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by

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INTRODUCTION

In Canada today, oil and natural gas are by far the most commonly used fuels for residential space heating. In 1976, oil and gas heating systems were installed in 85 per cent of the existing housing stock.[1] Electric heating systems of one form or another accounted for only 13 per cent,[1] about the same proportion as after their introduction in the late 1950's. In spite of a relatively low initial cost when compared with gas and oil heating equipment, electric heating was and still is expensive from an operating-cost point of view. In many regions of the country today, however, the differential between conventional fossil-fuel and electricity prices is narrowing and electricity is being used to a greater extent than in the past. One device, which utilizes electrical input to produce heat and promises to make electrical heating more economical from an operating-cost and prime energy-use standpoint, is the heat pump.

The heat pump employs the same basic principle as the common household refrigerator: heat is extracted from a space at low temperature and discharged to another space at a higher temperature. In fact, any process that results in transfer of heat from a low temperature to a higher one is referred to as "heat pumping." By this definition an air conditioner and a refrigerator are both heat pumps because heat is removed from a conditioned space and discharged to the high temperature surrounding.

The heat pump employs the vapor compression refrigeration cycle in order to transfer heat from a body at a low temperature to one at a higher temperature (Figure 1). The basic components employed in the cycle are two heat exchangers, compressor, expansion valve and interconnecting piping. A fluorocarbon refrigerant such as Freon 22 is usually employed as the working fluid in the cycle. The temperature at which the liquid refrigerant vaporizes can be controlled by reducing the pressure in the evaporator. Similarly, by increasing the pressure of the vapor in the compressor, the temperature is raised above the temperature of the sink and the vapor condenses, liberating both latent heat and heat of compression. The high temperature, high pressure liquid then passes through an expansion device, which reduces the pressure of the liquid to the pressure in the evaporator section and the cycle is repeated.

The unique advantage of this cycle is that the energy obtained from the condenser as heat can be appreciably greater than that associated with the electrical power required to drive the compressor. The measure of performance of such a device, called the coefficient of performance (COP), is defined as the ratio of the desired effect to the heat equivalent of compressor input. It is always greater than 1. When operating on the cooling cycle, the desired effect is the heat taken in at the evaporator; when operating on the heating cycle, the desired effect is the heat liberated by the condenser. By reversing the direction of the refrigerant flow through the heat exchangers, the heat pump can provide either heating or cooling as required (Figure 2).

AIR-TO-AIR HEAT PUMP

The most widely employed heat pump uses the outdoor air as a heat source or sink (Figure 2). In the heating mode, outside air is drawn through the evaporator heat exchanger, gives up some heat to the vaporizing refrigerant, and is discharged from the outdoor section. The compressor increases the pressure and temperature of the refrigerant vapor allowing it to condense in the indoor heat exchanger where its heat is transferred to the air from the room. The heated air is then returned to the room.

The air-to-air heat pump was originally introduced in the southern United States, where cooling and dehumidification were of prime importance and heating seasons were short and mild. The units were sized to cope with the cooling load and in most applications such units also provided sufficient heating capacity. However, heat pumps marketed in more northern regions had to be equipped with supplementary electrical resistance heaters. This is still the practice today.

PERFORMANCE OF AN AIR-TO-AIR HEAT PUMP

Unlike conventional heating systems using natural gas or oil, the heating capacity of an air-to-air heat pump depends on the outdoor temperature (Figure 3). Both the capacity and COP of the heat pump are reduced as the outdoor temperature drops. The outdoor temperature at which the heat pump can just satisfy the heating requirement of the space is referred to as the "balance" point. The balance point is a function of the building heat loss and the capacity of the heat pump.

Below the balance point the heat pump has insufficient capacity to meet the demand, and the deficit is made up by supplementary heaters. The supplementary heaters are staged to come on in such a manner that only that portion actually required to make up the difference between the heat pump capacity and the space heating requirement is activated. The COP, including the electrical input to the heaters, is always greater than 1, but decreases as more stages of supplementary heaters are brought on-line. Above the balance point the opposite is true; the unit has surplus capacity and is made to match the building heat loss by cycling on and off.

It is interesting to examine the performance of a heat pump on a seasonal basis. With the following information, one may estimate the contribution of the heat pump to the total heating requirement:

(i) the manufacturer's performance data (i.e., capacity, input power, and COP versus outdoor temperature),

(ii) the building heat loss characteristic,

(iii) the frequency of occurrence of outdoor temperature requiring heating for the area of interest (<18°C; <65°F).

As an example, consider a heat pump with a nominal cooling capacity of 7 kW (2 tons) and a net heat loss of 8.6 kW at -26°C (-15°F) as shown in Figure 4, for a home in the Ottawa area. The balance point is approximately -1°C (30°F). The cross-hatched region in the figure is the heating requirement made up by the supplementary heaters. Figure 5 shows the frequency distribution of hourly dry bulb temperature for intervals of 3°C (5°F). This chart is based on an average for a 10 year period between 1957-1966 for Ottawa International Airport.

The next step is to determine average heat loss, heat pump input and output, and the supplementary heat requirement (Table 1) for each of the temperature intervals. For this example, the seasonal performance factor (SPF) calculated is 1.6, which represents a 37 per cent kilowatt-hour saving over straight electrical resistance heating. (The actual energy saving would be somewhat less since the manufacturer's ratings are obtained at steady-state full-load conditions, while in practice the unit cycles on and off above the balance point.)

For illustrative purposes, the heating energy inputs to the residence at various outdoor temperatures were plotted (Figure 6). The total area (A+B+C) approximates the annual heating energy requirement of the house used in the example. Area A + B is a fraction of the annual heating energy requirement supplied by the heat pump. Area B is the energy input to the heat pump required to run the compressor and fans. Area C is the supplementary heating energy supplied to the house and area A is a fraction of the total heating energy obtained from the outdoor air (~37 per cent).

POSSIBLE PERFORMANCE IMPROVEMENTS

To the right of the balance point the heat pump has excess capacity relative to space heating demands. If the heat pump capacity could be matched to the net building heat loss above the balance point, a significant performance improvement would be realized. This would also result in increased reliability because the effects of cycling on and off are eliminated. In theory, this could be achieved by using an infinitely variable speed compressor. In practice, a two-speed compressor would improve performance significantly.

In addition to the improvements possible through capacity modulation, there exist a number of areas where modifications will improve heat pump performance in cold climate regions.

1. Depending on the relative humidity, frosting or icing of the surface reduces the air flow through the face of the coil when the outdoor air-to-refrigerant coil is at or below 0°C (32°F). The result is a gradual reduction in capacity to the point where the heat pump must be reversed (to cooling mode) to defrost the coil. The frequency of defrost required is governed to a large extent by the fin spacing on the coil. The greater the number of fins per inch, the more frequently the machine must be defrosted. By increasing the primary surface area (tube area) and reducing the secondary surface area (fin area), a reduction in defrost cycles should be possible.

2. In present split-system heat pumps (one part inside and the other outside), the outdoor section contains the compressor, outdoor coil, reversing valve, outdoor fan motor and fan. Liquid and vapor refrigerant piping interconnects indoor and outdoor sections. Heat losses from the vapor line, reversing valve and liquid line could be reduced by moving high temperature components indoors. In some designs the condensed refrigerant is cooled below the condensing temperature by leaving the liquid line uninsulated. This is obviously a compromise in favour of the cooling mode because it has always been central air-conditioning practice to locate the compressor, condenser, fan and motor outdoors to readily dissipate heat to the environment. This heat could be put to better use to improve the heating cycle.

