total heat demand. At lower outdoor temperatures, the heat demand is covered by the boilers, which means that the boiler capacity has to correspond with the maximum heat demand at the lowest outdoor temperature. With such systems, it is possible to cover up to 90% of the annual space heating consumption by the heat pump, if the heating water temperature is governed by the outdoor temperature.

Another important point regarding industrial heat pump application is the temperature of the heat source, which is important if the system is to be evaluated from an economic point of view. In order to evaluate and to select from the various heat sources, the

"Utilization Temperature" of each heat source is obtained. Fig. 58 shows various utilization temperature ranges for heat pumping, direct space heating and storage, and for cooling and heating. The "Utilization Temperature" range was defined as the optimum range at which heat can be absorbed and rejected for providing heating or cooling simultaneously.

It is evident from Fig. 58 that for temperatures between 120^{0}F to 300^{0}F (49^{0}C to 150^{0}C) that even if it is available as a free source, heat pumping is not required and not economical unless a higher temperature utilization is needed. Bank I 50^{0}F to 120^{0}F (10^{0}C to 50^{0}C) is suitable for heat pumping and can be elevated to the optimum utilization temperature by heat pumping for heating or can be used directly if it exceeds 120^{0}F (50^{0}C). Bank I utilization temperature range for heat pumping can be obtained from internal heat, building ventilation exhaust, outside air, sewage waste, solar and waste heat from heat-producing equipment.

5.2. Case Studies of Large Scale Industrial Heat Pumps

5.2.1. Heating of Greenhouses by Waste Heat of Power Stations

In order to meet food demands of the population, especially vegetables and fruit in countries with very harsh winter conditions, a great number of greenhouses have been built. Most of them are heated by conventional means (oil, gas fired heaters) and due to the large heat demand, the operating costs for these installations are very high. For this purpose the Soviet and Hungarian governments (4) developed a greenhouse, whose conditioning is furnished by low temperature heat rejected from condensers of power stations. The heating and conditioning of the whole system is supplied solely by waste heat from the condensers of the power station and the power station uses the air heaters of the greenhouse as air coolers for the condensing system. In addition to this, by means of a large heat exchanger, a much better vacuum was produced for the turbine, resulting in a higher output of the power station, even during the hottest days of the summer period.

For the growing of vegetables usually 15 - 25°C (60 - 77°F) air temperature and 80 - 90% relative humidity is required. To maintain those conditions in a greenhouse, whose average heating demand is 600 - 700 Kcal/m^2 (U.S.S.R.) means that the yearly consumption of such a greenhouse is approximately 66,000 tons of crude oil. The new heating concept would utilize the 35 - 40°C from the power station condensers and use it to heat the greenhouse of area 34 hectacres, so that an air temperature of between 15 - 25°C can be maintained.

The operating schematic of the greenhouse heating system is shown in Fig. 59. From the condenser warm water is delivered by means of circulation pumps to the heating system of the greenhouse. The amount of water corresponding to the exhaust steam quantity gets delivered by the condensate pump through low pressure pre-heaters to the feed pumps and through the high-pressure pre-heaters to the boiler.

The condenser water flows through the finned surface heat exchangers at the two ends of the greenhouse where it exchanges its heat to air which is re-introduced into the greenhouses via louvres. The power plant is equipped with two steam turbines, each of 200 MW capacity. Rejected heat of one unit gets transferred to the dry cooling tower, while the heat of the other one passes through the heat exchangers of the greenhouse. With this set-up the heating of the greenhouse can be ensured even if one of the two turbine units is out of service. In this case it is only necessary to connect the warm water pipes to each of the condensers, in this way the waste heat of the power station is rejected by the heat exchangers of the greenhouse and the dry cooling tower can be shut off. As the heating of the greenhouse had to be ensured, even in the case of unexpected breakdown of the whole power station, certain pipeline connections within the installation had to be completed. These connections ensured that even in the case of an emergency shutdown, steam can be supplied by the boiler to the greenhouse.

A detailed study of the economics involved in this operation was carried out which included the thermal investigation of the operation, as well as studies with regard to the agrotechnical part of the greenhouse. The results of the economic evaluation of the project is illustrated in Fig. 60. These results refer to a 34 hectors greenhouse operating under Soviet climatic conditions. As can be seen from Fig. 60, the combined operation of the power station and the greenhouse offers approximately 66,000 tons of fuel oil savings over a separate power station and greenhouse. Referring the economy to the production costs of the goods produced in the greenhouse, it was found that in the case of the combined operation, the prime costs of greenhouse products were approximately 25% lower than in the case of conventional greenhouses operated with separate heating.

5.2.2. The London Tower's Heat Pump Installation

A particular case history which illustrates the substantial savings that can be achieved by the use of heat pumps is the system installed in the London Tower's complex (7) in London, Ontario. This consists of 22 floors and an 18 storey tower with a total floor area of 339,000 ft^2 (33,900 metres^2). It contains 220 deluxe apartments and 51,000 ft^2 (5,100 m^2) of commercial mall area, and requires 700 tons of cooling. After three years of closely monitored energy costs, the complex used 24 Kw.h/ft^2/year substantially less than the average 75 Kw.h/ft^2/year for a similar building.

The savings are accomplished through immersion electric elements with controls which maintain a thermal storage heat sink at the proper temperature to supply temperature controlled water to the building system and the heat pumps. Monthly savings on electricity bills by controlled load factor, plus an initial reduction of 50 - 75% in the connected electrical load for water heating, were the major factors which contributed to the energy saving.

The thermal storage heat sink that supplies 67^{0}F (20^{0}C) water for the building comprises of two Model H31-600-70/70 Megatherm Units, fitted with three-way valves on the by-pass. The total storage capacity of the hot water system is comprised of two 3,100 Imp. [illegible] (11.8m^3) tanks. The total capacity of the heat exchanger [illegible]tem is 8 litres/sec.

Heated water from each unit is mixed through a three-way valve and the 62^{0}F (17^{0}C) water from the return main in the system is mixed with the 110^{0}F (43^{0}C) water from the storage tanks, resulting in a supply temperature of 62 - 70^{0}F (17^0 - 21^{0}C) in the feed main. The peak temperature of the water in the tanks is 280^{0}F (138^{0}C) and the minimum is 130^{0}F (54^{0}C). This ensures a 110^{0}F (43^{0}C) supply to the mixing valve. Mixing in the by-pass pipe is at a ratio of 6:1 for 70^{0}F (21^{0}C) water in the system. At this temperature the pipes do not require insulation which provided a saving of over $50,000.

The load control of the 600 Kw of power in each space heating unit and the 175 Kw in the domestic water units were controlled by a "Gentec" load controller. This is a solid state minicomputer, which maintains a stabilized condition in the total use of kilowats and the monthly kilowatt use demand. It automatically detects when essential loads come on and takes off non-essential loads, thus reducing energy consumption at peak load periods. Another energy conserving feature of the peak load controller is that it can detect available energy on the utility system. If 40 Kw are available from the grid, the controller can cycle this power into the complex's system for space heating, domestic water and ventilation.

The total installed cost for the combintion heat pump/thermal storage/load control system was $1.24 million, which works out to approximately $3.66/ft^2 for an energy saving of more than 50%.

6. SOLAR ASSISTED HEAT PUMPS

6.1. Problems with Direct-Utilization Solar Collectors

The sun is the source of energy of our planet. It supplies us every year with an amount of energy which, if accumulated and saved in a useful form, would be sufficient to supply the energy needs of the world for thousands of years. However, since the energy reaching the earth is very diffused and comes only during the daytime, and even then changes constantly according to the day and the season of the year, this poses great difficulties in attempting to capture solar energy for use in our daily lives.

Considerable interest has been manifested in solar energy as a heat source either on a primary basis or in combination with other sources. Many direct-utilization solar collectors for comfort heating have been developed to provide maximu retention of the heat and to minimise heat loss to the ambient air. However, only a relatively small number of these collectors have been installed because of their complexity and high capital cost, and as a result solar energy obtained from this source tends to be expensive as compared to conventional energy sources. This type of installation tends to be very large, because the intensity of solar readiation is low and thus large collection areas are required in order to harness a reasonable amount. Moreover, the sun is not present at night, nor during cloudy days and, therefore, a storage system is necessary which renders the system even more expensive. The option of using a back-up heating system for nights or cloudy days is also expensive, and increases the cost of the energy obtained.

The heat pump offers several possibilities for overcoming some of the present handicaps facing the direct-utilization collector because of its ability to absorb the solar heat at a relatively low collector temperature. Operating at these lower collector temperatures reduces the transmission losses, which materially increases the collector efficiency. At the same time, these low collector temperatures permit obtaining a larger percentage of the lower intensity solar-energy occurring on cloudy days and during early morning and later afternoon hours. The principal attraction for using solar radiation in this manner is the possibility of providing a higher temperature heat source than the other more common sources.

6.2. Problems with Air-to-Air Heat Pumps

The greatest problems associated with present air-to-air heat pumps are performance and reliability. Due to the newness of the industry, several correctable problems have arisen, such as skimping on the heat exchanger size at the expense of decreased efficiency, poor installation and maintenance procedures have given rise to heat pump failure rates of up to 30% in military base installations. Undoubtedly, the economic reality of expensive energy will cause a premium to be placed upon good design and energy efficiency, and a rapidly expanding market develops a competent service industry, and as a result improved service and lower maintenance costs are likely.

However, there are other problems with air-to-air heat pjmps, which are inextricably bound up with the device itself and the way it is presently used. For example, it can be noted from Fig. 61 that the C.O.P. decreases as the source temperature drops, thus a heat pump which must extract heat from a low temperature source must necessarily have a low C.O.P. Besides being economically unpleasant, the low C.O.P. means that the capacity of the heat pump drops with the ambient source temperature. Since heating demand increases linearly as the ambient temperature decreases, there is a temperature called the "balance point" below which the heat pump capacity is not adequate to meet the heating demand. This causes two problems. First, auxiliary heat must be supplied, which is usually an electric resistance type with a C.O.P. of 1 and is, therefore, expensive. Secondly to avoid using supplemental heating the balance point is kept as low as possible. This is done by increasing the capacity and hence the cost of the heat pump.

Another interesting point to note in Fig. 61 is that the C.O.P. of the heat pump levels off for temperatures above 40^0F (5^0C). That is to obtain a low balance point and good efficiency below 40^0F (5^0C) most air-to-air heat pumps sacrifice efficiency above 40^0F (5^0C). This is not very desirable if the heat pump is to be used in conjunction with low cost solar collectors which are generally designed to collect solar energy at temperatures of 40^0F (5^0C) to 120^0F (50^0C).

Another major problem with air-to-air heat pumps is the defrost cycle. When the ambient temperature is in the range of 20^0F (-5^0C) to 40^0F (5^0C) water in the air freezes to the outdoor coil since the outdoor heat exchanger coil temperature is about 30^0F (17^0C) lower. Frost builds up and eventually blocks the air flow across the exchanger which prevents heat from being transferred. Frost removal is accomplished by temporaily reversing the refrigerant flow in order to use heat from indoors to heat the outdoor coil and melt the frost. During the "defrost cycle" the heat pump is drawing electrical energy, but is not providing heat. Typically a heat pump can spend up to 5% of the time in this mode when the ambient temperature is below 40^0F (5^0C) which causes a corresponding decrease in the performance of the heat pump.

The reverse cycle defrost technique causes severe reliability problems with hermitic type compressor units. Upon defrost superheated refrigerant gas from the indoor coil now flows through an accumulator which is a storage device for liquid refrigerant. This boils the refrigerant in the accumulator and also boils the refrigerant saturated in the lubricating oil in the oil sump. This causes "foaming" which drives off the oil with the refrigerant, and when this occurs the compressor is inadequately lubricated which can result in severe damage to the compressor.

The defrost process proceeds slowly because the cold outdoor temperature produces a low heat pressure. This means that only a trickle of liquid refrigerant flows back into the indoor coil to pick up defrosting heat. So during defrost, the low ambient temperatures causes the liquid phase of the refrigerant liquid - gas phase equilibrium to be favoured in the outdoor coil and liquid refrigerant gathers there. At the termination of defrost, refrigerant flow is reversed and this liquid is sucked back towards the compressor. The presence of an imcompressible liquid in the compressor cylinder can cause tremendous mechancial shock and also break valves. To prevent this the "accumulator" is installed to catch this "flood back" of liquid before it can reach the compressor. Although the accumulator eliminates this problem, its presence in the line during the defrost cycle leads to the additional problem of foaming as discussed above.

6.3. Solar Assisted Heat Pump Systems

There are two major designs for solar assisted heat pumps. These are the parallel and the series systems. The parallel system is shown in Fig. 62. In this system, solar collectors provide heat energy which is placed in a storage device and when the storage is at a high enough temperature, it is used to heat the load. If not the ambient source heat pump is used and if this means is inadequate, electrical resistance heating is used. This system can be regarded as a solar energy system which uses a heat pump for auxiliary heating instead of fossil fuels. It is subject to the unpleasant economic realities of the solar system and also to the performance and reliability difficulties inherent in ambient source heat pumps. As both components function best during warm, sunny weather and worst during cold cloudy weather, they do not complement one another, and as a result, expensive electrical resistance heating is often required as a back-up system.

In the series system, shown in Fig. 63, the solar energy is provided to the storage device which heats the load when possible, as in the parallel system. When this is not adequate, the heat pump removes water from storage and delivers it to the load. For two main reasons the series system has better performance and cost characteristics than that of the parallel system. Firstly, cheaper solar collectors can be used. The dominant economic factor in all solar energy systems is the cost of buying and installing the solar collectors. In the series arrangement, the collectors do not have to have to deliver heat at high enough temperatures to carry the load to be useful. Relieved of this distribution requirement, cheaper collectors which would be inadequate in the parallel system can be used. Secondly, the system efficiency can be greatly increased. If the storage temperature can be maintained above ambient temperature, the heat pump can operate with a higher temperature source and a corresponding higher efficiency. A secondary benefit is that the solar collectors also operate at a higher efficiency. This is because the heat pump removes heat from storage, which lowers the storage temperature. The fluid which is then circulated through the solar collectors is thus also at a lower temperature. All collectors have the property that their efficiency increases as the average circulating fluid temperature decreases for given ambient temperature. Storage losses are also reduced because of the lowered storage temperature.

6.4. Heat Storage in Solar Assisted Heat Pump Systems

Practical storage heating systems must employ thermal storage together with an independent 100% back-up system for those periods experiencing insufficient solar radition. Systems using air or water to transfer heat between the energy source and the storage unit are generally used. Where air is the heat transfer medium, sensible heat storage consisting of a high density material such as beds of crushed rock have been employed. When water is used as the transfer medium, water is also generally used as the storage medium. In many cases water is the preferred choice over air as a water storage system occupies less space than is required for rockbed storage and water refrigerant heat exchangers can obtain smaller "splits" than are possible with air-refrigerant exchangers. Thus, the heat pump source temperature is effectively increased which raises the heat pump C.O.P.

Due to the potential value of a compact, high capacity, economical storage device a large amount of work has been carried out in the field of latent heat storage. On the average, liquid-gass phase chagers involve the most energy, however, due to the containment problems created by the production of gasses, most efforts towards latent heat storage have used the solid-liquid phase change. Materials of special interest include salt hydrates, various paraffins and other organic chemicals. Typical problems presently common in liquid-solid phase change devices are:

1) Supercooling - The liquid continues to cool below the nominal freezing temperature instead of freezing. Nucleating agents have been used to rectify this problem.
2) Incongruence - The two phases separate due to gravity, which reduces the heat transfer rate.
3) Complex Melting - Repeated cycling results in less materials undergoing the phase change after repeated cycling.
4) Reduced Energy Density - Various schemes to alleviate the above problems, result in enlarged devices which are no longer smaller than equivalent sensible storage devices.

When water is used as the storage medium in a series solar heat pump system, there is evidence to suggest that the introduction of thermal coupling between the storage tank and the ground can improve the performance and recue the initial cost of such a system. The ground acts in two roles to achieve these improvements. When the storage temperature is below the ground temperature, the ground provides heat and thus behaves as a buffer to help raise the storage temperature. This smooths out the storage tank temperature fluctuations and raises the annual minimum storage temperature. As a result of the latter, resistive heating is reduced or eliminated. The ground is also used as a quasi-annual storage device, which permits greater usable energy collection and storage. This means that smaller collectors can be used, which reduces the capital cost of such an installation. The annual average storage temperature is also elevated, which makes more direct heating possible and raises the heat pump C.O.P.

6.5. Case Histories of Solar Assisted Heat Pump Installations

6.5.1. The Performance of a Residential Solar Assisted Heat Pump in Colorado Springs

The heating system of the house was classed as a solar boosted (12) heat pump system which was located in Colorado Springs 6,730 feet above sea level. The house was designed to accommodate a family of five and had a confirmed heat loss of 55,000 Btuh (16 Kw) at an outdoor temperature of -9°F (-24°C) with a 7 mile/hour wind (3 metres/sec). This heat loss corresponded to a 24 hour heating load per degree day (65°F base, 18°C) of 17,838 Btu/deg. (5.22 Kw hr/deg). The house peak air-conditioning load was 42,000 Btuh (12.2 Kw).

System utilizes a flat plate pressurized fluid solar collector, a large variable volume uninsulated, underground storage tank and a heat pump for space heating and cooling. The house is oriented south with two banks of collectors fixed at a 55° angle. In addition the roof was covered with white quarty to reflect sunflight falling on it, into the upper bank of collectors. All the windows were double glazed and the insulation value of the walls and the ceiling were R11 and R15 respectively.

Fig. 64 shows the components of the system, and the operating of the system when solar energy is being collected and transferred to storage as heat energy. Whenever the air temperature in the collector is higher than the water temperature in storage, pump A operates and transfers heat from the collector to the storage system.

Heat energy is stored both in the contained water and the ground around the storage tank. Fig. 65 illustrates how the system operates when the house thermostat calls for heat. Pump B circulates hot water from storage through the coil in the house air duct while the house air fan in the heat pump circulates air through the house. This mode always operates when the storage temperature is above 100°F (38°C). Figures 66 demonstrates the system operating mode, when the house thermostat calls for heat and the storage temperature is less than 100°F (38°C).

The object is to have the heat pump always operating with an ideal evaporator temperature, or to optimise the heat pump. In this system, the heat pump utilizes outdoor air as a heat sink, when the outdoor air is above 45°F (7°C) the storage system is utilized as a heat sink for the heat pump. When the heat pump operates using the storage system as a heat sink, the 3-way valve in the collector-to-storage piping system diverts the flow of heat transfer fluid, so that it is circulated between the heat exchanger in the storage system and coil B on the evaporator side of the heat pump, thus simulating a fluid-to-air heat pump connected to the storage system. Dampers are closed in the outdoor air ducts during this operation. The heat pump is equipped with a house air stream electric resistance heater, and the storage system has two electric immersion heaters in it as a back-up system.

Since the system went into operation in September 1974, the collectors have performed in a very satisfactory manner, with the upper or reflective boosted collector collecting 30% more energy than the lower collector array as shown in Table 5. The storage system with a capacity of approximately 2,000 u.s.g. (6,050 litres) has a large thermal inertia, so it takes a long time to heat up. The storage system was used extensively from January through May 1975, to act as a heat sink in optimising the heat pump. Fig. 67 compares the net energy of the storage system to the stored water temperature while the storage system is being used to optimize the heat pump. This figure demonstrates how the inertia of the storage system becomes greatly beneficial in optimizing the heat pump, and its long range capacity to perform this function due to the ground acting as an additional heat sink.

Fig. 68 shows the performance characteristics of the heat pump versus evaporator temperature and the heat loss of the house versus outdoor air temperatures. Using this data, for the optimised heat pump system, a comparison was made between the energy requirements of all electric heating system additions and for equivalent heat pump systems, for all the additional residential customers of Colorado Springs from 1978 to 1985. This information shown in Table 6 illustrates that by using the heat pump system a saving in new generating capacity of 88,000 Kw can be achieved. With an anticipated plant construction cost of $530/Kw generating capacity, a saving of $46 million would be realised.

6.5.2. Comparison of a Number of Heating Systems for a Philadelphia Residence

In this paper the performance of six space heating systems for a 1,500 ft^2 (140 m^2) Philadelphia single-family (20) residence were compared. They were: Electric Resistance Heat, Combustion Furnace, Direct Solar Heating with Auxiliary Combustion Furnace, Conventional Air-to-Air Heat Pump, and two different Solar Assisted Heat Pump Systems.

The home located in Philadelphia at 40^0 north latitude was choosen as being representative of current construction in the north eastern United States. Its design heat loss was calculated to be 53,000 Btuh (61 Kw) and its design cooling load was 32,000 Btuh (10 Kw). The performances of the six different heating systems for this residence were calculated each day and compared for an eight-month heating season from October 1, 1972 through May 31, 1973. These systems were defined as follows:

1) Electric Resistance Heating, 16 Kw Capacity.
2) Combustion Furnace, 100,000 Btuh (30 Kw) input, 60% seasonal efficiency assumed constant throughout.
3) Direct Solar Heating: Vertical, South Facing Solar Collectors
   two glass covers, flat black absorber ($\alpha = \beta =$ 0.95)
   Collector Areas of 375, 500 and 750 ft^2 (35,47,70 m^2)
   Thermal Energy Storage Capacities of 0, 100, 250 and 500 thousand Btuh (0,30,75,150 Kw-hours).
   Auxiliary Heat: Combustion Furnace 60% efficiency.

4) Conventional air-to-air 3 ton Heat Pump.

5) Solar Assisted Heat Pump (A) (See Fig. 69).

6) Solar Assisted Heat Pump (B) (See Fig. 70).

The conventional systems 1, 2 and 4 were operated in their conventional modes, based on instantaneous demands. For System #3, the daily heat load was multiplied by the fraction of the day during which the sun shined, in order to obtain the average heat load which can be supplied directly from the solar collector to the residence. Solar heat collected in excess of this amount is transferred from the collector to thermal energy storage, until the storage device capacity is reached.

Any solar heat exceeding storage capacity is dumped. Next, the remaining heat load of the building is supplied from the thermal storage until it is depleted or until the heat load is satisfied. If the device is depleted prior to satisfying the heat load, supplementary heat is supplied by a combustion furnace. System #5, as shown in Fig. 69, operates as a conventional air-to-air heat pump when the ambient temperature is above 45^{O}F (7^{O}C). For lower ambient temperatures, the refrigerant circuit receives energy from an evaporator coil inside the thermal storage unti. The temperature of that unit is maintained above 60^{O}F (15^{O}C) either by solar energy from the collector or from auxiliary combustion heat. Supplementary electric heating in the supply air duct was also provided in conditions when the heat pump capacity was exceeded. Thermal storage of 200,000 Btu (60 Kw-hr) was effected though the latent heat of fusion of paraffin wax. System #6 as shown in Fig. 70 permits direct solar heating from a 600 ft^2 (56 m^2) collector or from the 10,000 lb water/Glycol thermal storage unit (4,500 Kg) provided the storage temperature is above 105^oF (40^oC). Below that temperature the heat pump is used to boost solar heat from storage to a temperature level sufficient to heat the living room. If the storage temperature drops below ambient temperature the refrigerant evaporation is switched to a conventional ambient air evaporator coil. Supplementary heat was supplied by electric resistance heaters.

The results of this comparison is shown in Figs. 71 to 74. In calculating these results a conversion transmission of 31% was assumed for electrical energy from fuel to the circuit breaker at the residence. Electric resistance heating was assumed to be 100% efficient and the energy consumption of internal fans and pumps was not included.

System #5 consumes 64% less resource energy than electric resistance heating, 30% less than the combustion furnace and 18% less than the conventional air-to-air heat pump. However, it consumed 15% more than the direct solar heating system. This is caused by the depletion of thermal storage from heat pump operation and the inability of the system to utilize the direct solar heating mode on warm, sunny spring and fall days. It was because of this drawback System #6 was designed in order to eliminate this problem. As a result the system consumed 71% less resource energy than electric resistance heating, 44% less than the comustion furnace, 34% less than the heat pump and 15% less than direct solar heating.

The decreased heating capacity of a conventional heat pump during periods of low ambient temperature, when the heating load is large, causes greater proportions of the load to be met by supplemental electric resistance heaters in the early and late months of the heating season. Fig. 74 illustrates this sharply peaking electricity demand during January and February for the conventional heat pump. The solar assisted heat pump System A demonstrated a smoother electricity demand curve. The electric load factor for heat pump System B was 0.57 which was 10% better than the conventional heat pump. It should be noted that these values were monthly averages, which are not directly comparable to the instantaneous load peak used by utilities to calculate load factor and demand charges.

