UNIVERSITY OF ALBERTA

ANALYSIS OF A SOLAR-ASSISTED HEAT PUMP SYSTEM

BY

FREDERICK C. OOI

A thesis submitted to the Faculty of Graduate Studies and Research in partial fulfillment of the requirements for the degree of Master of Science.

DEPARTMENT OF MECHANICAL ENGINEERING

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Frederick C. Ooi

Frederick C. Ooi
51, Kelawai Road, Penang,
Malaysia, 10250.

7th JUNE, 1993

UNIVERSITY OF ALBERTA

FACULTY OF GRADUATE STUDIES AND RESEARCH

The undersigned certify that they have read, and recommend to the Faculty of Graduate Studies and Research for acceptance, a thesis entitled **ANALYSIS OF A SOLAR-ASSISTED HEAT PUMP SYSTEM** submitted by **FREDERICK C. OOI** in partial fulfilment of the requirements for the degree of **MASTER OF SCIENCE**.

(J.D. DALE, PROFESSOR)

Gerald W. Sadler

(G.W. SADLER, PROF. EMERITUS)

(G.S.H. LOCK, PROFESSOR)

(J. FEDDES, PROFESSOR)

10th JUNE, 1993

For My Parents

ABSTRACT

The main objective of the present study is to analyze the performance of a 1 kW solar-assisted vapor compression heat pump installed in module six at the Alberta Home Heating Research Facility to determine the characteristics, limitations, and feasibility of the heating system. The investigation also focuses on the performance of the heat pump evaporator, a solar collector, and presents a detailed set of daily performance data on the heat pump as a basis for future reference.

The analysis was performed based on data collected during the 1986-87 heating season (September '86 - March '87). A computer program was used to process and handle the hourly data. Results showed that the average coefficient of performance (COP) of the heat pump was 2.0 with a maximum of 2.6 at solar radiation levels of at least 600 W/m^2. For the period of study, the minimum COP of 1.4 occurred during the night at an ambient temperature of -19 °C. The performance of the unit was indirectly limited by the compressor due to a suction pressure regulator which protected the component from excessive pressures. The solar collector efficiency was found to vary from 0.2 to 0.6 at solar radiation levels of 1000 W/m^2 and 300 W/m^2, respectively.

An economic analysis based on 1993 fuel prices in Alberta, Canada ($17.67 per GJ of electricity, $2.639 per GJ of natural gas), indicated that solar-assistance was not feasible for this particular heat pump in this region. Overall, the heat pump was found to be more economically feasible than the electric resistance heater but significantly more expensive than the natural gas furnace.

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INTRODUCTION

In cold climate countries such as Canada a significant portion of the region's available energy supply is used for heating. This is particularly evident in the prairie regions where winter temperatures occasionally plunge to -40 °C. Buildings in cities like Edmonton, Alberta are equipped with exceptional heating systems to brace against any such extremes. The availability and affordability of natural gas has long deterred manufacturers from shifting their focus from the current heating technology using this fuel. However, the need to explore other sources of energy, different methods of heating, and methods of improving system efficiency have been the basis of research and development in this field.

One major problem is the lack of quantitative data available for study. In 1979, the University of Alberta in Edmonton proposed and developed a test site called the Alberta Home Heating Research Facility (AHHRF) to be used as an aid in the research and development of optimum insulation-heating strategies for cold climates. The facility is located at the University of Alberta Ellerslie Research Station, in a rural location, and has six single storey modules of approximately 6.7 x 7.3 meters in plan, with full basements, constructed side-by side in a single east-west row. All six modules have the same exterior, interior, and roofing finish, a different insulation level ("standard" to "superinsulated") and, except for the passive module and the module which is "superinsulated", no south facing windows. All modules are electrically heated to permit a direct determination of the energy consumption and are uninhabited to prevent the complication of measurements due to the different lifestyles of people. Among the various heating methods implemented were a passive solar heating system, an active solar heating system, and a solar-assisted heat pump system. This study will be concerned with the latter.

1.1 HEAT PUMPS

In the environment thermal energy is available in abundance. It is present in the soil, in ground water, in the surrounding air, and even in the sunshine. This energy can be extracted from a material or space and moved to an environment at a higher temperature through the use of a heat pump. The refrigerator and the heat pump are devices which are used to perform such a task and are actually one and the same. However function dictates their identity. The function of the refrigerator is to remove heat from a space at a low temperature resulting in a cooling effect while the heat pump is defined as a device which extracts heat from a "source" at a low temperature **(T_L)** **(Qin)** and discharges it as useful heat to a "sink" at a higher temperature **(T_H)** with the addition of external energy **(Win)** for a heating effect **(Qout)**. Nevertheless, since the heat pump's warm and cold sides can be used alternatively, the device can also act in a cooling mode. This dual function characteristic makes it a very attractive device for household or commercial building cooling or heating depending on the season. Figure 1.1 illustrates the function of a vapor compression heat pump or refrigerator. Figure 1.2 illustrates the schematic of the cycle showing the four main components in a vapor compression heat pump (the evaporator, the condenser, the expansion valve, and the compressor).

Unlike the refrigerator which did not encounter any opposition since few processes (e.g. absorption cooling) for continuous cooling exist, the heat pump for heating purposes has had to compete with conventional heating methods which utilize inexpensive, abundant, and readily available fossil fuels. It is therefore not surprising that the device has been slow in gaining acceptance among consumers as a practical heating instrument. In today's energy and environmentally conscious society many experts believe the heat pump represents an economical and efficient alternative to heating systems which utilize fossil fuels such as oil, gas, coal, etc. which are not only

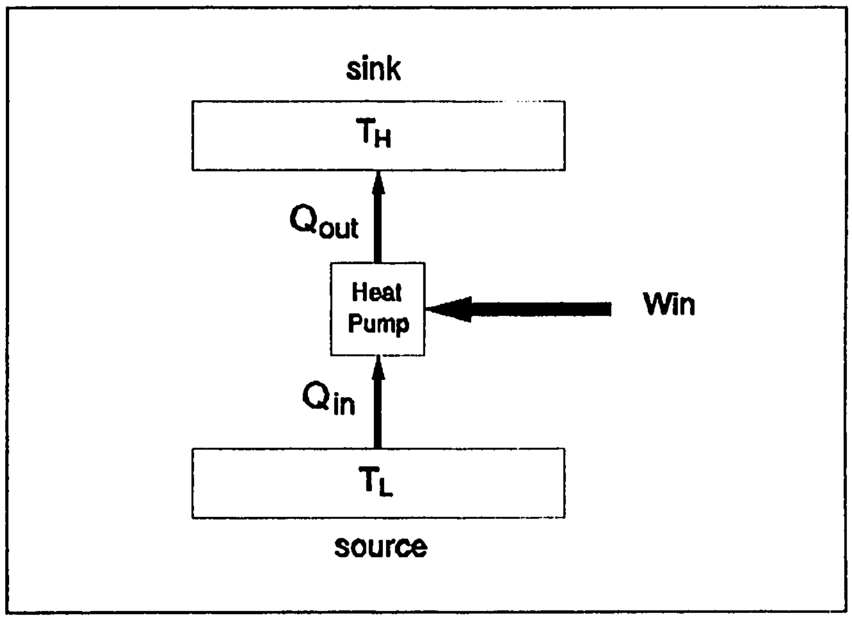

Figure 1.1 - The Heat Pump

depletive and pollutive to the environment but also unstable in price and availability and are only found in certain regions. However, there are some regions (such as Western Canada) where fossil fuels are abundantly available and electricity is relatively expensive (due to a lack of hydro power).

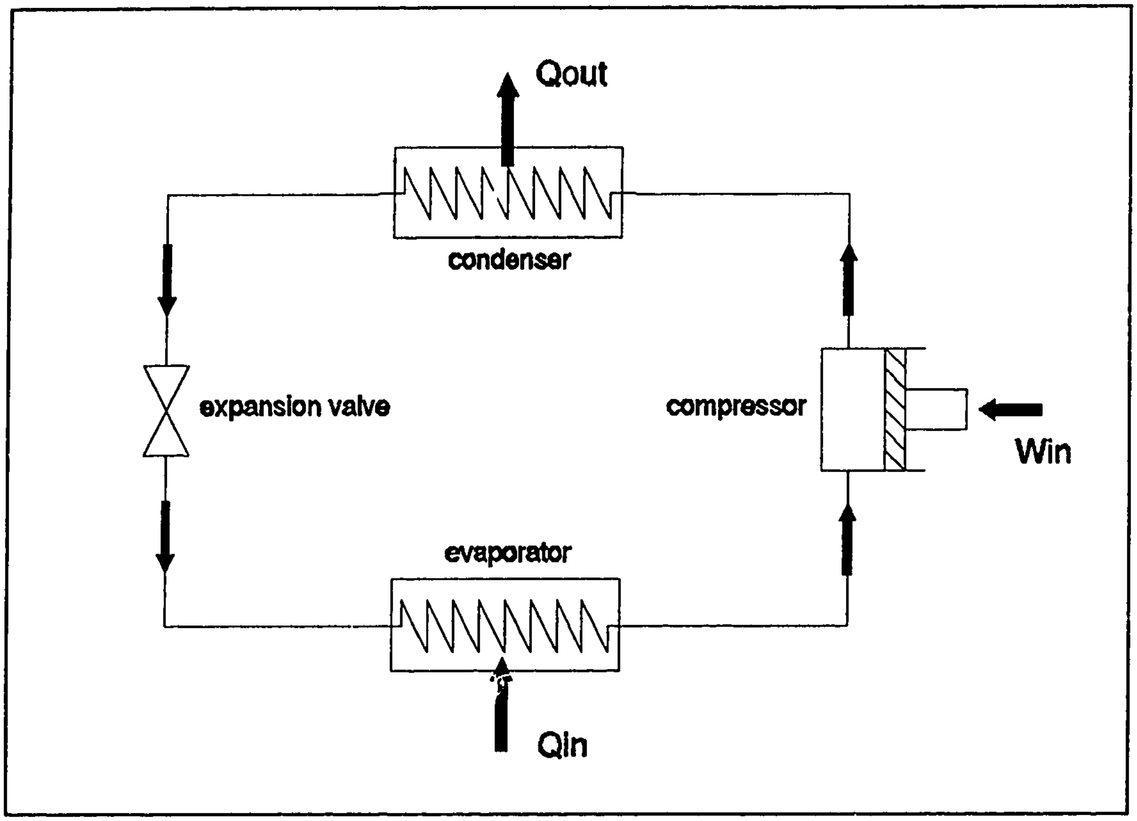

Figure 1.2 - Basic Components of the Vapor Compression Heat Pump

1.2 TYPES OF HEAT PUMPS AND HEAT SOURCES

In today's market not only are there numerous types of heating systems, there are also several different types of heat pumps. To make a proper selection, a consumer has to weigh carefully the advantages and disadvantages of each system and match it to his or her requirements. A knowledge of the various types of heating systems available, in this case the types of heat pumps, is therefore valuable.

Heat pumps can be classified in different ways. In the USA and Canada it has been customary to classify them according to their heat source(s) and sink(s). However, it is also useful to consider the type of thermodynamic process used. Among the different types are compression heat pumps, absorption heat pumps, and vapor jet heat pumps. Within these classifications further distinctions can be made according to the heat source(s) or sink(s) used. Some examples of natural heat sources are soil, surface water, ground water, solar radiation, and air. Water and air are also common heat sinks. Unnatural heat sources are waste heat sources such as exhaust air from heated spaces, waste water, and other fluids from industrial and commercial buildings. An air-water heat pump, for example, would therefore be a heat pump which uses air as the heat source and water as the heat sink.

Heat sources largely determine the operating characteristics and economics of a heat pump since the temperature difference between the source and sink must be overcome by adding energy. For economic operation of heat pumps the ideal heat source should have as high a temperature as possible and should be available at any time and anywhere (independent of geographical location and climate). Soil was considered the best source of heat during the 1940s and the majority of research was devoted toward it. However, the difficulties in placing coils in the ground forced manufacturers to turn to air which is readily available as the alternative although the resulting coefficient of performance (COP) is considerably lower. Combined heat sources, where two or more different heat sources are used to complement each other, might result in a better performance. The unit under study is a vapor compression heat pump which uses a combination of air and solar radiation (outside air and solar radiation collected on a solar array) as heat sources and air (the heated space within the building) as the sink. According to Ambrose (1966), outdoor air offers a universal, inexpensive, and abundant heat source-sink for the heat pump and the air to air design is by far the most universally used, particularly for residential and smaller commercial installations.

1.3 ADVANTAGES OF THE HEAT PUMP

There are several advantages in using heat pumps rather than more conventional heating systems. The most obvious and basic advantage is that the heat pump can be used to both heat and cool. In countries such as Canada this characteristic (heating and cooling) is invaluable because the equipment can potentially be used throughout the year resulting in a lower operating cost. According to Langley (1983), "because the equipment is in use all year long, a lower cost per hour of use on the original equipment purchase is attained".

A second advantage is the fact that the heat pump is capable of yielding a higher heating output per unit of fuel than any of the conventional fossil fuel heaters. This factor not only results in cheaper electric bills for the owner but also in more sensible and efficient use of the world's swiftly depleting fuel resources. Although this may be less true when considering power plant efficiencies, the emphasis toward solar energy makes the heat pump the better choice for space heating in the future.

1.4 DISADVANTAGES OF THE HEAT PUMP

The heat pump is not without its drawbacks. Its major disadvantage, probably the chief reason for its slow acceptance by the public, is its high capital cost compared with conventional equipment. Consumers will not readily abandon their performance proven heating equipment for new technology which carries a high initial cost. In addition, Sherratt (1984) states that low fuel prices would result in long payback periods. The frequency of compressor failures in the past will also raise doubts for potential customers although today the component is more reliable and is usually guaranteed for several years.

Another weakness is the limited operating range of the heat pump. In mild climates this would not be a significant factor. However in relatively cold climates like Alberta's a supplementary heat source may be necessary. Consumers would naturally choose devices which have a higher operating range, such as the natural gas furnace, to cover the total heating load and which would, as a consequence, not require secondary heating.

Recent findings in the research in ozone depletion have led to the banning of R-12, the most widely used refrigerant today, in North America by the year 1996. The refrigerant's harmful effect on the earth's protective layer has forced manufacturers to turn to a different alternative. Currently refrigerant R-134a has been designated as the substitute. A comparison between these two refrigerants in a residential exhaust air heat pump was performed by Linton et al. (1989). For R-134a to be a worthy stand-in for R-12, new systems would have to be designed.

1.5 THE SOLAR-ASSISTED HEAT PUMP

A solar source heat pump is basically a vapor compression heat pump which uses solar radiation as a heat source. Because solar radiation is not always available, such units often require either an auxiliary source of energy for heating or a heat storage method. However, "solar-assisted" heat pumps combine solar radiation with other heat sources and are capable of continuous operation. Previous works on solar-assisted heat pumps include Terrell (1979), Hume (1980), and Balaban (1977).

To efficiently collect and absorb heat energy from solar radiation, a solar collector array is required. The collector surface is painted black or treated to have a high absorptance and a low emittance and may be covered with one or more sheets of glass or other transparent material to trap air and decrease convection losses. Efficiency is increased when the component is kept at a temperature below that of the ambient air. This would prevent heat loss to the surrounding air (since heat cannot be transferred by convection from the plate to the ambient air if the plate temperature is below the ambient temperature), increase the convection heat transfer from the ambient air, and would allow for the use of diffuse radiation which is present even under overcast skies.

Research and development in solar-source heat pumps has been concerned with direct and indirect systems. The former has evaporator tubes containing refrigerant incorporated in a flat solar collector array while the indirect system uses water or air circulated through the solar collector. Evaluation of the performance of a solar-assisted heat pump is concerned partly with the efficiency of the evaporator in absorbing heat energy. Of course the available solar radiation is dependent on the season and the locale (latitude) of the unit and the orientation of the evaporator. In Edmonton, Alberta for example, on average, solar radiation in winter is only available for about eight hours. In summer however, this value may sometimes increase to sixteen hours (but of course heating would not be required during the summer). Some of the important factors involved in solar array performance are

(a) the angle of inclination of the array

(b) the shape and size of the array

(c) the color or absorbing characteristics and type of material used

The most common shape used is the flat plate profile. This form is designed to capture as much radiation as economically and as feasibly possible. Such a form would also allow for heat transfer from the air or water (rain) flowing over it. For small buildings the array could either be mounted on the inclined roof or on the ground beside the structure served by the heat pump. The former would require that the roof be inclined in a north-south direction to optimize solar collection. However, if the building has a flat roof (as most large commercial buildings do) the array could be installed there. It is of course important to set the collector at an inclination angle to the sun such that the optimum performance is attained.

The type of material employed should be a conductor of heat and the collector surface should have a high absorptivity factor at solar wavelengths. The efficiency might be improved further with an array which is variable in angle of inclination so that the optimum angle of inclination could always be achieved with the aid of a tracking mechanism. Such a mechanism however would carry with it a high cost which might not justify the unit's improvement in COP.

1.6 RESEARCH TOPIC

An investigation of the operating characteristics of a solar-assisted vapor compression heat pump that utilizes a solar collector as its evaporator and which has an electric resistance heater for auxiliary heating is the subject of this research. The main objectives of this study are:

- to determine and evaluate the performance characteristics of the heat pump unit
- to evaluate the performance of the solar collector evaporator in the heat pump unit
- to compare the heat pump unit's energy consumption characteristics to those of a standard electric resistance heater unit installed in a similar module and to the heat pump system's own electric resistance heater
- to investigate the economic feasibility of the heat pump as compared to other heating methods
- to investigate the feasibility of solar-assistance for the heat pump
- to present a detailed set of performance data on the solar-assisted heat pump as a basis for future reference

A previous study done on the system, Sadler et al. (1987), concentrated on the effects of solar radiation and ambient temperature on the system performance and determined the equipment limitations under conditions of high ambient temperature and/or high radiation levels.

CHAPTER 2

THE HEAT PUMP SYSTEM

2.1 INTRODUCTION

Energy consumption in houses due to space heating is a major concern in cold climate countries such as Canada. A lack of quantitative data and quality research facilities impeded the study and evaluation of various heating methods available in the market. The Alberta Home Heating Research Facility (AHHRF) is located at Ellerslie, just south of the city of Edmonton. Extensive and relatively inexpensive research is conducted on various heating systems for home or commercial building use in a northern climate. The heat pump system under consideration is housed in module six of the facility and consists of a solar-assisted vapor compression heat pump, a solar collector array, an electric resistance heater to provide supplementary heating, and ducts to distribute the heated air throughout the module. Partly because of the low average temperature in the Edmonton area, the heat pump was installed for heating purposes only. By maintaining a single operating mode, the complications and added expense of reversing the refrigerant flow in the heat pump unit are also avoided.

To effectively study the heat pump unit, it is essential to have some knowledge of its environment, the construction of the building it is serving, the heat pump unit and its individual components, the thermodynamic cycle of the operation, and the auxiliary heating system. This section presents information which was also given by Gilpin et al. (1980), on these areas.

2.2 THE SYSTEM SITE

The AHHRF facility consists of six single storey modules of approximately 6.7 x 7.3 meters in plan, with full basements, constructed side-by-side in a single east-west row. All modules have the same exterior and interior finish and the same roofing finish. However, each is equipped with a different heating method and a different insulation level (from "standard" to "superinsulated"). In addition, the modules also vary in window area and window distribution. All modules are electrically heated to permit a direct determination of the energy consumption without having to consider furnace efficiencies. To allow for an accurate measure of performance without occupant influence, the modules are uninhabited. Performance of the test modules have been reported by Ackerman et al. (1985).

2.2.1 ENVIRONMENT

The test site is located at the Ellerslie Research Station of the University of Alberta. The six test modules are constructed in a single east-west row and the land is open except for the farm buildings located to the west and north of the site. To the north and south, the land is used for growing crops only. Geographically, the site is at a latitude of 53.57 degrees.

The Edmonton region has an annual climate of approximately 5700 Heating Degree Days and experiences about four to five months of winter where the average ambient temperature drops below zero °C. According to Duffie and Beckman (1980), the monthly average daily radiation on a horizontal surface H (kJ/m^2), the monthly average clearness index K_T, the 24 hour average ambient temperature Ta (C), and the average number of degree days in the month DD (degree Celsius days) for the city of Edmonton are as in Table 2.1. The average number of hours of bright sunshine per month, **hrs**, was obtained from an annual meteorological summary for the Edmonton International Airport (1992).

Table 2.1 - Pertinent Meteorological Information for Edmonton, Alberta.

	H	hrs	K_T	Ta	DD
	kJ/m^2			°C	°C days
Jan	3726	97.7	.54	-14	1006
Feb	7368	118.5	.60	-11	844
Mar	13063	172.1	.64	-5	739
Apr	17291	232.8	.58	4	425
May	21310	283.5	.57	11	222
Jun	21478	286.7	.52	14	123
Jul	22022	313.1	.56	16	41
Aug	17124	284.3	.52	15	100
Sep	12476	183.3	.53	10	228
Oct	7871	162.9	.54	5	410
Nov	4647	102.5	.57	-4	675
Dec	2763	77.5	.50	-10	891

Module six is shielded on the east side by a wind barrier, partially on the south wall where the solar collector is located, and the west wall which is the side where module five is located. Thus module five is partially shielded on its east and west walls by adjacent test houses.

2.2.2 MODULE SIX

Module six is a single storey house measuring 6.7 x 7.3 x 5 meters. The unit has a poured concrete basement which extends almost 2 meters below grade, wood frame wall construction, and a gable roof on elevated roof trusses (the basic construction of all the modules). The elevated roof trusses allows for the installation of varying thicknesses of insulation without the need to perform structural modification. The insulation levels of module six are shown in Table 2.2 (along with those of module five) and are typical of current Alberta housing stock in 1979. The figures in parentheses are the R values.

Table 2.2 - Insulation Levels of Modules Five and Six

Module	Insulation Level RSI, (R)			Window Area
	Ceiling	Walls	Basement	(m^2)
5	2.11 (12)	1.76 (10)	1.76 (10)	5.8
6	5.64 (32)	1.76 (10)	1.76 (10)	5.8

Module six has an insulated metal door which is approximately 0.9 meters x 2 meters. Three double glazed windows are located on the north, east, and west sides of the module. The south wall does not have a window and is partially shaded by the adjacent ground-mounted solar collector array which is used as the evaporator for the heat pump unit. Table 2.3 lists the details on module six's construction.