ECONOMICS OF THE AIR-TO-AIR HEAT PUMP

It has been generally accepted that a heat pump could not be justified economically unless it could be used for summer cooling. In Canada, the need for central air conditioning may not be a major factor, therefore the decision whether to use a heat pump should be based on heating energy savings alone.

The installed cost of an air-to-air heat pump, suitable for residential space heating varies from $2000 to $3500 depending on capacity (5.3 to 10.5 kW) (1 1/2 to 3 ton). This assumes that both the duct work and electrical service will not have to be upgraded in the case of a retrofit application. Consider the example where a central electric furnace and heat pump are being compared for the residence mentioned previously. The duct work and electrical service would be the same in either case. The comparative costs of the two systems are shown in Table 2. In this instance, is the heat pump a better investment than the electric furnace? If the useful service life of the heat pump is assumed to be 15 years, a constant annual saving of \$166 would only justify an investment of \$1262, assuming an interest rate of 10 per cent, whereas the extra capital investment required for a heat pump is \$1500. The unit cost of electricity, however, has been increasing steadily in the past few years. This is expected to continue and should be taken into account in the analysis.

Assuming that the unit cost of electricity escalates at a constant rate, the maximum investment that can be justified is determined by the equation:

$$\text{Maximum Investment Justified} = S \times \frac{1}{1+i} \left(\frac{a^n-1}{a-1}\right)$$

where S = first year saving (\$)

i = interest rate

$a = \frac{1+e}{1+i}$ (e = electricity unit cost escalation rate)

n = useful service life (years).

For i = 10 per cent, e = 8 per cent, S = \$166 and n = 15 years, the maximum investment justified = 12.03 x \$166 = \$1997. Thus the heat pump would be a better investment than the electrical furnace if the extra cost was less than \$1997. What effect would changes in the various assumptions have on the analysis? Consider the following cases:

(1) an SPF of 1.4 rather than 1.6 with other assumptions the same. This would reduce the first year savings to \$117 (~30 per cent);
Maximum Investment Justified = \$1407.

(2) an electricity unit energy cost escalation rate of 12 per cent and reduction in the SPF as in (1), with other assumptions the same;
Maximum Investment Justified = \$1815.

(3) as in (1), but in addition a maintenance cost difference of \$50 per year;
Maximum Investment Justified = \$1165.

A reduction in the SPF, from that calculated, would be likely, for the reasons outlined previously, i.e., because of on-off cycling above balance point and defrost energy requirement. Case (3) raises the question of heat pump reliability, which in the past has been a problem. Recent indications are that manufacturers have solved a number of problems that plagued early air-to-air heat pumps. The average annual service cost reported in a recent study was approximately \$50 per year.[2]

Comparing case (1) and (2), one can see the importance of knowing energy unit cost trends. In the case of electricity, the unit cost escalation rate will, in all likelihood, exceed the inflation rate on other goods and services. Factors such as the utility generation mix, i.e., the percentage fossil fuel, nuclear or hydro generation, is a major factor determining the wholesale unit cost increases and subsequent rate increases to the consumer.

ALTERNATIVES TO SUPPLEMENTARY ELECTRIC RESISTANCE HEATERS

The SPF calculation of the air-to-air heat pump showed that the supplementary heaters provided only 27 per cent of the total heating requirement. This reduced the heat pump efficiency by 23 per cent (i.e., from a COP of 2.11 to a SPF of 1.61). More significant is the impact on the electric utility peak demand. The heat pump has the same peak power demand as electric resistance heating, although it uses 37 per cent less kilowatt hours than the resistance heaters over the course of the heating season. The utility must still provide generating capacity based on peak demands. In other words, heat pumps have a poorer load factor for the utility than electrical resistance heating and thus would contribute to higher rates for electricity.

By eliminating the supplementary resistance heaters, the heat pump would present the utility with a desirable type of load because input power for the compression cycle drops off at reduced outdoor temperature (Table 1). There are alternative ways of providing the supplementary heating. Today a number of manufacturers offer packages which may be installed on a new or existing oil or gas furnace. When located in the position of a normal cooling coil, the heat pump can provide all the heating above the balance point, while the furnace provides the heating below it.

One drawback to the heat pump furnace combination is that the heat pump and furnace cannot be operated simultaneously (i.e., the heat pump indoor coil is located downstream of the furnace discharge). The reason for this is that the temperature of the return air (air entering the heat pump indoor coil) must be limited, otherwise compressor power input increases resulting in a high refrigerant discharge temperature. This would result in a reduction in performance (COP) and heating capacity, and would impose high stress conditions on the compressor. Other methods allow the heat pump to run below the balance point. By employing a two-stage thermostat,[3] the furnace cycles on at the balance point temperature (sensed by a thermistor located in the air stream entering the outdoor coil). When the furnace supply air reaches a predetermined temperature the heat pump cycles off. After the furnace has satisfied the second stage and cycled off, the indoor blower continues to run until supply air drops to a temperature below the previous set point. A few minutes later the heat pump comes back on-line and attempts to satisfy the demand. If it cannot, the furnace repeats the cycle.

One other method is to employ a hydronic system,[4] with a water-to-air heat exchanger in the supply duct downstream from the heat pump condenser coil. This would allow the heat pump to operate continuously below the balance point. The hydronic supplemental heat may be provided by direct combustion of fossil fuel, limiting the fossil fuel to that required to make up the difference between building load and heat pump capacity.

If the cooling function of the heat pump is not required or desired, the indoor coil could be located upstream of the furnace. This would allow continuous operation of the heat pump below the balance point. Current practice reflects the concern that, in the cooling mode, the heat pump dehumidifies the return air from the conditioned space and any condensate entrained in the air stream could contribute to corrosion in the heat exchanger of the furnace.

ALTERNATIVE HEAT SOURCES FOR THE HEAT PUMP

Subsurface Ground

Throughout the late 1940's and 50's, several electric utilities, universities and other research organizations spent considerable effort and money assessing the potential of the ground as a heat source and sink for the heat pump.[5] The concensus of opinion at the time was that an air source heat pump would provide inadequate capacity under cold ambient conditions. By the late 1950's, however, the work was discontinued for a number of reasons some of which will be outlined here:

1. The air, as a heat source, is more universally predictable. A manufacturer can develop a product line which can be used over a wide range of climatic conditions. However, in the case of a ground source, soil properties tend to vary considerably from one region to another, requiring on site soil analysis for each installation. In particular, the amount and configuration of the ground heat exchanger depends on the depth of the water table in the ground.

2. Where summer cooling is of primary importance the ground is far from an ideal heat sink. The heat dissipated from the conditioned space to the soil drives the soil moisture from the vicinity of the coil. In time, voids and cavities, due to drying

action around the heat exchange surface, result in a dramatic reduction in heat sink capacity.

3. The cost of installing the heat transfer coil was perhaps the most important reason for the limited application of the ground source heat pump. In some instances, however, depending on soil type, moisture content or proximity of ground water, the ground source heat pump may offer performance advantages over the air source heat pump, which might offset the greater capital investment required for the ground coil.

One of the most definitive studies was conducted in Port Credit, Ontario between 1949-52.[6] This was a joint study undertaken by Ontario Hydro and the University of Toronto. An Ontario Hydro employee's home was used for the study. Three separate ground coils were laid, each with an equivalent length of 94.2 m (309 ft), at a depth of 1.5 m (5 ft), on a lot 30.5 by 45.7 m (100 by 150 ft) (Figure 7). The soil on the site was described as a very fine sand with a shallow water table. An ethyl alcohol and water anti-freeze solution was circulated in the ground coil system which was interfaced with a water-to-air heat pump. The performance of the installation was monitored for three winters (Table 3).