It was concluded that electric resistance heating requires the least capital investment, but had the highest operating cost and resource energy consumption. The additional cost required to go from electric heating to a conventional heat pump could be amortized in five years through savings in electrical energy alone. However, the additional investment in a solar augmentation system for a conventional heat pump, produced an annual saving of only 1.5% for an electricity price of 4¢/Kw-hr. This concluded the author was not sufficient to entice the homeowners to invest in this type of system.

7. MODELLING OF A SIMPLE HEAT PUMP

7.1. Introduction

The heat pump is a refrigeration machine that is designed to extract heat from a sink and deliver it to a load. The useful effect is the heat delivered rather than the heat extracted as for a typical refrigeration machine. Refrigeration machines have normally been compared with regard to their energy efficiency through the magnitude of their coefficient of performance or C.O.P. For a heating machine the coefficient of performance heating (C.O.P.H.) is defined as the ratio of the heat received from the machine by the load (sink) to the net work absorbed by the machine.

$$\text{Thus:} \quad \text{C.O.P.} = \frac{\text{Heat Delivered}}{\text{Work Absorbed}} = \frac{Qd}{W} \quad -(1)$$

The coefficient of performance for the cooling function can similarly be defined as the ratio of the heat extracted from the source by the machine to the work absorbed by the machine.

$$\text{Thus:} \quad \text{C.O.P.C.} = \frac{\text{Heat Extracted}}{\text{Work Absorbed}} = \frac{Qe}{W} \quad -(2)$$

A thermodynamically reversible heat pump has a heating coefficient of performance.

$$\text{C.O.P.H.} = 1 + \frac{TL}{TH - TL} = \frac{TH}{TH - TL} \quad -(3)$$

Where TH is the absolute temperature of the heated space or fluid and TL is the absolute temperature of the cooled fluid. The term $\frac{TL}{TH - TL}$ is the ordinary cooling C.P.O.C. of refrigerating machines and the unity term is the driving energy conserved and transferred to the heating-load. The ratio of the attained cooling C.P.O.C. of a machine to the term $\frac{TL}{TH - TL}$ is a measure of the thermodynamic performance of the machine. Thus the excellence of a heat pump is measured not by the C.O.P.H. but by the C.O.P.C. For example a heat pump with a C.O.P.H. of 2.0 is not merely twice as good as a resistance heater (C.O.P.H.= 1.0) but is infinitely better.

7.2. Heat Pump Circuit

The heat pump unit to be modelled is an air-to-air unit as shown in Fig. 75. Both the evaporator and the condensers are multiple pass heat exchangers whose tubes are finned and cross-flow air is forced through the exchangers shell by fans. An accumulator has been included in the circuit, since it is being used by more and more heat pump manufacturers for vital compressor protection. Most common heat pump systems use the reciprocating compressor, mainly because their available capacities are below the economic flow range of centrifugal and axial compressors.

7.2.1. Reciprocating Compressors

In compressors handling refrigerants, the capacity depends on piston displacement per minute and on volumetric efficiency. The volumetric efficiency is defined as the ratio, in per cent, of the actual delivered volume flow rate (measured at inlet conditions) to the piston displacement. This value is used to compute the actual capacity of a compressor, based on its geometry. Volumetric efficiency may be estimated by the following equation (26).

$$N_v = 100\left[(1 - L) - C\,(r^{1/K} - 1)\right] \qquad .....(4)$$

where N_v = Volumetric efficiency, per cent

C = Cylinder clearance per cent; it is expressed as a decimal fraction of the piston displacement per stroke, and is dimensionless.

L = Loss factor, dimensionless. L is estimated in different ways by various compressor manufacturers. However an accepted value for L is 2r.

r = Compresson ratio, dimensionless

K = Ratio of specific heats, dimensionless

Since the compressor selected has non-cooled cylinders, the discharge temperature will be higher than that corresponding to simple isentropic compression, by an amount caused by the extra ratio imposed by valve pressure drop and by irreversibility. The cooling effectiveness of the cylinder and therefore the discharge temperature are influenced by many factors, which are not adequately defined during the design phase, thus it is not practical to estimate discharge temperature on any rational basis other than simple isentropic compression.

(i.e.) $$T_2 = T_1 (r)^{\frac{K-1}{K}} \quad .....(5)$$

where T_2 = Discharge Temperature

T_1 = Suction Temperature

In order to obtain the adiabatic horsepower input required for the compressor, the following formula can be used.

$$H.P. = \frac{0.01\ Q1\ P1X}{D} \quad .....(6)$$

where X = X factor, which can be obtained from tables

$= r^{0.283} -1$

Q_1 = Gas flow rate, ft^3/min

d = 2.292 (K - 1)/K

The size of the compressor required for a particular service can be determined by equation 7;

$$Va = \frac{VWr}{Nv} \qquad ....(7)$$

where Va = Actual displacement of the compressor, (C.f.m.)

v = Specific volume of refrigerant entering the compressor

Wr = Mass flow rate of the refrigerant, lbs/min.

The compressor shell heat loss can be calculated from the following equation:

$$Q\ shell\ loss = Hcm\ Acm\ (Tcm - TMBCM) \qquad .....(8)$$

where: Acm = Effective shell surface area, (ft^2)

Hcm = Effective shell heat transfer coefficient $(Bth/hr - Ft^2 - {}^{o}F)$

Tcm = Average shell surface temperature, $({}^{o}F)$

Tmbcm = Average ambient temperature around the compressor shell $({}^{o}F)$

In general, an estimated value of Q shell loss is normally used in any calculations, because little manufacturers data is available for evaluating the actual shell heat loss or the average shell temperature.

7.2.2. Suction Lines and Accumulator Pressure Drop

All refrigerant piping may be classified as liquid line, low-pressure gas line, and high pressure gas line. On heat pumps these lines often serve dual purposes, for example on units that interchange for function of the two heat exchangers to reverse the cycle. The piping to the heat exchangers carried low-pressure gas on one cycle and high pressure gas on the alternate cycle. The size of the refrigerant piping is critical, especially on long lines, if the piping is too small, there will be excessive pressure drops which will impair the efficiency of the system. On the other hand, if the piping is unnecessarily large, a greater amount of refrigerant is required. This together with the additional cost of the piping will result in an unnecessarily expensive unit.

For the prediction of the pressure drop in the piping from the evaporator outlet to the inlet of the accumulator a modified fanning equation can be used. As the flow of fluid in this piping is essentially two-phase, the following equation was proposed, to cover such a situation (29).

$$\Delta P = \frac{4\ f'g\ L\ Pg\ V^2g}{2\ gcD}$$

where: ΔP = Pressure Drop, P.s.i.

L = Pipe length, Ft.

Pg = Gas Density, lbs/ft^3

Vg = Gas Velocity, Ft/sec

D = Inside Diameter of Pipe, ft.

f' = Modified Fanning Friction Factor, obtained from graphs in Reference 29

The pressure drop in the compressor suction piping can be estimated using the fanning equation:

$$\Delta P = \frac{f \times (Q)^2}{30,302,351 \times D \times Pm \times A^2} \qquad \ldots\ldots(10)$$

where: Pm = Average pressure in the pipes, p.s.i.a.

Q = Volumetric flowrate of gas, c.f.m.

D = Inside dia. of pipe in feet

A = X^n Area of pipe in ft^2

f = friction factor, for pipe

In order to estimate the pressure drop across the accumulator, the following equation will provide a value which should be accurate enough for modelling purposes (30).

$$h = K \frac{V^2}{2g} \qquad .....(11)$$

where: h = Head loss, feet or fluid

V = Velocity of fluid, feet/sec.

K = Empirical resistance factor, varying with the type of fitting

For the accumulator, this is approximately the exit loss analogous to a pipe entering a receiver or large volume.

Thus: $K = 1.0$

7.2.3. Evaporator and Condenser

An air-to-air heat pump when operating on the heating cycle requires the transfer of the heat from the outside air to the refrigerant vapour in the evaporator and from the high temperature refrigerant vapour in the condenser to the air being conditioned. Such a system involves two heat-transfer surfaces under conditions of relatively poor heat transfer. As a result the heat transfer surfaces required in a heat pump are large, which contributes to the cost, size and the weight of the unit. Consequently considerable ingenuity must be employed in the design of these heat transfer units, in order to keep the size of the heat pump system moderate.

The calculation of heat transfer rates in finned tube heat exchangers involves consideration of both conduction and convection heat transfer. Heat is transferred from the refrigerant to the walls of the heat exchanger and to the fins principally by conduction. Convection then transfers heat from the walls of the tubes and the fins to the air, passing through the heat exchanger. The rate of heat transfer of the heat exchangers can be expressed in general by the equation:

$$q = AU\,\Delta Tm \qquad .....(12)$$

where: q = Rate of heat flow, Btu/hr

A = Heat transfer area,

perpendicular to the air flow, ft^2

U = Overall Heat transfer coefficient, Btu/hr/ft^2/^0F

ΔTm = Log mean temperature difference, ^0F

In order to determine the heat exchange area required to transfer a certain amount of heat, there are certain conditions which have to be fixed such as temperature difference, air velocity/refrigerant velocity through the heat exchanger, number of rows of tubes, and fin density. The conditions in the refrigerant side are more complex in that the refrigerant going to the heat exchanger will probably be superheated and the refrigerant leaving the heat exchanger will probably be subcooled. Since, however, most of the heat picked up by the air comes from the latent heat of condensation, the temperature of the refrigerant entering the condenser can be taken as that of the saturation temperature of the refrigerant, at the pressure at which the refrigerant is supplied to the condenser.

The air velocity is a very important factor in the selection of a finned heat exchanger, since this determines the heat transfer coefficient, the noise requirement and the fan power requirements. Where space permits, air velocities of 600 ft/min are used, if higher values are required, provision for noise absorption must be made in the case of residential units. Heat transfer coefficients based on transverse fin heat exchangers use Colbums heat transfer factor to determine values of U, which coupled with heat exchanger manufacturers data from condensing refrigerant 22, and based on the log mean temperature difference, give the following equations for a typical (6,13) condenser.

$$\text{2 Row:} \quad UAC = 20.15 \left(\frac{CFMC}{AFC}\right)^{.374} X \left(\frac{FPI}{8}\right)^{.59} X\ (1-SF)(AFC)$$

$$\text{3 Row:} \quad UAC = 9.29 \left(\frac{CFMC}{AFC}\right)^{.3764} X \left(\frac{FPI}{8}\right)^{.59} X\ (1-SF)(AFC)$$

$$\text{4 Row:} \quad UAC = 6.95 \left(\frac{CFMC}{AFC}\right)^{.3834} X \left(\frac{FPI}{8}\right)^{.59} X\ (1-SF)(AFC) \qquad \ldots\ldots(13)$$

$$\text{5 Row:} \quad UAC = 6.0 \left(\frac{CFMC}{AFC}\right)^{.3688} X \left(\frac{FPI}{8}\right)^{.59} X\ (1-SF)(AFC)$$

where: UAC = fixed condensing condenser surface area X overall convecting heat transfer coefficient, Btu/hr - ^{o}F

CFMC = air volume through condenser, C.F.M.

AFC = condenser coil face area, Ft^{2}

FPI = fin density, fins/inch of condenser coil

SF = fraction of condenser area devoted to subcooling

0.24 MAC = mass air flow through the condenser X specific heat of air (Btu/hr^{o}F)

For counterflow finned heat exchangers, the condenser temperature can be estimated using the following relationship: (27)

$$TD = TAI + \frac{MR\ X\ (H3 - H1)\ Xe^{UAC/0.24\ MAC}}{0.24\ MAC\ X\ (e^{UAC/0.24\ MAC} -1)} \qquad .....(14)$$

where: Td = Condenser Temperature, ^{o}F

TAI = Air temperature leaving the subcooling portion of the condenser, ^{o}F

H_3 = Enthalpy of refrigerant vapour at compressor discharge, Btu/lb

H_1 = Enthalpy of refrigerant at exit of condenser, Btu/lb

MR = Refrigerant mass flow rate, lb/hr

TAI can be computed from the following equation:

$$TAI = TR + (TD - TE) \times C.P.R. \times MR/0.24\ MAC \quad .....(15)$$

where: TR = Return air temperature, ^{o}F

CPR = Specific heat of saturated liquid leaving the condenser, Btu/lb^{o}F.

$$TE = TR/(1 + Z \times CPR \times MR) + \frac{Z \times CPR \times MR \times TE}{(1 + Z \times CPR \times MR)} \quad ......(16)$$

$$Z = X \bullet Y/(1 - Y) \quad ......(17)$$

$$X = \frac{0.24 \times MAC - CPR \times MR}{0.24 \times MAC \times CPR \times MR} \quad ......(18)$$

$$Y = \frac{e^{UAS/0.24 \times MAC}}{e^{UAS/CPR \times MR}} \quad ......(19)$$

$$UAS = 5.29 \left(\frac{CFMC}{AFC}\right)^{.505} \times \left(\frac{EFPI}{8}\right)^{.7} \times (CNR) \times (SF) \times (AFC)$$

$$......(20)$$

where: CNR = Number of rows in condenser coil

UAS = Surface area of condenser used for condensing X overall convection heat transfer coefficient based on log mean temperature difference (Btu/hr - ^{o}F)

Based on data obtained from heat exchanger articles and from manufacturers data for evaporating refrigerant R22, the overall heat transfer coefficient for the evaporator (6,13) can be estimated from the following equations:

$$\text{2 Row:}\quad UAE = 20.44\left(\frac{CFME}{AFE}\right)^{.372} \times \left(\frac{EFPI}{8}\right)^{.59} \times (AFE)$$

$$\text{3 Row:}\quad UAE = 31.0\left(\frac{CFME}{AFE}\right)^{.3698} \times \left(\frac{EFPI}{8}\right)^{.59} \times (AFE) \quad \ldots\ldots(21)$$

$$\text{4 Row:}\quad UAE = 38.24\left(\frac{CFME}{AFE}\right)^{.3813} \times \left(\frac{EFPI}{8}\right)^{.59} \times (AFE)$$

where: UAE - Evaporator area X overall heat transfer coefficient, (Btu/hr - ^{o}F)

0.24 MAE - Mass air flow through the evaporator X specific heat of dry air, (Bth/hr - ^{o}F)

CFME - Air volume flow rate through the evaporator, (CFM)

AFE - Evaporator coil face area, (Ft^2)

EFPI - Fin density of evaporator coil, (fins/inch)

The ambient air temperature can then be calculated from the following equation: (13)

$$TA = \left[\frac{MR\ (H_2 - H_1)e^{UAE/0.24\ MAE}}{0.24\ MAE\ (e^{UAE/0.24\ MAE} - 1)}\right] + T_G \quad \ldots\ldots(22)$$

Note: See Fig. 75 for T_G.

It should be noted that the above equations are applicable to dry coils only. However, they have been used conservatively to estimate the heat transfer for wet coils, but are not applicable to heavily frosted coils.

It should also be noted that the equations governing the condenser and the evaporator assume that the refrigerant liquid leaving the condenser is subcooled by 10°F (5.6°C) and that the refrigerant vapour leaving the evaporator is superheated by 10°F (5.6°C). However, since the changes in enthalpy brought about by these assumptions is small in comparison to the total change in enthalpy, these enthalpy contributions will be ignored and the vapour leaving the evaporator will be assumed saturated.

7.2.4. Pressure Drops Across the Evaporator and Condenser

Using information obtained from articles and manufacturers data, the following equations estimate the pressure drops across the evaporator and the condenser: (6,13)

$$\text{2 Row:} \quad \Delta P = 3.84 \times 10^{-6} \left(\frac{CFM}{AF}\right)^{1.7} \times (0.235 + 0.0638\ FPI)$$

$$\text{3 Row:} \quad \Delta P = 4.94 \times 10^{-6} \left(\frac{CFM}{AF}\right)^{1.7} \times (0.235 + 0.0638\ FPI) \quad ....(23)$$

$$\text{4 Row:} \quad \Delta P = 6.26 \times 10^{-6} \left(\frac{CFM}{AF}\right)^{1.7} \times (0.235 + 0.0638\ FPI)$$

$$\text{5 Row:} \quad \Delta P = 7.49 \times 10^{-6} \left(\frac{CFM}{AF}\right)^{1.7} \times (0.235 + 0.0638\ FPI)$$

To compensate for velocity head loss and cabinet loss in the outdoor coil, it is good practice to add a further 20% to the pressure drop value calculated using the above equations. For the indoor coil the following additional pressure drops should be added (2).

Cabinet Loss: $\Delta P = 1.5 \times 10^{-7} \times (CFM)^2$ ....(24)

Ductwork Loss: $\Delta P = 5.5 \times 10^{-7} \times \frac{(CFM)^{1.84}}{D^5} \times L$ ....(25)

Where: ΔP = Pressure Drop, (inches of water)

L = Duct length, (Ft)

D = Duct Diameter, (Ft)

7.2.5. Fan Power Requirements

Using the equations set out in the previous section for pressure drops across the evaporator and condenser coils, the fan motor shaft power can be calculated from the following equation: (2)

$$SHP = 4.6 \times 10^{-4} \times CFM \times \Delta P \quad .....(26$$

where: SHP = Fa[illegible]haft horsepower, (H.P.)

Permanent split capacitor motors vary in size and efficiency, and using manufacturers data for this type of motor, the efficiency can be estimated using the following correlation: (2)

$$P = \frac{\text{Watts input}}{\text{S.H.P.}} = 1000 + 2000 \times (0.6 - SHP) \quad .....(27)$$

NOTE: This equation is only applicable to the following range of S.H.P.

$$^{1}/_{12} < \text{SHP} < ^{1}/_{2}$$

where: P = Motor Efficiency

The fan SHP is then multiplied by P to obtain the input watts as follows: (2)

$$\text{Fan Motor Watts} = \text{P X S.H.P.} \qquad .....(28)$$

7.3. Stepwise Performance Program for a Simple Heat Pump Circuit

Details of the heat pump equipment must be specified, in order that certain parameters such as heat transfer coefficients, air flows, fin densities, etc. can be defined.

Step #1

Assume an air velocity of 600 ft/min and a specific heat transfer capacity, select an evaporator which will meet these requirements. Manufacturers data list the face area and fin density for each coil, thus the heat transfer coefficient for the coil can be found using the equations listed in Section 7.2.3. The volumetric air flow can be found as follows: (23, 24, 25)

$$\text{S.C.F.M.} = \text{Coil Face Area X Air Velocity (600 ft/min)} \quad \ldots.(29)$$

In manufacturers data, for each coil there is a wet bulb temperature listed for the entering and departing air. If the enthalpy difference of the air entering and leaving the coil is calculated, the capacity of the coil can be determined as follows:

$$\text{Capacity of Evaporator} = \text{Enthalpy Difference X S.C.F.M. X Density of Air Btu/min} \quad \ldots.(30)$$

This capacity value should approximate the heat required to be transferred by the heat pump circuit, which could be obtained from heat loss calculations for a home into which the heat pump is to be installed.

NOTE: The heat loss for a particular home will assume a specific value for the ambient temperature.

Step #2

Assume a value for evaporator pressure (a good indication can be found in manufacturers data).

Step #3

Select an initial value for the refrigerant mass flow (MF). Select an initial value for TAI (Air temperature leaving the sub-cooling portion of the condenser).

Step #4

Assume a value for the condensing pressure.

Step #5

Using equation (9) in Section 7.2.2., the pressure drop from the outlet of the evaporator to the entrance of the accumulator can be calculated. In order to determine the pressure drop, the vapour leaving the evaporator must be assumed saturated. A nominal pipe diameter and length must be selected, upon which ΔP can be found.

Step #6

Flow through the accumulator is assumed to be gaseous only. Thus the pressure drop across the accumulator can then be calculated using Equation (11) listed in Section 7.2.2.

Step #7

Only gas flow exists in the compressor suction piping as a result the volumetric gas flow can be calculated assuming a nominal pipe size, using equation (10).

Step #8

The pressure drops calculated in Steps 5, 6 and 7 can be checked against manufacturers data or this data can be used directly.

Step #9

Using the pressure drop values calculated in steps 5,6 and 7, the pressure at the entrance to the compressor (PA) can be calculated.

Step #10

Assume a trial value for the condenser temperature T_D (since a condensing pressure was assumed in Step 4, a value for T_D can be obtained from manufacturers data).

Step #11

Assuming a specific length and a nominal diameter for the piping connecting the compressor outlet to the condenser, calculate the pressure drop in the piping using equation (10) outlined in Section 7.2.2.

Step #12

From the estimated value of P_2 and the piping pressure drop calculated in Step #11, the discharge pressure of the evaporator (P_B) can be found.

Step #13

Since the pressure ratio of the compressor is now known, using equation (4) in Section 7.2.1. the suction temperature can be found. This allows the points A and B as shown in Fig. 75 to be fully defined (temperature and pressure) and as a result the enthalpy of the fluid at these points can be defined from tables.

Step #14

If a particular compressor is specified, the displacement (C.F.M.) is known. The mass flow rate of the refrigerant can then be calculated from equation(7) in Section 7.2.1. and the required horsepower from equation (6).

NOTE: Steps 13 and 14 can be combined and the capacity and horsepower calculated using calometric compressor curves supplied by manufacturers which only require the points A and B to be defined (Pressure and Temperature).

Step #15

If manufacturers data is not available to calculate the shell heat loss of the compressor, a specific fraction of the compressor capacity can be used. However, the suction gas temperature plays an important factor in the selection of this fraction (e.g. Trane F-230 unit at $40^{O}F$ suction temperature, shell loss is approximately 2%).

Step #16

Assume a value for TAI (air temperature leaving the sub-cooling portion of the condenser).

Step #17

Check the condenser capacity, to see if the heat load (MR $(H_3 - H_1)$) can be transferred to the return air with the condensing temperature of T_D. This is done by using equation (14) in Section 7.2.3. Calculate the value of T_D. Compare this value with the value assumed in Step 10. If they do not agree Steps 2 through 16 are iterated until agreement is achieved.

Step #18

Compute the value for TAI from equation (15) in Section 7.2.3. If this value agrees with the initial value assumed in Step #3, proceed to Step #19. However, if agreement is not achieved, review the initial guess and repeat Steps 3 to 17 until tolerable agreement is achieved.

Step #19

The ambient air temperature TA is calculated using equation (22) in Section 7.2.3. Compare this value with the desired value, if agreement is achieved proceed to Step #20, if not revise the guess for P1 and return to Step #2.

Step #20

Compute the C.O.P.H. of the heat pump, accounting for shell heat loss in the compressor, and the heating effect of the fan motors using equation 1.

7.4. Defrost Cycle

7.4.1. Defrost Problems of Heat Pumps

The heat pump defrost problem can be summarized by the fact that air-to-air heat pumps having an excellent steady state C.O.P. for average winter temperatures have poor seasonal performance factors. The C.O.P. is degraded by long defrost cycles cancelling out the gain of heat extracted from the outdoor air. Long defrost cycles with their functional problems have convinced some dealers that an air-conditioning unit plus straight electric heat is a more practical service free and cost-effective heating and cooling system than the air-to-air heat pump (22, 28).

Frost deposits on the outdoor coil surface under weather conditions, during which the surface temperature is below the freezing point and below the dew point temperature of the entering air. Frost does not accumulate uniformly on the fin-tube structure since it is not at a uniform surface temperature. As the air passes through the coil, it is cooled and the convection temperature difference between air and refrigerant declines as does the heat exchange per square foot of fin surface and thus the average fin temperature. However, if the coil is frosting the specific humidity of the air declines also, so that the gross dew point depression of the surface does not change significantly from front to back. As a result, frost deposits rather uniformly in the gross sense, but preferentially near the tubes as opposed to far from the tubes. The frost surface is relatively rough, so that as frosting progresses air flow is restricted near the tubes and is diverted to the space between the tubes. The heat exchange burden moves away from the tubes and must be conducted in fin metal, through a larger average path, so the effective fin efficiency declines. As a result the coil friction factor increases slowly at first, then more and more rapidly for each increment of frost laid down. The coil thermal conductance (UA) increases at first because frost is rougher than a clean surface, then declines rapidly because of the foregoing. Air flow falls off as a result of equilibrium between the coil flow resistance factor and the fan ΔP versus air flow characteristic.

Most texts which attempt to model the performance characteristics of a coil using the "Number of Transfer Units", and "Exchanger Effectiveness" approach, show that the refrigerant temperature (T_R) and the ambient temperature (TA) are related by the following equation: (6, 13)

$$TR = TA - \frac{Q}{0.24\left[MA\ (1 - e^{-UA/.24\ MA}\right]} \qquad \ldots\ldots(31)$$

Where: Q - Heat burden, (Btu/hr)

Early in frosting MA decreases, but UA increases and the exponential term dominates the denominator (increasing) so the refrigerant temperature rises. As frost build-up continues UA will peak and then decline. When UA has returned to its clean coil value both air flow and conductance effects are in the same direction and the refrigerant temperature is slightly depressed and drops rapidly with further frosting. This event marks the critical frost loading of the coil, since it is the now rapidly declining T_R which reduces the system capacity. In densely finned coils, this occurs at about 85% clean coil air flow and with wide fin apacings at about 80% clean coil air flow.