Table 2.3 - Design Details of Modules Five and Six

Floor Area:	6.7 m x 7.3 m
Wall Height:	2.4 m
Basement Height:	2.4 m, 1.8 m below grade
Walls:	9.5 mm Prestained Rough Tex Plywood 89 mm Fiberglass Batt Insulation 50 mm x 100 mm studs, 400 mm o/c 4 mil poly vapor barrier 13 mm drywall painted
Wall area/Floor area:	1.39/1
Windows:	N 0.99 m x 1.9 m double glazed sealed S None E 1 m x 1.9 m double glazed opening W 1 m x 1.9 m double glazed opening
Window area/Floor area:	11.9%
Ceiling:	Batt insulation (see Table 2.2) 9.5 mm plywood sheathing 4 mil poly vapor barrier 13 mm drywall painted
Roof:	CMHC approved trusses with 0.75 m stub 210# asphalt shingles 9.5 mm plywood Ext. GD sheathing
Basement:	13 mm preservative treated plywood to 0.6 m below grade 50 mm rigid insulation, to 0.6 m below grade 0.2 m concrete wall 0.1 m concrete slab on 6 mil poly vapor barrier
Door:	0.9 m x 2 m
Electric Furnace Capacity:	7.5 kW

2.2.3 MODULE FIVE

Module five is located beside module six and is of similar size and construction to the latter with the exception of ceiling insulation. Module five is heated with an electric resistance heater similar to the one used for supplementary heating in module six. The physical similarities, including the type of insulation, of module five to module six was the main reason for installing the heat pump system in the latter. Relatively reliable comparisons can be conducted between the heating systems of the two modules. Incidentally, module five is only used for comparative measurements and is regarded as the "standard" for comparing the relative performances of the other modules. No alterations of any sort are conducted in it.

2.3 THE HEAT PUMP

The heat pump under consideration is a 1 kW solar-assisted vapor compression unit. Solar radiation collected on its evaporator array and ambient air are its heat sources. The working fluid is refrigerant-12 (R-12) and the compressor is powered by electricity. Cube et al. (1981) claim that R-12 is the oldest, best proven and most used refrigerant. The heat pump also has a pressure limiting regulator to prevent excessive suction pressures in the compressor.

Schematics of a basic heat pump and its four main components (compressor, evaporator, condenser, expansion valve) and of the actual unit and its components are shown in Figures 2.1 and 2.2, respectively. The components found in the heat pump unit considered are (for Figure 2.2):

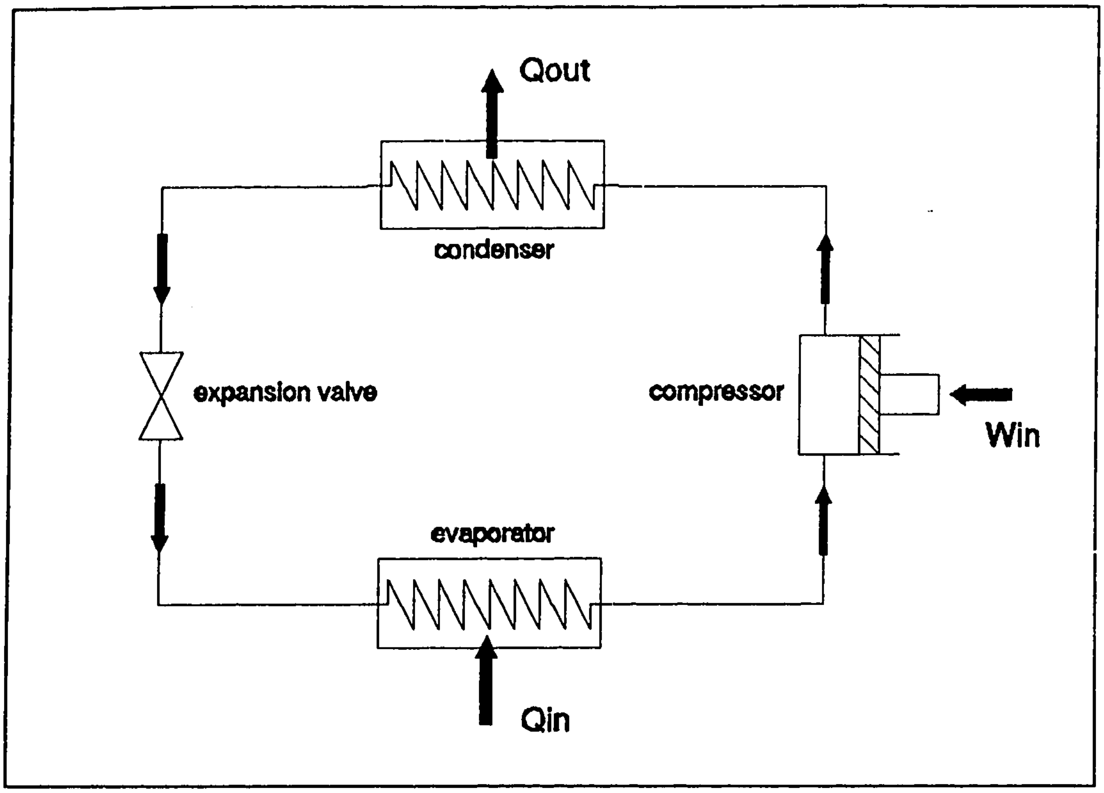

Figure 2.1 - Schematic of the Heat Pump

(a) compressor

(b) direct-exchange (D-X) air-cooled condenser

(c) receiver

(d) rotolock

(e) driers

(f) sight glass

(g) pump down solenoid

(h) thermostatic expansion valve (T-X valve)

(i) solar evaporator array

(j) accumulator

(k) suction pressure regulator (SPR)

As illustrated in Figure 2.2, these components are interconnected with pipes to form a circuit for the working fluid. An in depth look into the capabilities of the heat pump and the function of the individual components will be beneficial in understanding the system.

2.3.1 THE ACTUAL CYCLE

For all thermal machines (including the heat pump) the Carnot process is the ideal thermodynamic cycle and represents an upper limit of the performance of any compression heat pump. However, the actual process of a vapor compression heat pump differs in a number of ways from the ideal process. At the compressor for example, during intake the refrigerant is at a lower temperature than the compressor walls. Subsequently heat transfer to the fluid from the compressor occurs resulting in an increase in entropy. By the end of the compression stage the refrigerant reaches a temperature higher than that of the compressor. This in turn causes heat to flow back from the fluid to the compressor resulting in a decrease in entropy. The net result of this process can be either an increase or decrease of entropy as illustrated in Figure 2.3 (either 8-1 or 8-1') which shows the actual cycle. The state points for the process are:

8,1 - compressor inlet, outlet

2,3 - condenser inlet, outlet

4,5 - T-X valve inlet, outlet

6,7 - evaporator inlet, outlet

The various stages are:

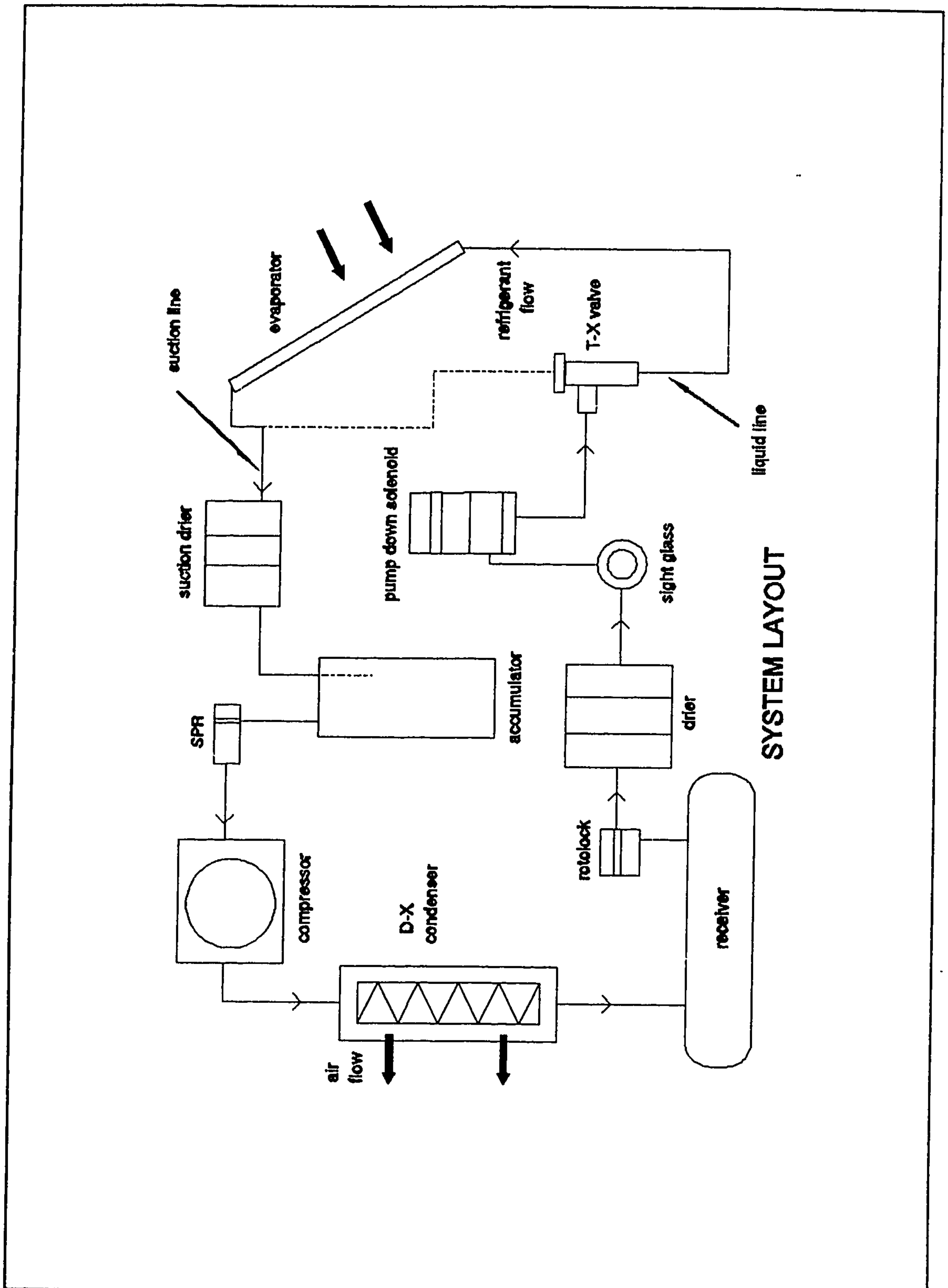

Figure 2.2 - The Heat Pump and its Components

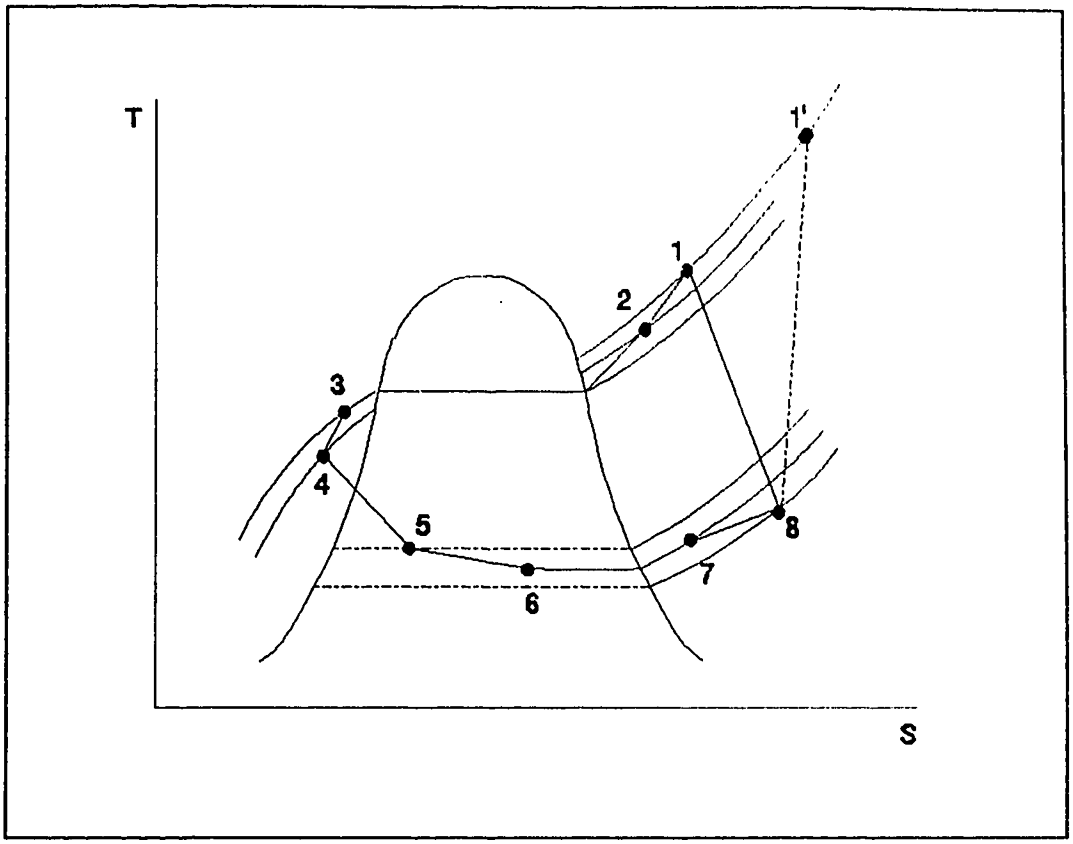

Figure 2.3 - The Actual Thermodynamic Cycle of the Heat Pump

1-2 heat rejection with pressure loss due to friction

2-3 heat rejection by condensation with pressure loss

3-4 heat rejection with subcooling

4-5 isenthalpic expansion

5-8 heat absorption with pressure loss and superheating

8-1 compression with a variable polytropic exponent

2.3.2 COMPRESSOR

To raise the pressure of the refrigerant from the evaporator to the condenser pressure, a compressor is required. It provides the necessary force to circulate the fluid through the refrigerant circuit. The larger the compression, the higher the discharge pressure, and, consequently, the greater the heating output. Various types of compressors are available. The model used is a 1 kW hermetically-sealed reciprocating compressor which is electrically powered. The motor and compressor are located within a metal casing. To prevent excessive motor temperatures which are inherent in heating modes due to the low refrigerant density at low suction pressures, low temperature refrigerant vapor is passed over the winding coils. The reciprocating compressor is single-acting and is designed to handle only vapor. To protect the compressor from liquid refrigerant, an accumulator (which will be discussed later) is used.

2.3.3 CONDENSER

The type of condenser used is a remote air-cooled direct exchange finned condensing coil in a forced air system. The condensing stage involves desuperheating, condensing, and subcooling. Superheated vapor enters the condenser and is cooled by convection to the saturated vapor temperature. Condensation then occurs theoretically engaging approximately 85% of the condenser surface area. At the saturated liquid point, subcooling begins on the liquid refrigerant to lower the vapor content of the fluid entering the evaporator to enhance the capacity of the fluid.

A fan circulates the air through the condensing coil and throughout the building in ducts. By increasing the fan speed, and therefore the mass flowrate of air across the coil, efficiency is enhanced. In an earlier experiment (Doskoch et al.(1986)) the volume flowrate of air across the condenser coil was found to be 0.212 m^3/s (450 cfm). This value is used in setting up a heat balance on the condenser to determine the refrigerant mass flowrate.

2.3.4 REFRIGERANT RECEIVER

To accommodate the fluctuations in the liquid refrigerant flow due to varying loads, a refrigerant receiver is required. A receiver stores the excess liquid refrigerant and also manages swelling and foaming of the liquid. The component used in the system is a horizontal gas/liquid separator and is located downstream from the condenser.

2.3.5 SYSTEM DRIERS

A drier is a device which removes moisture from the refrigerant. Some modern driers also execute such functions as filtering and acid removal. It contains a desiccant, a substance (usually activated alumina or silica gel) used to collect and hold moisture. Moisture is a detrimental factor in a refrigeration system. Moisture in a refrigeration system contributes to the formation of acids, sludge, and corrosion, and causes the freezing-up of flow control devices, compressor burnout, and other mechanical malfunctions.

The heat pump unit has two system driers. One is situated before the thermostatic-expansion valve in the liquid line, while the other is placed in the suction line downstream of the evaporator.

2.3.6 SIGHT GLASS

The sight glass is a miniature glass located on the refrigerant line. In the system concerned it is placed in the liquid line after the system drier to indicate the refrigerant level and to detect any restriction in the drier. A restriction will produce bubbles in the sight glass caused by flash gas.

2.3.7 PUMP DOWN SOLENOID

Building temperature is controlled by a thermostat which activates the pump down solenoid. The pump down solenoid is an electromagnetic coil with a moving core that operates a valve. It controls the compressor by controlling the refrigerant flow. For example, if heating is required, the pump down solenoid is opened and liquid refrigerant flows to the thermal expansion valve and to the evaporator causing a rise in the compressor inlet pressure until the compressor start setpoint (at which point the compressor turns on). Likewise when heat is not required the thermostat closes the pump down solenoid. The compressor continues pumping the fluid from the low side to the high side until the suction pressure (compressor inlet pressure) drops below the setpoint. Since the compressor is unable to handle this low pressure the pressure switch will initiate a shutdown. The pump down solenoid is located just before the expansion valve.

2.3.8 THERMOSTATIC EXPANSION VALVE (T-X VALVE)

In order to reduce the amount of warm liquid refrigerant to the evaporating pressure, an expansion valve is required. The system concerned employs a thermostatic expansion valve (T-X valve). A T-X valve lowers the pressure by metering the refrigerant to the evaporator through an orifice in the valve. The flowrate is modulated as required by a plunger and seat which changes the flow opening.

The thermal expansion valve is a precision device which accurately meters refrigerant at the exact rate that fluid is evaporated in the evaporator in response to a set degree of superheat. It does this by sensing the suction line temperature (the evaporator outlet). A remote bulb containing a small amount of refrigerant is attached to the evaporator outlet line. The pressure assumed by this fluid, P1, as a consequence of the superheat vapor in the suction line, acts on the diaphragm in the opening direction. In equilibrium this force is exactly balanced by the evaporator pressure, P2, and the superheat spring pressure, P3. When the temperature of the refrigerant in the evaporator outlet increases due to superheating, P1 increases and moves to open the valve allowing an increase in refrigerant flowrate. This consequently raises P2 and results in a closing movement of the valve. Likewise, a decrease in the evaporator outlet temperature will decrease P1 and will cause the valve to move in a closing direction and as a result reduce the refrigerant flowrate.

The expansion valve's mechanism is also capable of handling load fluctuations. An increase in the evaporator load will result in an increase in the refrigerant superheat in the suction line. Sensing this the remote bulb will then exert a higher P1 on the valve's diaphragm to increase refrigerant flowrate. This action will in turn increase P2. Precision flowrate control is required to maximize evaporator capacity at all load levels and minimize liquid exiting the evaporator.

The thermal expansion valve has its drawbacks as well. As the remote bulb senses a temperature change it calls for the increase or decrease in refrigerant flowrate. However, the bulb does not respond to sense any change in the flowrate until the refrigerant completes its journey through the evaporator. The time delay that ensues causes a continuous fluctuation in the flowrate (which produces fluctuations in temperature and pressure of the refrigerant), a phenomenon known as "hunting", as the controls attempt to establish an equilibrium condition. To minimize this problem oversized valves should be avoided.

2.3.9 SOLAR EVAPORATOR ARRAY

An evaporator absorbs heat energy from a source by evaporating cool refrigerant liquid. The type of evaporator chosen is dictated by the kind of heat source used. A ground source unit, for example, may employ the use of pipes in the ground as an evaporator while an air source unit may have fins on its evaporator surface. Evaporators must be designed to provide balanced system performance and maximum efficiency during both heating and cooling cycles.

The heat pump unit concerned uses a solar array, as an evaporator, to collect energy from solar radiation and ambient air. The component consists of four flat plate collectors connected together to yield a total surface area of 8.4 square meters. It is made of aluminum and anodized black to increase absorptivity and decrease reflectance. The unit was mounted on an adjustable stand facing southward, placed adjacent to the south wall of the module, and angled at a 68 degree incline (as recommended by ASHRAE(1981)) from the horizontal.

Heat is absorbed from the incoming radiation falling on the evaporator's surface and from the surrounding ambient air during the day. At night the evaporator acts as a large flat plate to absorb heat from the air or moisture present on the surface. Frosting on the evaporator surface occurred several times during the heat pump's operation due to low surface temperatures but was not a significant problem.

2.3.10 ACCUMULATOR

Liquid refrigerant is sometimes present in the suction line. Since compressors are designed to compress vapor not liquid, it is essential to remove it from the compressor inlet line. Failure to do so would result in compressor breakdown because excessive liquid refrigerant dilutes the lubricating oil, washes out the bearings, and may cause a complete loss of oil in the compressor crankcase. This is achieved with an accumulator. An accumulator acts as a reservoir for the excess oil-refrigerant mixture and returns the fluid at a rate that the compressor can handle. The accumulator is situated before the compressor and the suction pressure regulator.

2.3.11 SUCTION PRESSURE REGULATOR

Excessive compressor inlet pressure is harmful to the compressor. High suction pressures result in gas at a higher density causing a higher mass per unit volume to enter the cylinder. The increase in mass means a higher work per stroke which may result in an overload of the compressor motor. Likewise, high starting torques result when the pressure difference across the compressor is large and can also produce motor failure. A suction pressure regulator is a device which, as its name suggests, regulates the pressure of the refrigerant entering the compressor to prevent these two situations from occurring. It is a spring-loaded valve with a seated disc. When the fluid pressure drops below the pressure exerted by the spring, the valve opens to allow for an increase in the refrigerant flowrate. Similarly, when the fluid pressure exceeds the spring pressure, the valve moves to decrease the flowrate.

2.4 THE HEATING CONTROL SYSTEM

As discussed earlier, heating for module six was provided by the heat pump and an auxiliary heater (electric resistance heater). The heat pump operated as long as heating was required. When the ambient temperature is such that the heat pump could no longer support the full heating load, the auxiliary heater would be turned on to provide the additional heat. The heating demands were monitored by two separate thermostats set at two different temperature levels. When the room temperature in module six dropped below the first thermostat setting the heat pump unit would be brought on-line. Similarly, when the room temperature dropped below the second thermostat setting the auxiliary heater would be turned on. Although the thermostat settings used during heat pump operation was not recorded the heat pump data strongly suggest that they were set at 22 and 21 °C for the heat pump and the auxiliary heater, respectively.

CHAPTER 3

DATA ACQUISITION AND PROCESSING

3.1 INTRODUCTION

In studying and analyzing the heat pump unit installed at the AHHRF a computer program was developed to handle and process the enormous amount of raw data. The program was used to convert recordings of temperature, pressure, etc., to thermodynamic quantities of enthalpy, entropy, and heat flowrate. The output of the program was then analyzed on a spreadsheet program. This chapter reviews the data processing program. In addition the data acquisition procedure will be discussed.

3.2 DATA PROCESSING PROGRAM

The program code was written in Fortran on the UNIX operating system. Fortran was chosen over other programming languages such as Basic and C because it contains highlevel input/output functions which is ideal for han[illegible]ng multiple large datafiles simultaneously. It was also a matter of convenience since the main functions of the program obtained from McMullen and Morgan (1981) were originally written in Fortran. The UNIX operating system was used because it provides a flexible and efficient environment for editing.

The program developed was used to calculate the state point properties of R-12 and the energy quantities in the system cycle given the temperatures and pressures. The overall performance parameters including the coefficient of performance of the heat pump were determined. Selected parameters were output in either hourly or daily formats to be analyzed in a spreadsheet environment or used for presentation purposes. The program also combines the energy consumption of module five with the calculated output parameters for comparison purposes. The following section will discuss the main functions used along with the program's input and output datafiles.