Because three separate ground coils were available, the effect of varying the installed ground coil capacity could be examined. For the winter of 1950-51 (Figure 8) all three coils were used and 48 per cent of the annual heating energy requirement was supplied by the heat extracted from the ground. A 3.73 kW (5 hp) motor was employed to drive the compressor. In the winter of 1951-52 (Figure 9) only one ground coil was employed and yet 43 per cent of the annual heating requirement was supplied from the ground. The 3.73 kW (5 hp) compressor was replaced with a 2.24 kW (3 hp) unit. In spite of a 66 per cent reduction in the effective ground coil length, the energy savings were reduced by only 10 per cent. This indicates that cost optimization studies are needed to select the most economical amount of ground coil and size of heat pump.

The Solar-Assisted Heat Pump

In the past, because of its intermittent nature and a need for a satisfactory means of collection, solar energy was passed by in favour of heat sources that were an indirect result of solar input (such as the ambient air and ground). By employing some form of thermal energy storage, the problems due to the intermittent nature of solar radiation can be largely overcome.

The efficiency of a flat plate solar collector increases as the absorber temperature approaches the outdoor air temperature (Figure 10).

To provide direct space heating, the collector would have to operate at a large temperature difference between absorber and outdoor ambient under typical midwinter Canadian conditions. However, by operating the collector as the heat source for a heat pump, the collection temperature can be reduced, resulting in higher collector efficiency and larger solar contribution over the course of the heating season (Figure 10).

Not unlike a conventional air-to-air heat pump, where the outdoor evaporator surface area limits the "free" heat available to the cycle, the effectiveness of the solar-assisted heat pump is governed by the collector area and thermal storage capacity. Increasing the size of the collector area (for a fixed storage capacity) seems to have a more pronounced effect on system performance than increasing the size of storage (with collector area fixed).[5,7] However, there are cost and space limitations that dictate the upper limit for collector area and thermal storage volume. For this reason, a number of options should be considered (see Figure 11). Provision could be made for an outdoor air-to-air heat pump mode.[5,7] On mild, sunny days, the heat pump can satisfy the heating demands of the space using outdoor air as a heat source, and the collector loop can charge the storage. Under mild, overcast conditions, this mode of operation will still satisfy the heating requirements, without depleting the thermal storage. A system with heat storage on both evaporator and condenser sides of the heat pump offers performance advantages over a single storage system.[5,7] This would allow for heat pump operation during "off-peak" hours only. It would also be possible to operate with a smaller capacity heat pump, running more or less continuously, not necessarily matching the heating demands of the space at any given time. Further studies are required to assess the potential of these and other solar-assisted heat pump systems under Canadian climatic conditions.

"ICE-MAKER" HEAT PUMP

The most commonly employed medium for thermal energy storage is water. For each kilogram (pound) of water, and C deg (F deg) difference between the upper and lower temperature limits of storage, 4.187 kJ (1 Btu) of heat can be stored. A thermal storage of 1.055 GJ (10^6 Btu) with a useful temperature range of 27.7 C deg (50 F deg) requires 9072 kg (20 000 lb) of water (9077 ℓ; 2000 gal.) and would occupy a space of 9.07 m^3 (320 cu ft). This would apply where only sensible heat of water could be utilized, as in a solar heating application, for example. Consider instead, operation between -1 to 10°C (30 to 50°F), which encompasses the freezing point of water. The heat capacity would be 379 kJ/kg (163 Btu/lb), the latent heat of fusion of water yielding 335 kJ/kg (144 Btu/lb) with an additional 44 kJ/kg (19 Btu/lb) sensible heat (ice has only half the specific heat of water). For this temperature range, a thermal storage of 1.055 GJ (10^6 Btu) would require approximately 98 ℓ (613 gal.) of water that would occupy a space of 3 m^3 (107 cu ft) when frozen.

One such system being developed at the Oak Ridge National Laboratory[8] is an "ice-maker" heat pump (Figure 12). An aluminum plate evaporator replaces the conventional cross-flow air-to-refrigerant heat exchanger of a typical air-to-air heat pump. A circulating pump delivers chilled water to a distributor located above the evaporator plate. The water freezes on the surface of the evaporator, which is maintained at approximately -7°C (20°F), liberating the latent heat of fusion to the evaporating refrigerant. In time, the ice builds up and owing to its insulating effect performance, tends to drop off. The ice is released from the evaporator plate by using warm liquid refrigerant that leaves the condenser to temporarily raise the plate temperature above 0°C (32°F). The ice subsequently melts during mild weather. In more northern regions, some type of solar collector may be required to melt the ice and provide an auxiliary source of heat for the heat pump to reduce the size of the ice storage bin.

SUMMARY AND CONCLUSIONS

1. In many cases today, the air-to-air heat pump is able to compete economically with electrical resistance heating systems. From a "prime" energy use point of view, it can raise the over-all efficiency of electrical heating from 28 to 42 per cent (Figure 13). If the seasonal performance factor could be improved, in this case by 35 per cent, the air-to-air heat pump would be more resource energy efficient than conventional oil and gas furnaces.

2. A number of alternative heat pump systems, listed below, require further investigation to assess their potential for residential application:

   (i) ground source heat pump,

   (ii) solar-assisted heat pump,

   (iii) "ice-maker" heat pump.

3. There is little doubt as to the performance improvements possible through the use of alternative heat sources for the heat pump but three important factors need to be considered:

   (i) initial cost to the consumer,

   (ii) system reliability,

   (iii) load factor and time of peak demand on the electrical utility.

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TABLE 1

Seasonal Performance Factor (SPF) Calculation

Outdoor Temperature Range °C	Outdoor Temperature Range °F	Heating Hours	Avg Bldg Heat Loss (kW)	Avg Heat Pump Capacity (kW)	Running Time (%)	Rate of Input to Heat Pump (kW)	Energy Input to Heat Pump (kW·h)	Suppl Heater Input (kW·h)	Heat Pump Input to Bldg (kW·h)	Heating Energy Required (kW·h)
16 to 18	60 to 64	710	0.3	8.2	3.6	2.3	58.8	0	213.0	213.0
13 to 15	55 to 59	691	0.85	7.55	11.2	2.2	170.3	0	587.3	587.3
10 to 12	50 to 54	633	1.4	6.9	20.3	2.2	282.7	0	886.2	886.2
7 to 9	45 to 49	540	1.9	6.1	30.6	2.1	347.0	0	1 026.0	1 026.0
4 to 6	40 to 44	565	2.45	5.2	47.1	2.0	531.1	0	1 384.0	1 384.3
2 to 3	35 to 39	670	3.0	4.35	68.9	1.8	830.9	0	2 010.0	2 010.0
- 1 to 1	30 to 34	728	3.5	3.9	89.7	1.8	1 175.0	0	2 548.0	2 548.0
- 4 to - 2	25 to 29	536	4.0	3.6	100.0	1.7	911.2	214.4	1 929.6	2 144.0
- 7 to - 5	20 to 24	563	4.55	3.35	100.0	1.7	957.1	675.6	1 886.0	2 561.0
- 9 to - 6	15 to 19	383	5.1	3.1	100.0	1.7	651.1	766.0	1 187.3	1 953.3
-12 to -10	10 to 14	348	5.6	2.8	100.0	1.7	591.6	974.4	974.4	1 949.0
-15 to -13	5 to 9	275	6.2	2.5	100.0	1.7	467.5	1 017.5	687.5	1 705.0
-18 to -16	0 to 4	210	6.7	2.2	100.0	1.6	336.0	945.0	462.0	1 407.0
-20 to -19	- 5 to - 1	145	7.25	1.9	100.0	1.5	217.5	775.7	275.5	1 051.0
-23 to -21	-10 to - 6	83	7.75	1.6	100.0	1.5	124.5	510.4	132.8	643.3
-26 to -24	-15 to -11	41	8.3	1.3	100.0	1.3	53.3	287.0	53.3	340.3
						TOTAL	7 705.6	6 166.0	16 242.9	22 408.7

$$\text{SPF} = \frac{\text{Heat Pump Input to Building (kW·h) + Supplementary Heater Input (kW·h)}}{\text{Energy Input to Heat Pump + Supplementary Heater Input}} = \frac{22\ 408}{13\ 871} = 1.61$$