7.4.2. Frost Penalty Modelling

The driving force for frost formation is the saturated humidity at ambient temperature times the actual minus the critical relative humidity (defined as the R.H. at ambient temperature at which the surface temperature equals dew points). The mass coefficient (for frosting) is the heat transfer rate divided by the product of the specific heat of air and the ambient to surface temperature difference. Thus, the frosting rate is proportional to the relative humidity excess over critical and the ambient saturated humidity. Because the latter factor ranges over a 6/1 ratio (72% at 40^0F to 84% at 0^0F) and the available range for the former over nearly a 2/1 ratio, high frosting rates are only possible on relatively mild humid days and the problem is acute in the 30^0 to 40^0F ambient range.

Calculation of the critcal frost deposit time has been cited in literature and is based mainly on the Prandtl model of boundary layer heat and mass transfer (28).

By the first low:

$$Q_H = 0.24 \text{ MAE}' \times \text{LMTD} = 0.24 \text{ MAE} (\text{EDB} - \text{LDB}) \qquad .....(32)$$

$$\frac{\text{MAE}'}{\text{MAE}} = 1 - \beta = \frac{\text{EDB} - \text{LDB}}{\text{LMTD}} \qquad .....(33)$$

by the Mass Heat Transfer Analogy:

$$1 - \beta = \frac{\text{EW} - \text{LW}}{\text{LMHD}} \qquad .....(34)$$

$$\frac{\text{LMHD}}{\text{LMTD}} = \frac{\text{EW} - \text{SW}}{\text{EDB} - \text{TS}} \qquad .....(35)$$

$$\text{LW} = \text{EW} - \text{LMHD} \frac{(\text{EDB} - \text{LDB})}{\text{LMTD}} \qquad .....(36)$$

$$\text{LW} = \text{EW} - (\text{EW} - \text{SW}) \frac{\text{QH}}{0.24 \text{ MAE} (\text{EDB} - \text{TS})} \qquad .....(37)$$

$$\text{LT} = \frac{\text{LF} \times \text{Face Area}}{\text{QM}} = \frac{\text{LF} \times \text{Face Area} \times 0.24 (\text{EDB} - \text{TS})}{\text{QH} (\text{EW} - \text{SW})} \quad ..(38)$$

where: QH - Heat Burden, (Bth/hr

LW - Coil leaving specific humidity, (lb/lb)

SW - Saturated specific humidity at TS, (lb/lb)

EDB - Ambient dry bulb, (^{o}F)

MAE - Air flow (lb/hr)

LMTD - Log mean temperature difference, (^{o}F)

LMHD - Log mean specific humidity difference (lb/lb)

LF - Critical frost loading, (lb/ft^2)

QM - Frost rate, (lb/hr)

EW - Ambient specific humidity (lb/lb)

TS - Coil temperature (^{o}F)

LDB - Leaving dry bulb (^{o}F)

MAE' - Equilibriated boundary layer air flow (lb/hr)

LT - Critical frost deposit time (hr)

β - By-pass ratio (defined as the fraction of the core flow which has not entered the boundary layer)

The critical frost loading is obtained from the fan characteristic and the flow impedance versus frost loading of the coil. Data compiled by Stoecker, which is based on a coil of 3.14 ft^2 face area, 5 rows on 1 - $^7/_{16}$" square pitch, $^5/_8$" tubes and nine fins per inch of 0.0011 inch stock correlates fairly well to (28)

$$\Delta P = K \times (CFM)^{1.7} \qquad \text{.....(39)}$$

where: $K = 8.4 \times 10^{-7} + 3.589 \times 10^{-7} (F) + 3.178 \times 10^{-7} (F)^2 + 8.19 \times 10^{-9} (F)^8$ .....(40)

F = Frost load, lb/ft^2 face area

ΔP = Coil pressure drop (inches water)

A more usable form of K which is valid for 1 X 0.86 inch tube pitch with about 0.005 inch stock is as follows:

$$K = [5.96 \times 10^{-6} + 1.09 \times 10^{-4} (F)^2] \times [N]^{.813} \times [0.234 + 0.085 \, FPI] \qquad \text{.....(41)}$$

where: F - Frost deposit, lb per row per ft^2 face area

N - Number of rows

FPI - Fins per inch

A stepwise procedure to calculate the energy consumed by the defrosting cycle during a typical heating season, can be carried out as follows:

Step #1

From the fan characteristics calculate the pressure drop across the clean coil.

Step #2

Determine the pressure drop at 85% of the clean pressure drop.

Step #3

Calculate K from equation (41) assumed F.P.M. = 600

Step #4

Solve equation (39) for F.

Step #5

Calculate the critical area frost loading as follows:

$$LF = N \times F(\text{critical}) \ \text{lb/Ft}^2$$

Step #6

Using data which can be obtained from Ashrae Charts, for a particular location, calculate the number of hours during the heating season for which conditions of temperature and humidity are conducive to frosting of the coil.

Step #7

For each new condition of temperature and humidity at which frosting occurs, find QH, TS EDB and SW. Calculate LT for each new temperature, humidity condition.

Step #8

Divide the number of hours spent at a particular set of frosting conditions (temperature and humidity) during the heating season, by the frosting time LT and the defrost cycle length, to find the number of defrost cycles. In order to obtain the net energy for that number of defrost cycles, multiply the number of defrost cycles by the kilowatt hours per defrost cycle, which should include the net energy applied to the coil plus any energy used to heat the home during the defrost cycle.

8. CONCLUSIONS

In view of the foregoing, the following conclusions can be drawn:

Due to the larger number of degree days, the seasonal performance factor of a heat pump principally developed for the United States is reduced, when operated under Canadian climatic conditions.

One of the greatest problems associated with heat pumps is performance and reliability. Due to the newness of the industry several correctable problems such as inadequate system design, poor installation and maintenance procedures have severely hampered the acceptance of heat pumps in the commercial market.

Decreased heating capacity of conventional heat pumps during periods of low ambient heat, when the heating load is largest, causes greater proportions of the load to be met by supplemental electrical resistance heating. As a result on a cold day, the electrical demand for a heat pump is virtually no less than it would be for an electrical resistance heating system, which reduces the C.O.P. of the system and makes such a heating system less attractive from an economical viewpoint. Only if electrical demand is penalized will this encourage further development work on lower balance point systems.

In the ambient temperature range of 20^0F (-5^0C) to 40^0F (5^0C) the C.O.P. of air-to-air heat pumps is degraded by long defrost cycles. Thus air-to-air heat pumps having an excellent steady state C.O.P. for average winter temperatures have poor Seasonal Performance Factors.

The "Series Solar Assisted" heat pump system is more cost effective than the "Parallel" system under northern climatic conditions.

Indications are that the trend towards "Solar Assisted" heat pumps will continue, because of the unique advantages which they offer over conventional heat pump systems. However, this growth will be slow particularly in eastern Canada, where the high incidence of cloud cover during the winter months requires the use of an expensive heat storage system, electric resistance heating systems or both as a back-up.

LIST OF FIGURES

FIG. 1

BASIC AIR-TO-AIR HEAT PUMP SYSTEM, COOLING AND DEFROST CYCLE

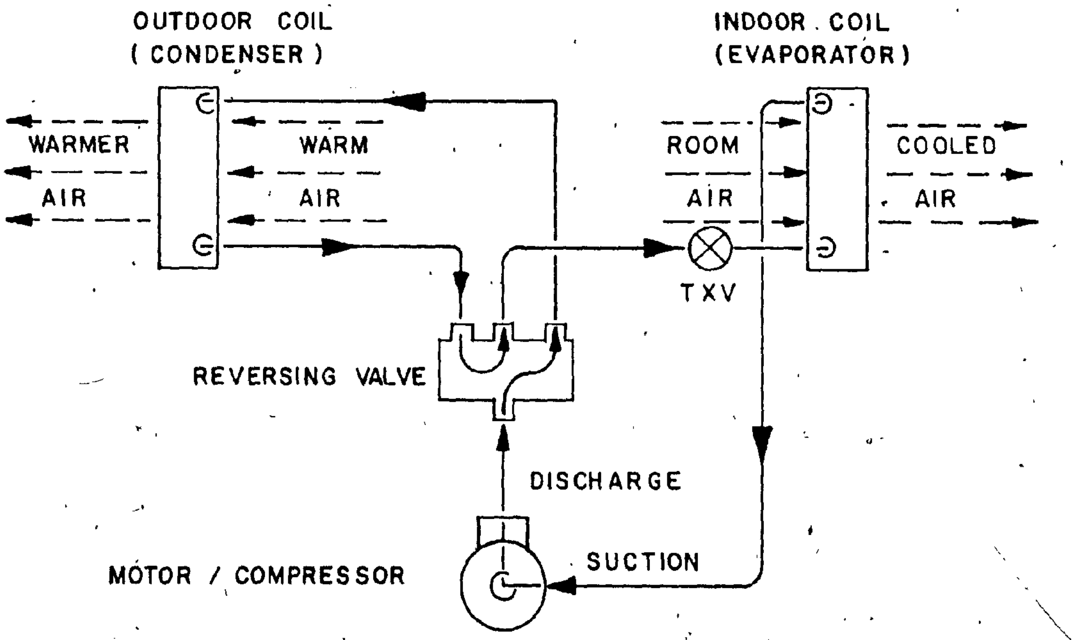


FIG. 2

BASIC AIR-TO-AIR HEAT PUMP SYSTEM, HEATING CYCLE

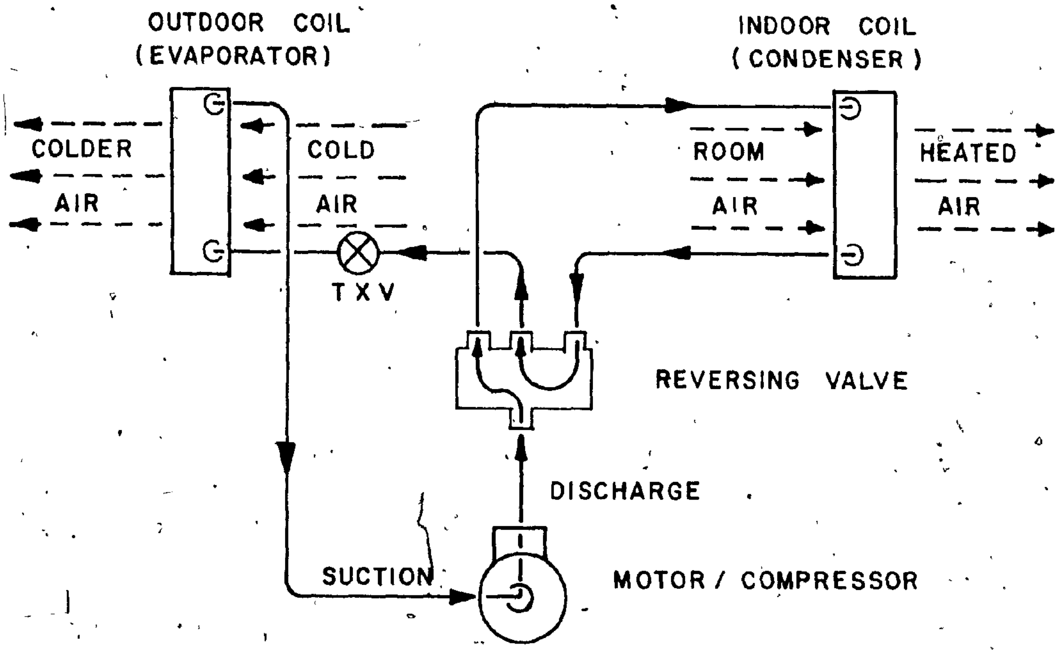
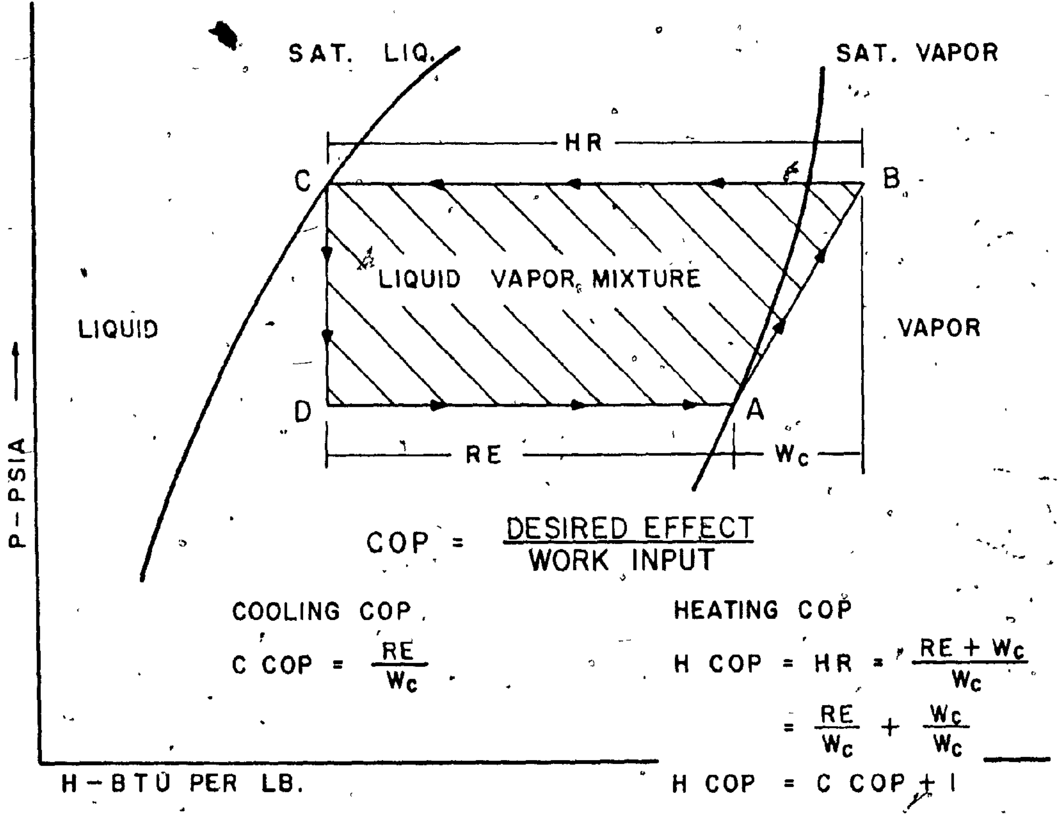


FIG. 3

SIMPLE REFRIGERATION/HEAT PUMP CYCLE



$$COP = \frac{\text{DESIRED EFFECT}}{\text{WORK INPUT}}$$

COOLING COP

$$C\ COP = \frac{RE}{W_c}$$

HEATING COP

$$H\ COP = HR = \frac{RE + W_c}{W_c}$$

$$= \frac{RE}{W_c} + \frac{W_c}{W_c}$$

$$H\ COP = C\ COP + 1$$

FIG. 4

HEAT PUMP CONDENSER AND EVAPORATOR PERFORMANCE

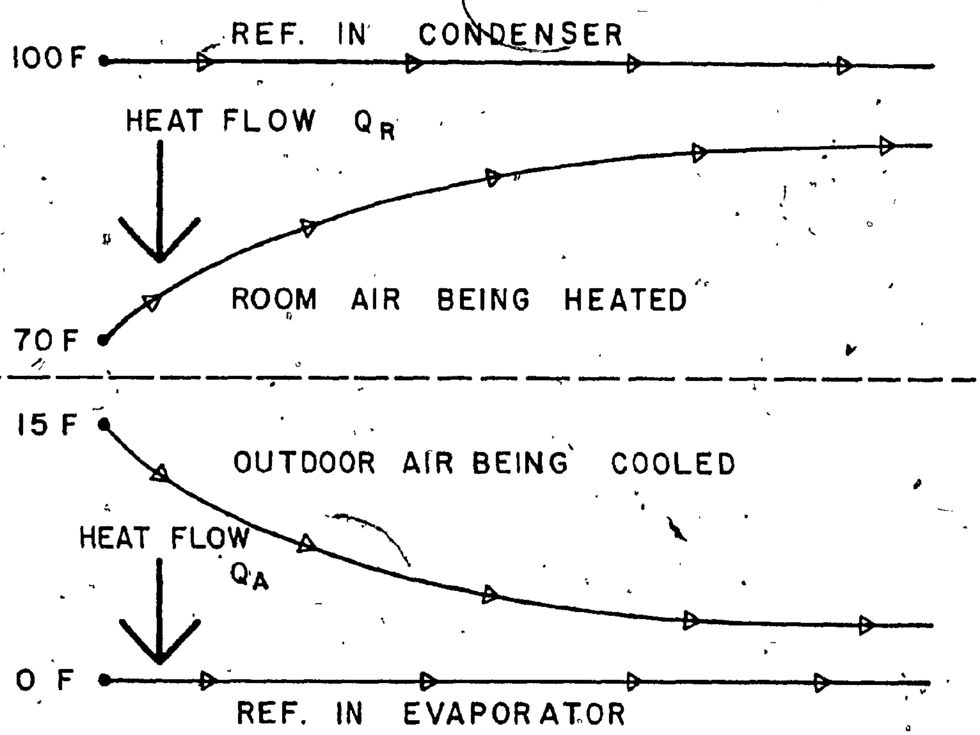


FIG. 5

ENTHALPY (BTU/LB ABOVE SATURATED LIQUID AT -40^{0}F). (E.I. DU PONT DE NEMOURS AND CO. INC.

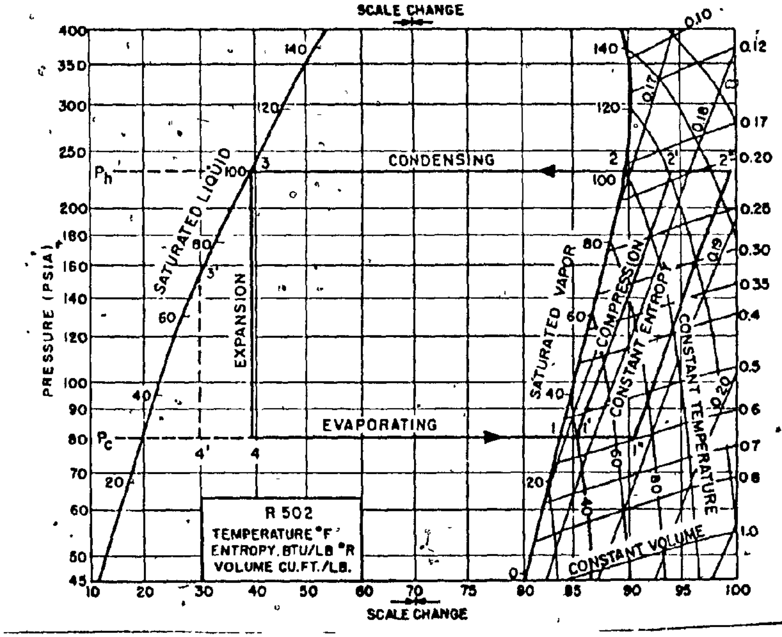


FIG. 6

HEATING CAPACITIES AT VARIOUS OUTDOOR AIR AND SATURATED REFRIGERANT TEMPERATURES FOR HEAT PUMP HAVING A 100-TON COOLING CAPACITY

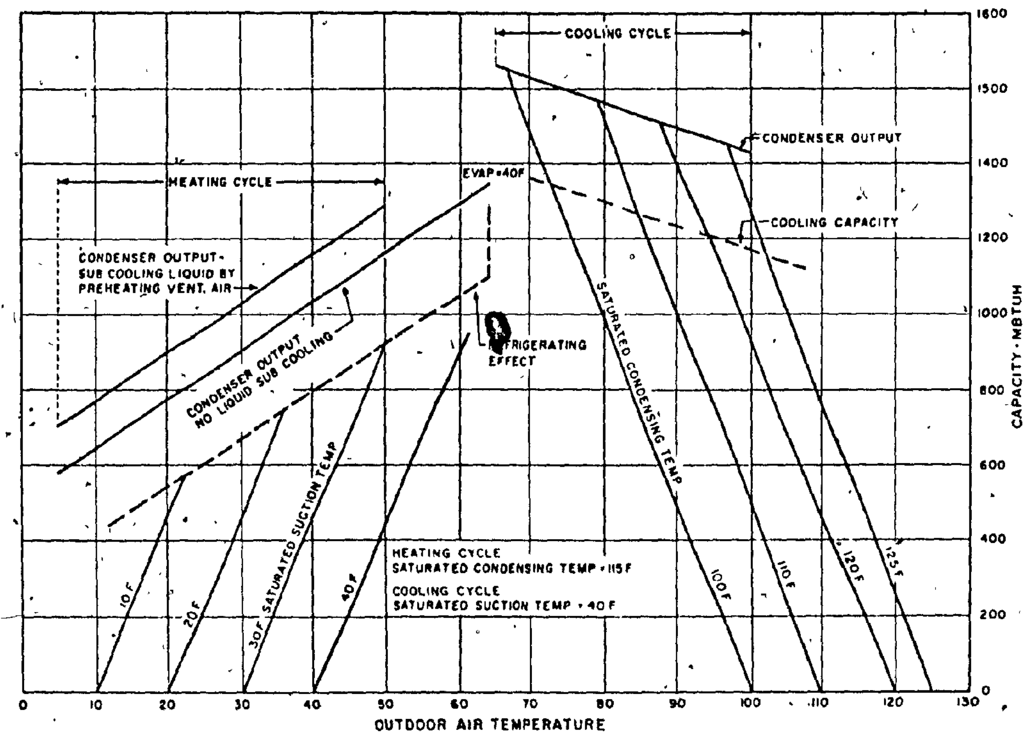


FIG. 7

EFFECT OF TEMPERATURE DROP IN HEAT EXCHANGER ON EFFICIENCY

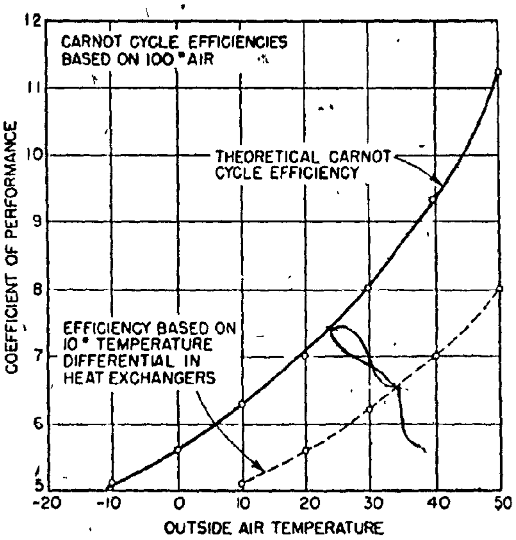


FIG. 8

AIR-FLOW REQUIREMENTS

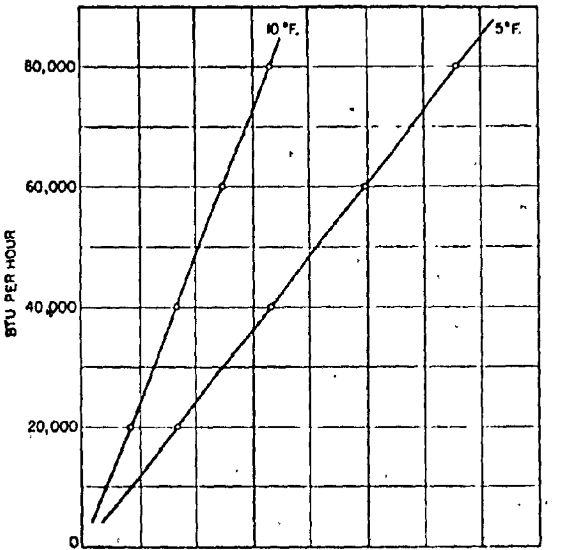
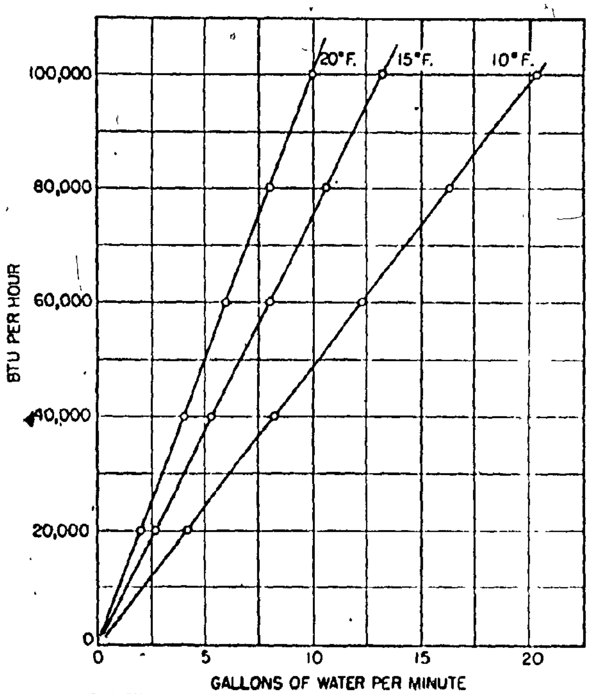


FIG. 9

WATER-FLOW REQUIREMENTS



2

FIG. 10

EARTH HEAT CAPACITY (HEATING AND VENTILATING)

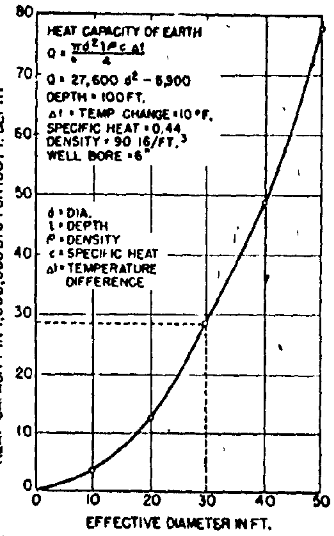


FIG. 11

WELL SYSTEM FOR EARTH-HEAT RECOVERY

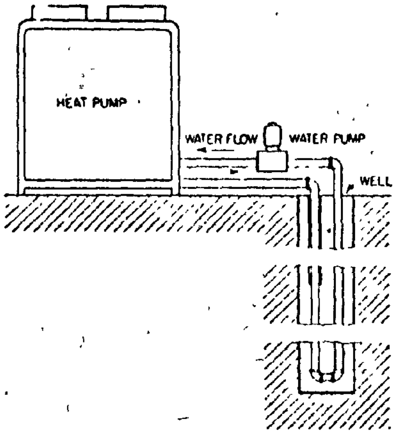


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FIG. 12

SUBMERGED EARTH SYSTEM.