3.2.1 INPUT DATA

Two different forms of the raw heat pump data were available. One form presents the data in approximately four minute intervals. The other is the processed version of these four minute intervals to hourly averages. The latter form is the version used in this study due to the abundance of available raw data. There is one other set of data which indicates the electrical energy consumption of each module. Together these datafiles are the initial input data for the program developed.

(a) INPUT PARAMETERS

The input parameters are the input data for the data processing program. Figure 3.1 shows the information flow diagram. **INPUT 1** contains data taken from module six only. This data file contains raw data, in an hourly format, and was collected with a data acquisition system. **INPUT 2** contains the power consumption recordings of each of the six modules. This datafile is required in order to conduct a performance comparison between the heat pump unit in module six and the electric resistance heater in module five (which is the "standard" module as discussed in the previous chapter) and is also in an hourly format.

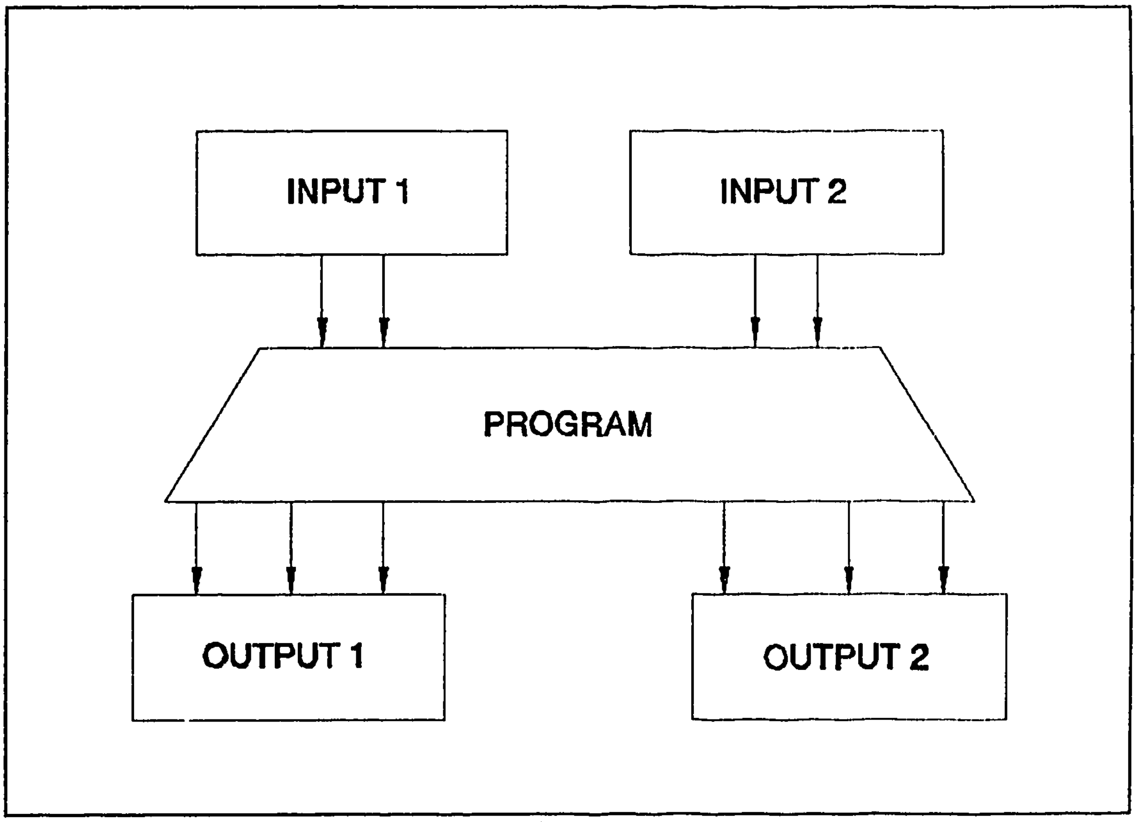

Figure 3.1 - Information Flow Diagram for the Data Processing Procedure

The parameters contained in the **INPUT 1** and **INPUT 2** datafiles are listed and described as follows. All parameters are based on the hourly format.

<u>**INPUT 1**</u>

(a) **date** - date (day, month, year)

(b) **time** - time (mountain standard time)

(c) **Tamb** - ambient temperature (°C)

(d) **Troom** - room temperature (°C)

(e) **Tcoil1** - upstream air temperature from condensing coil (°C)

(f) **Tcoil2** - downstream air temperature from condensing coil (°C)

(g) **T1, P1** - compressor outlet temperature (°C) and pressure (psig)

(h) **T2, P2** - condenser inlet temperature (°C) and pressure (psig)

(i) **T3, P3** - condenser outlet temperature (°C) and pressure (psig)

(j) **T4, P4** - T-X valve inlet temperature (°C) and pressure (psig)

(k) **T5** - T-X valve outlet temperature (°C)

(l) **T6, P6** - evaporator inlet temperature (°C) and pressure (psig)

(m) **T7, P7** - evaporator outlet temperature (°C) and pressure (psig)

(n) **T8, P8** - compressor inlet temperature (°C) and pressure (psig)

(o) **Qtot** - total electrical energy use of module six (W-hr)

(p) **Win** - heat pump electrical energy usage (W-hr)

(q) **Thp** - amount of time that heat pump is on (sec)

(r) **Rad** - radiation falling on the evaporator surface (W/m^2)

<u>INPUT 2</u>

(a) **date** - date (day, month, year)

(b) **time** - time (mountain standard time)

(c) **Qtot1,..,Qtot6** - total energy use of each module (W-hr)

(d) **Troom1,..,Troom6** - room temperature of each module (°C)

(e) **Tamb** - ambient temperature (°C)

The measured radiation energy falling on the evaporator surface is comprised of the direct, diffuse, and ground reflected solar radiation. A pyranometer was used to measure the solar radiation received on the evaporator surface. No readings were available for the net longwave radiation and therefore its effects were not considered.

The total energy use, **Qtot**, represents the total amount of electrical energy supplied to module six. This includes power for the heat pump compressor, the supplementary heating system, the data acquisition system, the lighting and the fans. Likewise, **Qtot5** is the total amount of electrical energy supplied to module five. However, this power is mainly consumed by the electrical resistance heater as discussed in the previous chapter. Note that **Qtot6** in **INPUT 2** is equivalent to **Qtot** in **INPUT 1.**

Win, the heat pump energy use, is the amount of electrical energy used by the heat pump to drive the compressor. The values were recorded with a meter leading solely to the heat pump compressor. The amount of time the heat pump was on, **Thp**, is a measure of the duration that the compressor was operating.

(b) AVAILABLE DATA

The heat pump system was installed in module six in the fall of 1984. It was test operated from January to June 1985 with limited recorded data. Full operation of the unit began in the fall of 1985 (September) and continued on till the fall of 1988. During this period, failures in equipment produced intervals with unusable data. Data for the electrical energy consumption of all modules was recorded on a continuous basis. To ease the task of organization, management, and analysis of the data and avoid the need for large output files, the two datafiles are broken down and grouped into months.

3.2.2 OUTPUT

From Figure 3.1 it can be seen that two different outputs, **OUTPUT 1** and **OUTPUT 2** are produced. These differ mainly in the time interval represented (either hourly or daily). Each also contain slightly different parameters. This section will be concerned with these output datafiles.

(a) OUTPUT 1

The output datafile **OUTPUT 1** is the most important output of the program and contains such parameters as the state point properties of the refrigerant throughout the cycle, the energy quantities in the system, and the system coefficient of performance. The output parameters are listed and described as follows.

(a) **date, time, rad, Tamb, Troom, Thp, Qtot, Win, Tcoil1,2, P_i, T_i**

- same as parameters in **INPUT 1**

(b) **Qtot5** - same as in **INPUT 2**

(c) h_i - enthalpy of the refrigerant at point "i" in the cycle (kJ/kg)

(d) s_i - entropy of the refrigerant at point "i" in the cycle (kJ/kg°C)

(e) v_i - specific volume of the refrigerant at point "i" in the cycle (m^3/kg)

(f) **X**$_i$ - quality of the refrigerant at point "i" in the cycle

(g) **Delh**$_i$ - (Δ h) change in enthalpy between points "i" and "i-1" (kJ/kg)

(h) **Irrev**$_i$ - irreversibility between points "i" and "i-1" (kJ/kg)

(i) **qout** - enthalpy change between points 1 and 4 (kJ/kg)

(j) **qin** - enthalpy change between points 8 and 1 (kJ/kg)

(k) **Qload** - energy required by the heated space (MJ)

(l) **COP** - coefficient of performance

(m) **SIrrev** - sum of the irreversibilities (kJ/kg)

(n) **FR12** - refrigerant flowrate (kg/s)

(o) **IHP** - heat pump on/off indicator (on=1, off=0)

These parameters are used primarily in the data analysis. The last parameter, **IHP**, indicates whether the heat pump was on or off. The conditions for on/off are presented in a following section. The output parameters produced in **OUTPUT 1** are, like **INPUT 1** and **INPUT 2**, on an hourly basis.

(b) OUTPUT 2

The second output datafile contains averages and sums of the hourly values of the input file and produces daily values of auxiliary energy, coefficient of performance, the heat pump heating output, etc. It is also used for presentation purposes in which case the results are tabulated in groups of months. This output datafile will be discussed in more detail in chapter five.

3.2.3 PROGRAM

In the past, equations were developed to generate thermodynamic tables of different refrigerants and describe their thermodynamic behaviour. Once the tables were established the tedious chore of calculating with equations was discontinued. However, with the advent of the computer, properties of a refrigerant along the various points of its cycle process can be easily and quickly determined using equations.

The developed program uses various equations to calculate the thermodynamic properties of refrigerant 12. The main equations used determine the refrigerant's:

(a) saturation pressure given temperature

(b) saturation temperature given saturation pressure

(c) saturation properties (entropy, specific volume, enthalpy) given saturation temperature

(d) specific volume, enthalpy, entropy of superheated refrigerant given temperature and pressure

(e) specific volume, enthalpy, entropy of subcooled liquid given temperature and pressure

Kartsounes and Erth (1971) successfully used the same equations and subroutines as the ones used in this investigation and found that they produce consistently accurate results. In particular, the calculated properties for refrigerant 12 were found to agree exactly with published tables in ASHRAE Handbook of Fundamentals (1981) (this is to be expected since these tables were generated using the same equations as presented here). Work was also done by Schofield (1970) to produce a computer program for the calculation of thermodynamic properties of fluorocarbon refrigerants with these equations. For more information regarding the data processing procedure, these equations and a listing of the main subroutines are presented in the appendix. Also presented is a list of the constants used in the program for refrigerant R-12.

After determining the various state point properties the program is used to calculate other significant parameters such as the refrigerant flowrate, the heating output of the heat pump, the energy gained from the compressor, the energy required by the auxiliary heating system and the lighting and measurement equipment in the module (for convenience this term will be referred to as the auxiliary energy from here on), the heating load of the heated space (module six), and the coefficient of performance of the heat pump. The refrigerant flowrate is calculated by performing a heat balance across the condenser and is a function of the following.

$$FR12 = f\,(h2, h3, Tcoil1, Tcoil2, Troom) \tag{3.1}$$

The parameters are the inlet and outlet enthalpies (points 2 and 3 in the cycle respectively), the upstream (**Tcoil1**) and downstream (**Tcoil2**) coil temperatures from the condenser, and the room temperature. The various state points are shown in Figure 3.2. For the heat balance across the condenser the value of the air flowrate through the condenser is required and was obtained from a 1986 student report. Using an anubar placed downstream of the system's air intake, the students found the air flowrate across the condenser to be 0.212 m^3/s (450 cfm). The details of the calculation are presented in the appendix.

The enthalpy drop of the refrigerant as it travels from the compressor to the

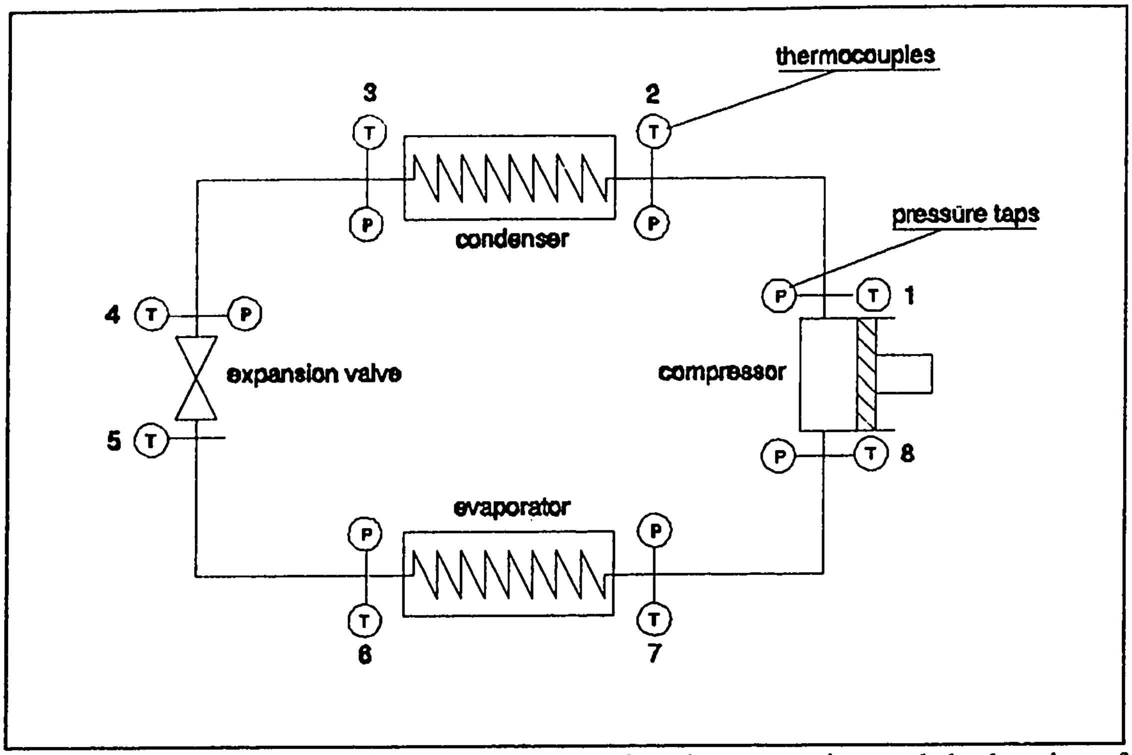

Figure 3.2 - Schematic of the Heat Pump showing the state points and the location of Thermocouples and Pressure Taps

thermal expansion valve, **qout**, is calculated as:

$$qout = h4 - h1 \ (kJ/kg) \tag{3.2}$$

Undoubtedly the bulk of the heat transfer process occurs at the condenser unit. However, since the lengths of pipe between the compressor and the condenser and between the condenser and the thermal expansion valve run within the module, the heat loss from these sections are heat gains to the heated space and are included in the calculation of the heating output. The amount of enthalpy change contributed to the refrigerant from the compression process is calculated as:

$$qin = h1 - h8 \ (kJ/kg) \tag{3.3}$$

where points 1 and 8 are the compressor inlet and outlet respectively. To calculate the hourly heating output of the system (**Qout**) and the hourly amount of energy added to the system in the compressor (**Qin**) in kJ respectively, **qout** and **qin** are multiplied by the refrigerant flowrate and the number of seconds each hour (3600):

$$Qout = qout \times FR12 \times 3600 \tag{3.4}$$

$$Qin = qin \times FR12 \times 3600 \tag{3.5}$$

Assuming that the thermostat in the module and the heating control system were functioning correctly, the heating load of the module, **Qload**, is the sum of the heat pump's heating output energy, the auxiliary energy, and the compressor heat loss (Figure 3.3 illustrates the energy flows in the module and its heating equipment):

$$Qload = Qout + Qaux + Qloss \tag{3.6}$$

The electrical energy to the compressor was assumed to translate into the energy gained, **Qin**, plus the compressor heat loss, **Qloss**. Therefore,

$$Qloss = Win - Qin \tag{3.7}$$

Qloss is included because the compressor unit is situated inside the module and any losses occurring at the unit will contribute to space heating. **Win** is, as stated before, the portion of the total electrical energy input to the module that is used solely to drive the compressor. **Qaux**, the auxiliary energy, is simply the portion of the total electrical energy used in the module that is not used by the heat pump (compressor):

$$Qaux = Qtot - Win \tag{3.8}$$

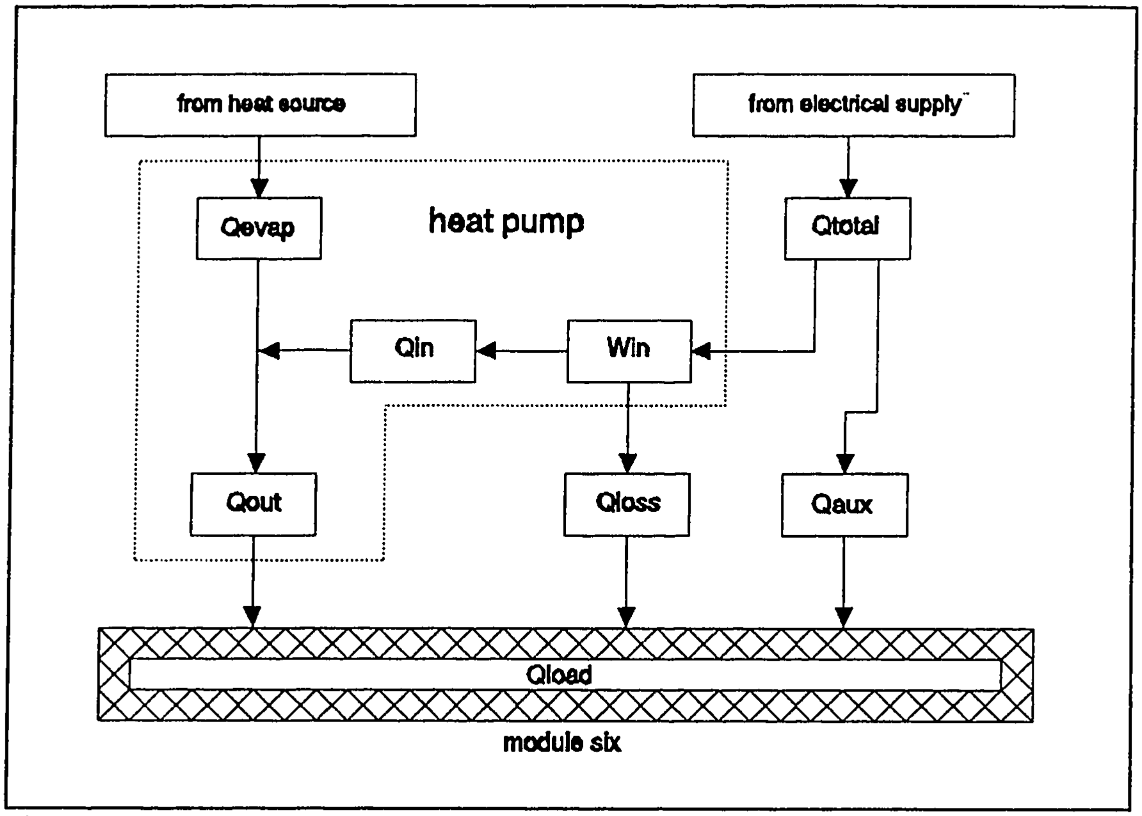

Figure 3.3 - Schematic of the Energy Flows in the System

Although the electric resistance heater is the only component in **Qaux** which is for the purpose of space heating, the entire term is included in the calculation of **Qload** because it is assumed that the other devices which utilize electricity such as the lights and the measurement equipment are inside the module and ultimately contribute to space heating.

The quantities **Qout** and **Win** are used to calculate the COP value of the heat pump:

$$COP = \frac{Qout}{Win} \tag{3.9}$$

The performance of a process or system can be expressed in terms of system irreversibilities. Irreversibility is the energy lost in a cycle due to non-isentropic processes such as friction (due to fluid flow along pipes) and heat transfers across a finite temperature difference. The system irreversibilities (**Irrev**) are calculated for the four main components (compressor, condenser, thermal expansion valve, evaporator) and for the lines connecting them. For the portion of the refrigerant circuit which is inside the module the irreversibility is:

$$Irrev_{i,i-1} = Tamb \times (\Delta(s)_{i,i-1} - \frac{\Delta(h)_{i,i-1}}{Troom}) \tag{3.10}$$

Here $\Delta(s)_{i,i-1}$ and $\Delta(h)_{i,i-1}$ are the change in entropy and enthalpy from point (i-1) to point (i) respectively. Equation 3.10 is valid for the condenser and the thermal expansion valve and the lines from the compressor to the condenser and from the condenser to the thermal expansion valve (i = 2,3,4,5 in Figure 3.2). For the compressor the irreversibility is calculated as:

$$Irrev_{comp} = Tamb \times [s(1) - s(8)] \tag{3.11}$$

For the suction line running from the evaporator to the compressor the irreversibility is calculated as:

$$Irrev_{8,7} = Tamb \times (\Delta(s)_{8,7} - \frac{\Delta(h)_{8,7}}{Tamb}) \tag{3.12}$$

After the refrigerant leaves the thermal expansion valve it may be unstable as it expands and might not be at equilibrium. Since the line from the thermal expansion valve to the evaporator is relatively short the evaporator inlet measurements of the temperature and pressure are suspect. Because of this uncertainty, the calculations for the state point properties of the evaporator inlet are not considered. This prevents the determination of the irreversibility for the line from the thermal expansion valve to the evaporator. To calculate the irreversibility of the evaporator the line from the thermal expansion valve to the evaporator will be assumed to be part of the evaporator. Therefore the irreversibility can be evaluated as:

$$Irrev_{evap} = Tamb \times (s(7) - s(5) - \frac{[h(7) - h(5)]}{Tamb}) \tag{3.13}$$

As discussed in the previous chapter, the heating systems in the module (the heat pump and the electric resistance heater) turn on and off in response to the thermostat setpoint. Calculations for the refrigerant flowrate, the heating load, the heating input and output energies, and the COP (i.e. all parameters which are concerned with the heat pump) are not performed if the heat pump is off. The device is considered off-line if the following conditions are true:

- **Thp** < 0 or **Thp** = 0, and **Qtot** > 0
- **Win** < 0 or **Win** = 0, and **Qtot** > 0

There are instances during the heat pump system's operation where the recording devices are either not working properly or are taken off-line for repairs. These conditions result in erroneous values recorded by the data acquisition system. Fortunately these instances can be identified with the data processing procedure. Readings are considered erroneous if:

- **X1** < 1, **X2** < 1, **X3** >0, **X4** > 0, **X7** < 1, **X8** < 1

- compressor work greater than total module energy use (**Win** > **Qtot**)

- **Qtot** < 0, **Qtot** = 0

These readings were not considered in the analysis.