Energy savings over straight resistance heating = 8 538 kW·h

TABLE 2

Comparative Costs of Air-to-Air Heat Pump versus Resistance Heating

Costs	Heat Pump 7 kW (2 ton)		Electric Furnace (15 kW)
Installed Cost	\$2500		\$1000
Difference in Installed Cost		\$1500	
Maintenance Cost	\$50		\$20
Difference in Maintenance Cost		\$30/year	
Energy Cost in First Year (@ 2.3¢/kW·h)	\$319 (13 870 kW·h)		\$515 (22 408 kW·h)
Net First Year Saving		\$166	

TABLE 3

Heat-Pump-System Performance Data for the Port Credit House

Heating Period	1949-50 Oct. 26-June 9	1950-51 Oct. 22-June 1	1951-52 Sept. 17-June 6
Degree Days, °C (°F)	3 816 (6 869)	3 634 (6 542)	3 993 (7 187)
Total Heat Energy Supplied, kW·h (Btu), from Ground Coil	14 910 (50 890 000)	14 883 (50 795 000)	13 671 (46 659 000)
Total Electrical Energy Supplied for Heating, kW·h			
To Compressor Motor	10 181	10 641	10 414
To Auxiliary Motors	2 330	3 031	3 926
To Resistance Heaters	5 189	1 998	3 937
Annual Coefficient of Performance (COP)			
Over-all	1.83	1.95	1.75
Heat Pump (Ground Coil and Compressor)	2.47	2.40	2.31
Heat Pump and Resistance Heaters	1.96	2.18	1.95
Heat Pump and Auxiliary Motors	2.18	2.09	1.95
Effective Length of Ground Coil, m (ft)	186.5 (612)	280 (918)	93 (306)
Total Heating Season, h	5 183	5 161	6 314
Total Operation of Heat Pump, h	3 789	2 724	4 068
Annual Heating Requirements Supplied, %			
By Heat Pump	77.0	83.8	75.4
By Ground Coil	45.6	48.0	42.8
By Resistance Heaters	15.8	6.5	12.3
By Auxiliary Motors	7.2	9.7	12.3