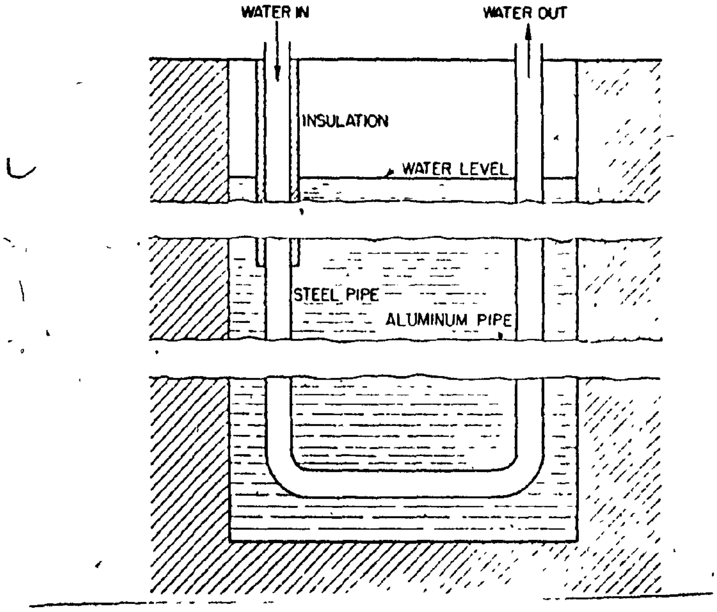


FIG. 13

HEAT PUMP ON COOLING CYCLE

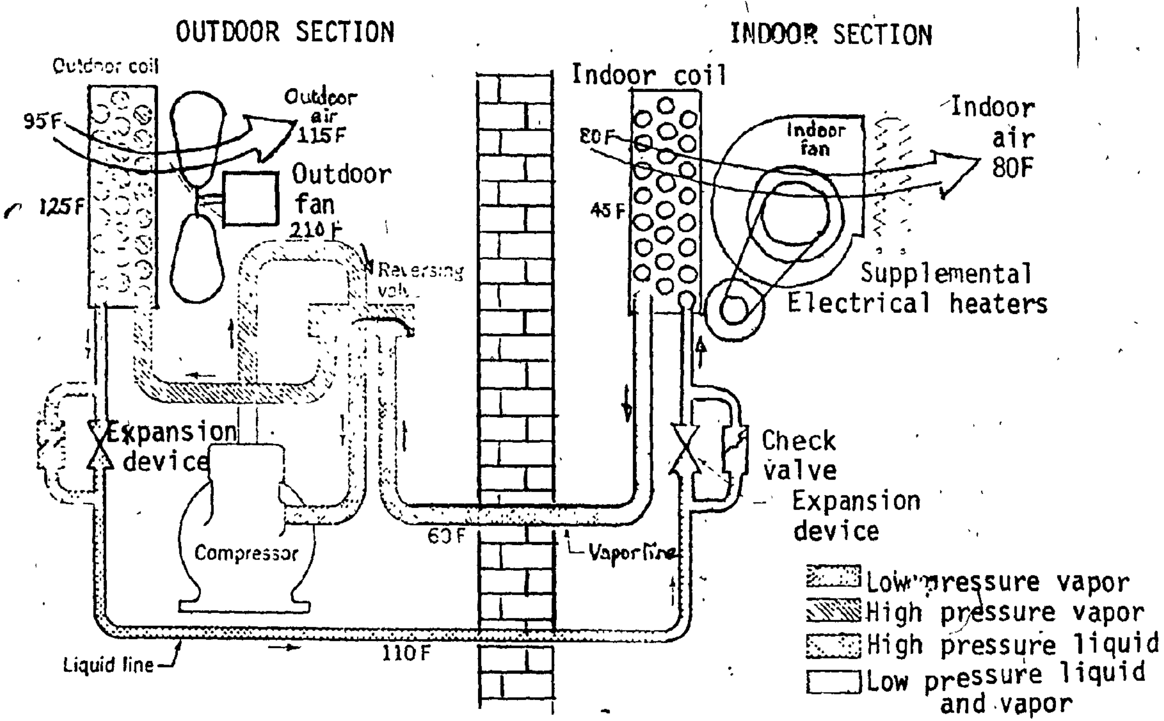


FIG. 14

HEAT PUMP ON HEATING CYCLE

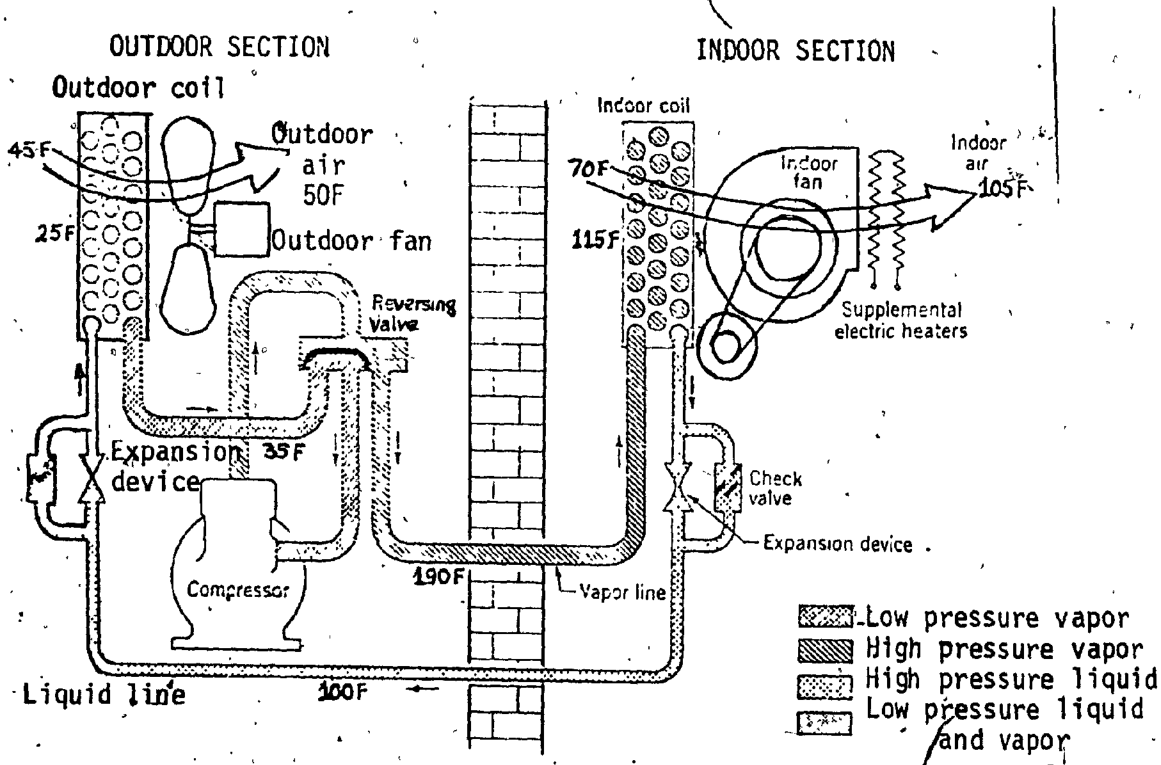


FIG. 15

POWER SAVER SCHEMATIC

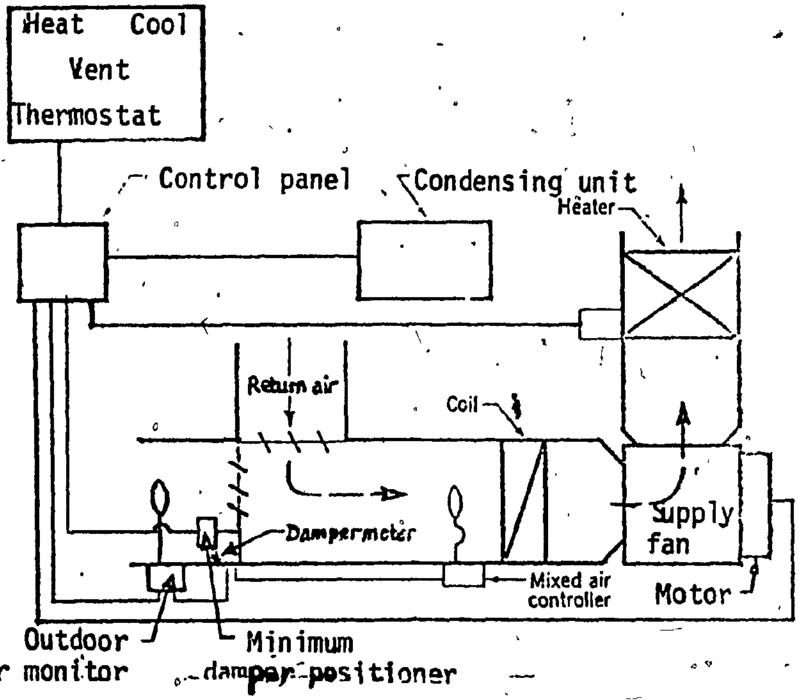


FIG. 16

VENTILATION SCHEMATIC

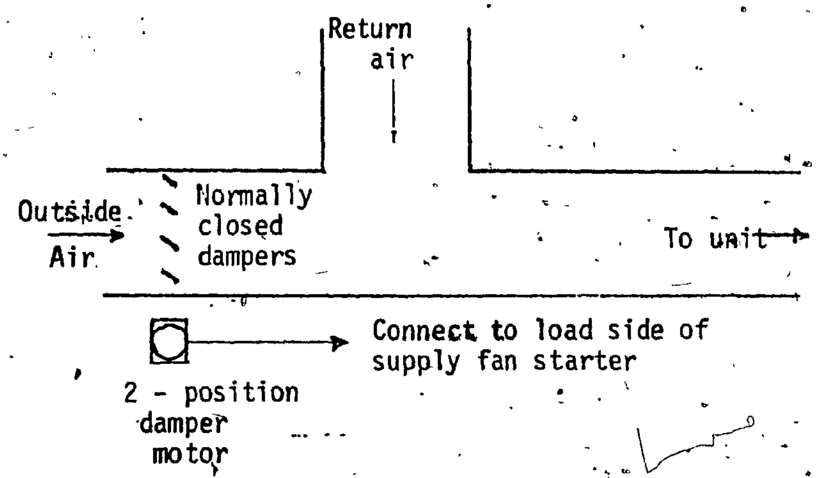


FIG. 17

VENTILATION SCHEMATIC

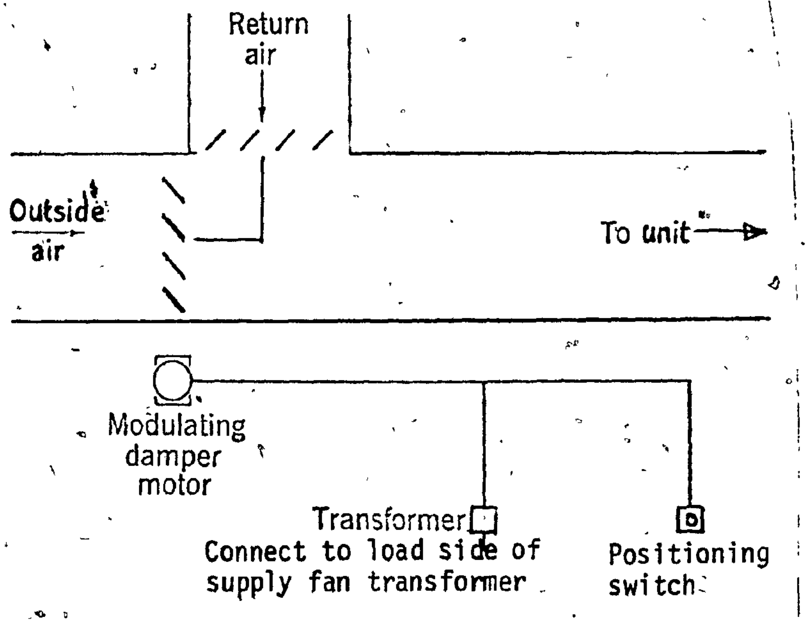
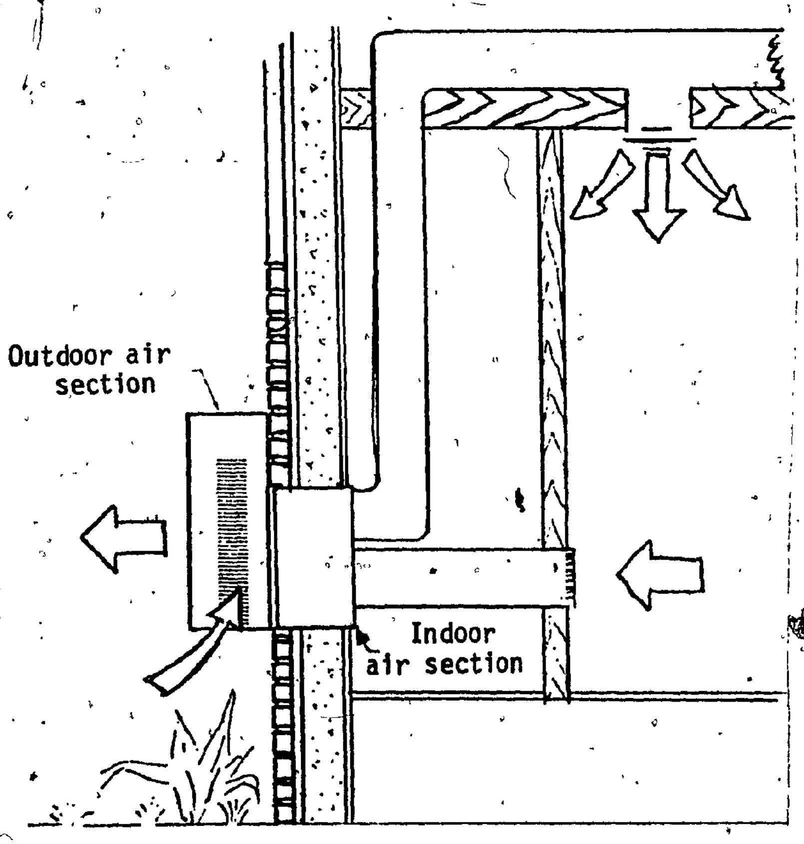


FIG. 18

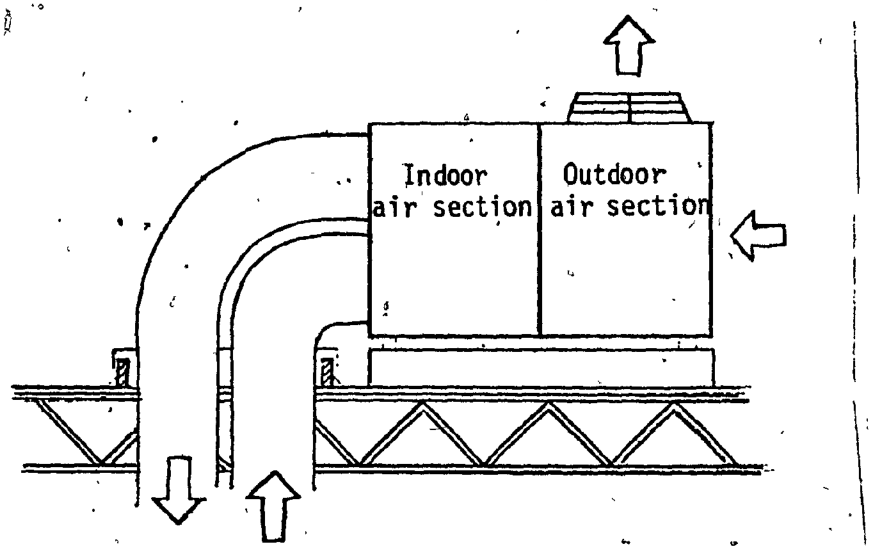
HORIZONTAL, SINGLE PACKAGE UNIT MOUNTED THRU THE WALL



9, 10

FIG. 18

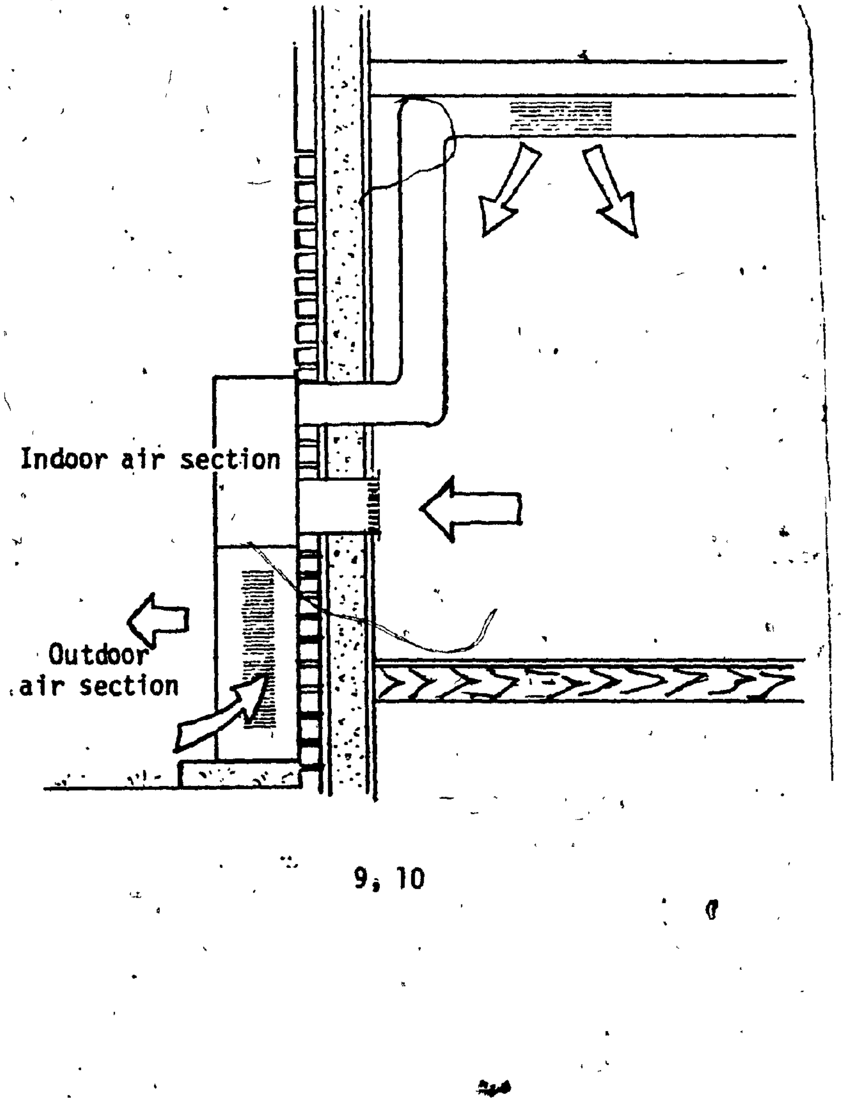
HORIZONTAL CABINET, SINGLE PACKAGE TYPE HEAT PUMP MOUNTED ON ROOF



9.10

FIG. 18A

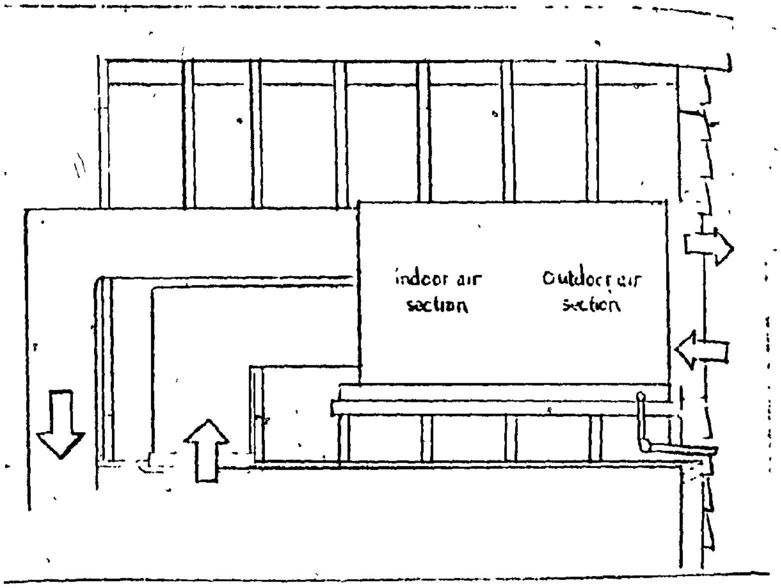
A VERTICAL CABINET, SINGLE PACKAGE TYPE HEAT PUMP MOUNTED OUTSIDE ON WALL OR SLAB