Thp is the amount of time the heat pump is on, **Qtot** is the total electrical energy usage of module six, **Win** is the heat pump's electrical energy usage, and $\mathbf{X_i}$ are the qualities of the refrigerant at the various points in the cycle (negative qualities indicate subcooled while larger than 1 qualities indicate superheated). For these readings the indicator parameter **HP** is set to 0 to denote that the heat pump is either off for that hour or the data recorded is erroneous. Otherwise this parameter is assigned a value of 1. This parameter is for the user's convenience to indicate the instances where the heat pump is on and serves no other purpose.

Although all the functions discussed handle and process data on an hourly basis, it is also often beneficial to have results in a daily format (**OUTPUT 2**). Such a format could be used in monthly summaries to allow for the observation of trends in the operation of the heat pump over a longer period. The program performs a daily average or sum of the values of **OUTPUT 1** to yield this output file. If the heat pump is off for a certain hour but the data are not erroneous, only the heat pump parameters such as the refrigerant flowrate, the heat pump "on" time (**Thp**), the heat output of the heat pump, the heat input from the compressor, and the electrical energy required by the compressor are not included in their daily averages or sums. Ambient and room temperatures and the total electrical energy usage of the module are still valid. However, if the readings are erroneous all parameters including ambient temperature and room temperature were ignored.

3.3 DATA ACQUISITION

The data acquisition system for the heat pump and its module consisted of various measuring and recording devices. Temperature of the refrigerant fluid was measured using type 'T' thermocouples (copper-constantan) which were referenced to an ice water bath at 0 °C and had an accuracy of ±0.5 °C. These temperature devices were strategically placed throughout the refrigerant circuit (as shown in Figure 3.2). In addition, the upstream and downstream condenser coil temperatures, the ambient temperature and the room temperature were also monitored.

For pressure measurements two sets of pressure manifold systems each having one pressure transducer, were created. Figure 3.4 shows the schematic of one such system. Connected to a transducer, each manifold had four separate intake lines leading to four different locations in the refrigerant circuit. Each intake line had a sampling solenoid which was controlled by the system computer. During the data acquisition process the solenoid valves would be opened one at a time to allow the pressure from their respective lines to be measured. The pressure transducers used had an estimated accuracy of ±5 %.

Solar radiation measurements were recorded with an Eppley precision pyranometer. The device was placed adjacent to the evaporator collector plate (so as not to be obstructed by the collector itself) and had an accuracy of ±0.5 %. Measurements combined both the direct beam and diffuse components of solar radiation.

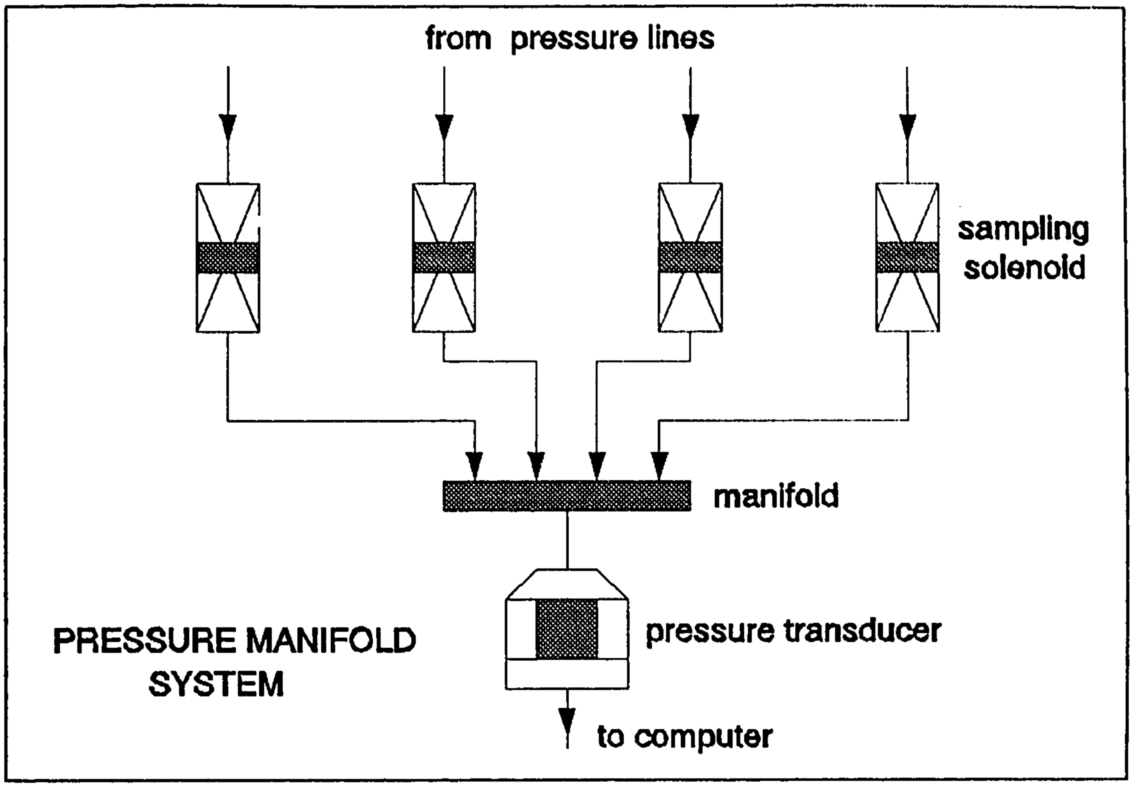

Figure 3.4 - The Pressure Manifold System

Electrical energy consumption was monitored for the module and the heat pump's compressor. These measurements were recorded with an accuracy of ± 1 %. The electrical energy consumption of module five was also available on a continuous basis.

All the measured data were collected with a data acquisition system which consisted of a HP85 computer with a 3490 HP scanner and the associated transducer and thermocouple controls. Data sampling was programmed to occur 14 times per hour (at 4 minute intervals). For the hourly set of acquired data the 14 data points were averaged. Also recorded were the total electrical energy consumption values of module five. These were recorded by Sangamo Watt-hour meters and are accurate to ± 1 %.

CHAPTER 4

PERFORMANCE ANALYSIS

4.1 INTRODUCTION

After being processed with the computer program developed the heat pump data were analyzed in more detail. The software package used to handle and analyze the various outputs of the program was the LOTUS 123 spreadsheet program. This section highlights the methods used in the analysis of the heat pump performance.

The study concentrates on various areas such as the energy consumption of the heat pump system and the efficiency of the solar-assisted evaporator. The total amount of heat pump data collected spans three years (September '85 to August '88). To simplify the analysis only the heating season of 1986-1987 (which presents the largest quantity of data of the three heating seasons), from September '86 to March '87, was considered. For cases where only a small amount of data is needed to illustrate a point, data from the month of January '87 was used since it is fully representative of the heat pump performance due to the extreme conditions present in that month.

4.2 SYSTEM ENERGY CONSUMPTION

One of the parameters obtained from the program developed is the measure of the amount of electrical energy consumed by module five ("QT5"). Since the module was maintained to represent an uninhabited house, all the energy consumed by the module is assumed to be dissipated as heating energy for the building. It was assumed that any energy used for interior lighting ultimately contributes to space heating.

The total energy used by module six, **Qtot**, is considered to be consumed by the heat pump compressor and the auxiliary heating system (which includes an electric resistance heater):

$$Qtot = Win + Qaux \tag{4.1}$$

The auxiliary energy, **Qaux**, is assumed to contribute to building heating with a conversion efficiency of 100%.

As mentioned earlier module five is very similar in design to module six and should be ideal as a standard for comparison purposes. Justification for the validity of the comparison between the two modules is further investigated in the following section by examining the module overall loss coefficient (UA) factors.

4.2.1 THE MODULE LOSS COEFFICIENT FACTOR

In a study by Gilpin et al. (1980) the average overall heat loss coefficient or "UA" factor of module five was found to be 139 W/ °C. Since the overall UA factor is an indication of the amount of heat dissipated from a module, the validity of a comparison between modules five and six would be assured if the latter bears an equivalent UA factor to that of the former. The UA factor of the module can be determined by examining the cumulative heating load of the building as a function of the cumulative heating degree hours. Heating degree hours are calculated as the temperature of the heated space minus the temperature of the ambient air for each hour (in °C) integrated over time. Since the building was unoccupied, the value of the indoor thermostat-setting, 21 °C, will be the value used for the temperature of the heated space.

Figure 4.1 illustrates the effect of the cumulative heating degree hours on the cumulative heating load of module six for the month of January '87. The heating load of the building (**Qload**), is the sum of the auxiliary heat output, the heat pump heating output, and the heat loss occurring at the compressor unit. The figure is represented by the following equation:

$$Qload = UA \times heating\ degree\ hours \tag{4.2}$$

The slope of the resulting curve would be the average UA factor of the module. Figure 4.2 shows the slope of the previous figure plotted against the cumulative heating degree hours. Ideally the resulting graph would be a straight horizontal line (a constant value of UA). The deviation from this ideal is due to factors such as solar energy gains during the day, shifting infiltration rates, and the time lag of the basement heat losses with respect to ambient temperature.

To obtain the UA factor for the month the last 100 UA factor data points were averaged. Table 4.1 shows the monthly overall UA factors for modules five and six for the heating season of '86-'87 (September - March). Along with these parameters is

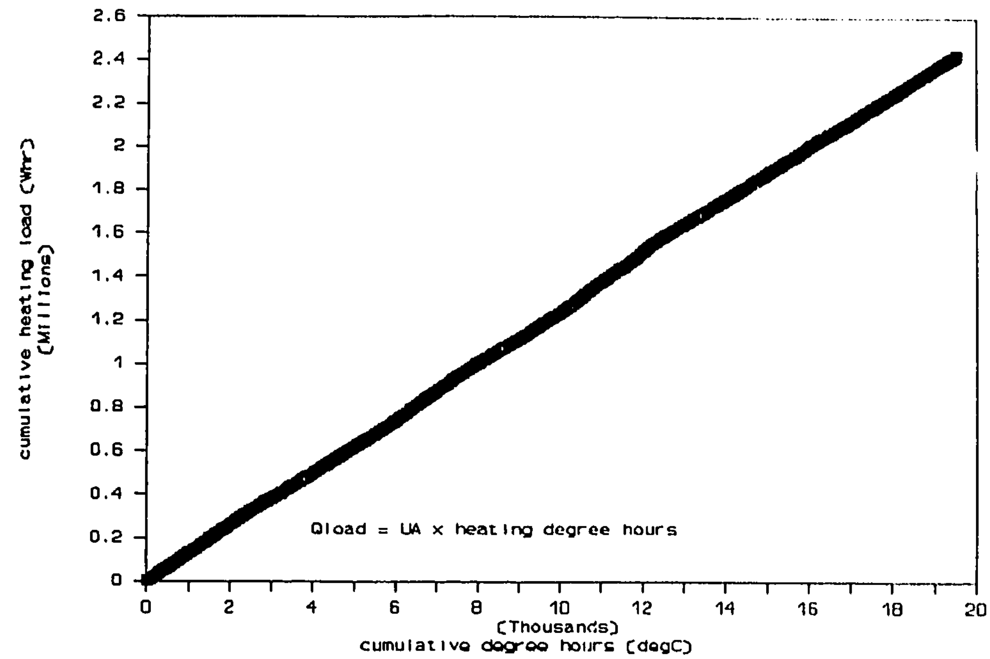

Figure 4.1 - Determination of the Overall UA Factor for Module Six for the month of January '87

shown the percentage difference module five's UA factors are from module six's.

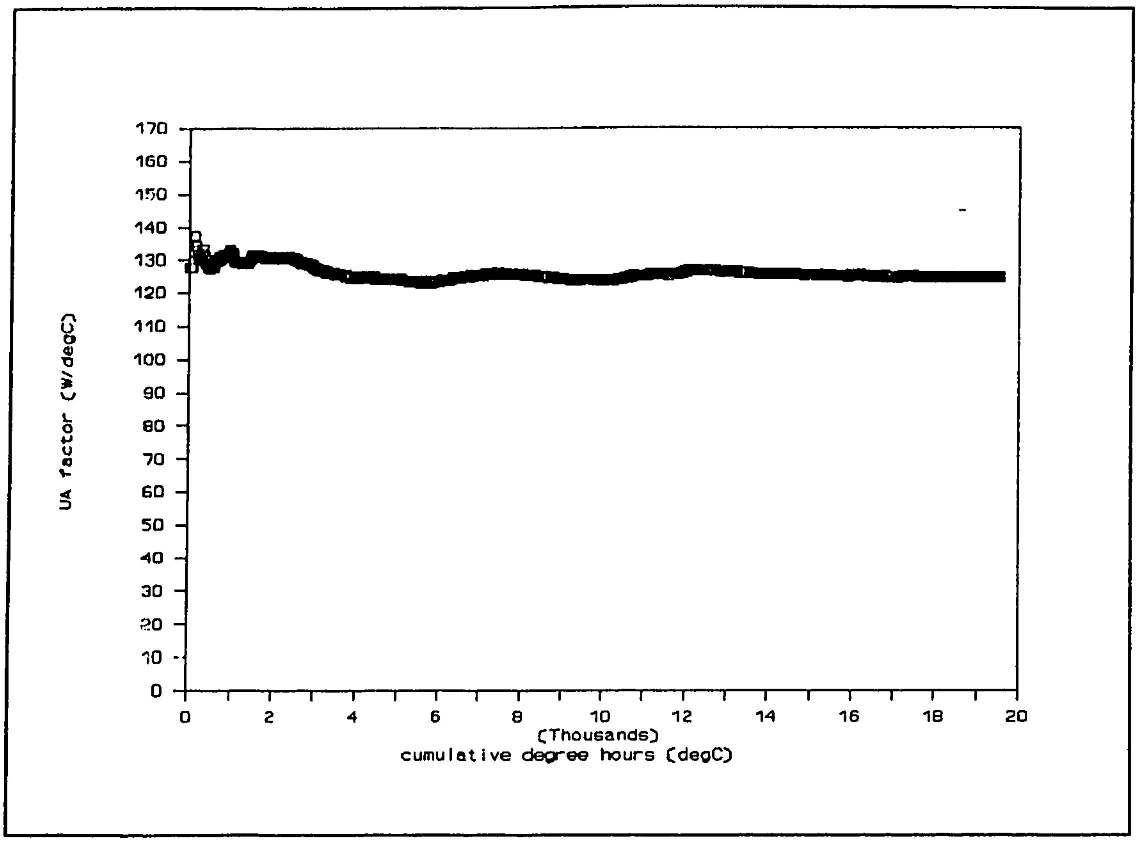

Figure 4.2 - Overall UA Factor for Module Six for the month of January '87

Table 4.1 - UA Factors for Modules Six and Five

	UA6	UA5	%diff
	W/°C	W/°C	
Sep'86	151	135	10.9
Oct'86	148	134	9.6
Nov'86	117	113	3.4
Dec'86	120	116	2.9
Jan'87	124	110	10.9
Feb'87	127	112	11.6
Mar'87	124	108	12.7

The monthly UA values determined for module six and module five range from 117 to 152 W/°C and from 108 to 135 W/°C respectively. Throughout the entire year the changes in the UA factors are quite similar. Differences in the two sets of UA values are not very high acknowledging that a comparison between the two modules is valid. The difference between the values is due to the difference in ceiling insulation levels (see Table 2.2). The UA range for module six agrees with the value of 139 W/°C for the UA factor for module five as found by Gilpin et al. (1980). It should also be noted that the UA factors for module six determined here agree with the value of 143 W/°C found by Gilpin et al. (1980).

4.3 EVAPORATOR COLLECTOR PERFORMANCE

Another area of analysis to be considered is the evaluation of the solar evaporator performance. As mentioned before, the heat pump unit utilizes a solar collector as its evaporator unit. Theoretically, it is expected that during the day the collector plate temperature will rise as a result of solar radiation and will, as a consequence, increase the temperature of (the heat input to) the refrigerant. To evaluate the collector performance the efficiency of the collector plate (its ability to utilize the low temperature heat energy present) and the benefits gained in the heat pump unit performance (if any) with its addition should be determined. In addition the collector surface area should be investigated to determine if it is over or under-sized for the application. This section outlines the methods used and the assumptions made to determine the collector plate efficiency based on collected performance data.

4.3.1 DETERMINATION OF THE COLLECTOR PLATE EFFICIENCY

The performance of collectors can usually be evaluated using the method originated by Hottel and Woertz (Hottel and Woertz (1942)) and also presented by several other investigators (Bliss (1959), Whillier (1967), Duffie et al.(1980)). However, for a flat plate solar collector undergoing phase change (Soin et al.(1979)), the basic equation is given as:

$$
\begin{aligned}
q_u &= I_{t\theta} \times (\tau \times \alpha)_\theta - U_L \times (T_p - T_{al}) \\
&= FR12\ \left(C_L(T_s - T_i) + lambda + C_V(T_o - T_s)\right)
\end{aligned}
\qquad (4.3)
$$

where

q_u = heat usefully gained by collector (W/m^2)

$I_{t(\theta)}$ = total irradiation of collector (W/m^2)

$(\tau,\alpha)_\theta$ = transmittance of cover, absorptance of plate at prevailing incident angle, θ

T_p, T_{at} = temperatures of the absorber plate and the atmosphere, (°C)

FR12 = fluid flow rate, (kg/s)

T_o,T_i = temperatures of the fluid leaving and entering the collector (°C)

C_p = specific heat (J/kg °C)

lambda = latent heat of vaporisation (J/kg)

T_s = saturation temperature (°C)

The evaporator collector efficiency is defined as the ratio of the amount of energy absorbed by the collector plate to the amount of solar radiation striking the collector plate:

$$\eta = \frac{q_u}{I_{t0}} = \frac{F'(\tau\alpha)_e}{L'} - \frac{F' U_L (T_f - T_{at})}{L' Rad} \tag{4.4}$$

where:

F' = collector efficiency factor

$(\tau.\alpha)$ = transmittance-absorptance product for the plate, and

Rad = total solar irradiation incident on the plate (W/m^2).

L' = fraction liquid level (dimensionless)

U_L = collector overall heat loss coefficient (W/m^2 °K)

However, the collector plate under investigation is different from the ones which equations 4.3 and 4.4 are derived for in that it also absorbs heat energy available in the ambient air in addition to solar energy.

Heat energy is delivered from the solar radiation and from the ambient air to the evaporator plate by the processes of radiation and convection heat transfers, respectively. Figure 4.3 illustrates the process occurring at the evaporator with the solar radiation energy, **Qrad** and the convection heat transfer energy, **Qconv**. **Qevap** is the evaporator input energy (the resulting heat energy entering the refrigerant at the evaporator (and therefore the heat pump unit)). **Qrad** consists of short and longwave radiation. Note that the convective heat transfer term is shown to be pointing in both inward and outward directions. When the temperature of the collector plate increases above the ambient temperature (such as on a sunny day) there will be a convection heat loss from the plate to the surrounding air (in which case **Qconv** would have a negative sign). **Qemit** represents the heat losses from the plate by radiation heat transfer and by conduction heat transfer through the plate's supports and is illustrated by an outward arrow. **Qref** is the reflected solar radiation from the plate. With these definitions the heat transfer process at the evaporator plate can be written as:

$$Qevap = Qrad + Qconv - Qref - Qemit \tag{4.5}$$

Here the conditions of the surroundings determine whether **Qconv** will be a loss or a gain. To simplify the analysis the heat losses by conduction heat transfer through the plate supports are considered negligible. In addition the emitted longwave radiation is assumed to be approximately equal to the incoming longwave radiation. The net longwave radiation would be relatively small and would have an insignificant effect on the heat transfer process at the plate. In their analysis, Bliss et al.(1959) assumed that the longwave emissivity and absorptivity could be considered equal. With these simplifications the **Qemit** term in equation 4.5 can be neglected and the radiation term

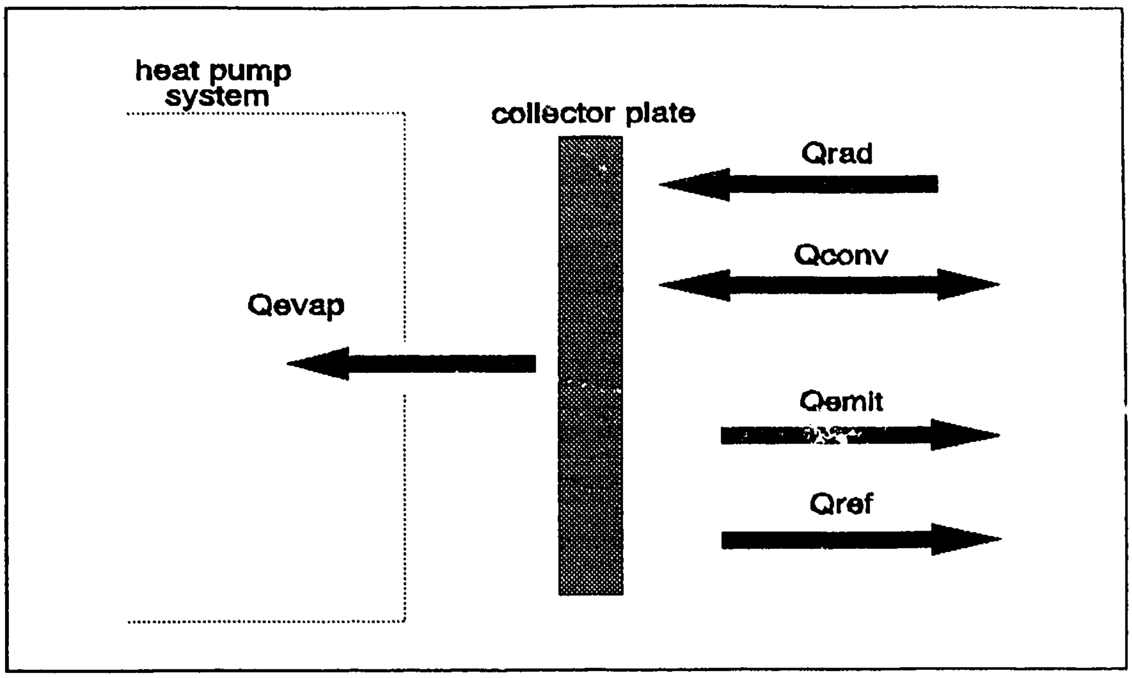

Figure 4.3 - Energy Flow at the Collector Plate

(Qrad) is equal to the solar radiation value as measured by the Eppley optical pyranometer (see section 3.3). The resulting equation then is:

$$Qevap = Qrad + Qconv - Qref \tag{4.6}$$

The total amount of heat energy absorbed by the system is actually more than just the quantity at the evaporator (Qevap). Between the T-X valve and the evaporator and between the evaporator and the heat pump's compressor (i.e. between points 5 and 6 and points 7 and 8 in Figure 4.4) , energy can be absorbed or lost through the pipes. The total absorbed, **Qabs**, is therefore:

$$Qabs = Qevap + Q_{87} + Q_{65} \tag{4.7}$$

The parameters **Q87** and **Q65** are the energy quantities gained or lost between the points