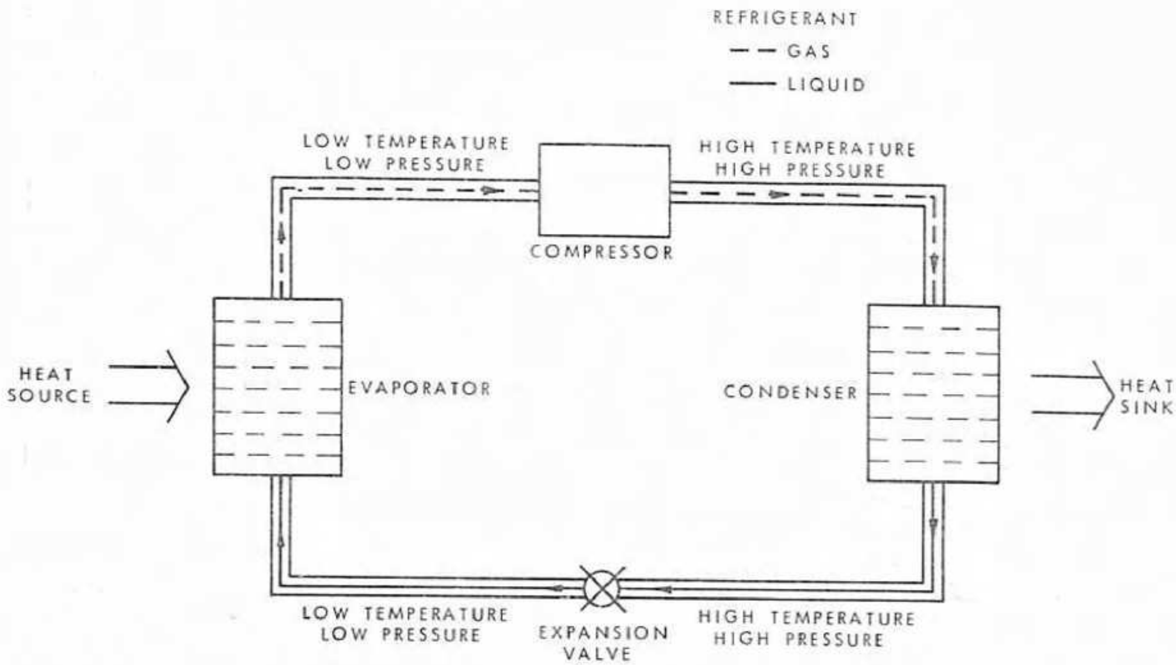


FIGURE 1

BASIC VAPOR COMPRESSION REFRIGERATION CYCLE

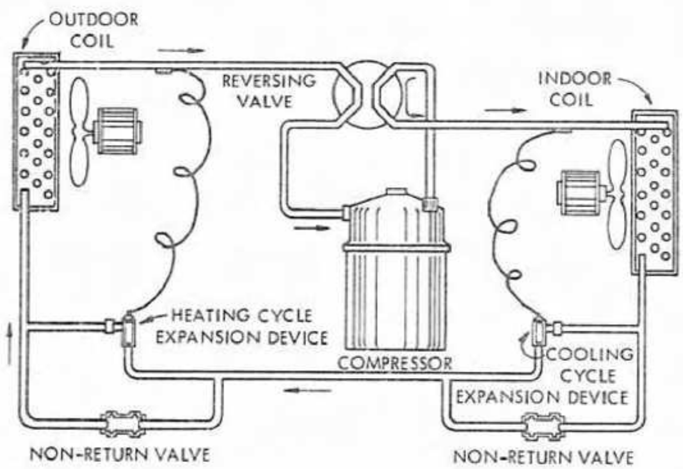


FIGURE 2

COMPONENTS OF AN AIR-TO-AIR HEAT PUMP

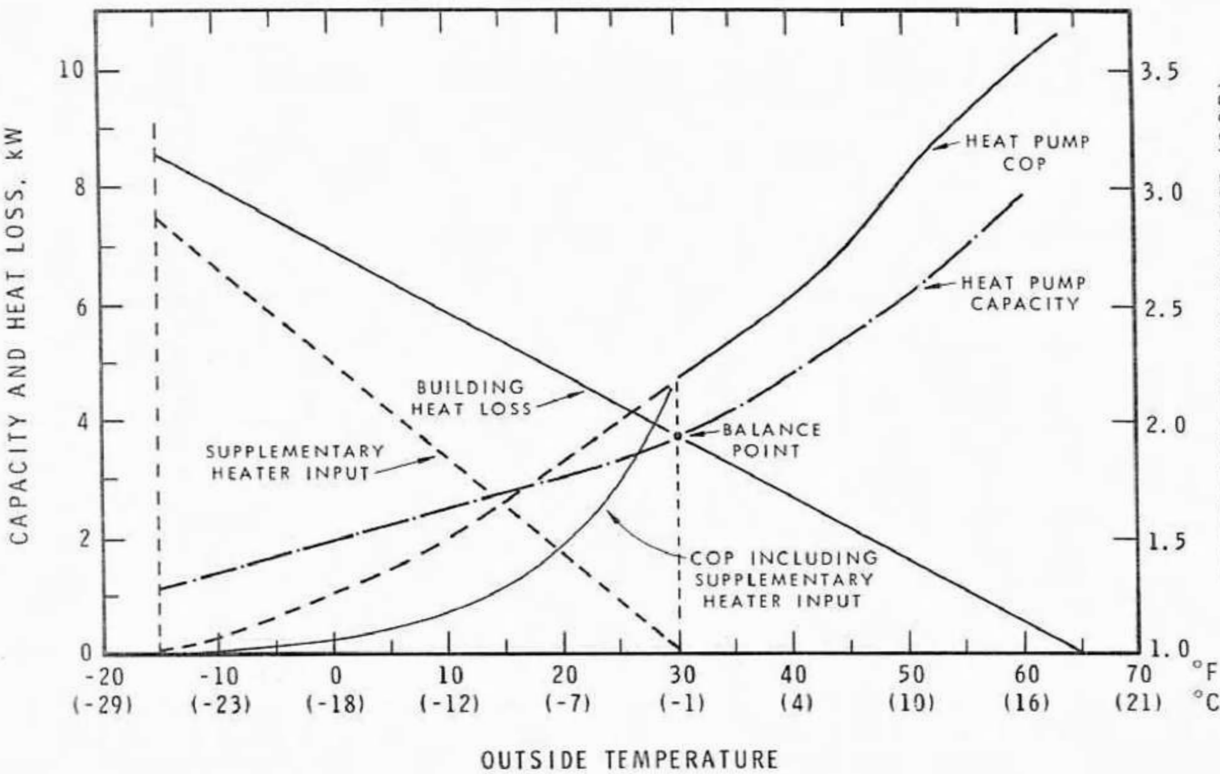


FIGURE 3

PERFORMANCE CHARACTERISTICS OF AN AIR-TO-AIR HEAT PUMP

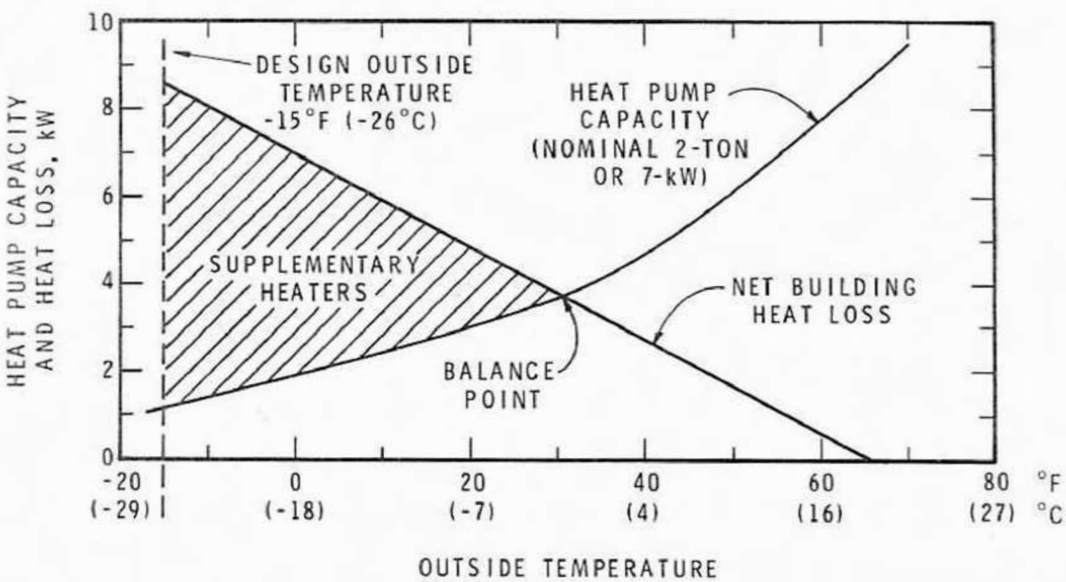


FIGURE 4

HEAT PUMP CAPACITY AND BUILDING HEAT LOSS FOR A HOME IN THE OTTAWA AREA

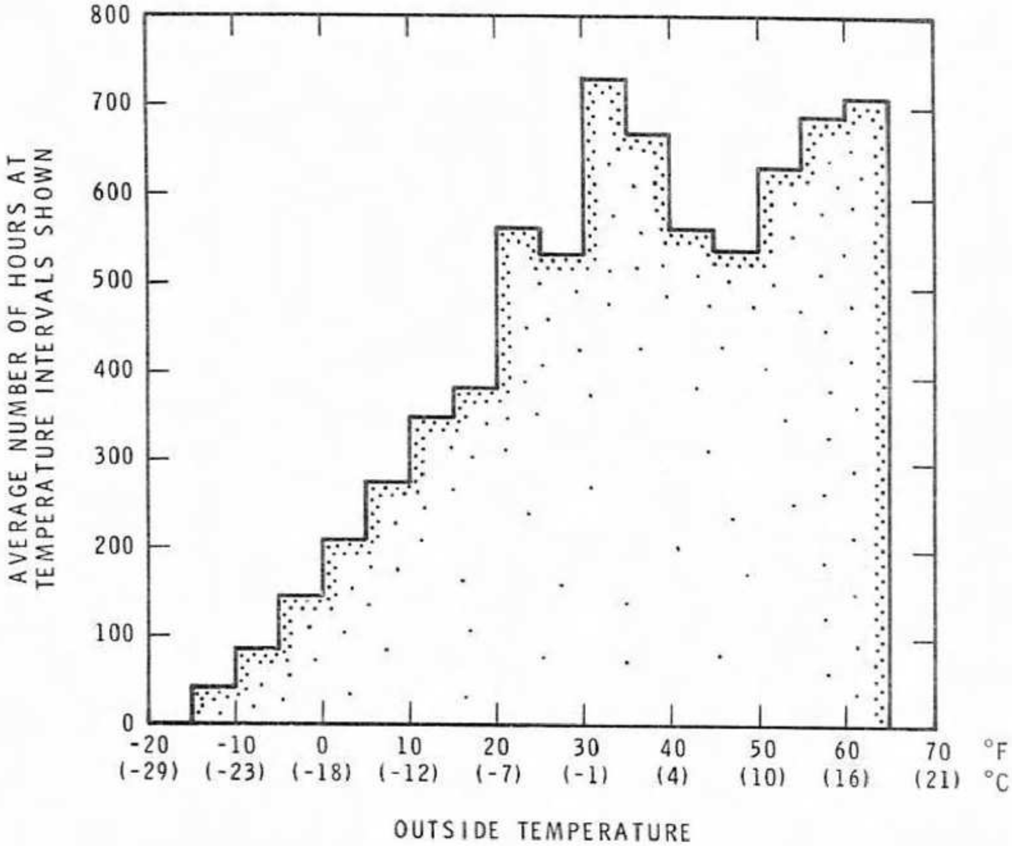


FIGURE 5

FREQUENCY DISTRIBUTION OF HOURLY TEMPERATURES IN OTTAWA BELOW 65°F (18°C)