9, 10

FIG. 19

HORIZONTAL, SINGLE-PACKAGE UNIT MOUNTED IN ATTIC, FLUSH TO OUTSIDE WALL



9, 10

FIG. 19A

VERTICAL CABINET, SINGLE PACKAGE, MOUNTED INSIDE CONDITIONED SPACE WITH OUTSIDE AIR CONNECTIONS THRU THE WALL

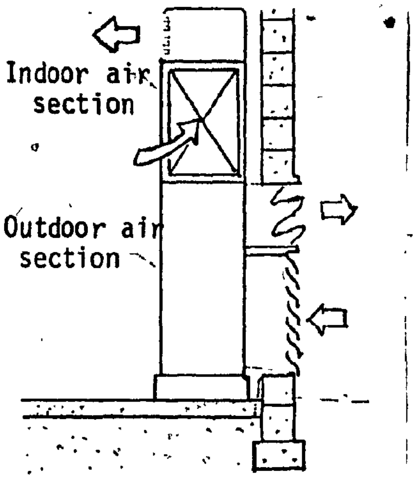


FIG. 20

HORIZONTAL, SINGLE PACKAGE LOCATED IN ATTIC
WITH BOTH INDOOR AND OUTDOOR AIR DUCTED

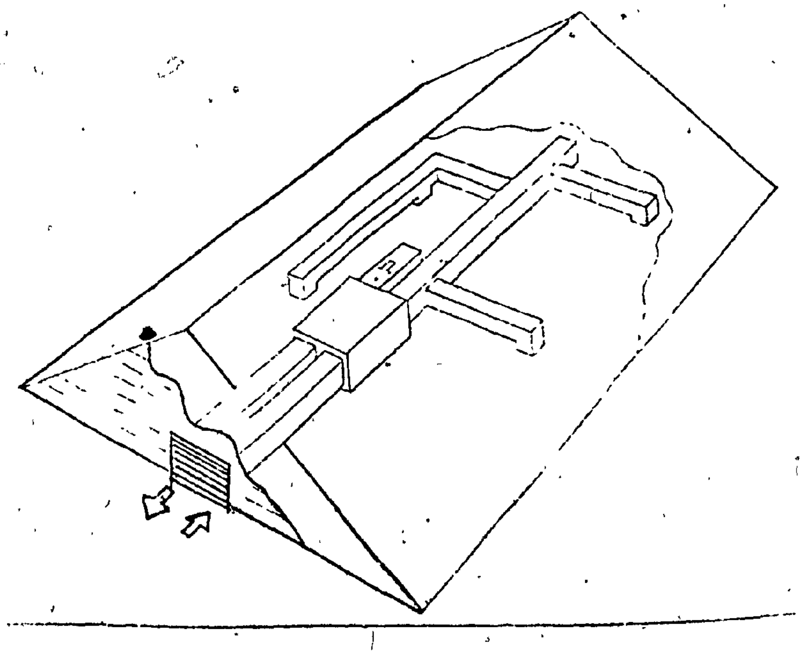


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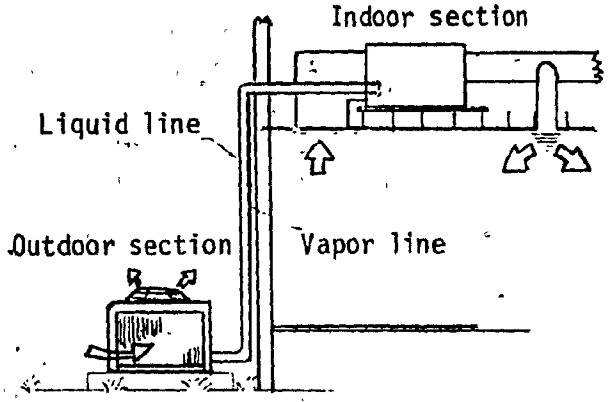
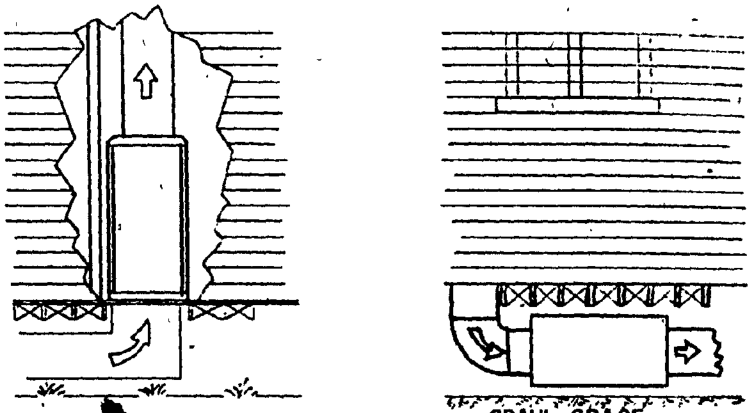
FIG. 21

TYPICAL EXAMPLES OF SLAB OR GROUND-LEVEL INSTALLATIONS



ATTIC INSTALLATION



CLOSET

CRAWL SPACE

9, 10

FIG. 22

HOUSE COOLING AND HEATING NEEDS VARY WITH OUTDOOR TEMPERATURE

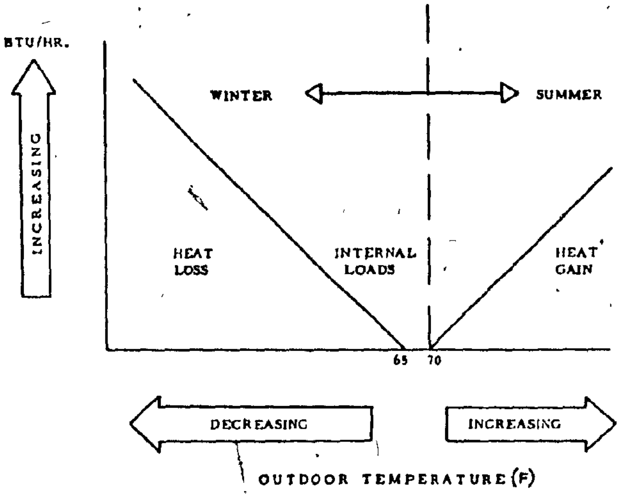


FIG. 23

HEAT PUMP CAPACITY AND EFFICIENCY VARY WITH OUTDOOR TEMPERATURE

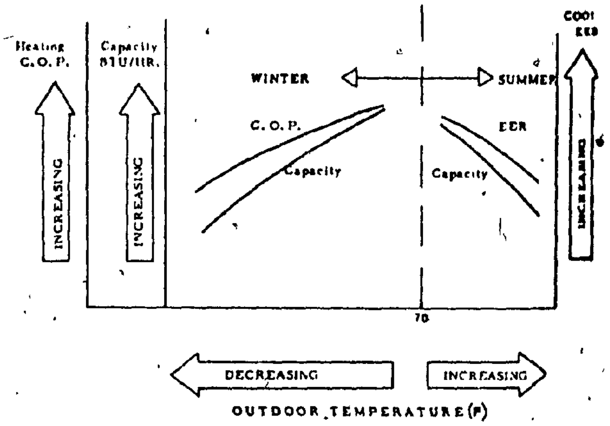


FIG. 24

HEAT PUMP PERFORMANCE IS INVERSE TO HOUSE NEEDS

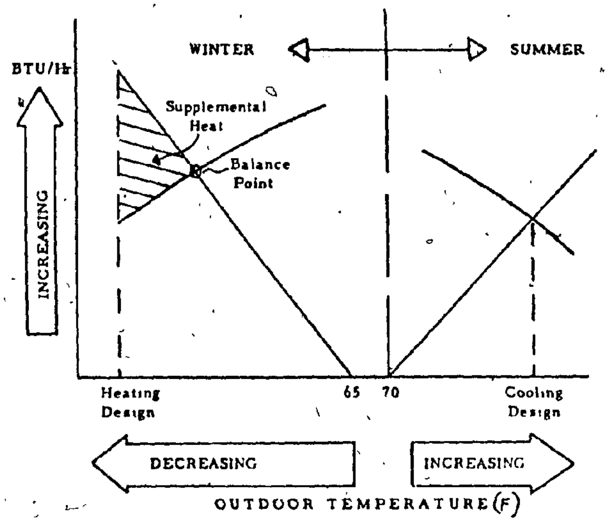


FIG. 25

HEAT PUMP APPLICATION

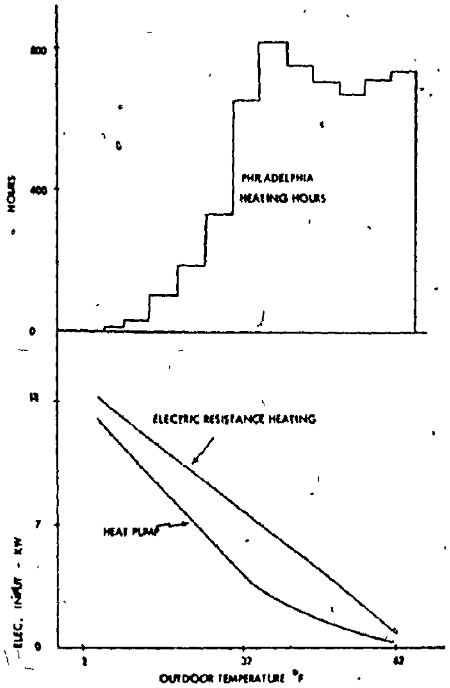


FIG. 26

HEAT PUMP APPLICATION - PHILADELPHIA

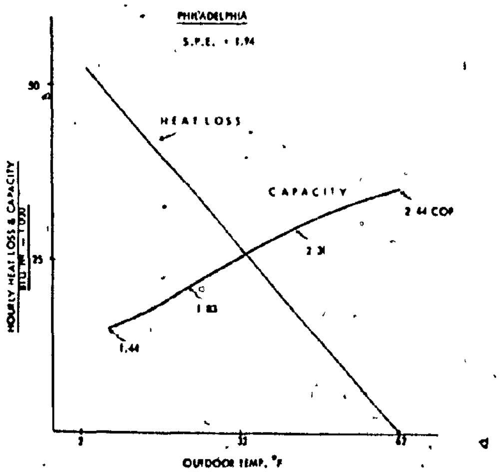


FIG 27

DERIVATION OF SEASONAL PERFORMANCE FACTOR (SPF)

PHILADELPHIA / 2100 Sq. Ft.
CALCULATION

by: R.C. Barnett date: 3/75

LOAD	29 469	52 424
	COOLING	HEATING

DESIGN TEMPERATURE		
I D	O D	DIFF
70	5	65

EQUIP	CFM
	1000

				HEAT PUMP ALONE					SUPPLEMENTARY HEAT			
OUTDOOR TEMP (5° INCREMENTS)	BTUH LOSS PER 1°F	HEAT LOSS – TD (OUTDOOR TEMP BELOW 65°F COLUMN A)	HEAT LOSS (BTUH) B × C	HEAT PUMP HEATING CAPACITY (BTUH) MFR DATA	HEAT PUMP RUNNING TIME (%) D ÷ E	HEAT PUMP INPUT (KW) MFR DATA	SEASONAL HEATING HOURS US WB	SEASONAL HEAT PUMP INPUT (KWH) F × G × H	RESISTANCE HEAT INPUT (BTUH) D minus E	RESISTANCE HEAT INPUT (KW) J ÷ 3413	SEASONAL RESISTANCE HEAT INPUT (KWH) H × K	DEGREE HOURS C × H
A	B	C	D	E	F	G	H	I	J	K	L	M
62	806.5	3	2.42	35.0	6.9	4.2	735	213				2205
57		8	6.45	34.1	18.9	4.1	710	551				5680
52		13	10.48	32.9	31.9	3.9	663	824				8619
47		18	14.52	31.4	46.2	3.8	701	1 232				12618
42		23	18.55	29.2	63.5	3.7	758	1 782				17434
37		28	22.58	27.0	83.6	3.6	818	2 463				22904
32		33	26.62	24.6	100	3.5	654	2 289	2.02	.59	386	21582
27		38	30.65	22.5		3.4	335	1 139	8.15	2.38	800	12730
22	CONSTANT	43	34.68	20.6		3.3	189	624	14.08	4.13	780	8127
17		48	38.71	18.5		3.2	100	320	20.21	5.92	592	4800
12		53	42.75	16.6		3.1	32	99	26.15	7.66	245	1696
7		58	46.78	14.8		3.0	9	27	31.98	9.37	84	522
2		63										
–3		68										
–8		73										
–13		78										
–18 & below		83										
								11 563	← TOTALS →		2 887	118917

		B		M (TOTAL)		P	
ANNUAL REQUIREMENT DUCTED RESISTANCE HEAT	=	806.5	×	118917 / 3413	=	28 101	KWH

		I (TOTAL)		L (TOTAL)		N	
ANNUAL REQUIREMENT HEAT PUMP SYSTEM	=	11 563	+	2 887	=	14 450	KWH

SEASONAL PERFORMANCE FACTOR	=	$\left(\frac{P}{N}\right)$	=	$\frac{28\ 101}{14\ 450}$	=	1.94	S P F

FIG. 28

AREA UNDER CURVE IS TOTAL HEAT REQUIRED
BY THE PHILADELPHIA HOUSE DURING YEAR

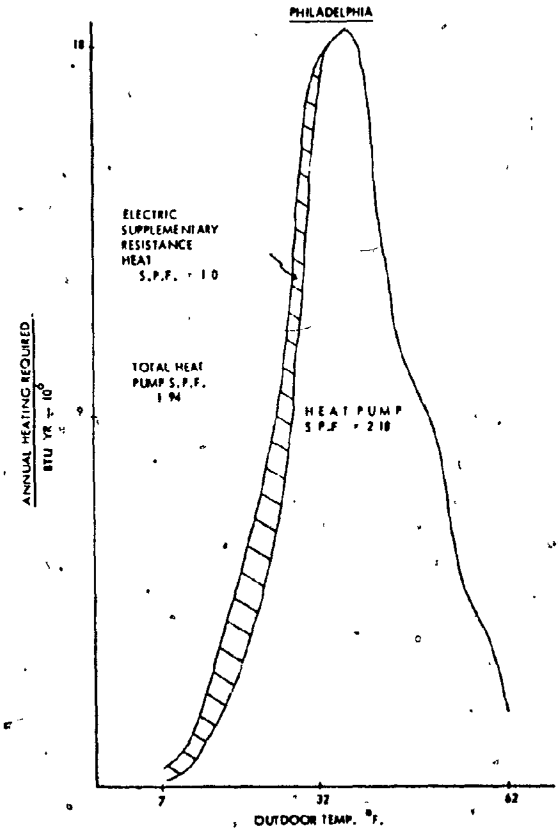


FIG. 29

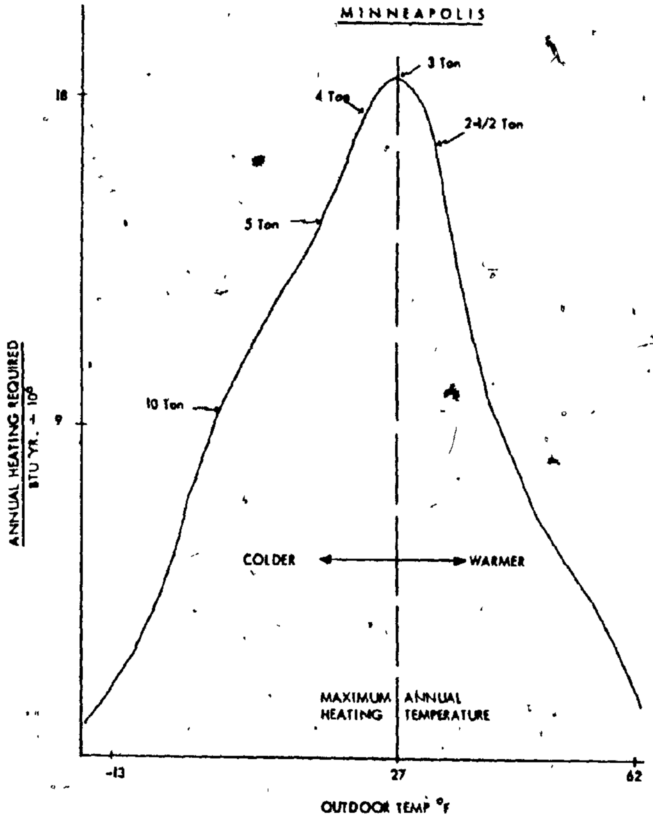
HEAT PUMP SIZING - PHILADELPHIA

NOMINAL CAPACITY (TONS)	S.P.F.*	KWH SAVED	BREAKEVEN YEARS
2-1/2	1.94	----	----
3	2.19	2 000	7
4	2.13	2 000	26
5	2.18	2 000	35
10	2.55	4 000	47

*Normally, the seasonal performance factor would increase with the use of over-sized units and not hold constant for the 3 to 5 ton sizes as shown. This particular examole involved climate, loading and unit size, and performance, which generated a special case and the S.P.F. remained virtually constant for the middle capacity size units shown.

Fig. 30

AREA UNDER CURVE IS TOTAL ANNUAL HEAT REQUIRED IN MINNEAPOLIS HOUSE



NOMINAL CAPACITY (TONS)	S. P. F.	KWH SAVED	BREAKEVEN YEARS
2-1/2	1.46	----	---
3	1.60	3 000	3
4	1.63	3 200	10
5	1.69	4 100	13
10	2.10	9 400	17

FIG. 31

HEAT PUMP APPLICATION - PHILADELPHIA PEAK POWER

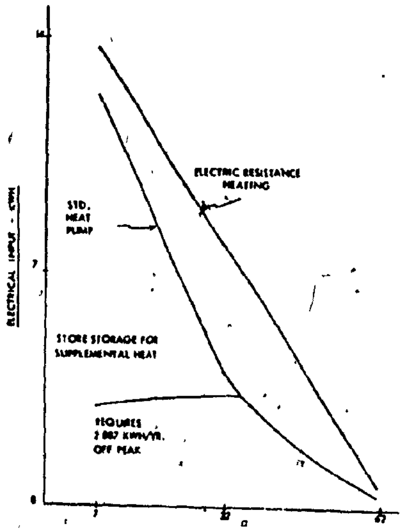


FIG. 32

50DR006 COOLING CYCLE

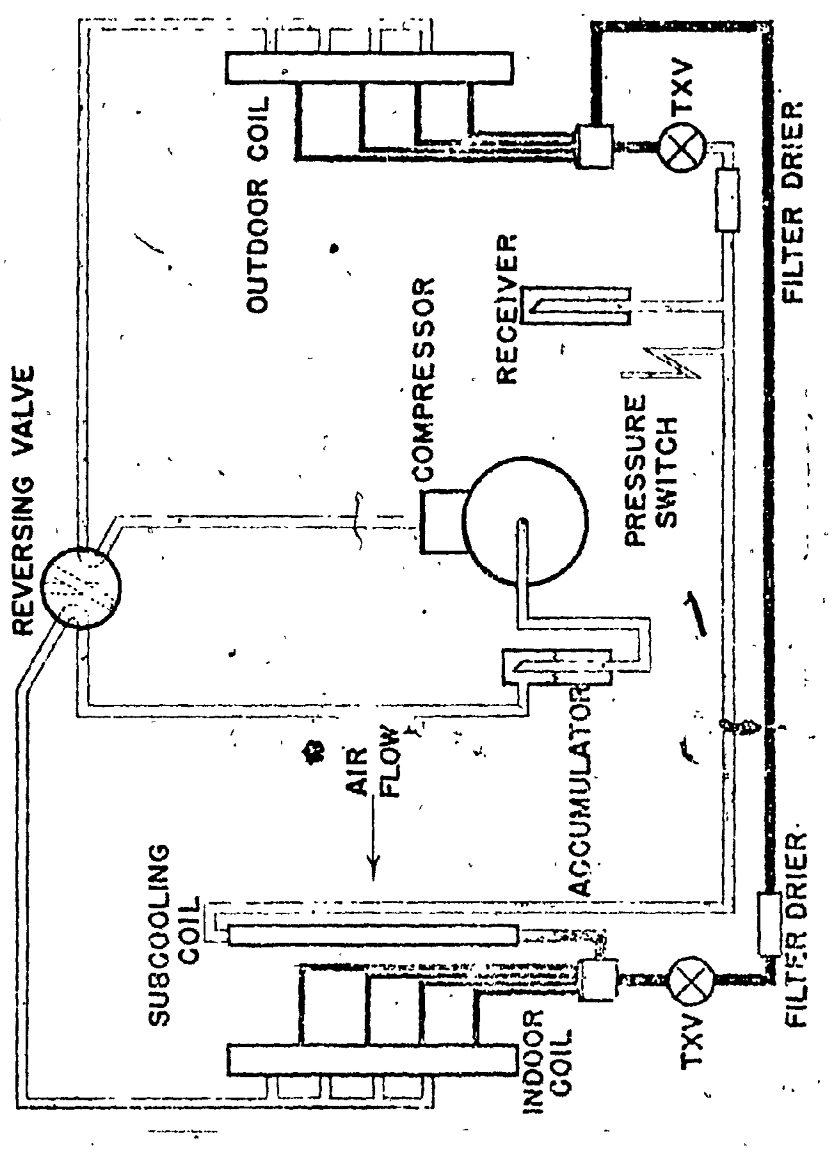


FIG. 32

50DR006 HEATING CYCLE

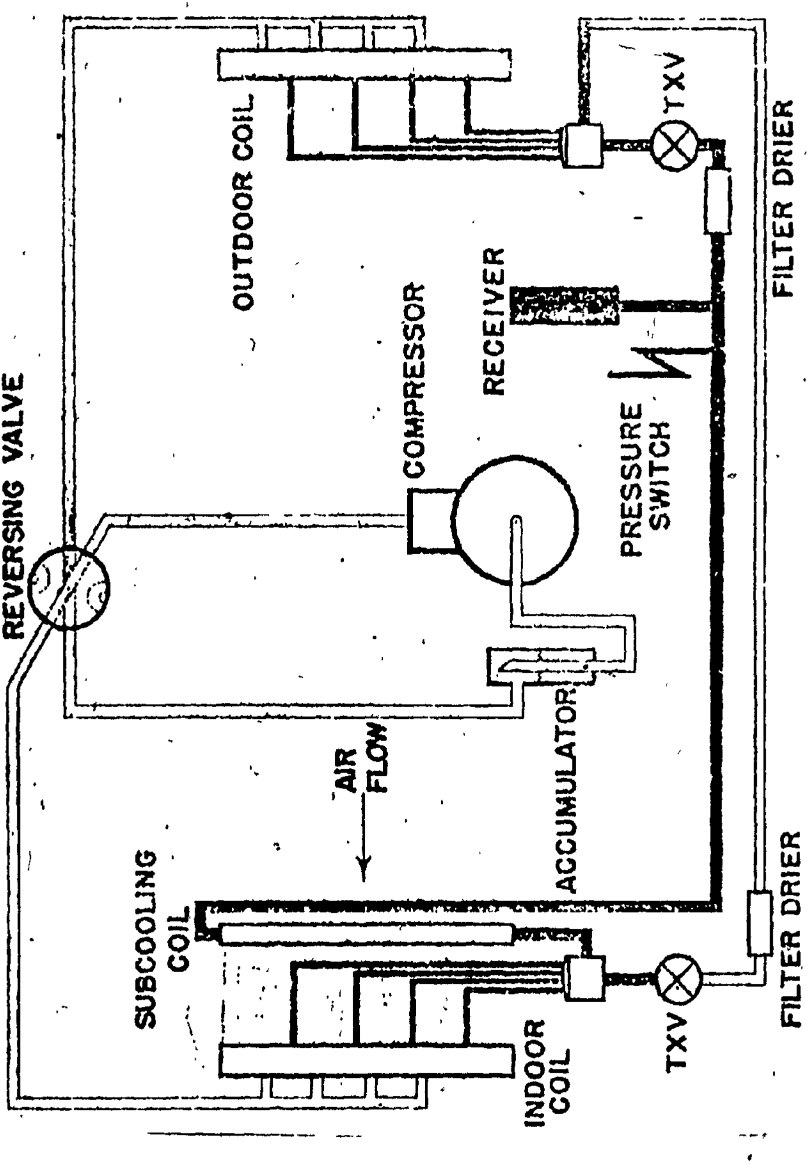


FIG. 33

SEASONAL PERFORMANCE OF 50DQ006

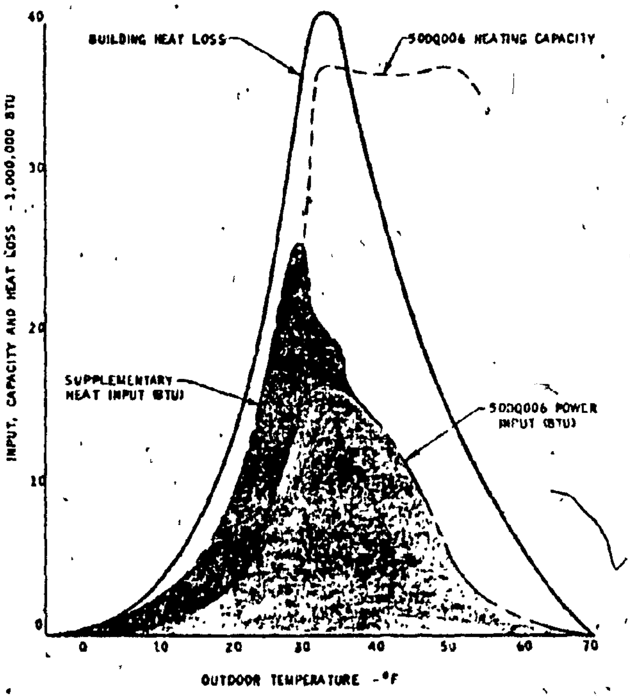


FIG. 34

SEASONAL PERFORMANCE OF 50DR006

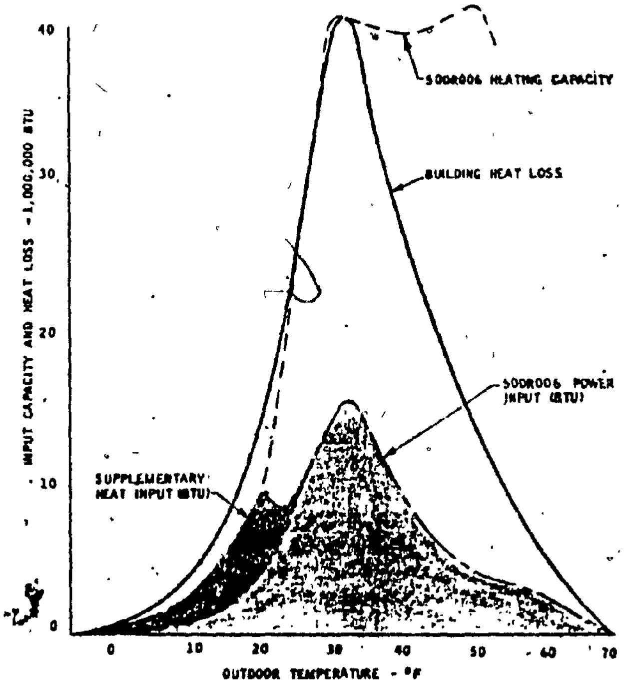


FIG. 35

HI/RE/LI CYCLE - HEATING

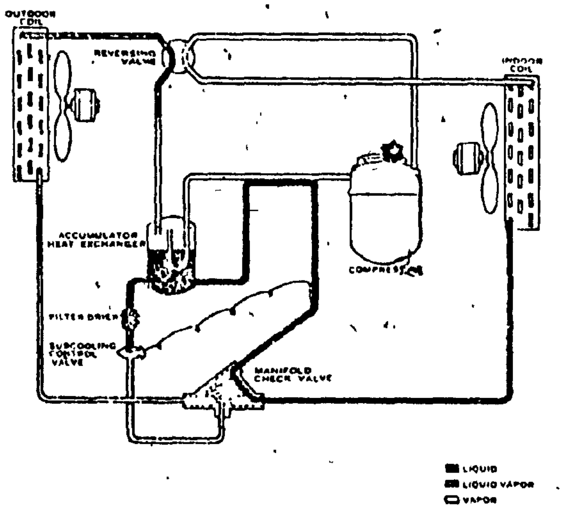
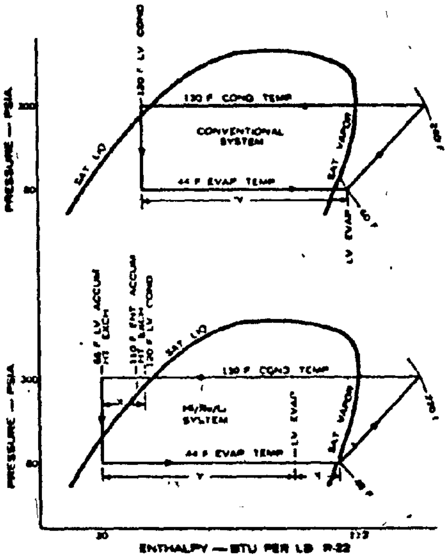