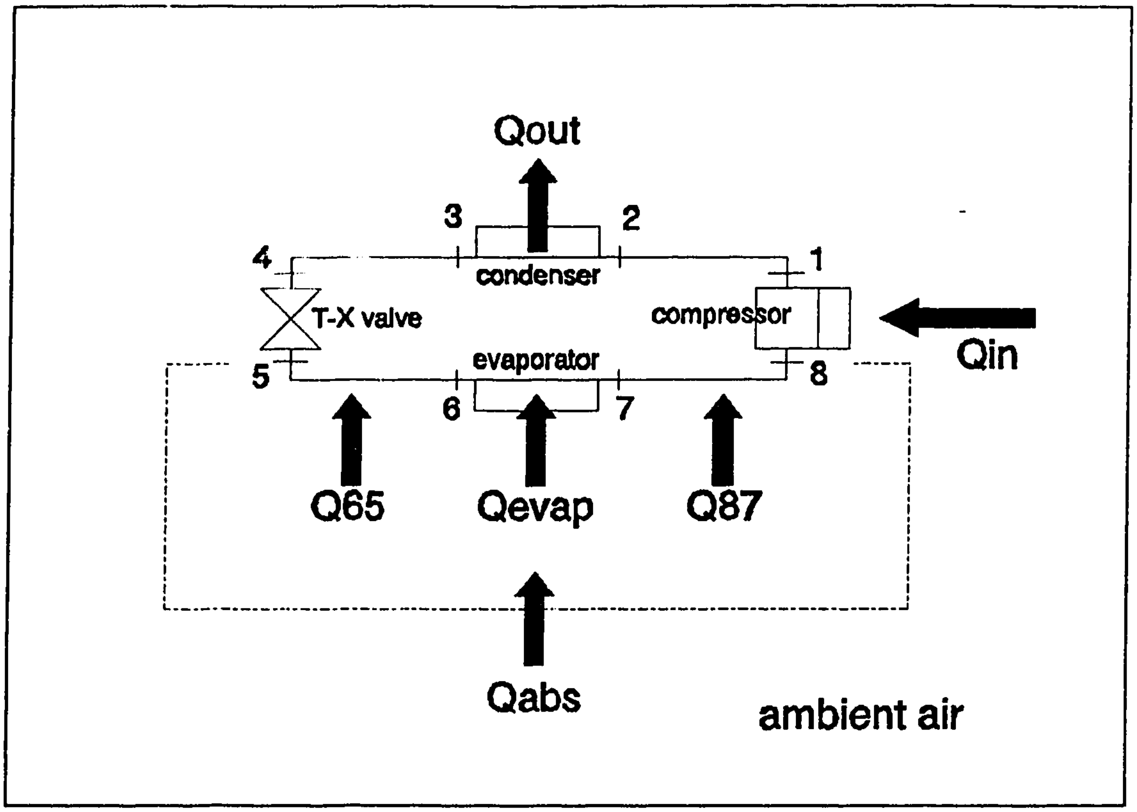

Figure 4.4 - Heat Input to the Evaporator

8 and 7 and 6 and 5 respectively (and can be either positive or negative) and **Qevap** is as defined earlier. These three parameters are functions of the state point enthalpies and the refrigerant mass flowrate (**FR12**):

$$Q_{ij} = FR12 \times (h_i - h_j) \tag{4.8}$$

$$Qevap = FR12 \times (h_7 - h_6) \tag{4.9}$$

However, because there is insufficient information to determine the enthalpy at point 6 (see section 3.2.3) an assumption has to be made to simplify the analysis. It was decided that the liquid line (from T-X valve to evaporator) would be considered as part of the evaporator. The **Qevap** parameter was therefore calculated as a function of the change in enthalpy between points 5 and 7.

In equation 4.6 the parameter **Qrad** is the total heat energy from the solar radiation incident on the collector plate. The parameter is defined simply as:

$$Qrad = Rad \times A \qquad (4.10)$$

where **A**, the surface area of the collector plate, is 8.4 m^2 on each side. The heat pump collector plate was mounted onto another collector unit, an active air solar collector, (about 20 cm above it) which was in operation prior to the installation of the heat pump. This was done as a matter of convenience. Although one side is shielded from the ground the 20 cm opening between the collector plate and its base allows some ground reflected solar radiation to strike the underside. However due to the limited opening area and because the plate sits at a 68 degree incline to the horizontal the amount of reflected solar radiation received by the underside of the plate is small and considered negligible. Therefore, since only one side of the collector plate is exposed to the sun's radiation the value of 8.4 m^2 was used in determining the total solar radiation energy impacting the plate (**Qrad**).

Figure 4.5 illustrates the effect of solar radiation on the difference between the average evaporator fluid temperature and the ambient temperature for the month of January '87. It appears that during the day, at solar radiation values of approximately more than 200 W/m^2, the average evaporator fluid temperature exceeds the ambient air temperature resulting in a convection heat loss (instead of a convection heat gain) from the plate to the surrounding air. The average evaporator fluid temperature is defined as:

$$Tave = \frac{(T_6 + T_7)}{2} \qquad (4.11)$$

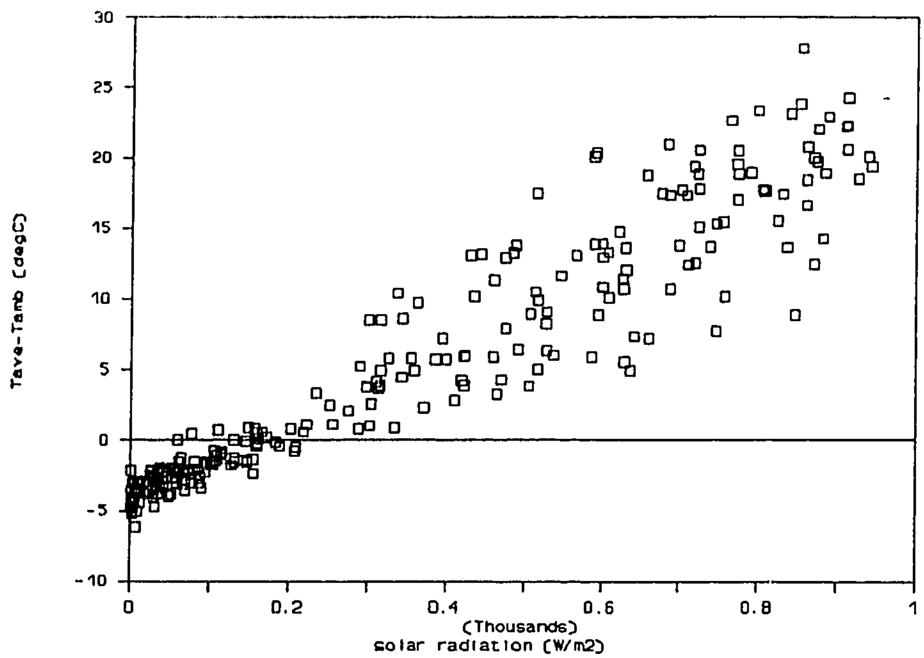

Figure 4.5 - Temperature rise of the Evaporator Fluid above the Ambient Temperature as a function of Solar Radiation for the month of January '87

Therefore, the evaporator plate is subjected to three different conditions each day:

- night
- day (solar radiation less than or equal to approx. 200 W/m^2)
- day (solar radiation more than approx. 200 W/m^2)

During the night the evaporator collects heat from the ambient air only. During the day when the average evaporator fluid temperature is less than the ambient temperature (at solar radiation of 200 W/m^2 or less) the evaporator plate absorbs heat from the ambient air as well as from solar radiation. While equation 4.6 is still applicable, since the plate temperature was not recorded, evaluation of the efficiency of the plate for this region is difficult or suspect at best.

However, when the average evaporator fluid temperature is higher than the ambient temperature (at solar radiation of more than 200 W/m^2), all the absorbed heat energy at the evaporator must come from solar radiation since convection heat transfer cannot occur from the air to the refrigerant (i.e. no heat gain by convection). For this region, with the assumptions discussed, equation 4.6 becomes:

$$Qevap = Qrad - Qconv - Qref \tag{4.12}$$

But since this is a similar case to the one derived by Soin et al., equation 4.4 will be used to approximate the efficiency of the plate for this solar radiation range. The results will be presented and discussed in the following chapter. The next section investigates the night-time case.

4.3.2 DETERMINATION OF THE NIGHT-TIME PLATE CONVECTION HEAT TRANSFER

As discussed earlier, during the night the only apparent heat transfer at the evaporator occurs through convection from the ambient air to the refrigerant in the evaporator. There may be some radiation from the plate to the night sky but since the amount was not measured it will not be considered. Because the average fluid temperature in the evaporator plate will be lower than the ambient temperature there will not be a convection heat loss from the plate to the air. For the night-time, equation 4.6 becomes:

It can be seen that the efficiency of the plate in collecting heat energy from the ambient

$$Qevap = Qconv \tag{4.13}$$

air during the night (i.e. in the absence of solar radiation) would be 1. For this case it is more useful to evaluate the convection heat transfer ability of the collector plate. Convection is a function of the ambient temperature, the plate temperature, and the overall heat transfer coefficient of the plate:

$$Qconv_{gain} = hplate \times \Delta T = hplate \times (Tamb - Tplate) \tag{4.14}$$

The ΔT term is the temperature difference between the ambient temperature (**Tamb**) and the plate temperature (**Tplate**). In the absence of solar radiation the plate temperature can be approximated by the average evaporator fluid temperature (as defined by equation 4.11). The **hplate** parameter is the overall heat transfer coefficient of the plate (which is similar to the UA factor discussed earlier) and includes the convection surface area of the plate. Bliss et al.(1959) used the overall heat transfer coefficient in calculating the convection heat transfer loss in their investigation of the performance of a similar flat plate collector. Combining equations 4.13 and 4.14:

$$Qevap = hplate \times \Delta T \tag{4.15}$$

Using only the night-time data and rearranging the above equation, the convective heat transfer coefficient-area product can be determined:

$$hplate = \frac{cumulative\ Qevap}{cumulative(Tamb - Tave)}\ (night) \tag{4.16}$$

Figure 4.6 shows a plot of the cumulative evaporator input energy (**Qevap**) for the night-time against the cumulative temperature difference between the ambient temperature and the collector plate temperature for the month of January '87. The linearity of the curve indicates a constant overall heat transfer coefficient.

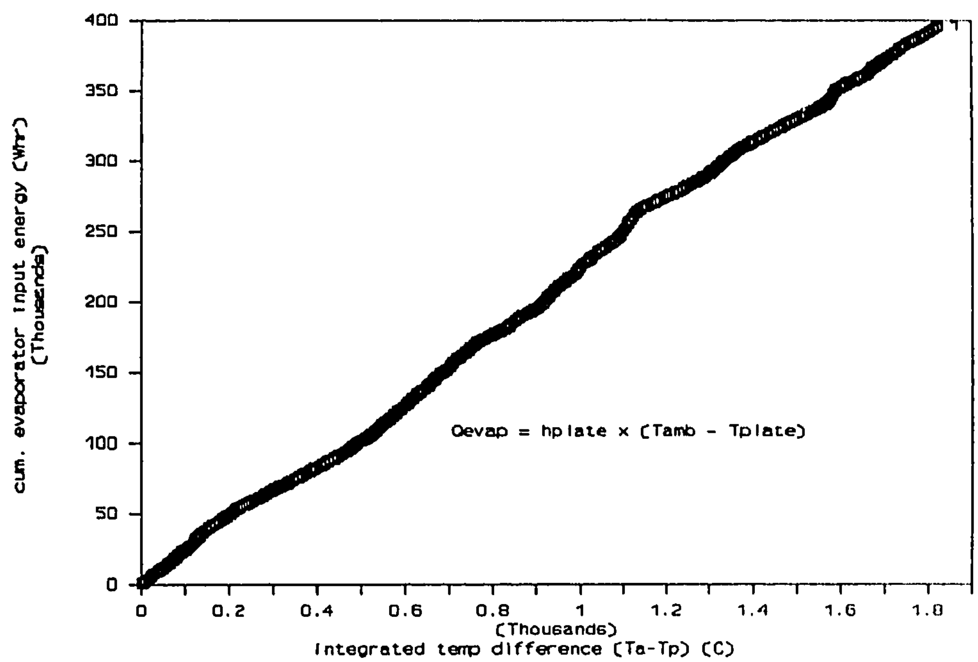

Figure 4.6 - The Cumulative Evaporator Input Energy (Qevap) plotted against the cumulative Temperature Difference between the Ambient Air and the Collector Plate for the Nighttime Periods of January '87

Figure 4.7 shows the value of **hplate** for the month of January '87. Averaging the last one hundred values on the graph yields an average convective coefficient area product value of 219 W/°C. Repeating this method the overall evaporator collector plate heat transfer coefficients can be determined for the other months. Table 4.2 shows the values determined for the heating season of 1986-'87 (from September to March). **Tamb** is the average ambient temperature for the month. It should be noted that the heat transfer coefficients are also influenced by wind speed and direction. It would seem that

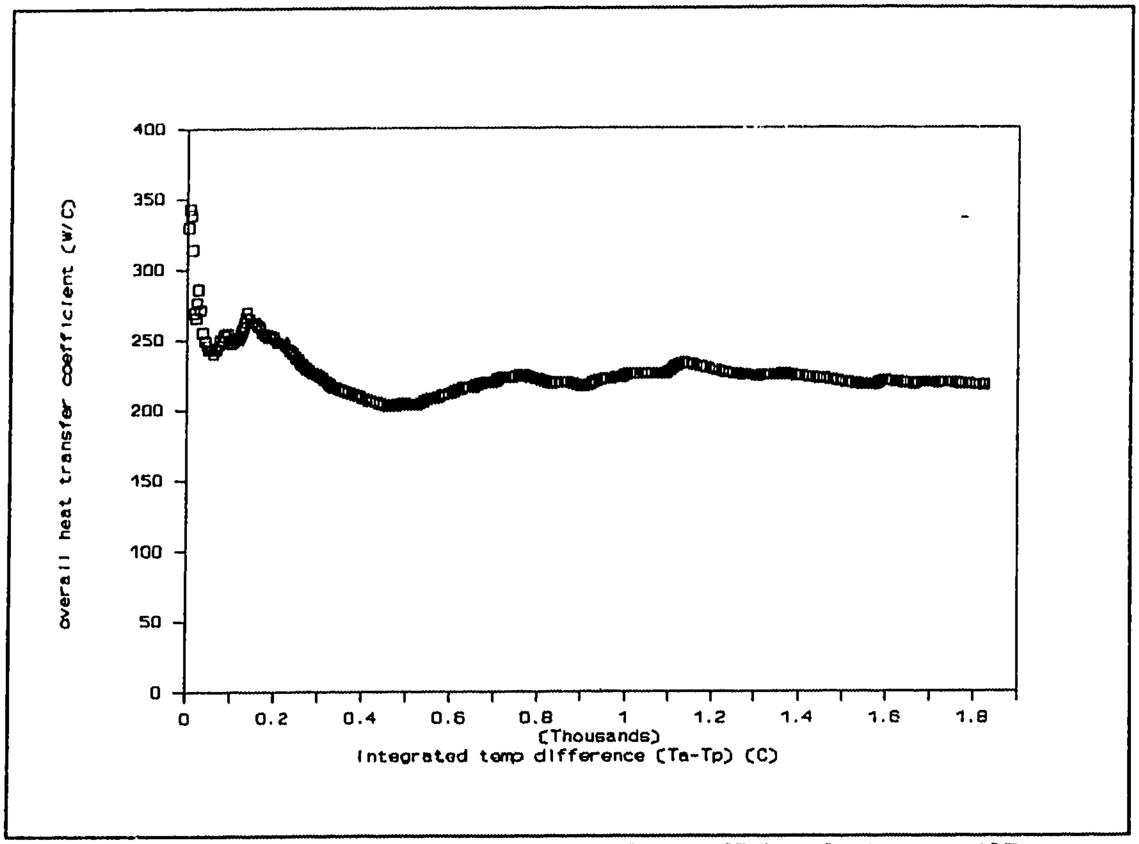

Figure 4.7 - Collector Plate Overall Heat Transfer Coefficient for January '87

on average the heat transfer coefficient of the collector is about 230 W/°C.

Table 4.2 - The Plate Convection Heat Transfer Coefficients for The '86-'87 Heating Season

	hplate	Tamb
	W/°C	°C
Sept'86	208	4.1
Oct'86	239	1.5
Nov'86	245	-7.0
Dec'86	220	-5.8
Jan'87	219	-6.3
Feb'87	218	-5.4
Mar'87	255	-5.5

4.4 AVERAGE HEAT PUMP COEFFICIENT OF PERFORMANCE (COP)

In Chapter 3 the coefficient of performance (COP) of the heat pump was defined as the ratio of the heating output of the device to the work input to the heat pump compressor. This value over the study period of September '86 to March '87 can be calculated by using the totals of the heating output (**Qout**) and the work input to the compressor (**Win**):

$$COP=\frac{\sum Qout}{\sum Win} \tag{4.17}$$

The average COP was found to be 2.0.

Since the heat pump unit under study was solar-assisted it would be helpful to be able to compare it to a heat pump that does not use solar radiation as a heat source. Assuming that the evaporator characteristics of the solar-assisted evaporator and an ordinary heat exchanger evaporator are the same, the COP of a heat pump without solar-assistance can be approximated by the COP of the solar-assisted heat pump during the night-time. With this value an economic analysis can be performed to compare the heating costs of the solar-assisted heat pump to an ordinary heat pump which utilizes a standard heat exchanger and fan as an evaporator.

CHAPTER 5

DISCUSSION

5.1 INTRODUCTION

In the last chapter some of the methods used in the process of evaluating the heat pump characteristics and performance were discussed. This chapter examines the results of those methods and determines the efficiency of the heat pump unit. Some significant issues include the effects of solar radiation and ambient temperature, the coefficient of performance, the energy consumption (along with its auxiliary heating unit), the efficiency of the solar evaporator unit, and the economics of the heat pump in comparison to other heating methods. In addition the format, and the justification of using the format, of the data presented will be discussed.

For the performance evaluation of the heat pump unit the analysis will be based on data collected during the 1986-87 heating season (September '86 - March '87). This particular period was selected because it contains the most continuous and complete data of the three heating seasons ('85-'86, '86-'87, '87-'88). The hourly data format was used since it presents a better picture of the characteristics of the heat pump than the daily data set. However, since the transient response of the system was not evaluated, the 4 minute interval data set was not used. It should also be noted that for the Sept'86 - March'87 period all data which were valid (not "erroneous" as defined in chapter 3) had the heat pump on-line. There were no instances where the heat pump was off-line for a full hour.

5.2 PERFORMANCE EVALUATION OF THE HEAT PUMP

The performance of the heat pump unit is determined by the coefficient of performance (**COP**) of the unit. As defined in section 3.2.3 the COP is a function of the heating energy output (**Qout**) and the electrical energy input to the compressor (**Win**). The heating energy output in turn depends on the amount of energy absorbed from the heat source and is therefore a function of the ambient temperature. Since the system employs a solar collector plate as the evaporator the amount of solar radiation will also affect the performance. It is appropriate to investigate the effects of these two parameters, the solar radiation and the ambient temperature, on the performance of the heat pump unit. The performance of the unit will also be compared to the heat pump manufacturer's specifications.

5.2.1 EFFECT OF SOLAR RADIATION ON PERFORMANCE

The unit, being solar-assisted, naturally utilizes solar radiation to supply part of the heat source. Throughout the day (and the year) the heat source will vary with the change in the angle of elevation of the sun. A higher angle of elevation will of course produce a higher amount of solar radiation on a horizontal surface. Ironically the time of year which requires the least amount of heating, summer, is when solar radiation is most abundant. Likewise the time of year when heating is most important, winter, yields the least number of hours of sunshine at a low angle of elevation in the sky (this was shown in Table 2.1). The evaporator collector is angled at 68 degrees (latitude + 15 degrees) and is faced southward. The collector is ground-mounted adjacent to the south wall of module six and has an unobstructed view of the sky.

The effect of varying solar radiation on the performance of the heat pump unit is investigated. Figure 5.1 shows the effect of solar radiation on the amount of time the heat pump was on per hour (**Thp**) for the '86-'87 heating season. There is little correlation between the two parameters with the majority of data at the full hour mark (3600 seconds per hour) throughout the solar radiation range. The fact that there are operation times below 3600 seconds however suggests the presence of system 'cycling' (where the heat pump turns on and off as the room temperature hovers near the control system's thermostat set point on warm days) for the period of study.

Figure 5.2 shows the effect of solar radiation (**Rad** - the solar radiation as measured at the evaporator inclination) on the heating output (**Qout**) and the heating energy contribution from the compressor (**Qin**). Heating output increases with increasing solar radiation. At around 600 W/m^2, heat output reaches a maximum of approxin 10 MJ (total for the whole hour) while the heating contribution from the compressor is about 3 MJ. Beyond this value of solar radiation neither parameter shows any increase. It would seem therefore that the heat pump reaches its peak heating output of 10 MJ per hour at a solar radiation value of 600 W/m^2. At zero solar radiation both parameters

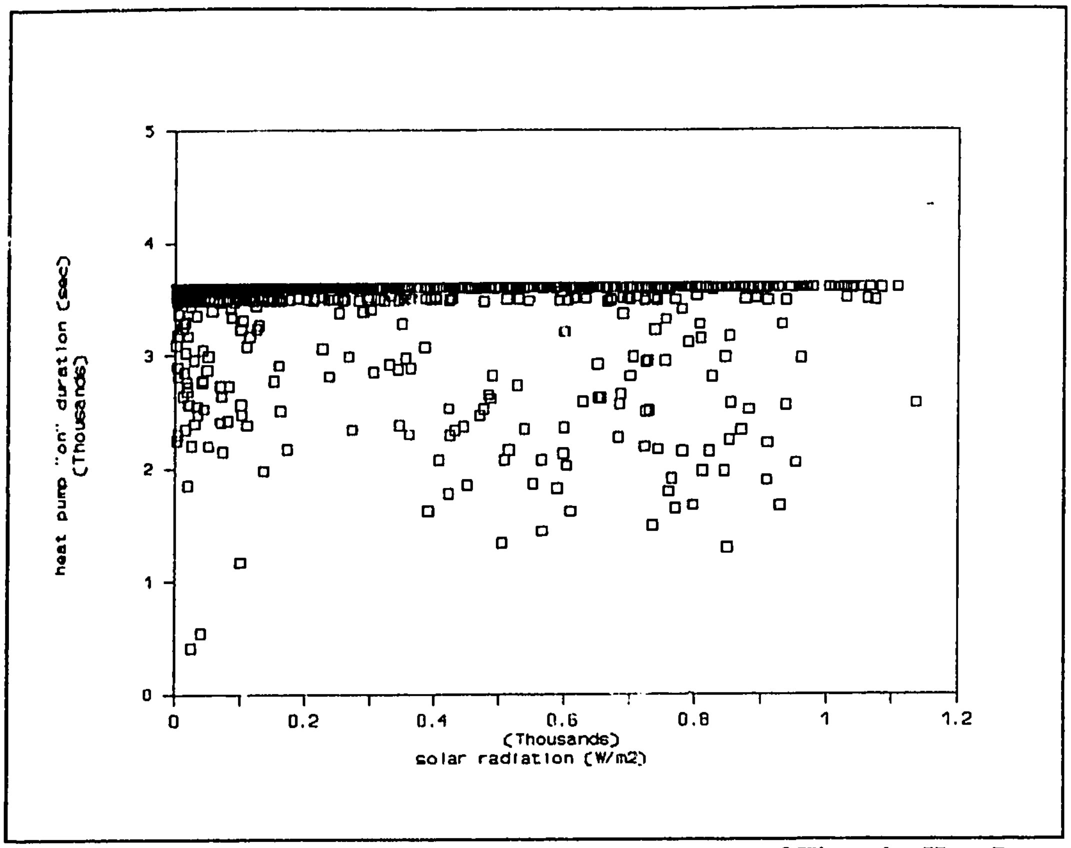

Figure 5.1 - The Effect of Solar Radiation on the Amount of Time the Heat Pump is on-line per hour for the '86-'87 heating season.

have a wide scatter with heat output varying from approximately 3 to 8 MJ and with heat input from the compressor varying from around 0.5 to 2.5 MJ. Since the graph represents data ranging from September '86 to March '87 a range of ambient temperatures are involved. The scatter is a result of the effect of ambient temperature which mainly dictates the performance during the night, early morning, and late evening (i.e. low or zero solar radiation). The narrowing of the data band with increasing **Rad** (i.e. better correlation) indicates that solar radiation dictates the performance of the unit during the day. It should also be noted that some of the scatter in the data may be due to the heat pump cycling on and off.