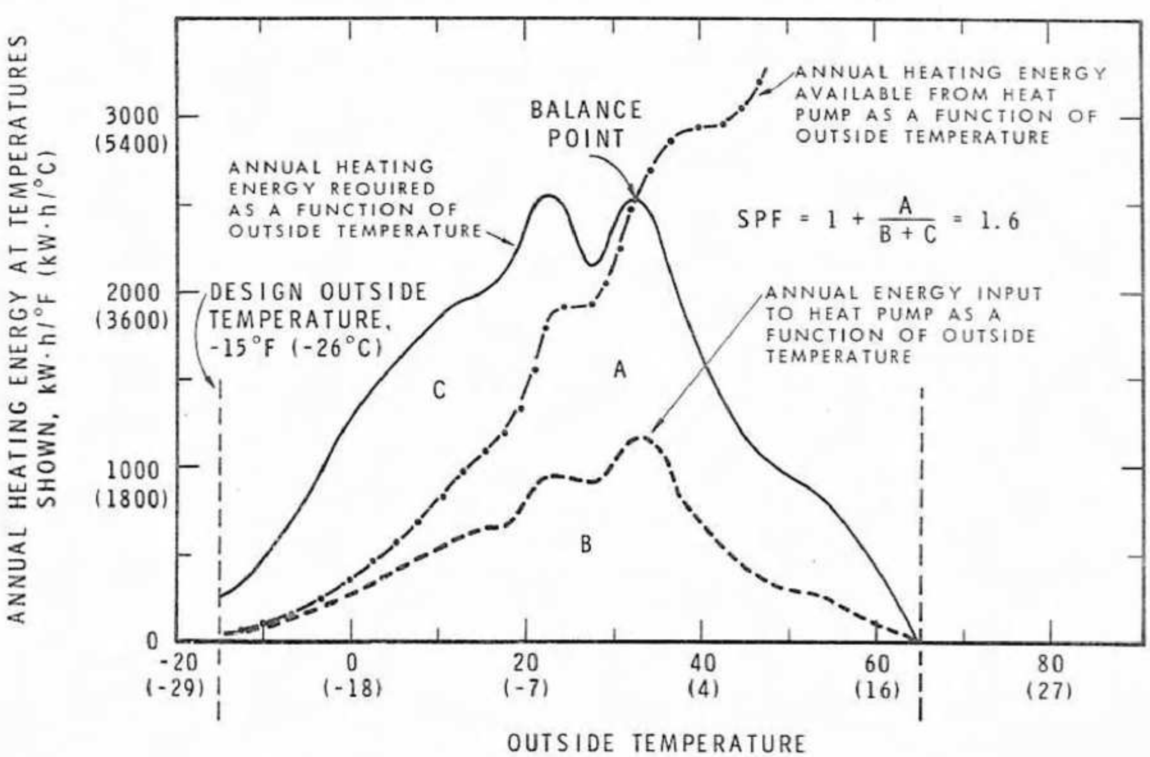


FIGURE 6

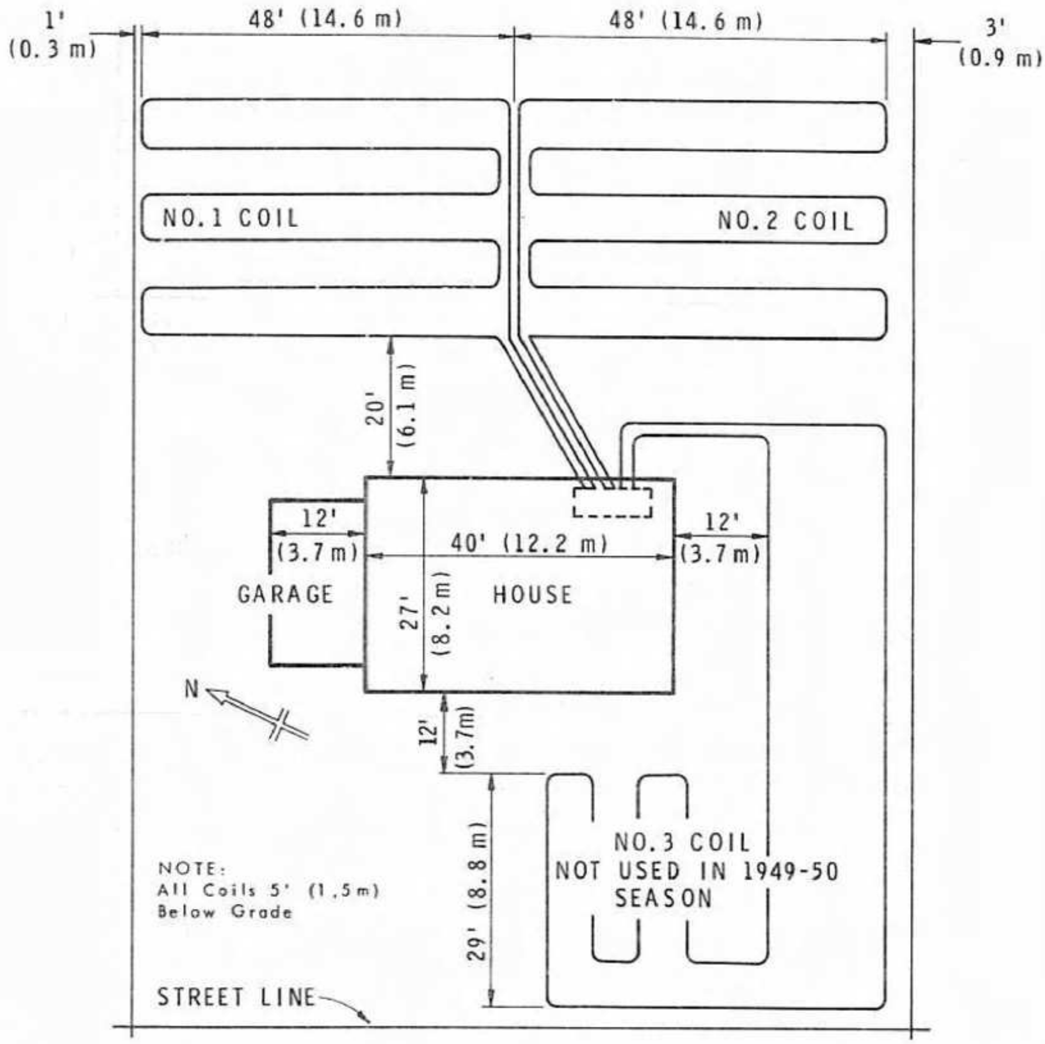


FIGURE 7

SITE PLAN OF GROUND SOURCE HEAT PUMP INSULATION AT PORT CREDIT

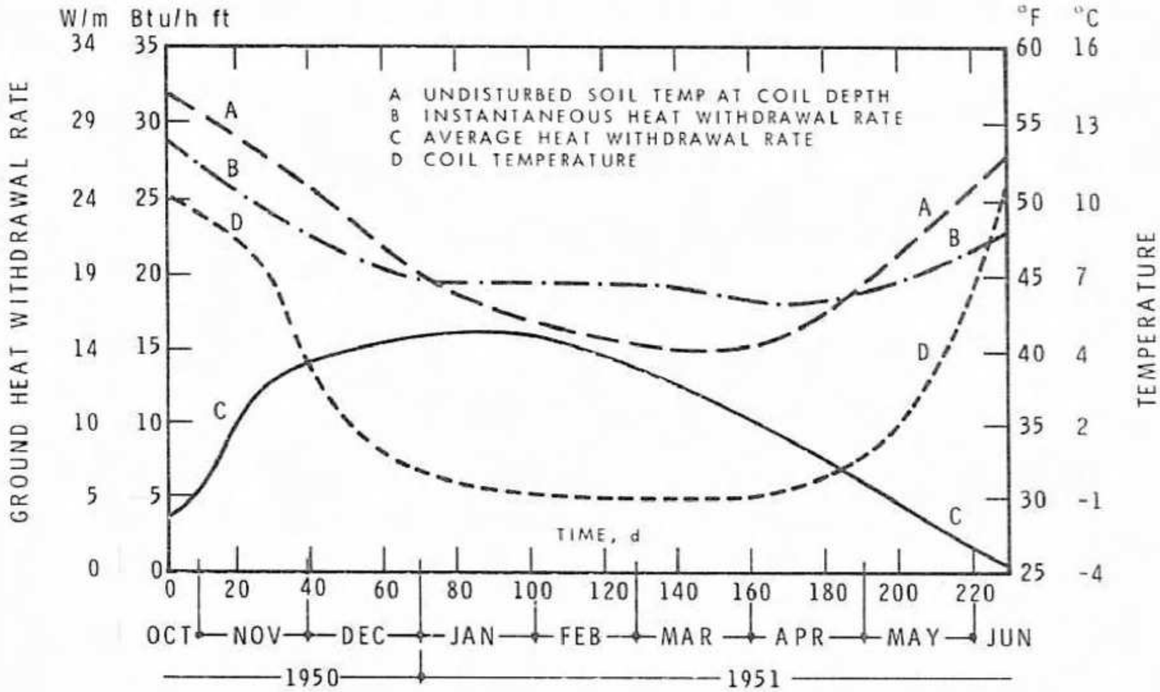


FIGURE 8