FIG. 36

PRESSURE-ENTHALPY DIAGRAM



x HEAT EXCAHNGE IN SUCTION LINE & ACCUMULATOR HEAT EXCHANGER
Y HEAT ABSORBED IN EVAPORATOR COIL

FIG. 37

KILOWATT HOURS PER KILOWATT OF DESIGN LOSS
PER DEGREE DAY DISTRIBUTION OF USE LEVELS

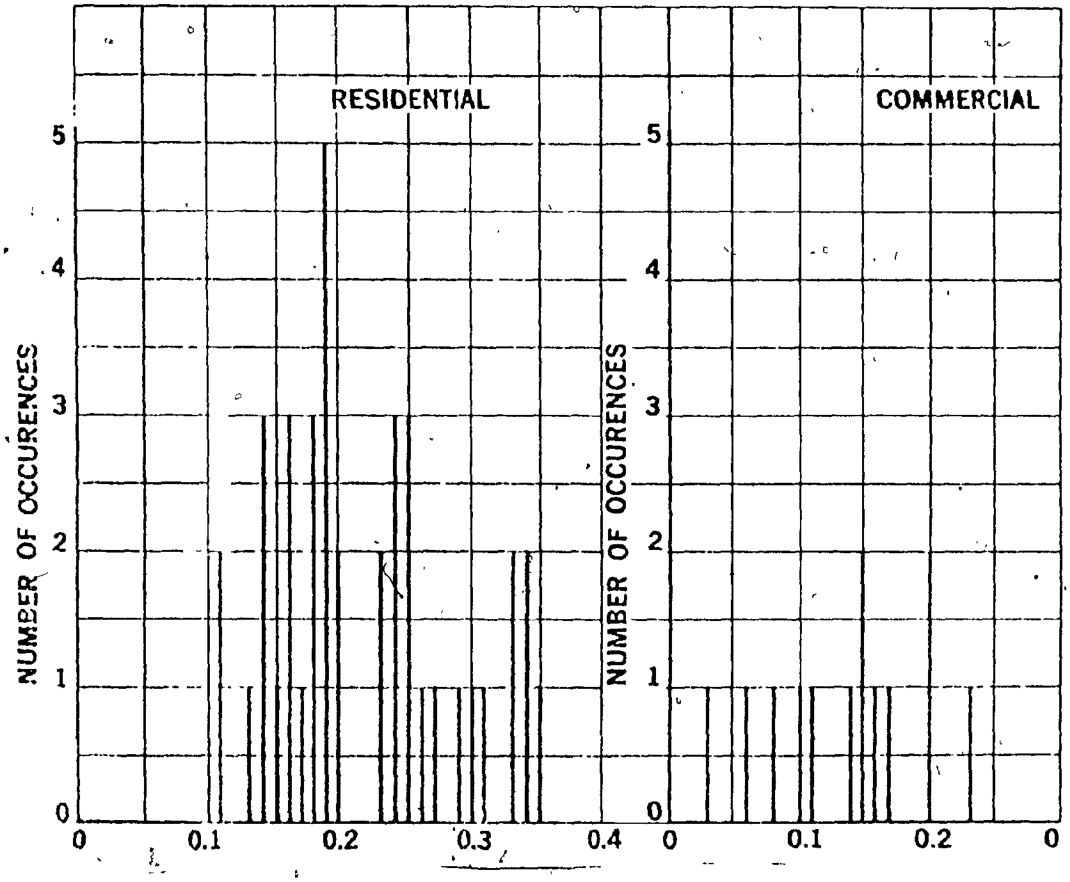


FIG. 38

OUTDOOR TEMPERATURE

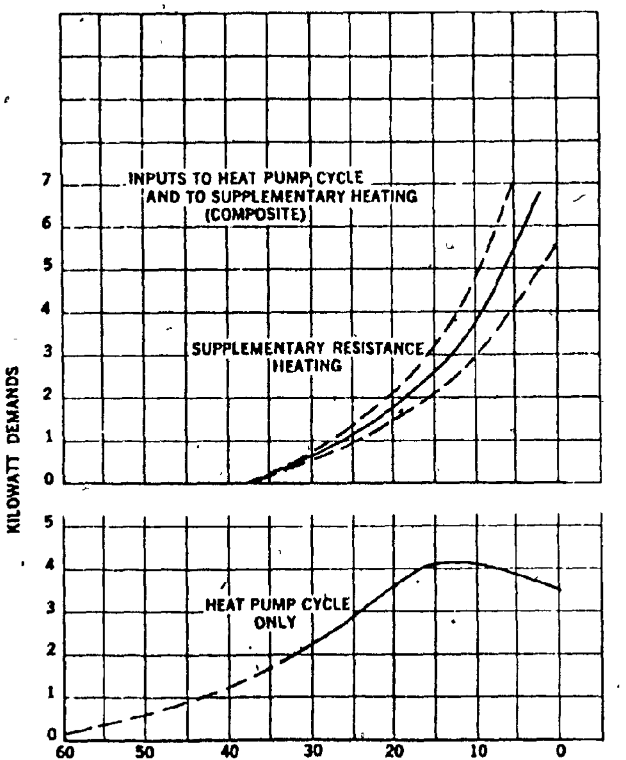


Fig. 39

WEEKLY ENERGY USAGE OF JERSEY CENTRAL P & L

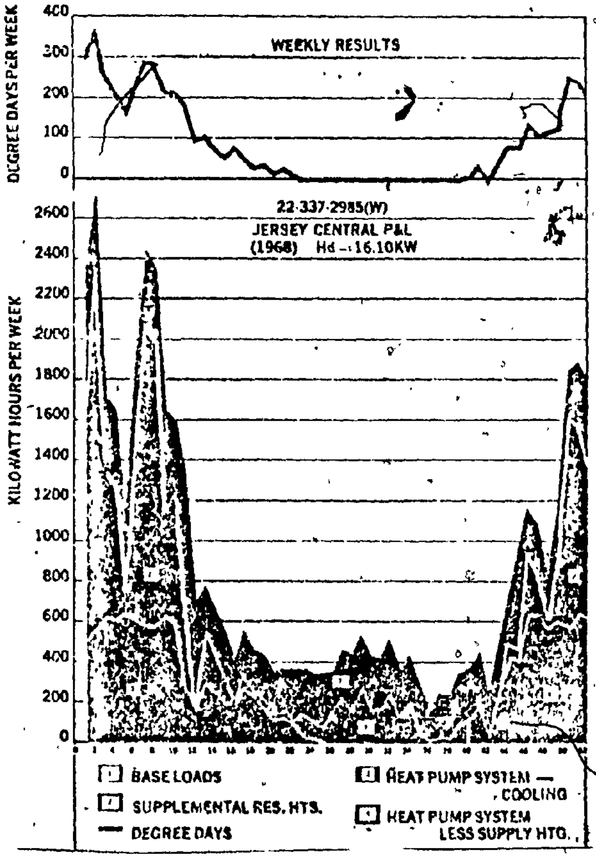


Fig. 40

WEEKLY ENERGY USAGE OF OHIO POWER CO.

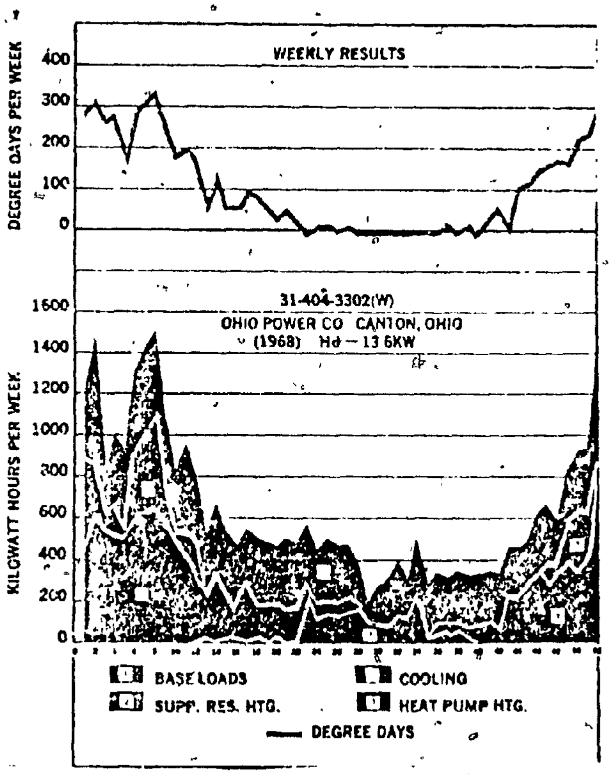
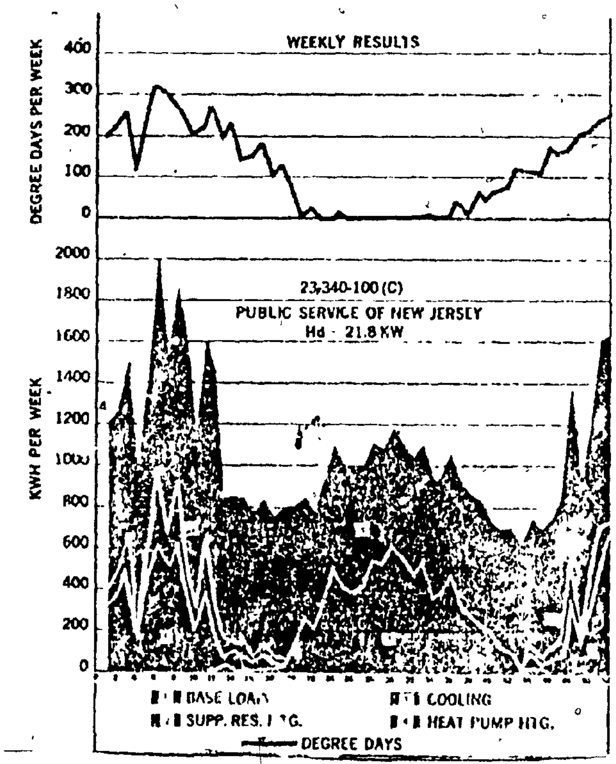


Fig. 41

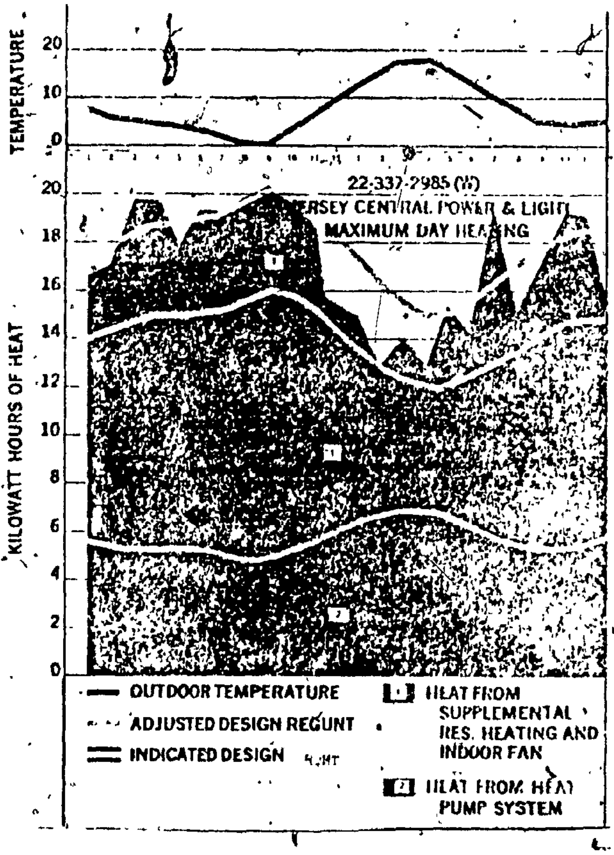
WEEKLY ENERGY USAGE OF PUBLIC SERVICE OF NEW JERSEY



15

Fig. 42

PERFORMANCE OF NEW JERSEY POWER & LIGHT HEAT PUMP



15

FIG. 43

AIR-TO-AIR HEAT PUMP PERFORMANCE

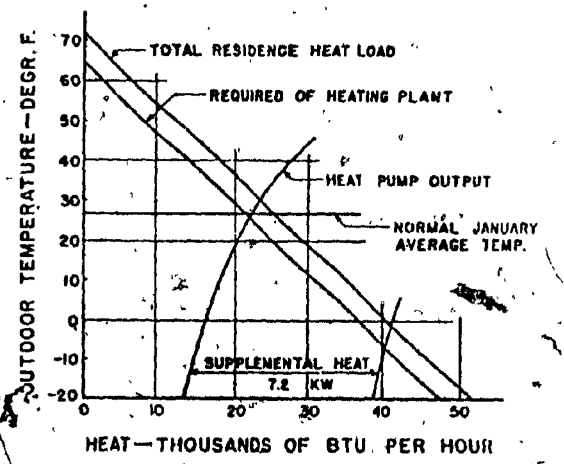


FIG. 44

HEATING PLAN KW DEMAND AND HOURLY TEMPERATURE FOR THREE COLD DAYS

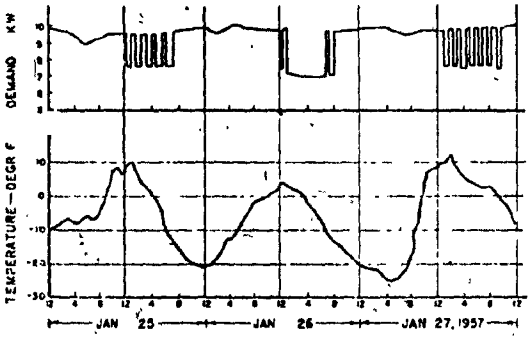
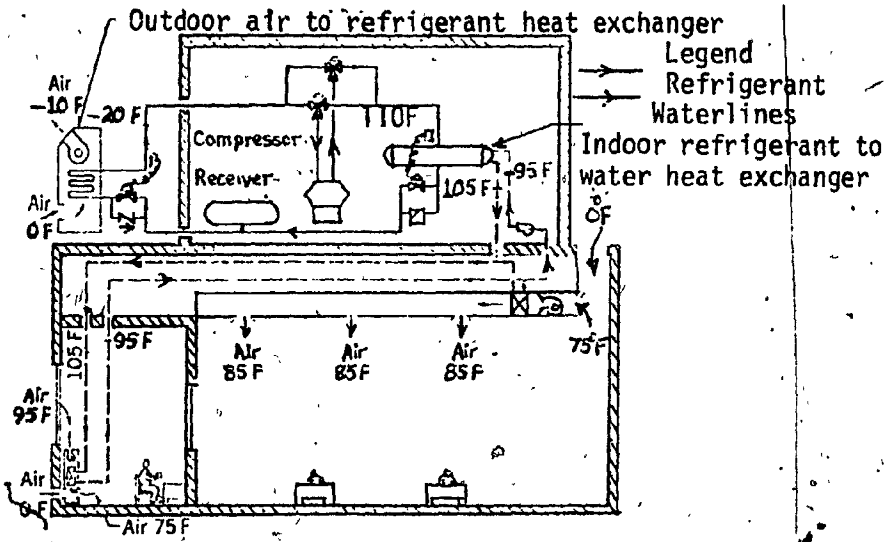


FIG. 45

AIR WATER HEAT PUMP SYSTEM (HEATING CYCLE)



9, 10

FIG. 46

WATER TO AIR HEAT PUMP (HEATING CYCLE)

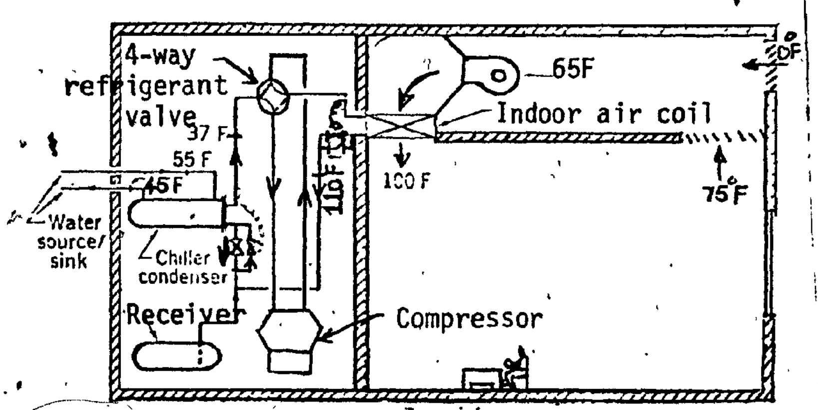
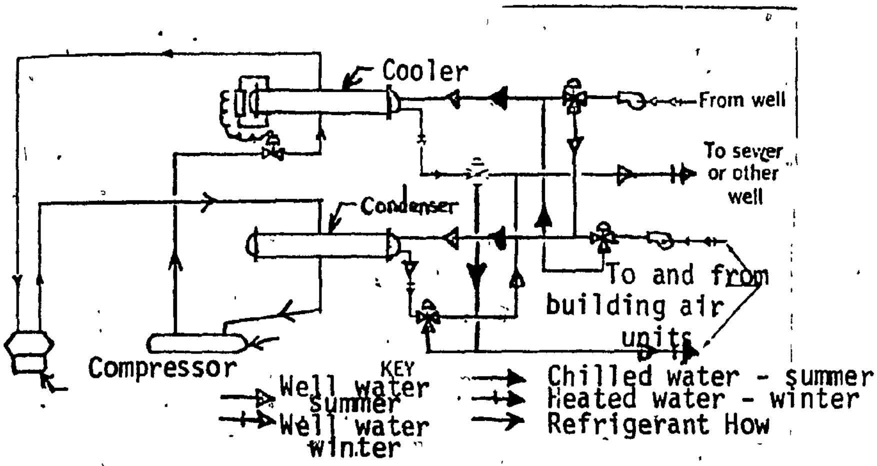


FIG. 47

WATER TO WATER HEAT PUMP



8

FIG. 48

LOAD ANALYSIS (SIMULTANEOUS HEATING AND COOLING WITH ECONOMIZER COOLING)

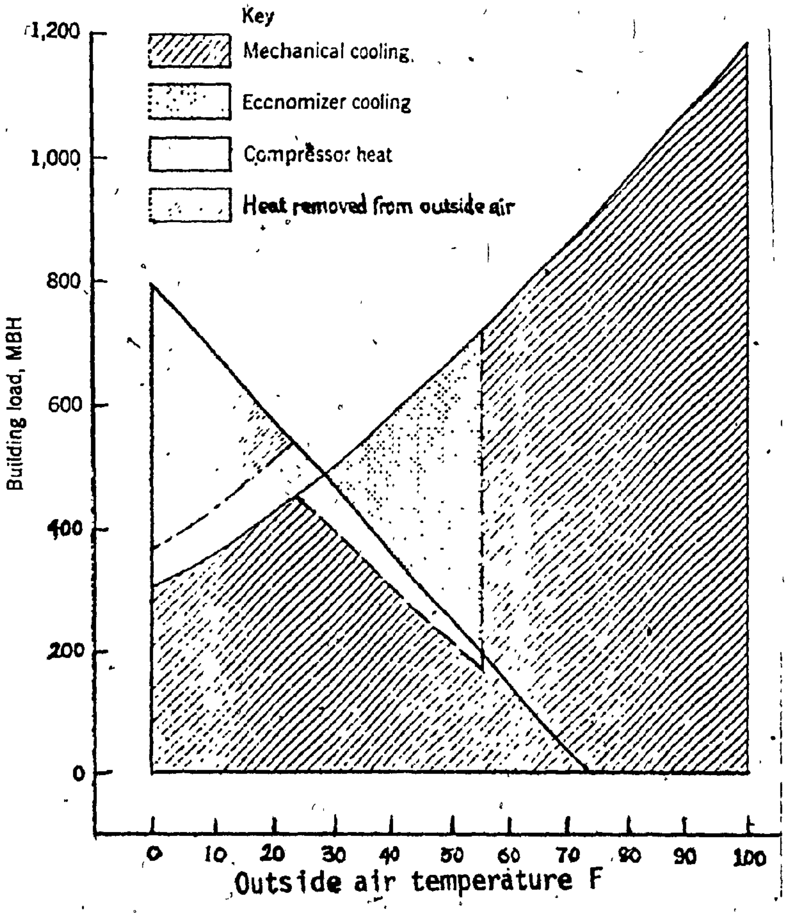


FIG. 49

AIR SOURCE HEAT PUMP ON DUAL-DUCT SYSTEM

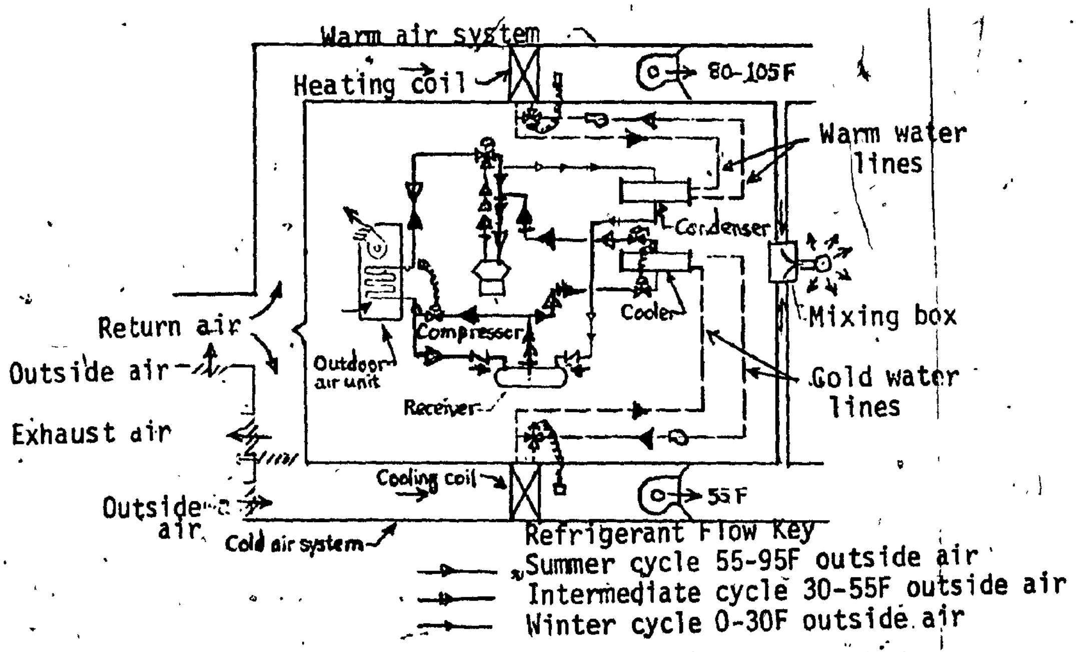


FIG. 50

HEAT PUMP SYSTEM WITH HEAT RECOVERY ACCESSORIES

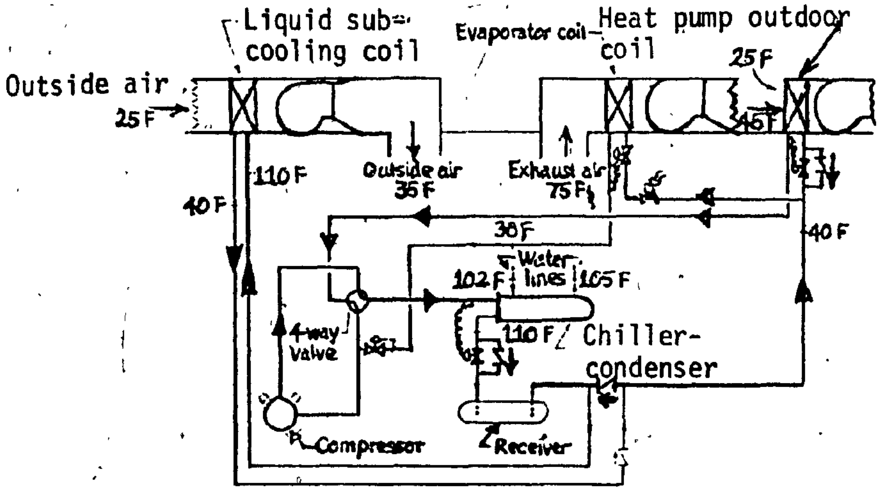
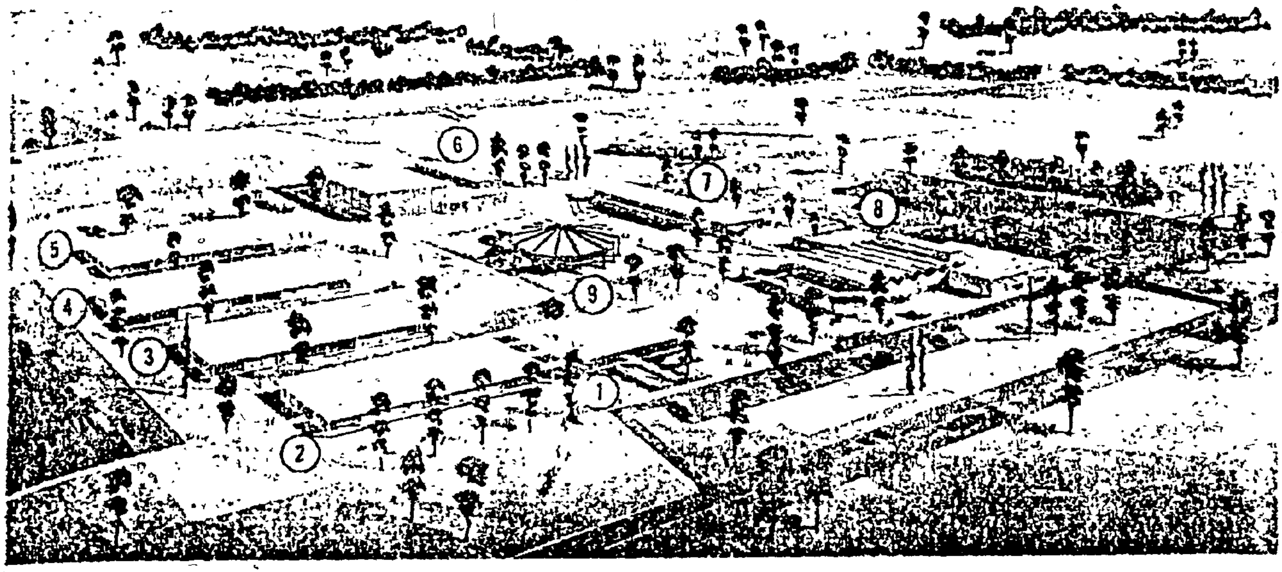


FIG. 51

KANSAS SCHOOL CAMPUS LAYOUT



Kansas school is housed in nine separate buildings, each a separate area of instruction.