Figure 5.3 illustrates the effect of solar radiation on the compressor work (**Win**) and of the module electrical energy consumption (**Qtot**). The amount of energy required to power the heat pump appears to increase slightly (by about 1 MJ in total) with increasing solar radiation. This characteristic may explain why the energy gained at the compressor (**Qin**) increases with radiation in the preceding graph. Like **Qout** and **Qin**, compressor work levels off at around 600 W/m^2 to a value of about 4 MJ. The module electrical energy consumption is relatively low at high solar radiation values averaging at approximately 5.5 MJ. However, at low values of radiation there is a jump in the energy consumption to about 10 MJ with occasional maximums of 16 MJ. This change in the total energy consumption of the module suggests some switching between the two heating systems (the heat pump and the electric resistance heater).

In Figure 5.3, the difference between the two graphs (total module and compressor energy consumptions) is equal to the auxiliary energy **Qaux** (the portion of the total module energy consumption that is not used by the heat pump). The control system switches on the electric resistance heater when the heat pump is unable to handle the full load (this process was discussed in section 2.4). Figure 5.4 shows the change in the heating load (**Qload**) and the auxiliary energy with solar radiation. For clarity the graph of **Qload** is shifted upwards by 3 MJ (the equation relating the two parameters was presented in section 3.2.3). Clearly both graphs exhibit similar behaviours with each having two plateaus particularly in the low solar radiation region. It should be noted that the similarity is reasonable since the heating load is calculated as a function of the auxiliary energy. At high solar radiation levels the values average about 1.5 MJ and

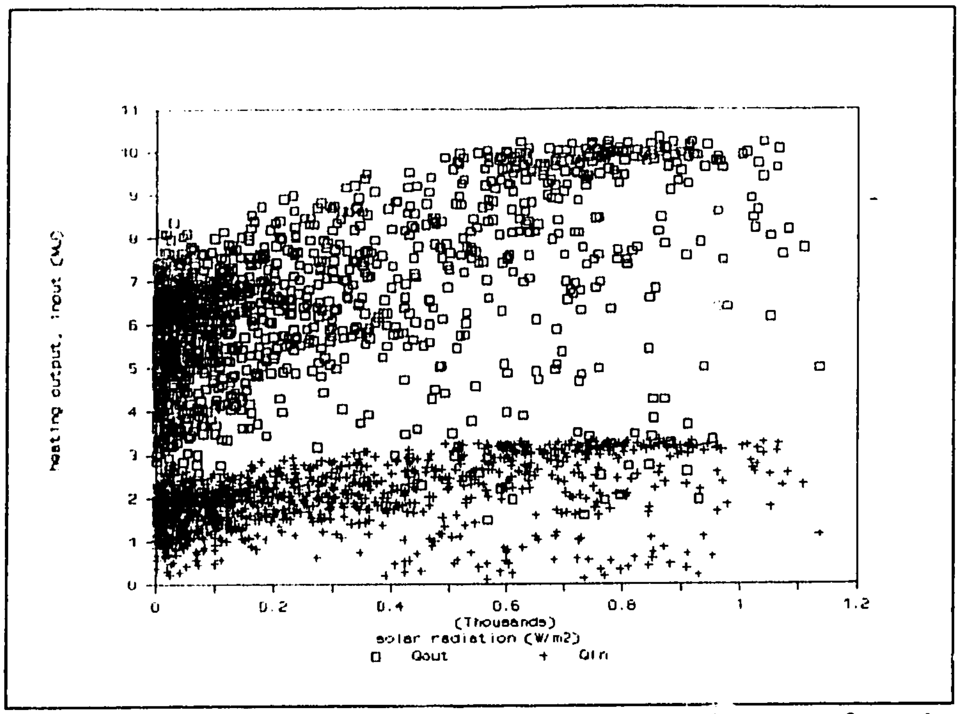

Figure 5.2 - Heat Pump Heating Output and the Heating Input from the Compressor as a function of Solar Radiation for the '86-'87 heating season.

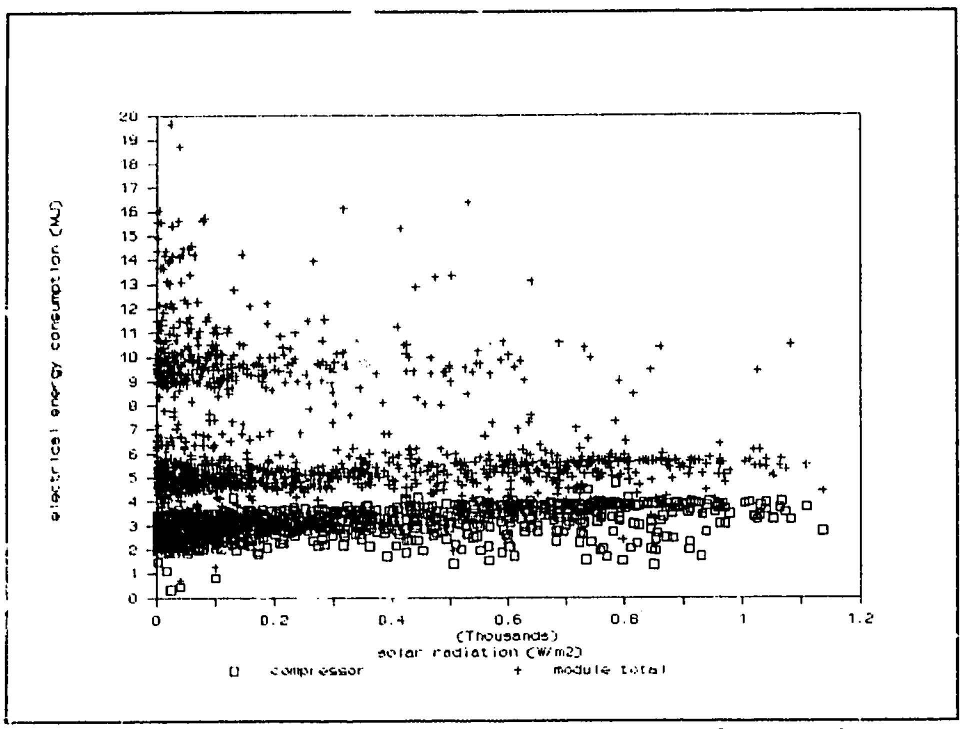

Figure 5.3 - Compressor and Total Module Energy Consumptions as a function of Solar Radiation for the '86-'87 heating season.

15 MJ for the auxiliary energy and the heating load, respectively. The lower plateau of the auxiliary energy represents the times when the electric resistance heater is off and the extra auxiliary energy is used only by the non-heating equipment such as the data acquisition system, lights, etc. (although these contribute to space heating indirectly). In the low solar radiation region the heat pump can no longer support the heating load and the electric resistance heater comes on-line. It should be noted that the heating load is a function of the ambient temperature as well and is the cause of the wide scatter in the data (the effect of ambient temperature will be investigated in the following section). A toggling effect is experienced by the electric resistance heater as the thermostat control forces it to maintain a preset room temperature resulting in the two plateaus in the auxiliary energy graph. This characteristic explains the nature of the total module energy consumption in the previous figure. The activation of the electric resistance heater produces an increase in the auxiliary energy of approximately 5.5 MJ per hour. Since the electric resistance heater has a heating output of 7.5 kW, this would mean that the heater was on-line for an average of approximately 12 minutes each hour.

As discussed in section 2.3.9, the thermostatic expansion valve (T-X valve) controls the amount of superheat to the refrigerant by modulating the flowrate. Figure 5.5 shows a relationship between the refrigerant mass flowrate (FR12) and solar radiation. It appears that the mass flowrate increases with increasing solar radiation. At high radiation values the thermal expansion valve senses the increase in the evaporator outlet pressure (due to an increase in the evaporator outlet temperature) and accelerates the flowrate to absorb the increased amount of energy available at the evaporator and minimize superheating of the fluid. Heat output and heat input from the compressor are functions of the mass flowrate and their increase with solar radiation (in the previous figures) is a direct result of the increase in flowrate with solar radiation. The increased mass flowrate presents the compressor with refrigerant gas at a higher density. As a result the compressor has to work harder to compress the refrigerant and therefore consumes a larger amount of electricity. This explains the reason for the increase in compressor work with solar radiation in Figure 5.3. The mass flowrate graph shows a levelling off of the flowrate at around 0.016 kg/s corresponding to an incident radiation level of about 600 W/m^2. At a certain superheat at the evaporator outlet the thermal expansion valve will be fully open to allow the maximum flowrate of refrigerant to the evaporator. Additional increases in the evaporator outlet pressure beyond this point will therefore have no further effect in increasing the flowrate. This might explain the levelling off characteristic of the heating output, heating input from compressor and compressor work graphs. However, another component could be responsible for the limitation of the refrigerant mass flowrate. The suction pressure regulator (discussed in section 2.3.12) prevents excessive inlet pressures to the compressor by regulating the refrigerant flowrate. A later section will try to determine which of these two components is the cause for limiting the flowrate to 0.016 kg/s.

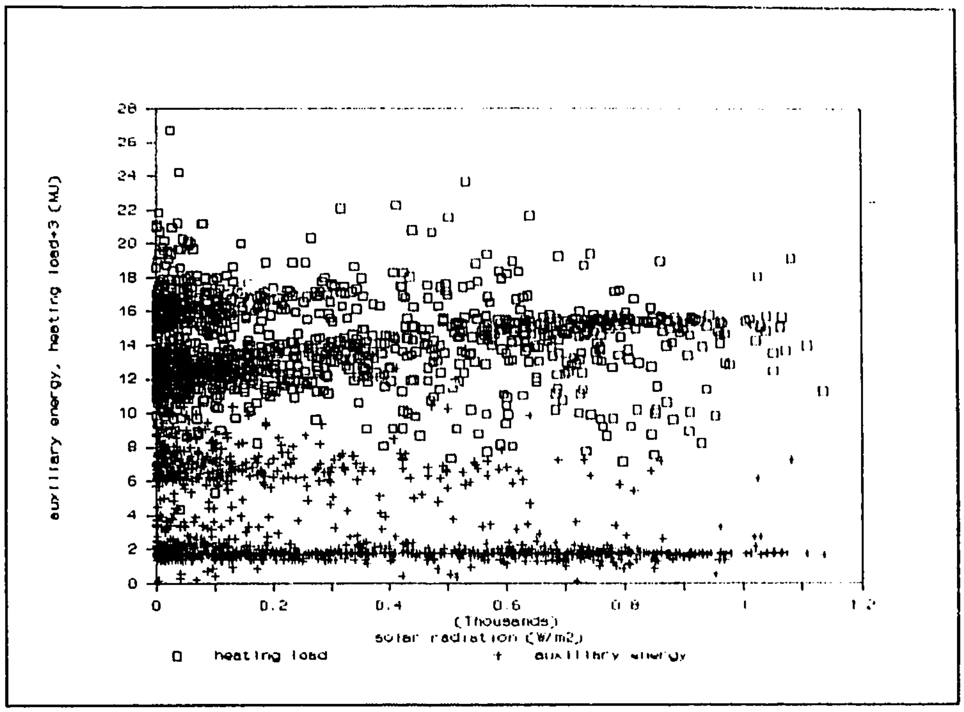

Figure 5.4 - Effect of Solar Radiation on the Heating Load and the Auxiliary Energy for the '86-'87 heating season.

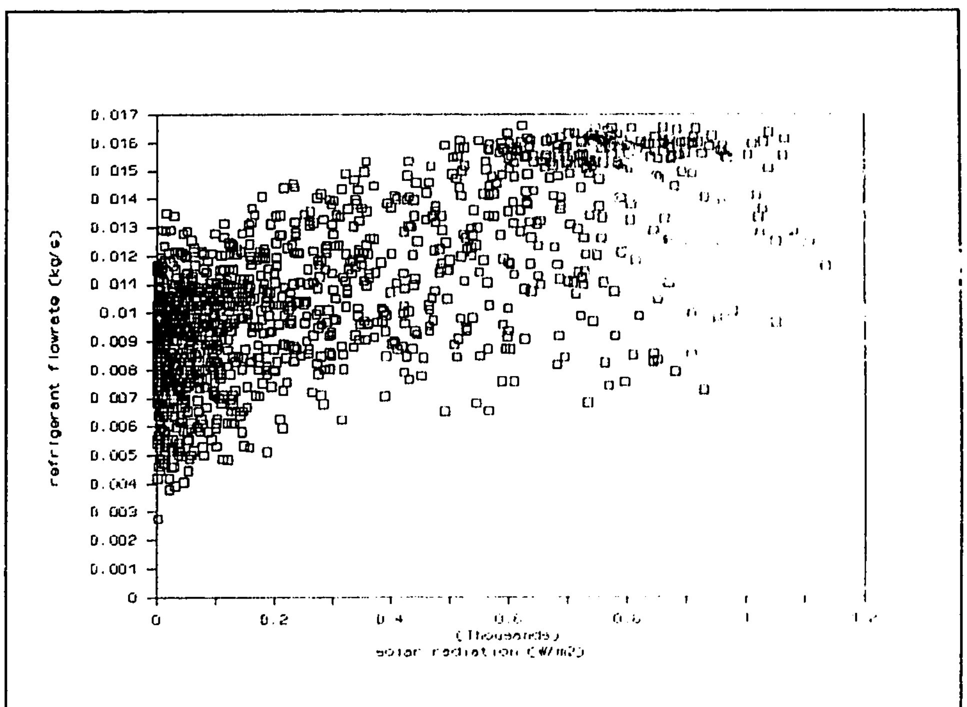

Figure 5.5 - Change in the Refrigerant Flowrate with Solar Radiation for the '86-'87 heating season.

At the evaporator the cool refrigerant liquid absorbs heat from the ambient air and, when available, the energy present in solar radiation. Figure 5.6 shows the effect of solar radiation on the amount of heat energy absorbed by the working fluid at the evaporator (**Qevap**) and at the suction line (**Q87**). The absorbed energy increases from an average of 3.5 MJ at low solar radiation to around 6.8 MJ at approximately 600 W/m^2 where it levels off. This radiation level corresponds to the value where the flowrate is at its maximum (0.016 kg/s) as determined in the previous graph. The amount of heat energy absorbed at the evaporator therefore is limited by the mass flowrate. The heat energy absorbed by the suction line appears to be independent of solar radiation most probably because it is more or less shaded by the evaporator.

The coefficient of performance (**COP**) of the heat pump is, as defined in section 3.2.3, the heating output of the heat pump divided by the energy input to the compressor. The effect of solar radiation on the COP is illustrated in Figure 5.7. At about 600 W/m^2, the value corresponding to the point where the maximum flowrate of 0.016 kg/s occurs, the COP levels off at approximately 2.6. The COP for low solar radiation ranges from about 1.3 to 2.3 due to the effect of ambient temperature. Solar radiation appears to dictate the COP at higher radiation levels. As discussed in section 4.4, the COP of this heat pump without solar-assistance can be approximated as the COP of the solar-assisted heat pump at night. Taking the average of the night-time values the COP of such a unit can be assumed to be 1.8.

The presence of solar radiation increases the heat energy input to the evaporator and (therefore) the heating output of the heat pump and reduces the total amount of

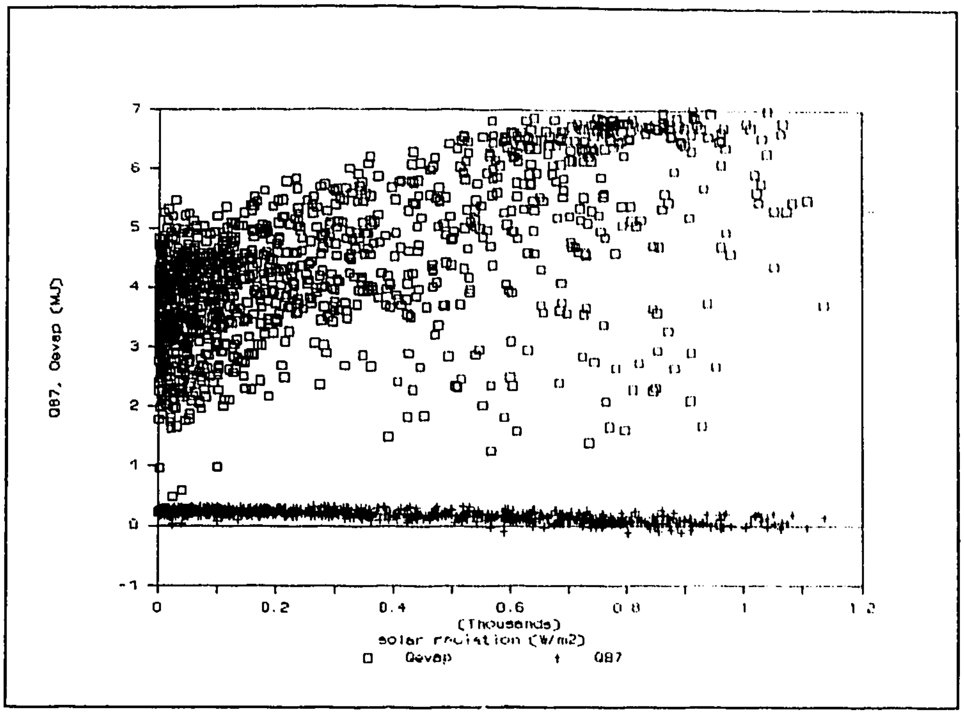

Figure 5.6 - Effect of Solar Radiation on the Evaporator and Suction Line Heat Absorptions for the '86-'87 heating season.

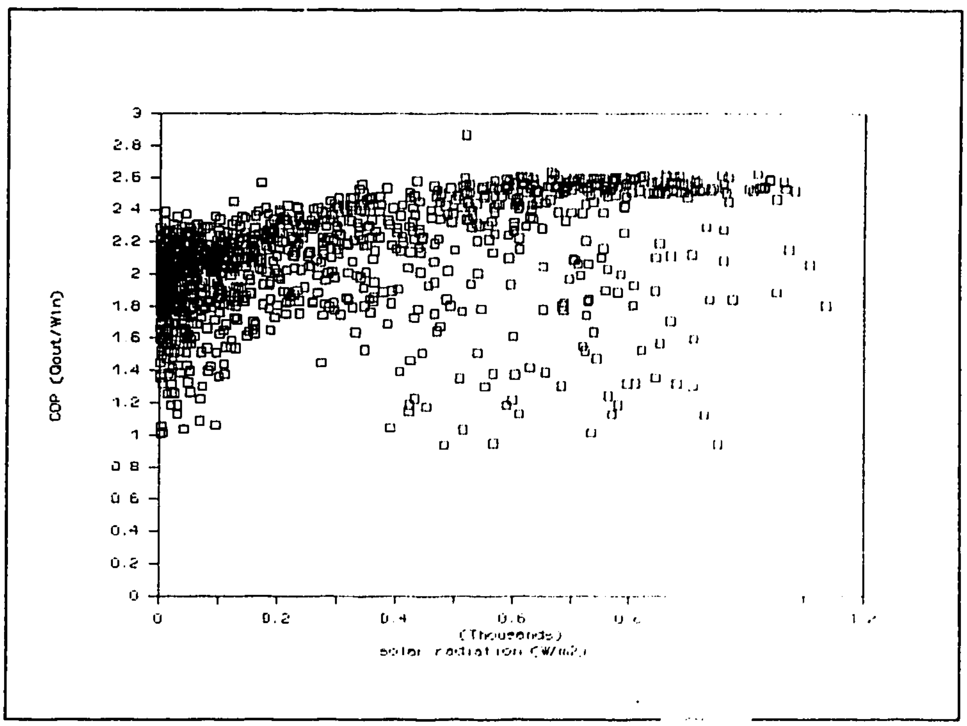

Figure 5.7 - Heat Pump COP as a function of Solar Radiation for the '86-'87 heating season.

electrical energy required for heating the module. The COP was also improved with solar energy by about 0.3 to 1.3. It remains to be seen then if the costs of these improvements (i.e. the cost of the solar evaporator) are justified.

5.2.2 EFFECT OF AMBIENT TEMPERATURE ON NIGHT-TIME PERFORMANCE

The other source of heat, the surrounding air, is like solar radiation in that it yields the most heat energy when it is least desired (summer) and yields the least heat energy when it is most needed (winter). Unlike solar energy however, it is always available. In the heat pump under investigation, which does not have a heat or energy storage device, air may be a more significant source of heat and solely dictates the operation of the unit during the night. Only the night-time data will be investigated for this section. The heat pump performance using air as a heat source is governed by air temperature and wind speed. The wind speed in the vicinity of the solar collector was not measured, although the wind speed at 10 meters elevation north and south of the houses was recorded. To simplify the analysis the effect of wind speed was not considered. Only the effect of ambient temperature on the unit performance was studied.