GROUND COIL PERFORMANCE - WINTER 1950 - 51

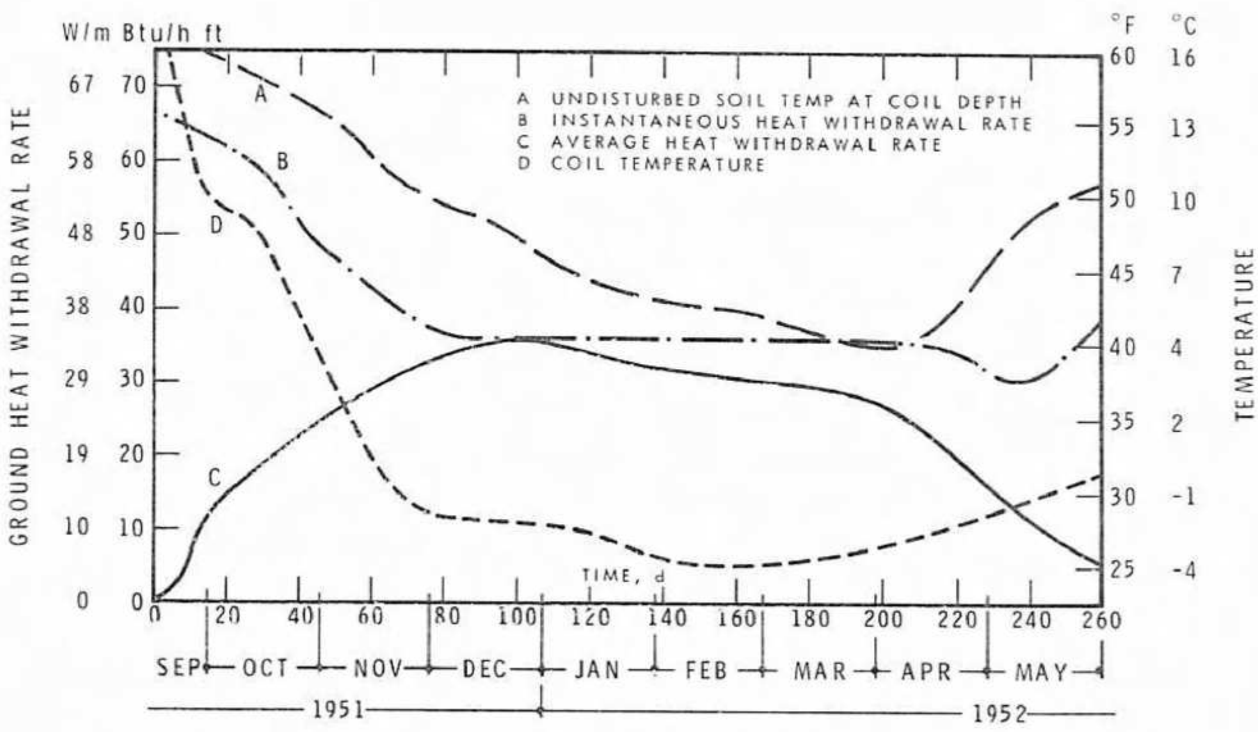


FIGURE 9

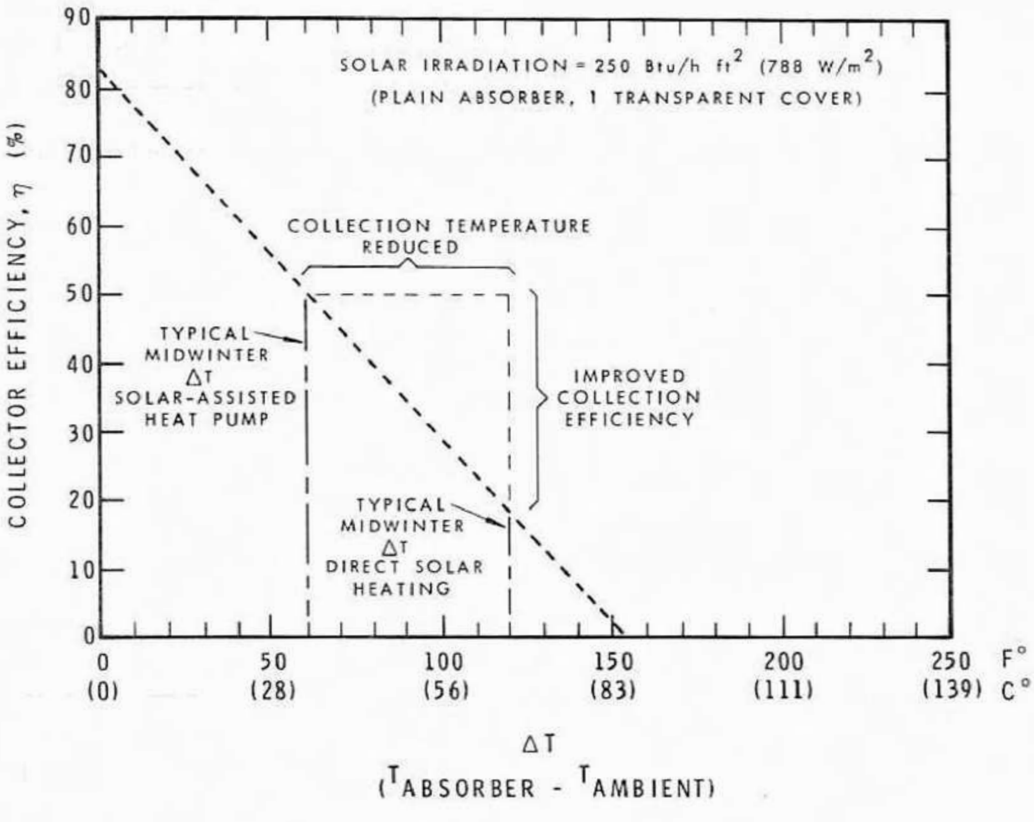


FIGURE 10

COLLECTOR PERFORMANCE WITH AND WITHOUT HEAT PUMP

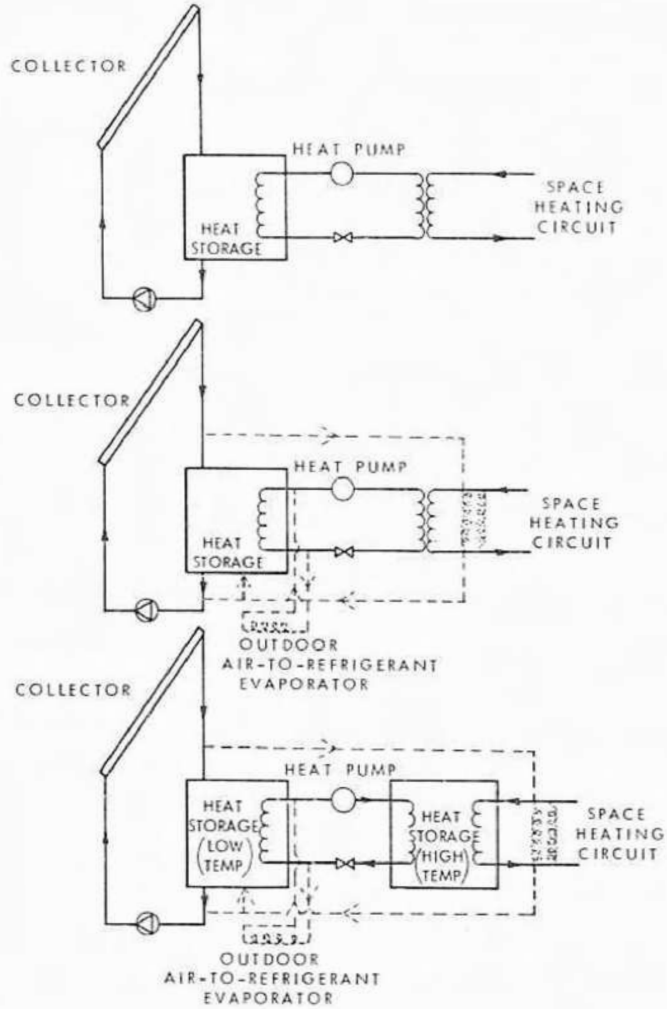


FIGURE 11