Unit 1 is the Administration Building, providing offices for the superintendent, principal, registrar and counselors. Units 2, 3, 4 and 5 are classroom buildings, providing space for social science, language, mathematics and science classes. Unit 6 is the gymnasium building, also housing the wrestling room, corrective gym, classrooms, physical education offices and swimming pool. Unit 7 is the shop building consisting of a metal shop, woodworking shop, electronics shop and classrooms. This building also houses the central heating and air conditioning plant. Unit 8 is the auditorium and fine arts building with space for instrumental rooms, stage work shop, art laboratory and craft shop. Unit 9 is a round building located as the focal point of the campus and serves not only as a cafeteria with kitchen and snack bar facilities, but also as a student union.

FIG. 52

Fuel	Unit Cost	Annual Cost
Non-interruptible natural gas	$.43 per 1000 cu ft	$ 7,600
Interruptible natural gas	$.29 per 1000 cu ft	5,147
No. 2 fuel oil	$.115 per gal	14,650
Electricity, direct resistance	$.0125 per kwh	26,700
Electricity, using heat pump, COP = 3.85	$.0125 per kwh	6,9

FIG. 53

WINTER OPERATION
(LAWLESS, HEATING, PIPING AND AIR-CONDITIONING)

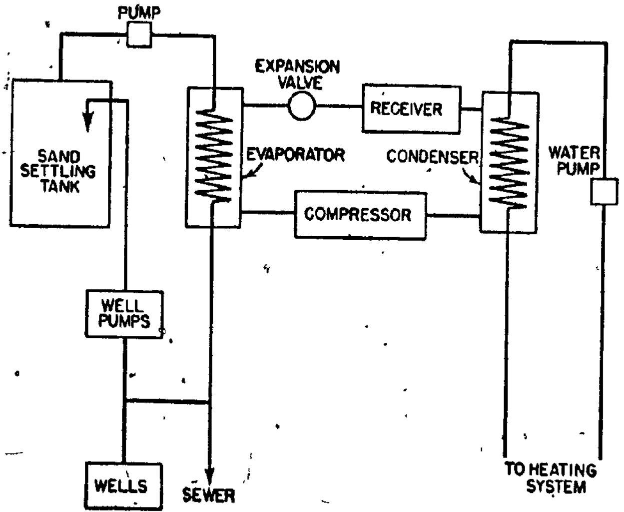


FIG. 54

SUMMER OPERATION
(LAWLESS, HEATING, PIPING AND AIR-CONDITIONING)

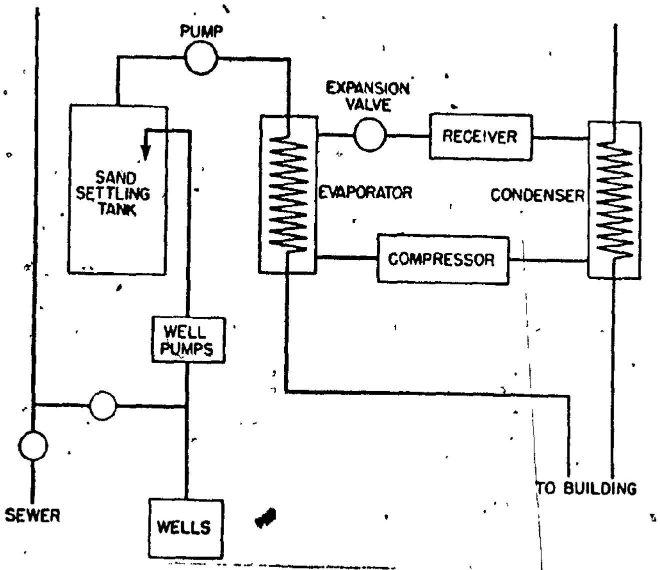


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FIG. 55

THREE-CIRCUIT AIR-CONDITIONING SYSTEM

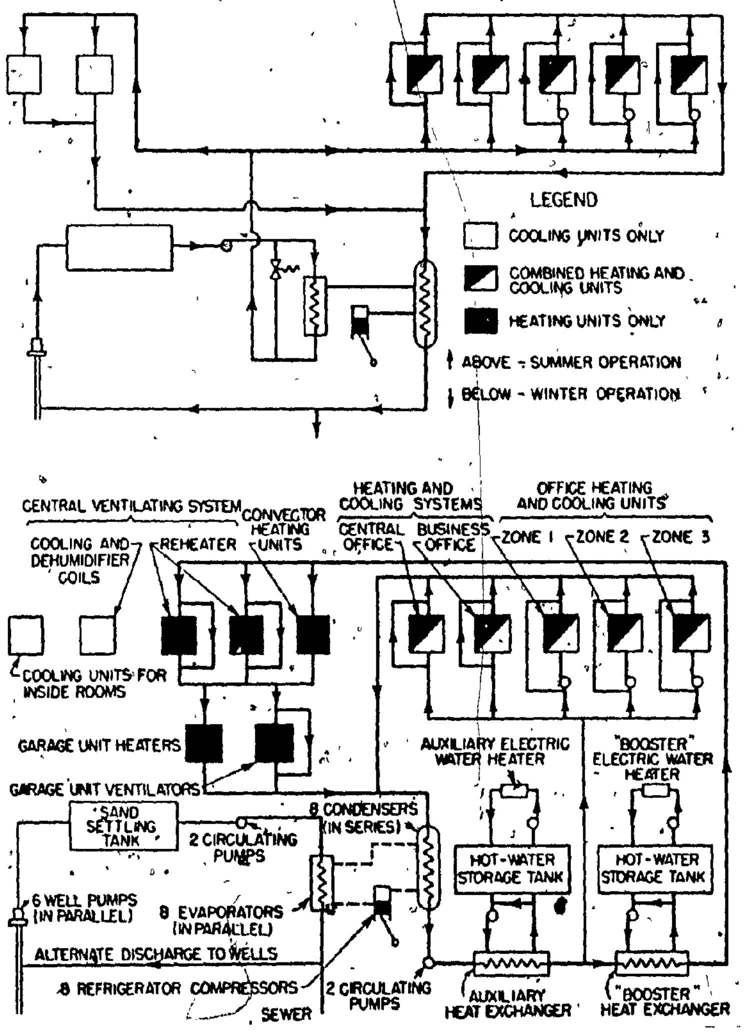

FIG. 56

CONSUMPTION OF ELECTRICITY IN KILOWATT-HOURS*

	1941		1942		1943	
	Heating	Cooling	Heating	Cooling	Heating	Cooling
January	99,100	....	95,100	....	77,700	
February	80,300	....	91,000	...	58,900	
March	84,000	......	63,400	.....	62,900	
April	16,900	......	26,000	.	44,500	
May	....	8,300	....	8,800	......	12,500
June	....	24,800	....	25,700	....	29,600
July	.....	32,900	....	37,300	....	29,600
August	..	28,800	....	27,600	..	26,800
September	....	18,100	.....	15,500	.	12,500
October	4,900	....	6,800	3,900	16,300	8,200
November	37,100	......	54,300	..	57,000	
December	72,100	....	89,900	..	81,300	.
Total	394,700	112,900	426,500	118,800	401,600	119,200
Degree-days	5,398	......	5,605	......	5,897	
Kwhr per degree-day	73.1	......	76.0	....	68.1	

FIG. 57

OPERATING COSTS OF HEAT-PUMP SYSTEM*

	Kwhr	Cost of electricity at 1 cent per kwhr	Operating labor	Maintenance†	Total
Heating season, 1941–1942	389,600	$3,896.00	$2,539.50	$ 704.79	$ 7,140.29
Cooling season, 1942	118,800	1,188.00	2,116.55	214.70	3,819.25
Total	508,400	$5,084.00	$4,956.05	$ 919.49	$10,959.51
Heating season, 1942–1943.	395,000	$3,950.00	$2,896.55	$ 856.90	$ 7,703.45
Cooling season, 1943	119,200	1,192.00	2,899.16	258.56	4,349.72
Total	514,200	$5,142.00	$5,795.71	$1,115.46	$12,053.17

FIG. 58

UTILIZATION TEMPERATURE CHART

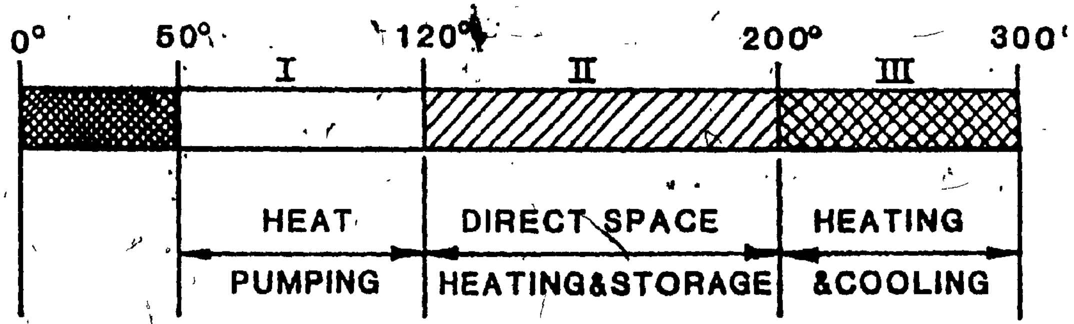

FIG. 59

OPERATING SCHEMATIC OF THE GREENHOUSE HEATING SYSTEM

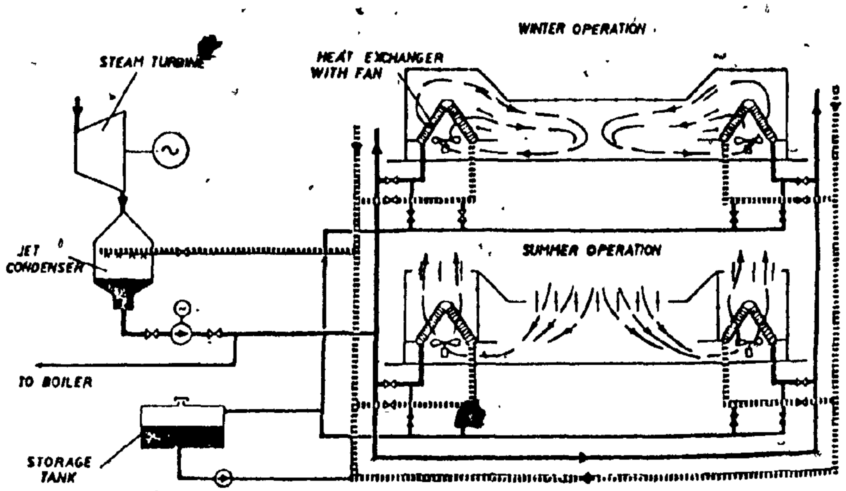

FIG. 60

ECONOMIC STUDY OF THE GREENHOUSE PROJECT

Investment costs		10^6 Ft*	
		A	B
		Separate power station and greenhouse	Power station and greenhouse combined
1.	Wet cooling tower for the power station 200 MW	120	-
2.	Greenhouse 34 ha, without heating	358	358
3.	Heating plant + pipeline to the greenhouse	338	-
4.	Heating installation inside of the greenhouse	119	-
5.	Combined condensing and heating installation	-	473
	Total investment costs:	935	831

Operational costs		10^6 Ft/year	
1.	13.5% of the investment costs	126.00	110.20
2.	Fuel consumption in the greenhouse	80.00	-
3.	Self-consumption of electric energy	3.15	23.9
4.	Make-up water for the power station	4.45	-
5.	Maintenance (surplus to A.)	-	3.97
	Total annual costs:	213.60	138.07

*Ft = Forint = unit of Hungarian currency.

FIG. 61

CARNOT THEORETICAL vs PRESENT HEAT PUMP HEATING COEFFICIENT OF PERFORMANCE

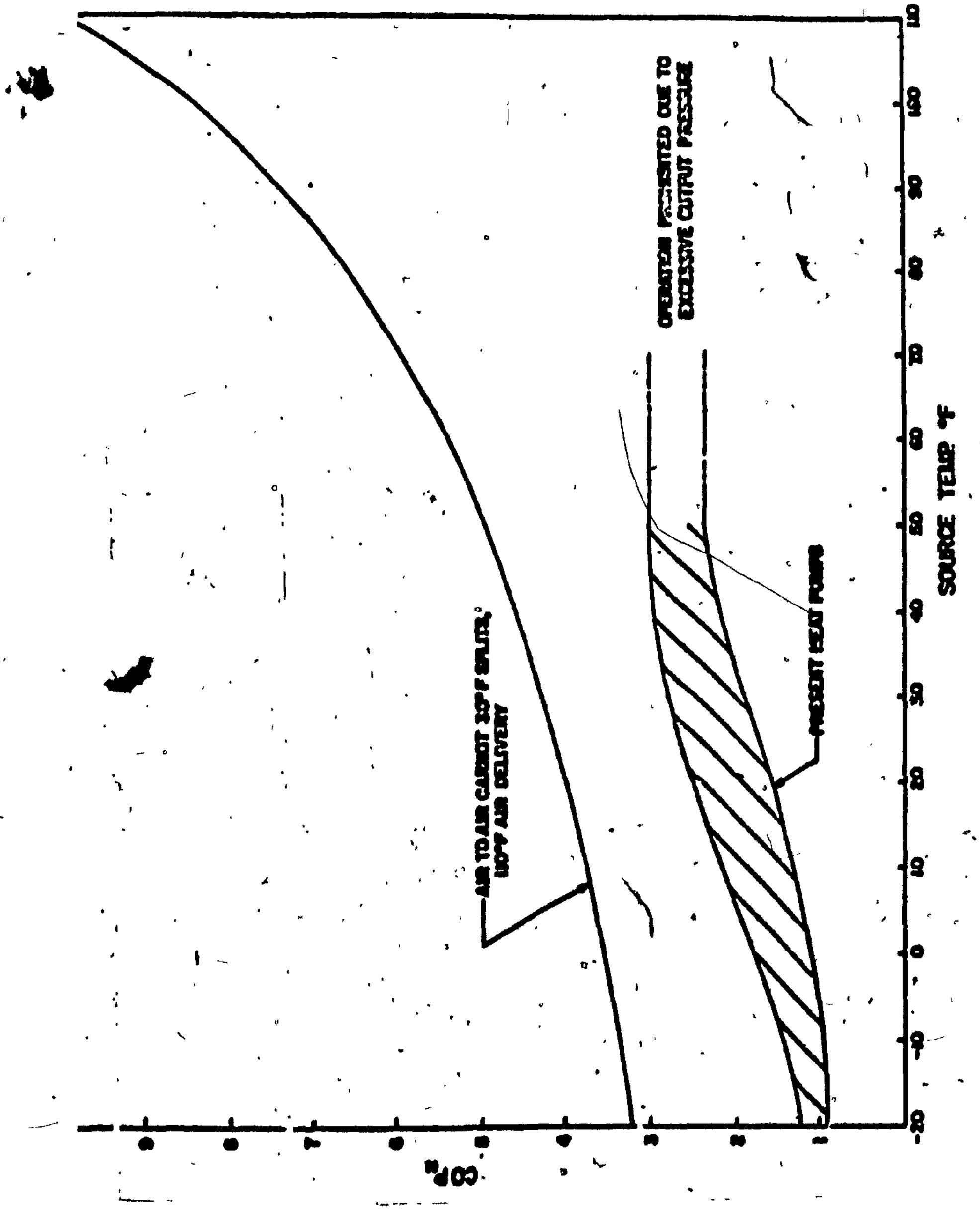

FIG. 62

THE PARALLEL SYSTEM

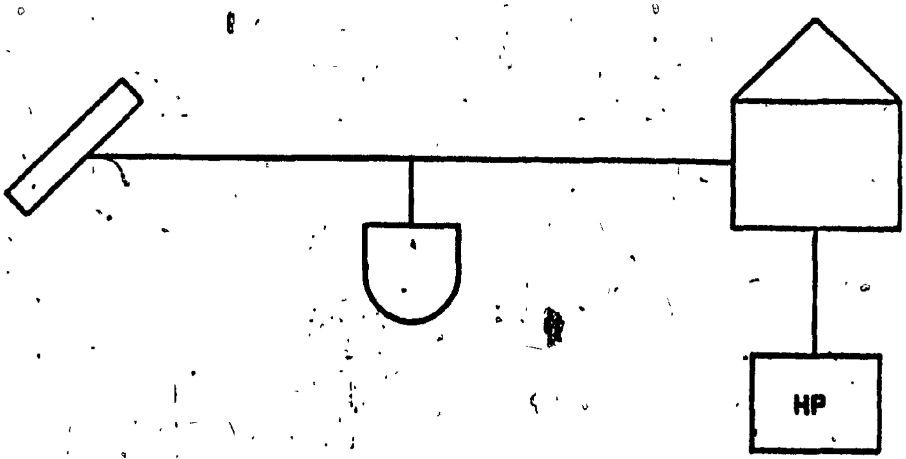

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FIG. 63

THE SERIES SYSTEM

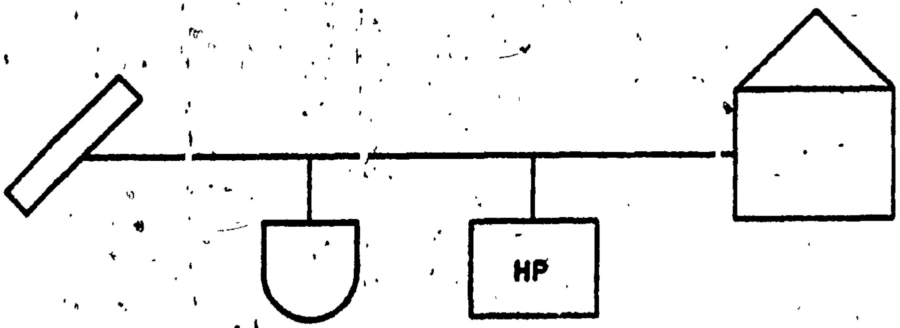

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FIG. 64

COLLECTION AND STORAGE OF HEAT ENERGY

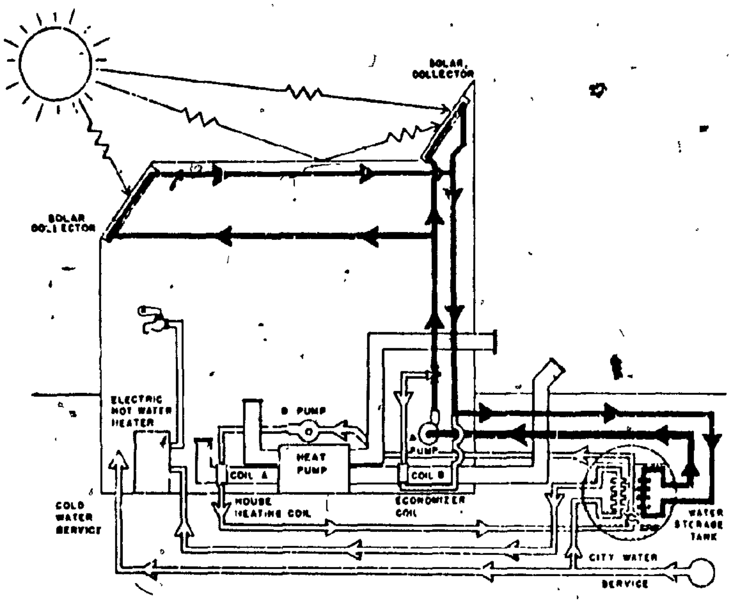
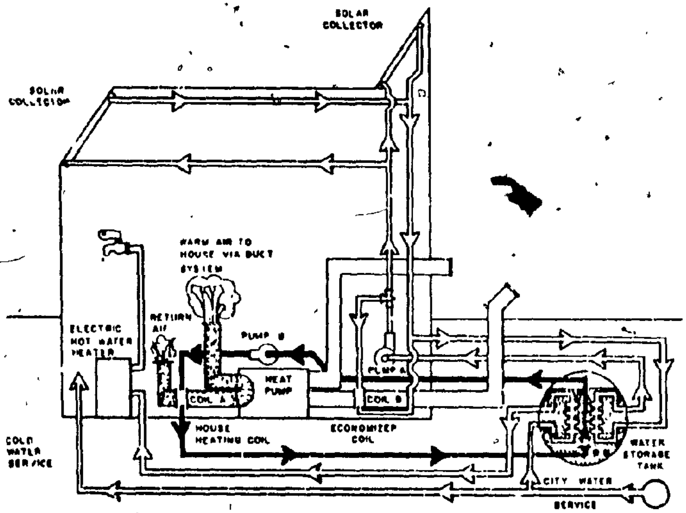


FIG. 65

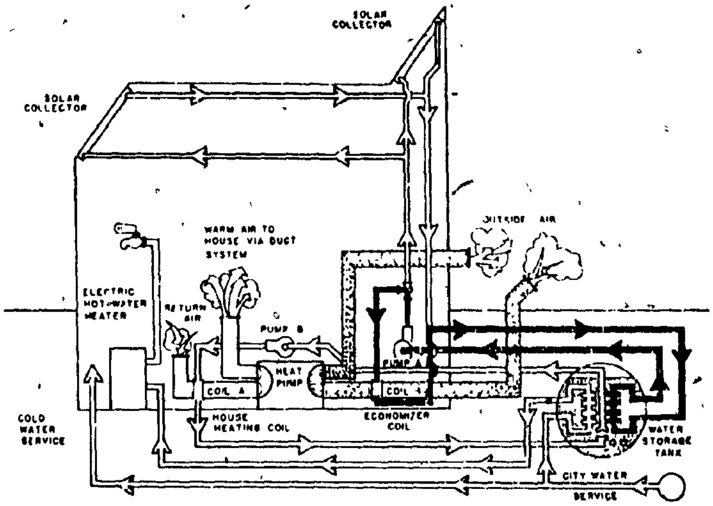
DIRECT USE OF STORED HEAT ENERGY FOR SPACE HEATING



12

FIG. 66

USE OF THE OPTIMIZED HEAT PUMP FOR SPACE HEATING.