Figure 5.8 shows the effect of ambient temperature (**Tamb**) on the amount of time the heat pump was on per hour (**Thp**) for the night-time periods of the '86-'87 heating season. Clearly there is a noticeable drop in operation times at warmer temperatures (around the 4 °C mark). The increase in ambient temperature lowers the heating requirements for the building, resulting in the intermittent operation of the heat pump as

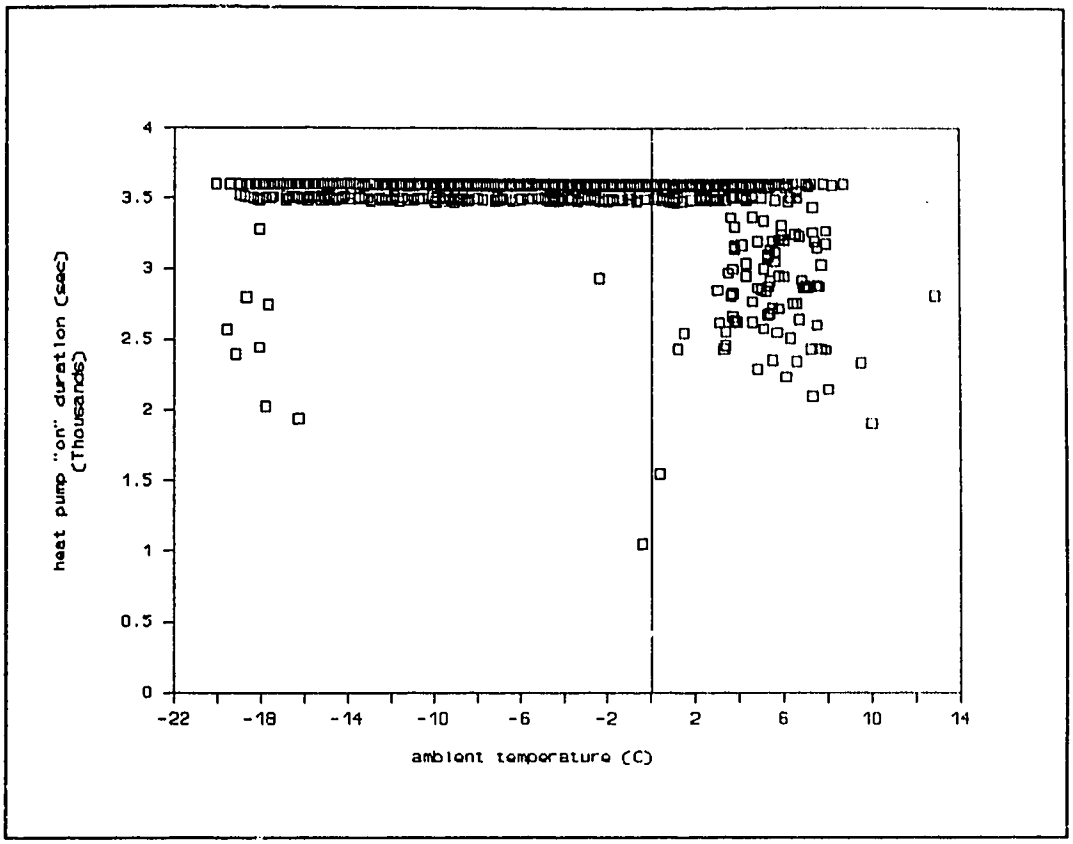

Figure 5.8 - Effect of Ambient Temperature on the Amount of Time the Heat Pump is on-line per hour for the '86-'87 heating season (night).

it cycles on and off. It is only below this ambient temperature that the heat pump must operate continuously to provide sufficient heating for the building. Also evident is the intermittent operation of the heat pump at around the -19 °C level. This suggests that the heat pump may not be capable of operating at very low ambient temperatures. This is due to the inability of the compressor to handle the low vapor pressure at such conditions resulting in the shutdown of the system initiated by a pressure switch. November '85 and February '86 were the coldest months in the study. They show evidence that the heat pump could not operate below approximately -23 °C corresponding to compressor inlet pressure of about 110 kPa. This inlet pressure is probably the compressor start set point discussed in section 2.3.8. The upper limit of the inlet pressure that the heat pump compressor could handle was not determined.

The effect of the ambient air temperature on the heating output from the heat pump and the heating input from the compressor parameters, **Qout** and **Qin**, for the '86-'87 heating season is shown in Figure 5.9. Increasing th[illegible] temperature causes an increase in the heating output of the heat pump. Th[illegible] mak[illegible] sense since a higher amount of heat energy available in the air allows for a higher amount of heat energy to be absorbed in the evaporator to yield a greater heating output. At about 4 °C the maximum heating output is reached at approximately 8 MJ. Higher temperatures seem to result in a drop in heat output and produce scatter in the data. This characteristic is due to the system cycling as discussed earlier. The lower limit of **Qout** is around 2.5 MJ at an ambient temperature of -19 °C at which point there is some falling off of data (scatter). As [illegible] this is because the heat pump is unable to operate at very low ambient tempera[illegible] It should be noted that the maximum heating output reached in the day (10 MJ) is about 2 MJ higher than the maximum in the night since two heat sources (solar radiation and air) are available during the day. Increasing ambient temperature also increases the heat energy input from the compressor. From -19 to 4 °C energy input from the compressor increases by approximately 1 MJ. Beyond 4 °C it undergoes a drop similar to the heat output case and for the same reason. Since the data under analysis does not include the cases when the heat pump was off there are no zero heat output or energy input from the compressor read[illegible] on the graph.

Figure 5.10 shows the impact of ambient temperature on the electrical energy

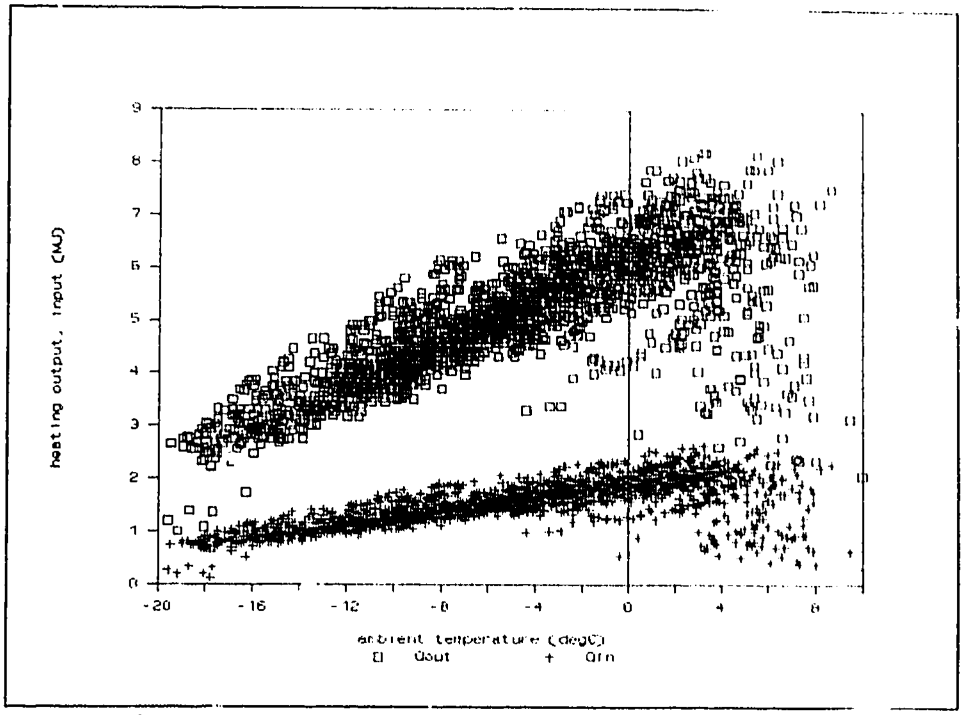

Figure 5.9 - Heat Pump Heating Output and Compressor Input as a function of Ambient Temperature for the '86-'87 heating season (night).

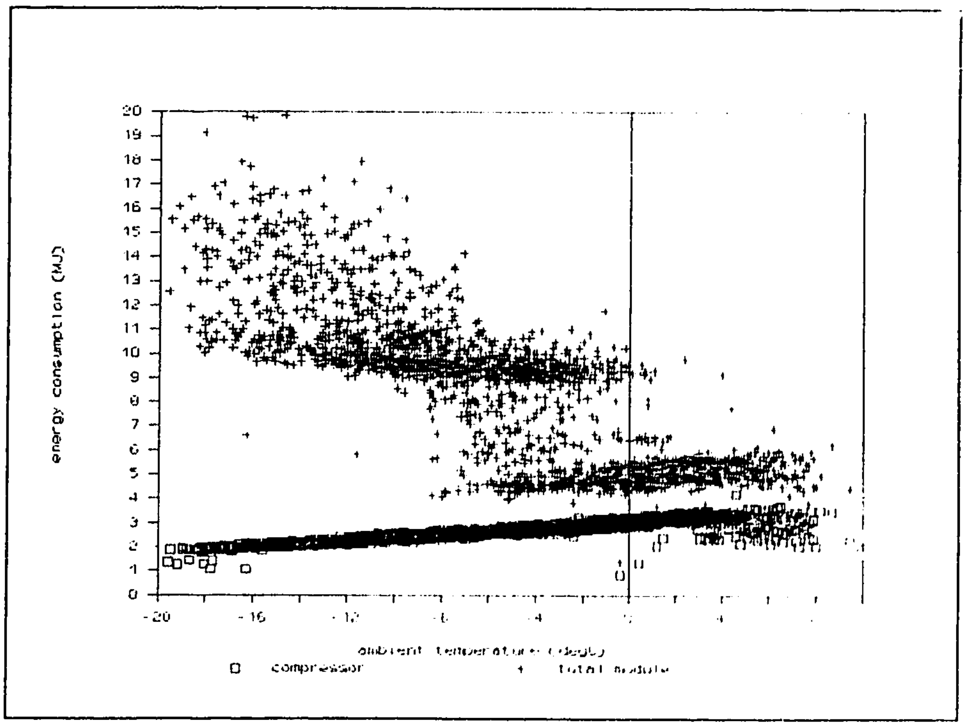

Figure 5.10 - Compressor and Total Module Energy Consumptions as a function of the Ambient Temperature for the '86-'87 heating season (night).

consumption of the compressor (**Win**) and the module (**Qtot**) for the night-time periods of the '86-'87 heating season. For the compressor work the effect is linear to 4 °C where it experiences a drop as the heat pump cycles. Compressor work ranges from about 2 MJ at -19 °C to about 3.5 MJ at 4 °C. Like solar radiation, the effect of increasing ambient temperature presents the heat pump with more heat energy and should cause the refrigerant flowrate, and therefore the electrical energy required by the compressor, to increase. The energy consumption of the module exhibits the same two plateau characteristic evident in the auxiliary energy graph in the last section. For ambient temperatures between 0 and 8 °C, module electrical energy consumption is around 5.5 MJ. Here the heat pump is capable of supporting the full heating load and the auxiliary heater is not required. Below this range the heating load becomes too great for the heat pump to handle solely and the electric resistance heater is brought on-line increasing **Qtot** to about 10 MJ. From about -8 to 0 °C the electric resistance heater operates no more than approximately 12 minutes per hour producing the scatter between the two plateaus. As the ambient temperature decreases further the amount of electric resistance heating per hour is increased.

The graphs of the heating load (**Qload**) and the auxiliary energy (**Qaux**) vs the ambient temperature for the '86-'87 heating season, as shown in Figure 5.11, show a similar pattern to the module energy consumption graph (Figure 5.10) with the two plateau regions. Note that the graph of the heating load is shifted upwards by 3 MJ for clarity. The auxiliary energy fluctuates between the 1.5 and 7 MJ values. In the previous section it was determined that the 1.5 MJ readings indicate the hours when the electric resistance heater was not on and the 7 MJ readings were for the hours when the electric resistance heater was on-line for about 12 minutes. The heating load at the lower plateau of the graph hovers at around the 10 MJ mark. When the auxiliary heating is turned on this value jumps to about 13 MJ (since Qload is calculated as the sum of the auxiliary energy plus the heat pump heating output plus the heat loss from the compressor. Heat loss from the compressor is simply the difference between the compressor work **(Win)** and the heat input from the compressor **(Qin)**).

The effect of ambient temperature on the flowrate during the night-time is linear as illustrated in Figure 5.12. As presumed, ambient temperature has a similar effect on the refrigerant flowrate as solar radiation does. The flow rates range from 0.003 kg/s at -19 °C to about 0.012 kg/s at 4 °C. It was found in the previous section that the maximum flowrate through the unit is 0.016 kg/s. Knowing this, Figure 5.12 reveals that the ambient temperature effect on the unit during the night within this range (-19 to 5 °C) is not restricted by either the thermal expansion valve or the suction pressure regulator. It would seem therefore that the energy absorbed at the evaporator should not level off as it did in the solar radiation case. Figure 5.13, a graph of the evaporator absorption vs the ambient temperature for the night-time, confirms this showing a maximum heat absorbtion of almost 5 MJ at 4 °C. This value is less than the 6.8 MJ achieved in the day when solar radiation is available at levels of 600 W/m^2 and higher.

The amount of heat absorbed by the evaporator **(Qevap)** and the suction line **(Q87)** as a function of ambient temperature for the '86-'87 heating season is shown in Figure 5.13. It should be noted that since the data considered are for night-time, the

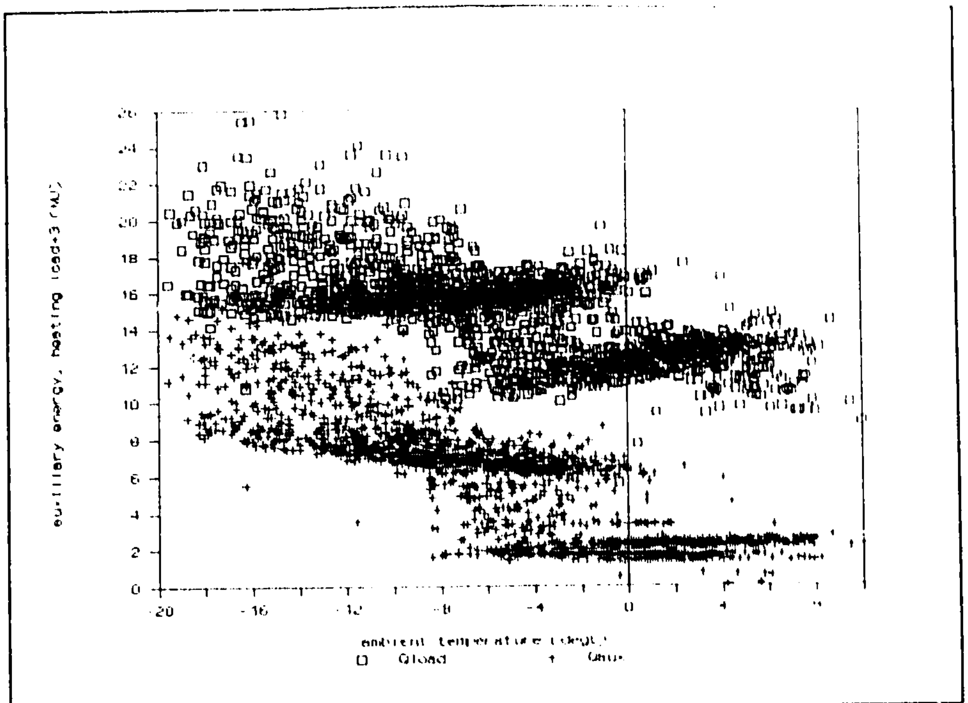

Figure 5.11 - Effect of Ambient Temperature on the Heating Load and the Auxiliary Energy for the '86-'87 heating season (night).

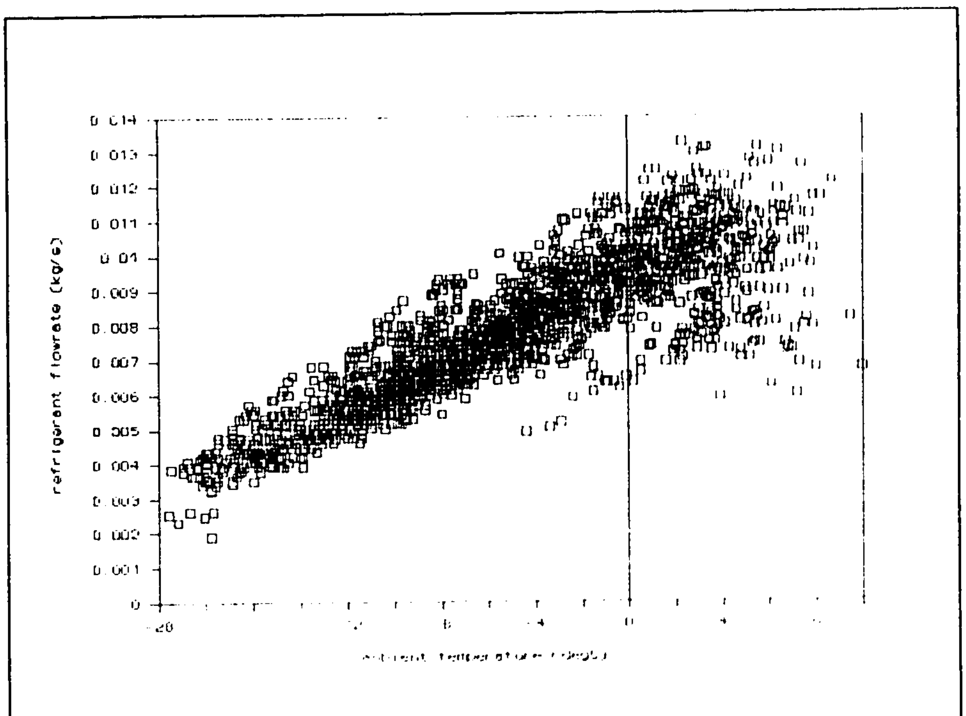

Figure 5.12 - Refrigerant Flowrate as a function of the Ambient Temperature for the '86-'87 heating season (night).

energy absorbed at the evaporator is comprised of only heat energy from ambient air and not from solar radiation. As with solar radiation, the amount of heat absorbed by the evaporator increases with increasing ambient temperature. At approximately 4 °C scatter in the data is evident due to the cycling of the heat pump. The maximum absorbed energy attained is about 5 MJ and the minimum (which occurs at about -19 °C) is about 1.7 MJ. Reasonably, the maximum and average energy absorbed by the evaporator per hour during the night is lower than during the day (which has a maximum of 6.8 MJ per hour). Stated another way, solar radiation provides the unit with a larger amount of heat than air does. It was determined in section 4.3.1 that for the majority of the day (solar radiation higher than 200 W/m^2) heat is rejected from the collector through convection heat transfer (with some reradiation) making solar radiation the only contributing heat source. The suction line heat absorption graph is magnified by five for clarity and appears to remain fairly constant with the ambient temperature at about 0.25 MJ.

The effect of ambient temperature on the coefficient of performance during the night for the '86-'87 heating season is shown in Figure 5.14. The COP increases steadily from 1.3 at -19 °C to about 2.3 at around 4 °C. Beyond this the COP drops off due to system cycling. Evidently during the night the ambient temperature does not improve the COP as much as solar radiation does (maximum of 2.6).

5.2.3 COEFFICIENT OF PERFORMANCE (COP) : HEATING PERIOD

The coefficient of performance of the heat pump for the period of September '86 to March '87 was found. As discussed in section 4.4, the average COP of the solar-

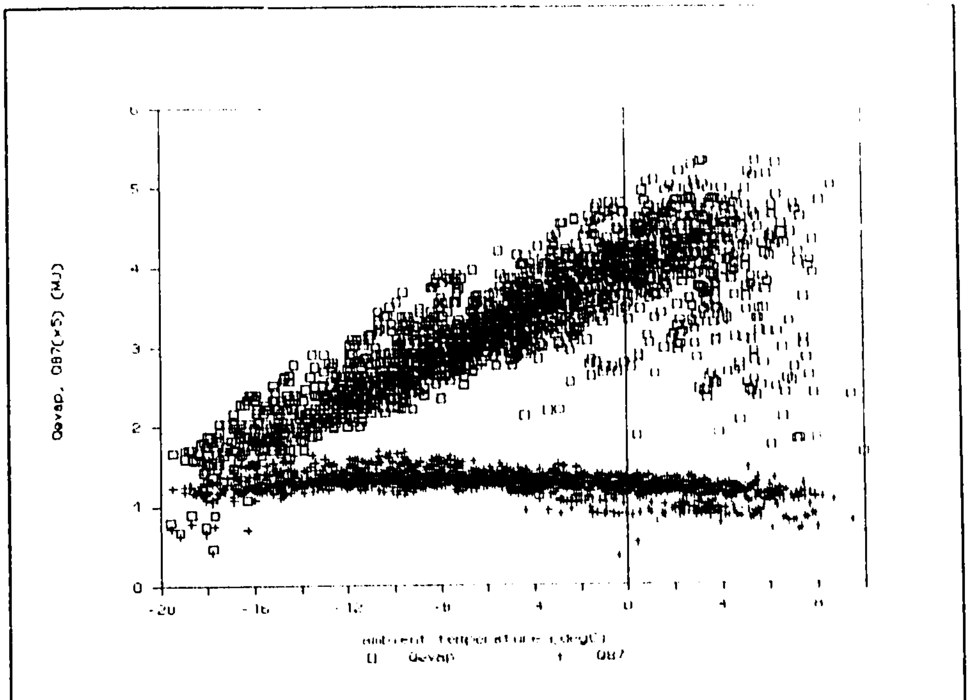

Figure 5.13 - Effect of Ambient Temperature on the Evaporator and Suction Line Absorptions for the '86-'87 heating season (night).

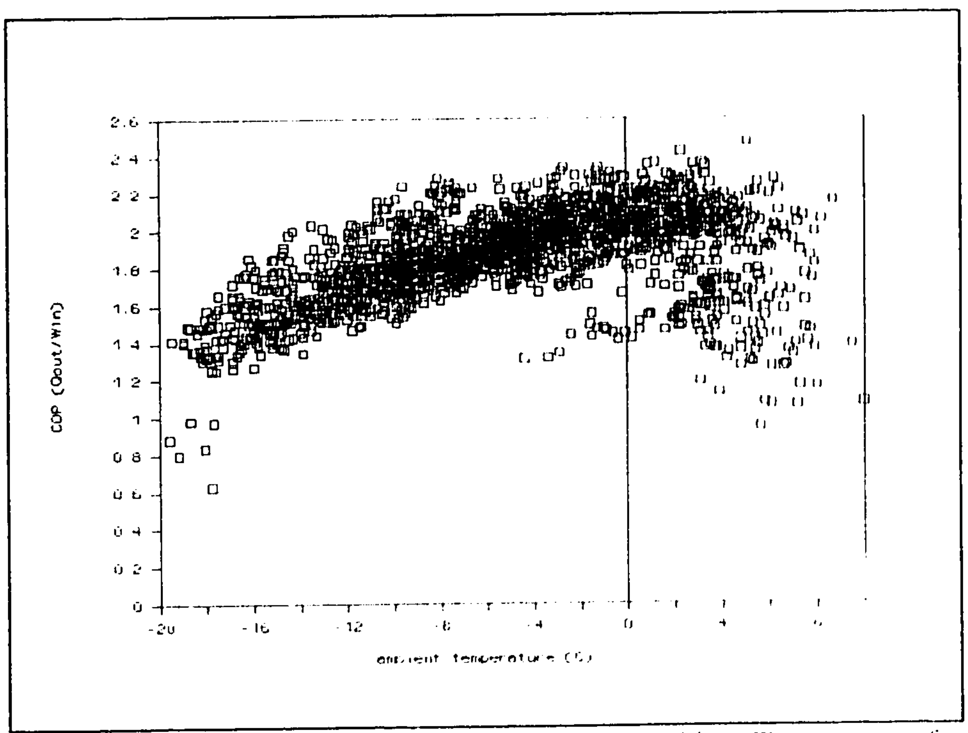

Figure 5.14 - Heat Pump COP as a function of Ambient Temperature for the '86-'87 heating season (night).

assisted heat pump for the entire period was found to be 1.98. Table 5.1 shows this and the day and night maximum COPs, the overall minimum (which occurs at night), and the average day-time and night-time COPs achieved during the September'86 March'87 period. The average values were calculated using the total heating output and the total compressor work for the entire period **(total)**, the day-time only **(day)**, and the night-time only **(night)**. The average COP during the night-time is used to approximate a heat pump unit without solar-assistance which has an evaporator with similar characteristics to the solar-assisted heat pump evaporator. An error analysis was performed (presented in the appendix). It was found that the difference between the day and night COP values are significant.