POSSIBLE CONFIGURATIONS OF SOLAR-ASSISTED HEAT PUMPS

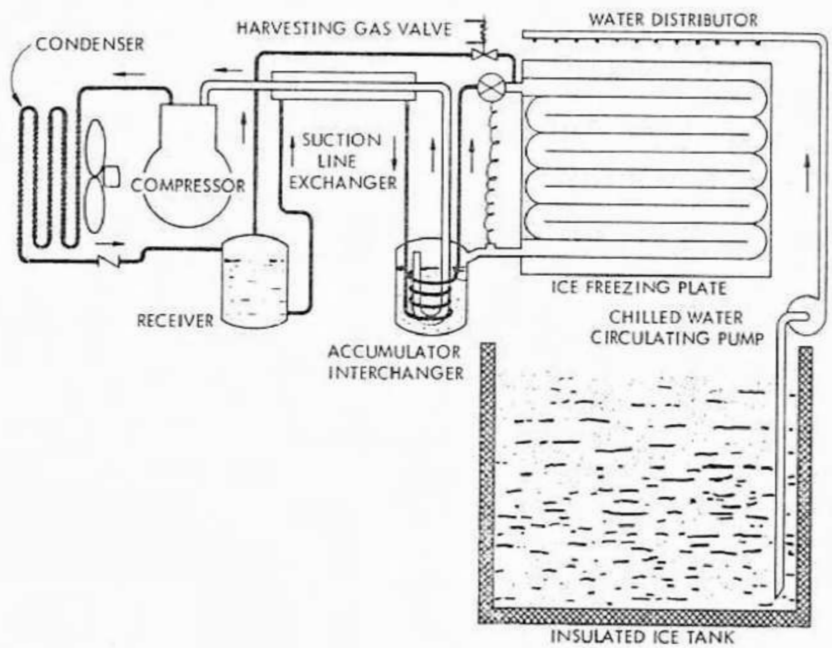
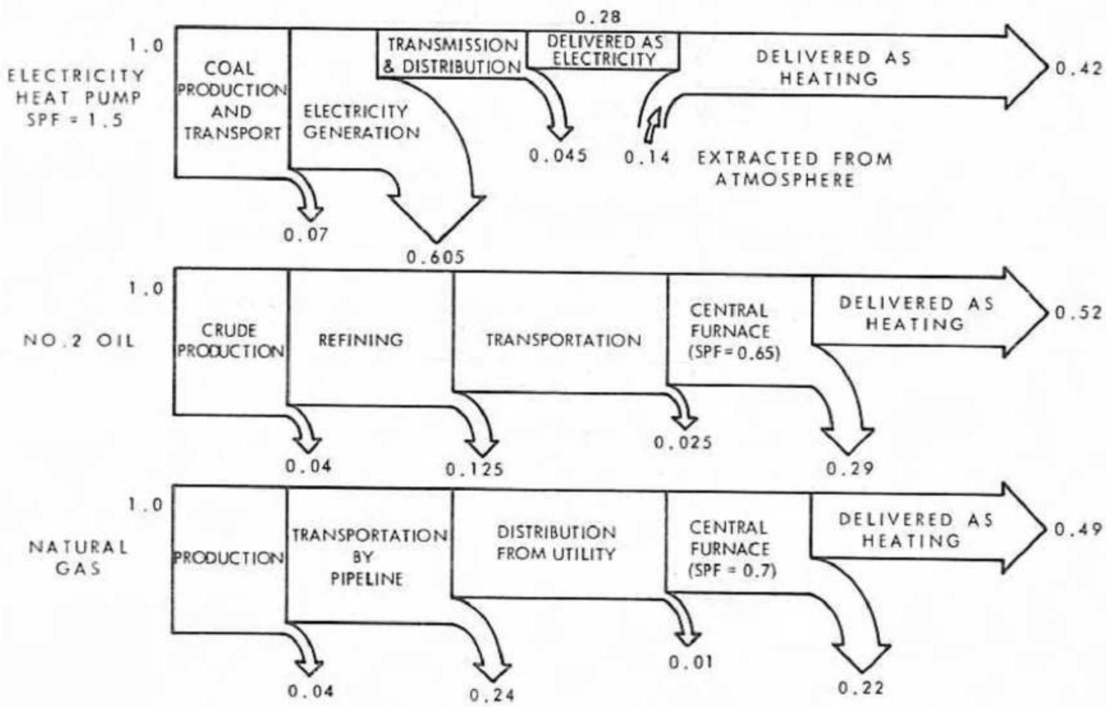


FIGURE 12



* Will Vary Depending On Pipe Line Distance, Capacity, Etc.

FIGURE 13

PRIME ENERGY REQUIRED FOR HEATING