12

FIG. 67

STORAGE SYSTEM NET ENERGY AND TEMPERATURE

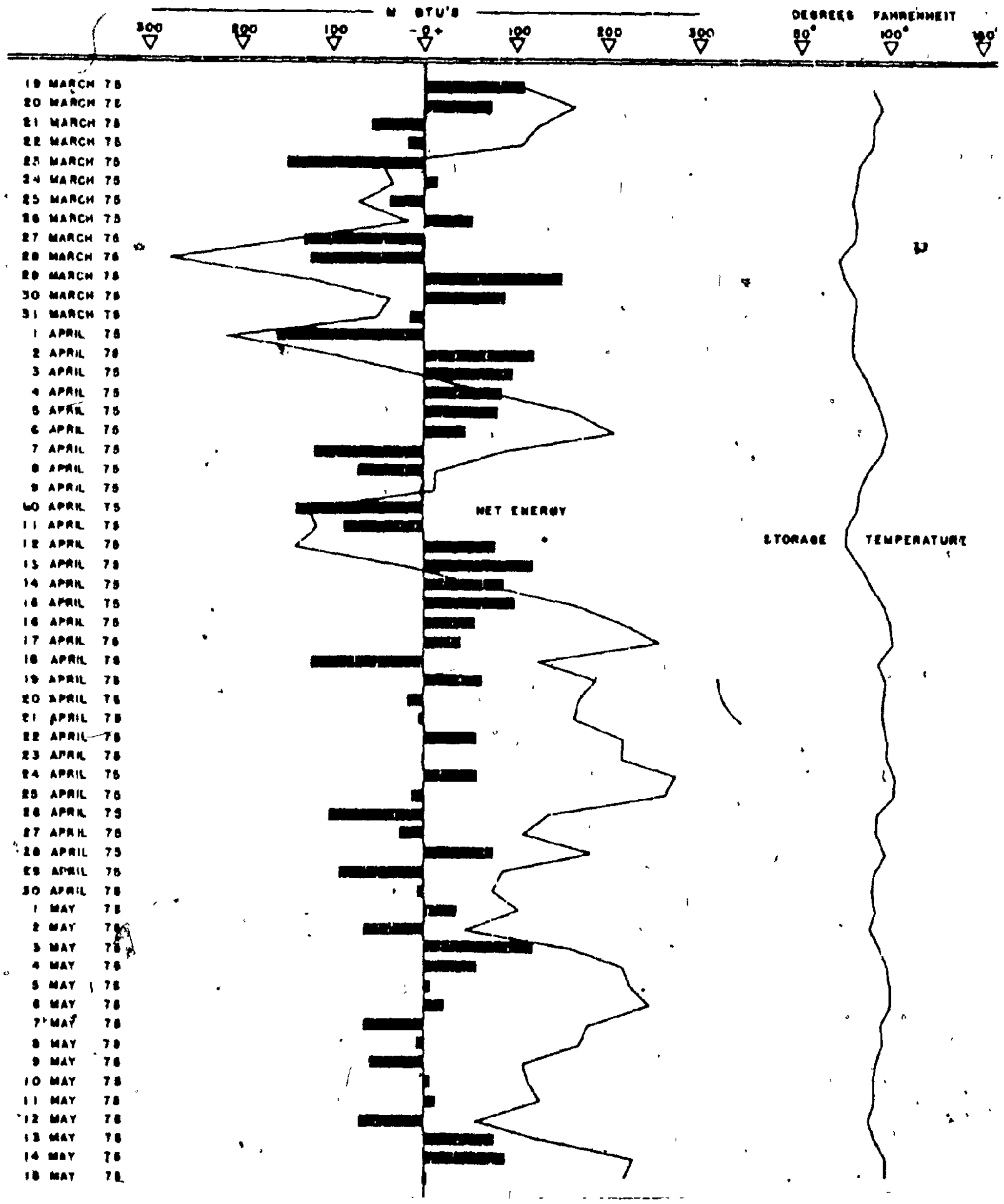


FIG. 68

HEAT PUMP PERFORMANCE

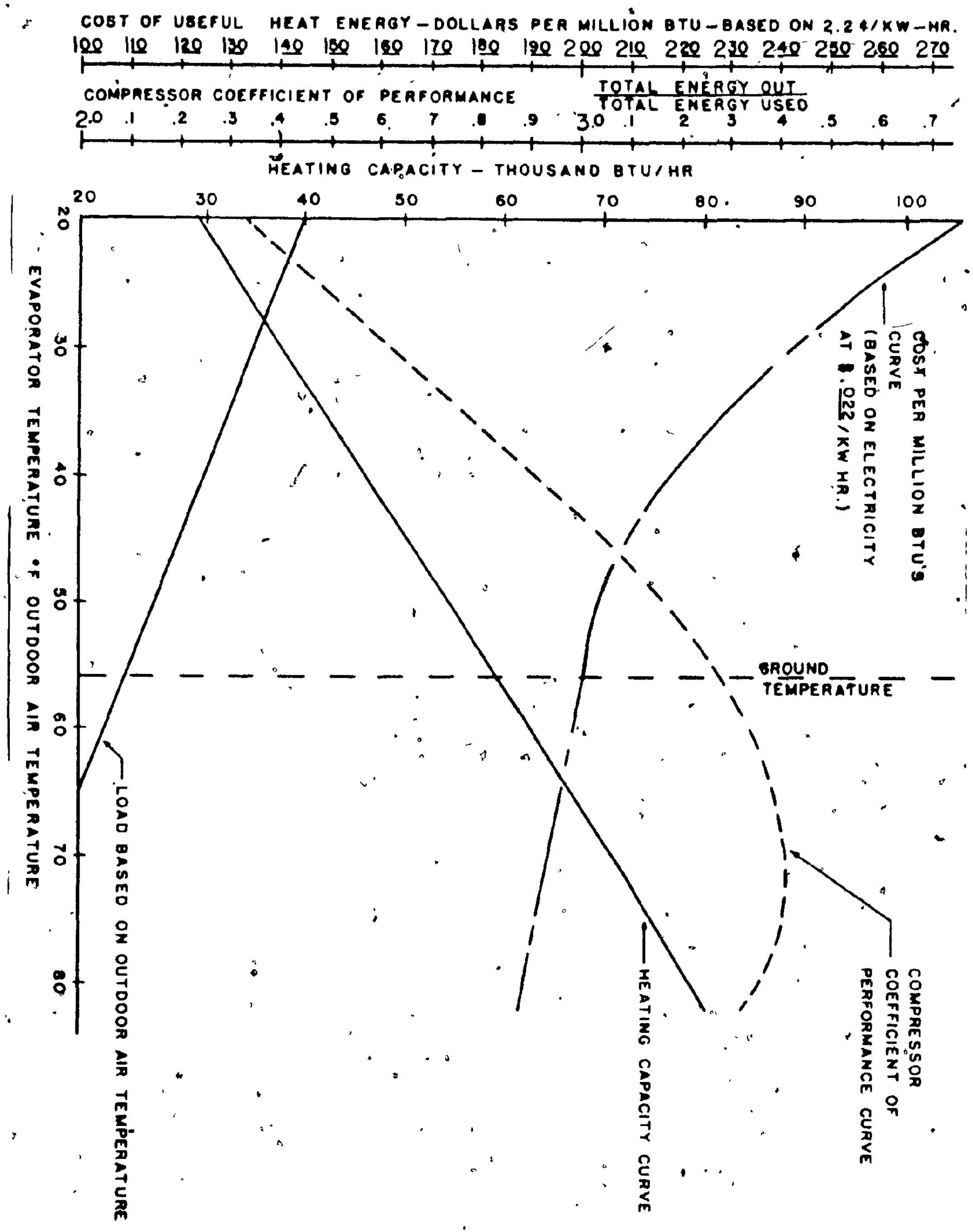
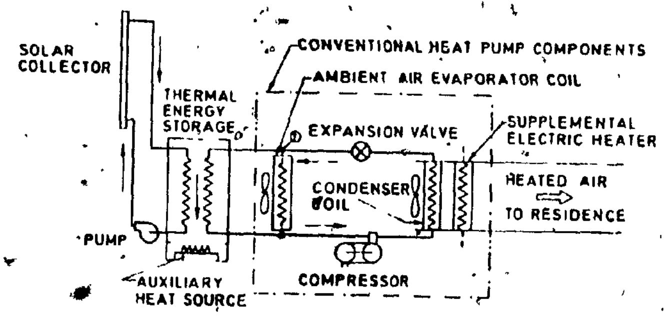


FIG. 69

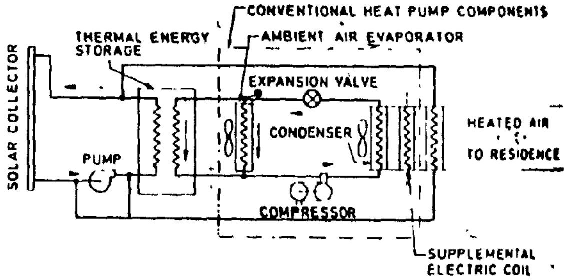
SYSTEM NO. 5, SOLAR ASSISTED HEAT PUMP A



20

FIG. 70

SYSTEM NO. 6, SOLAR ASSISTED HEAT PUMP B



20

FIG. 71

ENERGY CONSUMPTION OF THE DIFFERENT HEATING SYSTEMS

		1	2	3	4	5	6
HEATING SYSTEM		Electric Resistance	Combustion Furnace	Direct Solar Heating (see note below)	Conventional Heat Pump	Solar Heat Pump A (Fig. 1)	Solar Heat Pump B (Fig.2)
ANNUAL ENERGY CONSUMED	Electrical Energy (kWh)	29,600	-	-	13,000	9,080	8,430
	Fossil Fuel Energy (10^6 Btu)	-	168	110	-	30	- -
	Total Resource Energy (10^6 Btu)	326	168	110	143	130	94
Electrical Load Factor (Monthly Basis)		0.62	-	-	0.51	0.64	0.57

FIG. 72

VARIATION OF RESOURCE ENERGY CONSUMPTION WITH COLLECTOR AREA AND THERMAL STORAGE CAPACITY

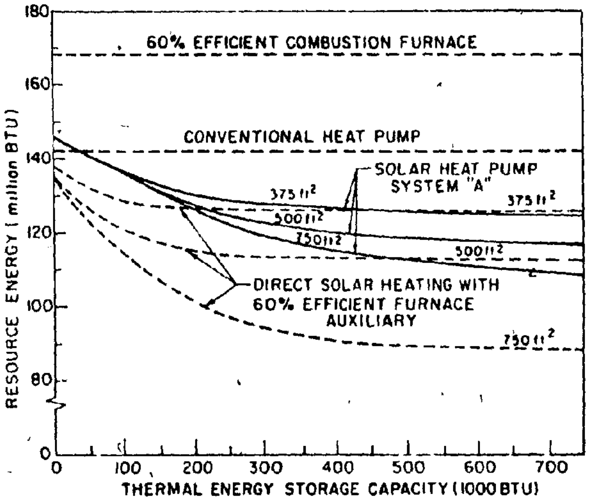


FIG. 73

ANNUAL RESOURCE ENERGY CONSUMPTION FOR DIFFERENT SPACE HEATING SYSTEMS IN 1500 SQ FT PHILADELPHIA RESIDENCE

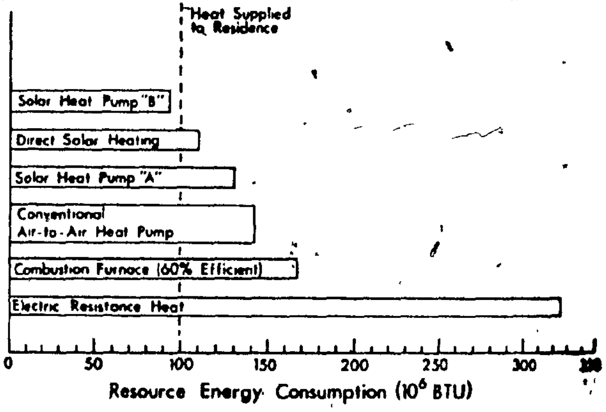


FIG. 74

MONTHLY ELECTRICAL ENERGY CONSUMPTION FOR DIFFERENT SPACE HEATING SYSTEMS IN 1500 SQ FT PHILADELPHIA RESIDENCE

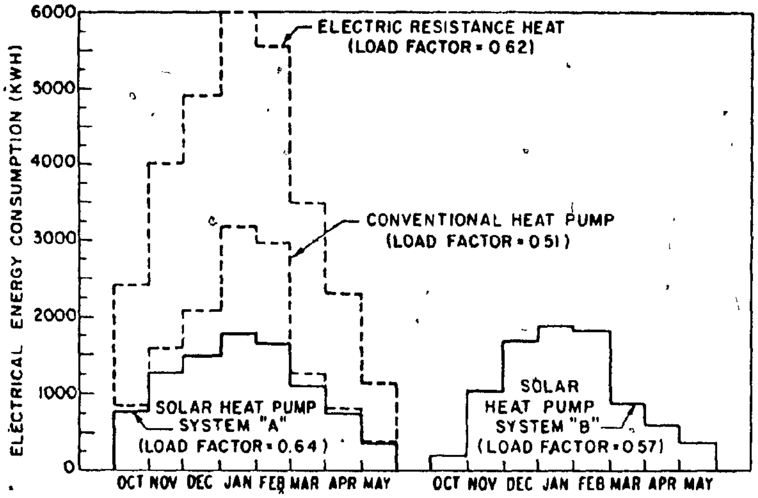


FIG. 75

SEASONAL PERFORMANCE FACTOR DERIVATION FOR AN AIR-TO-AIR HEAT PUMP INSTALLED IN MONTREAL

MONTREAL / 2100 Sq. Ft.
CALCULATION

LOAD:	29 469	52 424
	COOLING	HEATING

EQUIP	CFM
	1000

by: ______________________ date: ______________

DESIGN TEMPERATURE		
ID	OD	DIFF
70	-10	80

				← HEAT PUMP ALONE →					← SUPPLEMENTARY HEAT →			
OUTDOOR TEMP (5° INCREMENTS)	BTUH LOSS PER 1°F (HEAT LOSS ÷ TD)	OUTDOOR TEMP BELOW 65°F (65 − COLUMN A)	HEAT LOSS (BTUH) B × C	HEAT PUMP HEATING CAPACITY (BTUH) MFR DATA	HEAT PUMP RUNNING TIME (%) D ÷ E	HEAT PUMP INPUT (KW) MFR DATA	SEASONAL HEATING HOURS U S W.B.	SEASONAL HEAT PUMP INPUT (KWH) F × G × H	RESISTANCE HEAT INPUT (BTUH) D minus E	RESISTANCE HEAT INPUT (KW) J ÷ 3413	SEASONAL RESISTANCE HEAT INPUT (KWH) H × K	DEGREE HOURS C × H
A	B	C	D	E	F	G	H	I	J	K	L	M
62	806.5	3	2.42	35.0	6.9	4.2	840	243				2520
57	CONSTANT	8	6.45	34.1	18.9	4.1	480	372				3840
52		13	10.48	32.9	31.9	3.9	360	448				4680
47		18	14.52	31.4	46.2	3.8	600	1054				10800
42		23	18.55	29.2	63.5	3.7	720	1692				16560
37		28	22.58	27.0	83.6	3.6	400	1203				11200
32		33	26.62	24.6	100	3.5	500	1750	2.02	.59	295	16500
27		38	30.65	22.5		3.4	720	2448	8.15	2.38	1714	27360
22		43	34.68	20.6		3.3	1000	3300	14.08	4.13	4130	43000
17		48	38.71	18.5		3.2	300	960	20.21	5.92	1776	14400
12		53	42.75	16.6		3.1	120	372	26.15	7.66	919	6360
7		58	46.78	14.8		3.0	80	240	31.98	9.37	750	4640
2		63	50.81	12.90		2.9	50	145	37.91	11.11	556	3150
−3		68	54.84	10.80		2.8	22	62	44.04	12.90	284	1496
−8		73	58.87	8.90		2.7	9	24	49.97	14.64	132	657
−13		78	62.91	6.90		2.6	5	13	56.01	16.41	82	390
−18 & below		83										
								14326	← TOTALS	→	10638	167553

ANNUAL REQUIREMENT DUCTED RESISTANCE HEAT = (B) 806.5 × (M (TOTAL)) 167,553 / 3413 = (P) 39,593 KWH

ANNUAL REQUIREMENT HEAT PUMP SYSTEM = (I (TOTAL)) 14,326 + (L (TOTAL)) 10,638 = (N) 24,964 KWH

SEASONAL PERFORMANCE FACTOR = $\left(\frac{P}{N}\right)$ = $\frac{39,593}{24,964}$ = 1.59 SPF

FIG. 75

HEAT PUMP MODEL

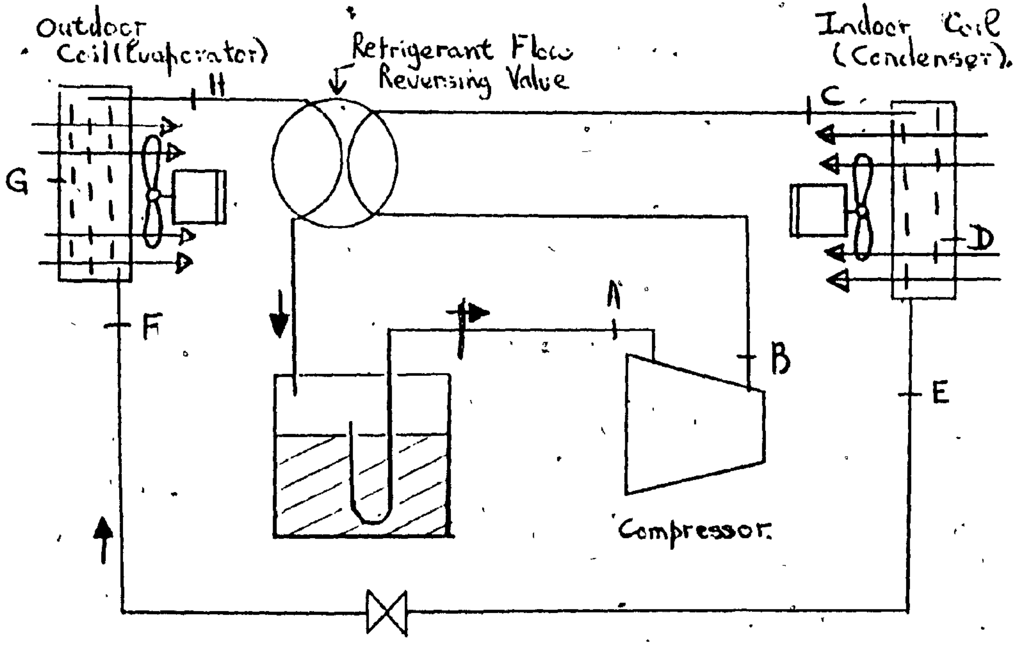


FIG. 76

COMPARISON OF THE PERFORMANCE OF AN AIR-TO-AIR HEAT PUMP INSTALLED IN PHILADELPHIA AND IN MONTREAL

Heat Pump Design Conditions	Philadelphia	Montreal
Area of Home, Ft^2 (m^2)	2100 ft^2 (195m^2)	2100ft^2 (195m^2)
Inside Design Temperature, ^oF (^oC)	70^{o}F (21^{o}C)	70^{o}F (21^{o}C)
Outdoor Design Temperature, ^oF (^oC)	5^{o}F (-15^{o}C)	-10^{o}F (-23^{o}C)
Temperature Difference	65^{o}F (18^{o}C)	80^{o}F (27^{o}C)
Heating Load, Btuh (Kw	52,424 (15Kw)	68,553 (20Kw)
Nominal Capacity of Air-to-Air Heat Pump, tons	2.5	2.5
Degree Hours	118,917	167,553
Seasonal Resistance Heat Input (Kwh)	2,887	10,638
Seasonal Performance Factor of Total Heat Pump System (S.P.F.)	1.94	1.59

LIST OF TABLES

TABLE I

ANNUAL NORMAL DEGREE DAYS

Cities	Degree Days
Spokane, Wash.	6852
Walla Walla, Wash.	4848
Lewiston, Idaho	5483
Pullman, Wash. ('53-'58 only) ..	6628
Pullman, Wash. (50 yr. average)	6938

TABLE II

WEATHER DATA FOR 1954 TO 1959

FISCAL YEARS FROM JULY 1 TO JUNE 30

Year	1954-55	1955-56	1956-57	1957-58	1958-59
Degree Days					
Weather Bureau	7194	7137	7026	5863	6292
Recorder on Site	7381	6882	6626	5890	6183
Minimum Temp. F	0	—12	—24	+17	—1
Heating Season Start	Sept. 9	Sept. 21	Oct. 7	Sept. 1	Sept. 14
Heating Season End	June 5	June 10	June 9	May 18	June 28

TABLE III
HEAT PUMP PERFORMANCE

	Heat Pump only Btu per hr	Including Supplementation Btu per hr.
Capacity at 45 F O.D. Temp.	29,000	53,200
30 F O.D. Temp.	24,100	48,300
10 F O.D. Temp.	17,700	41,900
—10 F O.D. Temp.	13,300	37,500

Coefficient of Performance — By Seasons
Based on all power for heating including fans & supplemental

Heating Season	C.O.P.
1954-55	2.10
1955-56	2.12
1956-57	2.17
1957-58	2.75
1958-59	2.67

TABLE IV

ENERGY REQUIREMENTS — ANNUAL FOR FISCAL YEARS, JULY 1 TO JUNE 30

SEASON		1954-55	1955-56	1956-57	1957-58	1958-59
POWER						
Heat Pump	kw hr	12,220[1]	10,425	9,665	8,236	7,622
Supplemental	kw-hr	904[1]	2,083	2,084	4	676
All Heating	kw-hr	13,124[1]	12,508	11,749	8,240	8,298
Domestic	kw-hr	10,358[1]	11,104	9,833	9,399	11,163
Total Residence	kw-hr	23,482[1]	23,612	21,582	17,639	19,461
DEMAND (15 min)						
Heat Pump	kw	3.8	3.5	3.6	3.5	3.5
All Heating	kw	10.6	10 6	10.1	5.4	7.7
Total Residence	kw	14.1	14.0	15.0	9.0	14.0
Domestic, no heat[2]	kw	—	5.3	5.3	6.0	10.2
LOAD FACTORS						
Heat Pump	%	36.7	34.0	30 6	26.8	24.8
Supplemental	%	1.5	3.3	3.7	0	4.8
All Heating	%	14.1	13.5	13 3	17.4	12.3
Total Residence	%	19.0	19.2	16.4	22.4	15.8
Domestic, no heat	%	—	23.9	21.2	17.9	13.0

[1] Corrected value after thermal calibration

[2] Summer Value

TABLE V AVERAGE DAILY HEAT ENERGY COLLECTED AND STORED (Btu/sq ft of collector/day)

Month	Year	Upper-Array	Lower Array	Average Total	Upper Array Yield — % above Lower Array
January*	1975	255	193	224	32.12
February	1975	327	292	292	27.43
March	1975	296	228	262	28.82
April	1975	319	238	279	34.03
May	1975	241	192	216	25.52

TABLE VI COMPARISON OF PEAK GENERATING CAPACITY REQUIRED FOR ALL OPTIMIZED HEAT PUMP HEATING LOAD ADDITIONS VERSUS ALL ELECTRIC RESISTANCE HEATING LOAD ADDITIONS FROM 1978 THROUGH 1985

Year	Kilowatts for All Resistance Heating Loads (ref. 2)	Kilowatts for All Optimized Heat Pumps*	Kilowatts Generating Capacity Saved with Optimized Heat Pumps
1978	50,000	22,000	28,000
1979	93,000	52,000	41,000
1980	140,000	80,000	60,000
1981	170,000	96,000	74,000
1982	200,000	120,000	80,000
1983	228,000	140,000	88,000
1984	254,000	170,000	84,000
1985	275,000	192,000	83,000

*Optimized heat pumps with a 3 to 1 coefficient of performance.

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