Table 5.1 - Heat Pump COP for the Period of Sept'86 - Mar'87

max COP		min COP	average COP		
day	**night**	**night**	**total**	**day**	**night**
2.6	2.3	1.4	2.0	2.1	1.8

5.2.4 MANUFACTURER'S SPECIFICATIONS

This section compares the measured results of the heat pump with the manufacturer's performance specifications. Table 5.2 shows the manufacturer's specifications for the solar-assisted heat pump with an 8.4 square meter, 4 panel solar collector array, and a water cooled condenser, when operating as a roof mounted unit in a temperate climate at a condensing temperature of 26.6 °C.

Table 5.2 - Manufacturer's Ratings

Tamb	Input	Output	COP
°C	W	W (Btu/hr)	
-6.6	870	3158.1 (3158.1)	3.63
-1.1	915	3669.2 (12523)	4.01
4.4	960	4214.4 (14376)	4.39
10	1010	4817.7 (16447)	4.77

Figures 5.15 and 5.16 compare the data collected to the specifications for the compressor input power (**Input**), the condenser heating output (**Output**), and the coefficient of performance (**COP**) with regards to the ambient temperature. Since the heat pump performance is fully represented by the month of January, and to reduce clutter in the graphs, the experimental data plotted are only from the first half of the month of January '87 (Jan 1-15). In the first graph the manufacturer's heating output values (assuming that linear extrapolation can be used) are much higher than the experimental ones given that the input power values for the two cases are almost the same. The manufacturer claims values of **Output** approximately one and a half times higher than were found. Note that some of the **Output** values almost reach 3000 W. These points correspond to the maximum heating output of the heat pump which occur at high solar radiation values and warm temperatures. This of course leads to the difference in coefficient of performance with the experimental data yielding significantly lower values with a maximum of about 2.6 (at the maximum heating output). A comparison based on solar radiation cannot be performed because the manufacturer did

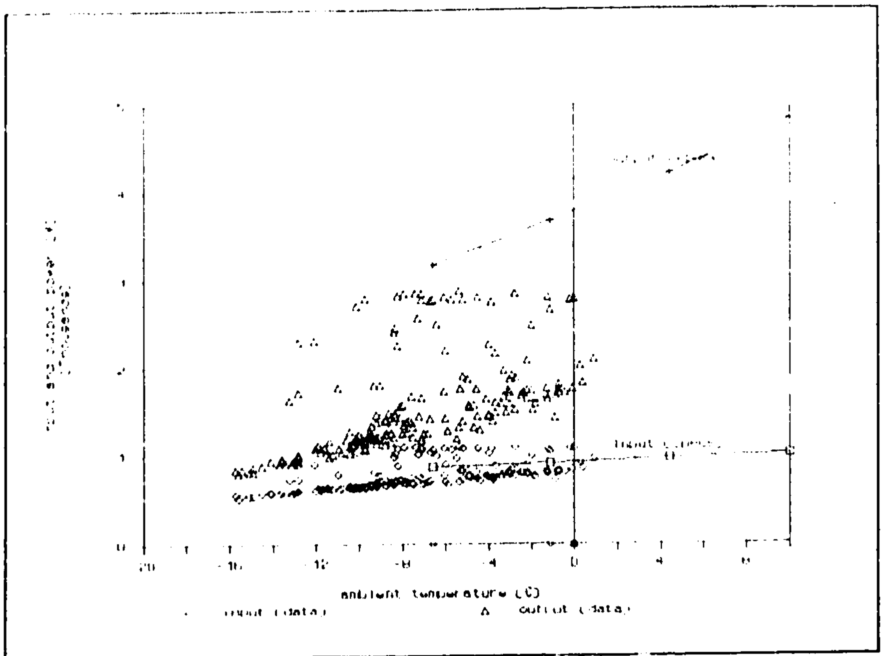

Figure 5.15 - Comparison of the Experimental Data to the Manufacturer's Performance Specifications for the Compressor Input Power and the Heat Pump Heating Output as a function of Ambient Temperature. Experimental Data are from January '87

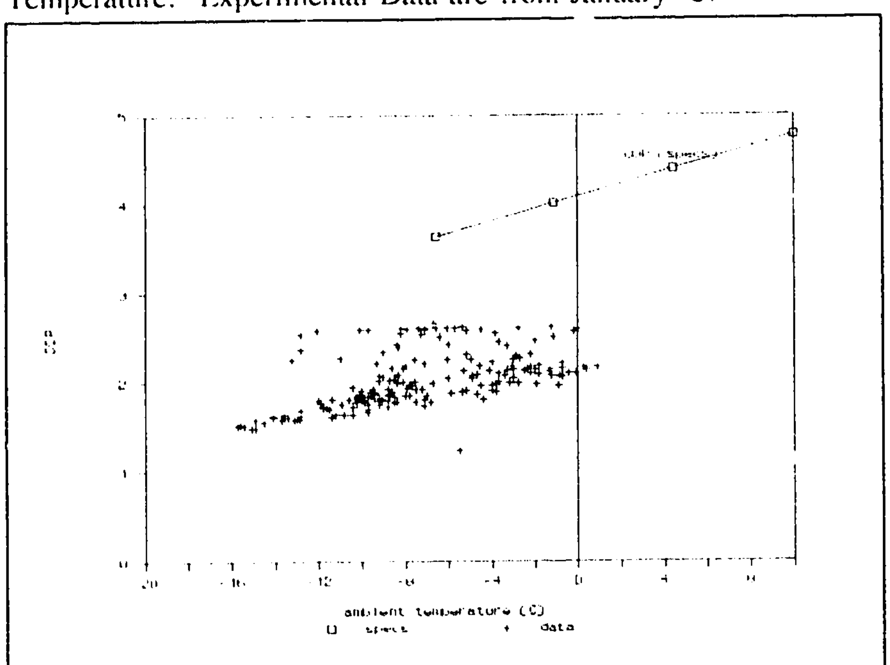

Figure 5.16 - Comparison of the Experimental Coefficient of Performance to that of the Manufacturer's Specifications as a Function of the Ambient Temperature. Experimental Data are from January '87

not provide solar values with the specifications. The experimental COPs are lower due to lower ambient temperature and the use of an air condensing coil. It cannot be concluded that the manufacturer overrated the heat pump unit since the original system was designed to heat water and not air.

5.2.5 THE THERMAL EXPANSION VALVE AND THE SUCTION PRESSURE REGULATOR

The refrigerant flowrate is affected by two components in the heat pump circuit, namely the thermal expansion valve and the suction pressure regulator (SPR). In the previous sections it was found that the maximum refrigerant flowrate achieved during heat pump operation was 0.016 kg/s. Any increase in the amount of solar radiation or evaporator pressure at this point did not have an effect on the flowrate. This characteristic could be due to the fact that the thermal expansion valve is fully open at 0.016 kg/s and is unable to increase the flowrate beyond this point or it may be that the suction pressure regulator limits the pressure to a certain value by limiting the refrigerant flowrate to 0.016 kg/s. The following analysis determines which of the two components is responsible (or more responsible) for this limitation in the system.

The thermal expansion valve controls the amount of superheat at the evaporator outlet at a preset limit. At values of superheat higher than the limit, the valve opens to increase the flowrate. Likewise when the amount of superheat is below the preset limit the valve closes to decrease the flowrate. Figure 5.17 shows the effect of the amount of superheat at the evaporator on the refrigerant flowrate during the day for the '86-'87 heating season. Since the line connecting the thermal expansion valve and the evaporator is considered as part of the evaporator for this investigation (see section 3.2.3) the amount of superheat is calculated as the rise in temperature between the expansion valve outlet and the evaporator outlet. As expected the general trend is an increase of the flowrate with the amount of superheat as the system strives to increase the rate of heat absorption. However the correlation between these two parameters is evidently not very good perhaps due to "hunting" or rapid change in cloud cover. The large scatter might also suggest that the thermal expansion valve may not be the main limiting factor in the refrigerant flowrate.

The working fluid encounters the SPR prior to entering the compressor. The SPR monitors and controls the incoming refrigerant fluid pressure to protect the compressor from extreme pressures. At pressures higher than a preset limit the regulator slows the flow of refrigerant into the compressor. The suction line which carries the refrigerant from the evaporator plate to the compressor is approximately 4 meters in length. The only significant drop in pressure in the suction line would be at the SPR when the preset pressure is exceeded. Figure 5.18 shows the correlation of the compressor inlet pressure with the evaporator outlet pressure (ie. the pressures after and before the suction pressure regulator respectively). The correlation between the two pressures appear to be about 1 to 1 until about 0.3 MPa. Further increases in the evaporator outlet pressure does not appear to have any effect on the compressor inlet pressure. This suggests that the SPR was set to limit the compressor inlet pressure to a maximum of 0.3 MPa.

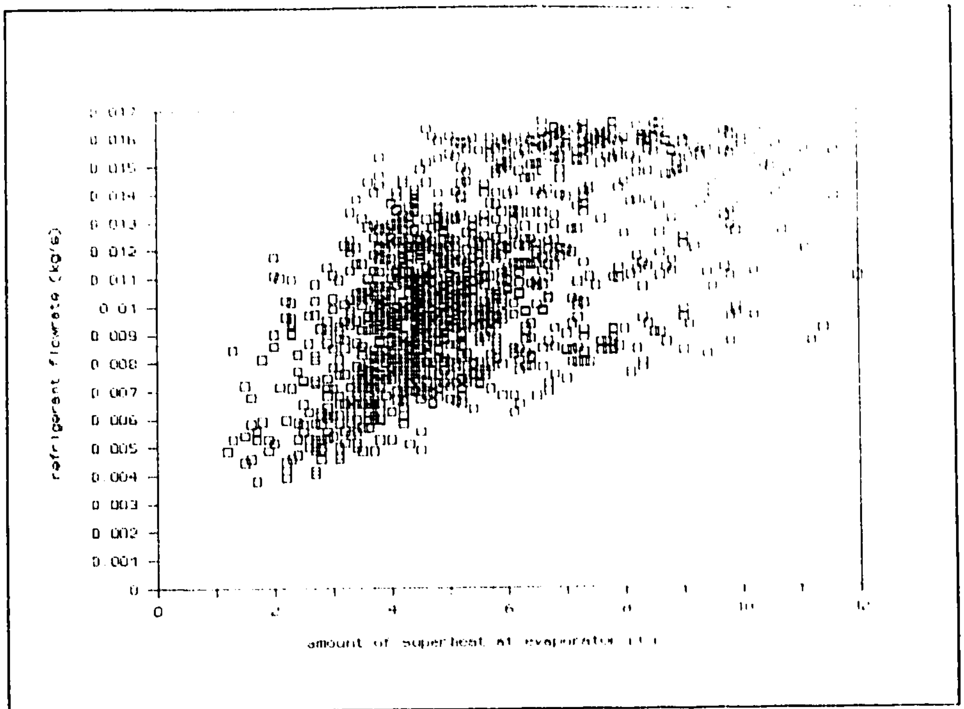

Figure 5.17 - Effect of the amount of superheat at the evaporator on the refrigerant flowrate during the day ('86-'87 heating season)

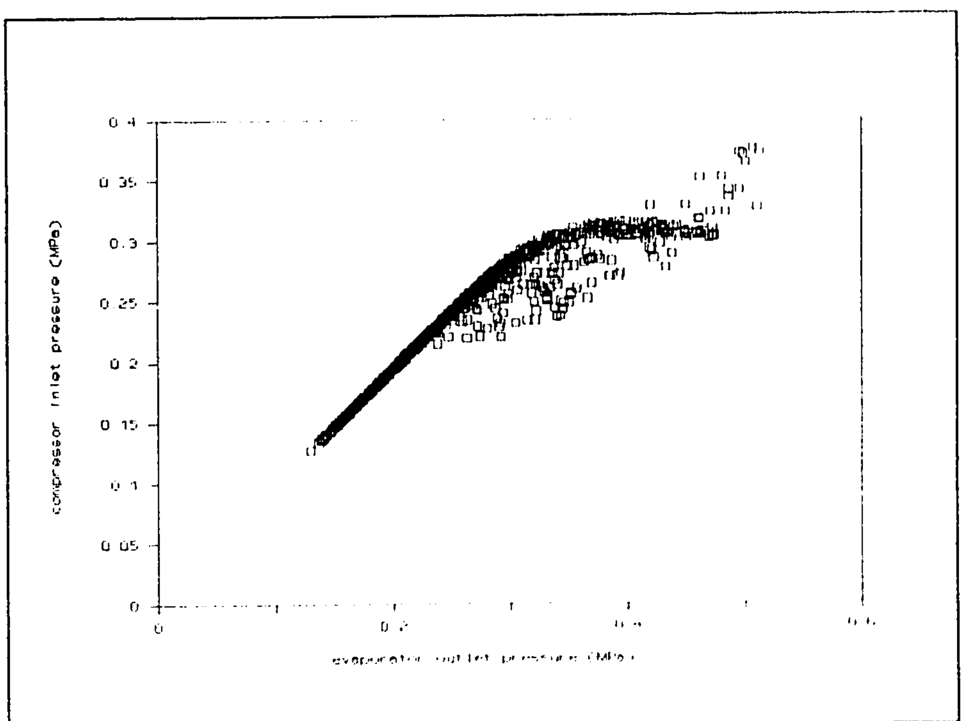

Figure 5.18 - Correlation of the compressor inlet pressure with the evaporator outlet pressure during the day ('86-'87 heating season)

Figure 5.19 shows the amount of pressure drop across the SPR at various flowrates during the day. Significant drops in pressure appear to occur only at the maximum flowrate value of 0.016 kg/s. Below this flowrate the pressure drop is small (about 10 kPa). Figure 5.20 shows the effect of the evaporator outlet pressure on the flowrate. Again the flowrate varies linearly with the pressure till about the 0.30 - 0.35 MPa point where it levels off at about 0.016 kg/s. Since the maximum flowrate is achieved only in circumstances where the evaporator outlet pressure is 0.3 MPa or higher and since the SPR only allows the fluid to have a maximum compressor inlet pressure of 0.3 MPa it can be concluded that the SPR is mainly responsible for limiting the flowrate to 0.016 kg/s. The thermal expansion valve controlled the flowrate up to the evaporator outlet pressure of 0.3 MPa.

It was determined that the maximum flowrate of 0.016 kg/s occurs at a solar radiation of about 600 W/m^2. It can be stated therefore that at this value of solar radiation the evaporator outlet pressure reaches 0.3 MPa and causes the suction pressure regulator to limit further increases in the refrigerant flowrate (at 0.016 kg/s). Since solar radiation levels of 1100 W/m^2 can occur and were measured in Edmonton, the evaporator would appear to be oversized in regards to the compressor for this particular region.

5.3 SOLAR EVAPORATOR PERFORMANCE

An important part of this study concerns the performance of the solar evaporator collector employed by the heat pump. In Chapter 4 the methods used to evaluate the

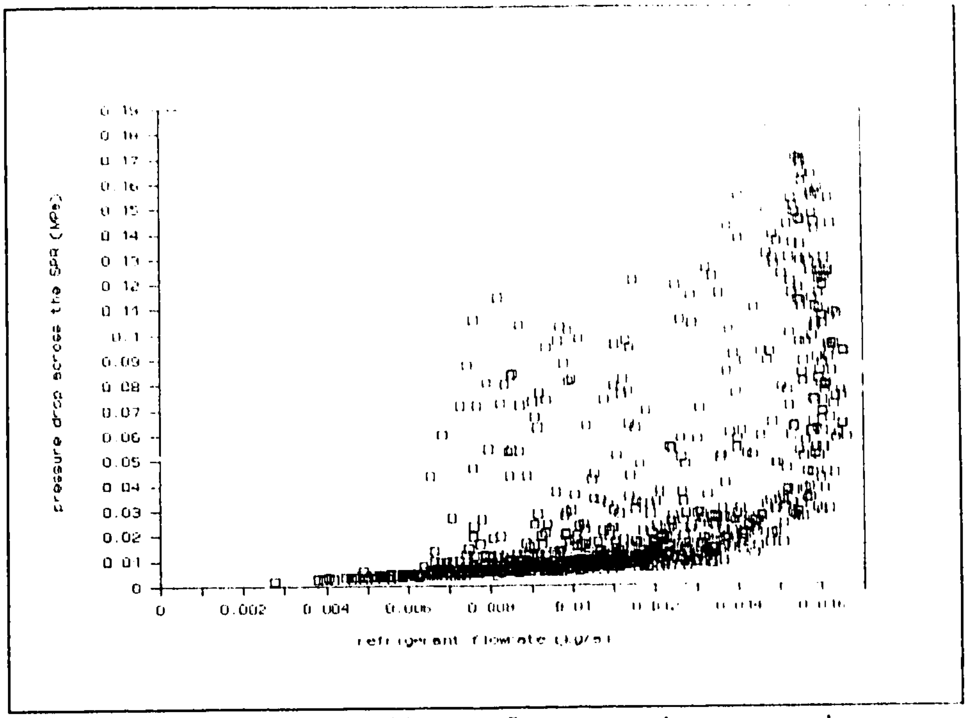

Figure 5.19 - Effect of the refrigerant flowrate on the pressure drop across the suction pressure regulator during the day ('86-'87 heating season)

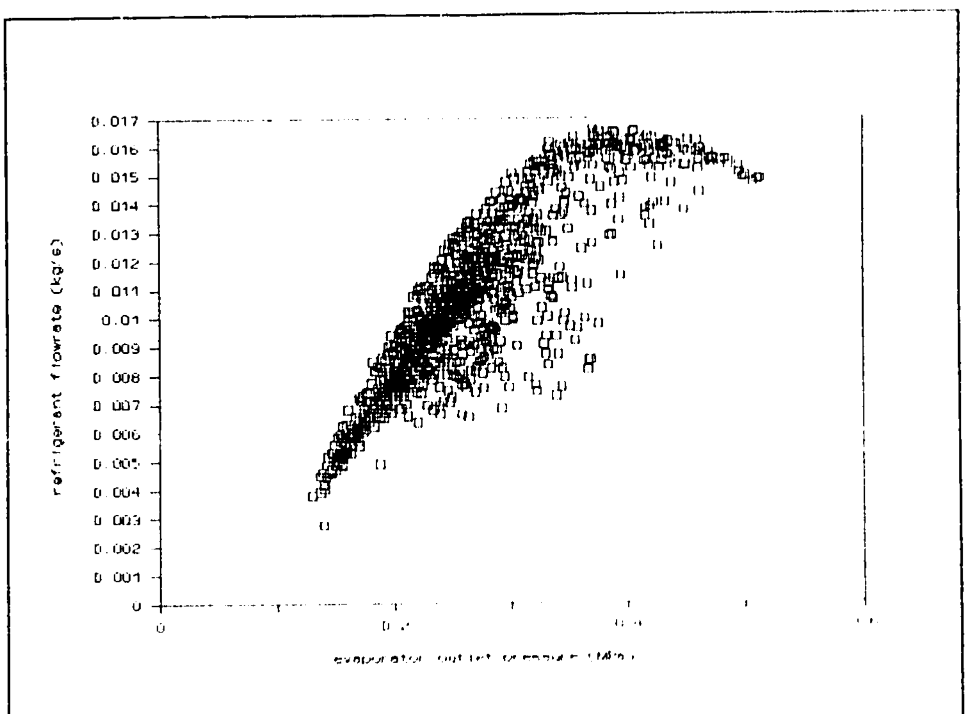

Figure 5.20 - Effect of the evaporator outlet pressure on the refrigerant flowrate during the day ('86-'87 heating season)

solar evaporator collector were described. As stated in that section, the three important questions to be answered are:

- how efficient is the evaporator collector?
- does the solar evaporator plate improve the performance of the heat pump?
- is the collector surface area over- or undersized?

5.3.1 COLLECTOR EFFICIENCY

The evaporator experiences three different conditions per day (as discussed in section 4.3.1), namely high solar radiation, low solar radiation, and no solar radiation (night). It was determined that the collector efficiency (the ratio of the amount of energy absorbed by the collector plate to the amount of solar radiation striking the collector plate) for the low solar radiation case would not be evaluated due to the complications involved with the presence of convection heat transfer to the plate from the air. However, with the methods discussed in that chapter, the collector efficiency can be evaluated for the case where solar radiation is high enough such that the plate does not absorb heat energy by convection (this occurs when the solar radiation is greater than 200 W/m^2). The collector efficiency as a function of solar radiation is shown in Figure 5.21 for the '86-'87 heating season. Although the plate temperature is on average equal to the ambient temperature at 200 W/m^2, only the range of solar radiation above the 300 W/m^2 mark is considered to minimize any uncertainty in the direction of convection heat transfer. The plate efficiency ranges from about 0.2 at a solar radiation value of 1000 W/m^2 to 0.6 at 300 W/m^2. It appears therefore that the evaporator is more effective in transmitting the absorbed energy to the working fluid during the periods of low sunshine.

The ASHRAE STANDARD (1977) suggests the evaluation of the collector efficiency at steady state or quasi-steady state conditions. It would therefore be more correct to evaluate the efficiency using data for bright sunshine days with no cloud cover. Data satisfying these conditions are few. For the 1986-'87 data, 21 bright sunshine days with conditions approaching no cloud cover were found. These are:

September'86: 3,5,6,7,19,21;

October'86: 10,12,15,16,17,18,21,22;

January'87: 5, 6, 7, 8, 22;

March'87: 27,28

Only data with solar radiation levels between 700 and 1000 W/m^2 were used for the analysis. The data is of the hourly type. As stated in section 4.3.1 the efficiency can be expressed as a function of the difference between the evaporator working fluid temperature and the ambient temperature divided by the solar radiation. Figure 5.22 illustrates this for bright sunshine days with low cloud cover for solar radiation levels between 700 and 1000 W/m^2. Using linear regression the best fit curve was found. By equation 4.4 (shown here again as equation 5.1 for convenience) it can be seen that the values of the y-intercept and the slope (magnitude) of this curve are approximately equivalent to the plate F'(τ,α)/L' and F'U_L/L' (collector loss) respectively.

$$\eta=\frac{q_u}{I_{t0}}=\frac{F'(\tau\alpha)_e}{L'}-\frac{F'U_L(T_f-T_{at})}{L'Rad} \qquad (4.4)$$

F'(τ,α)/L' was found to be 0.3 and is lower than typical $F_R(\tau,\alpha)$ values for glazed

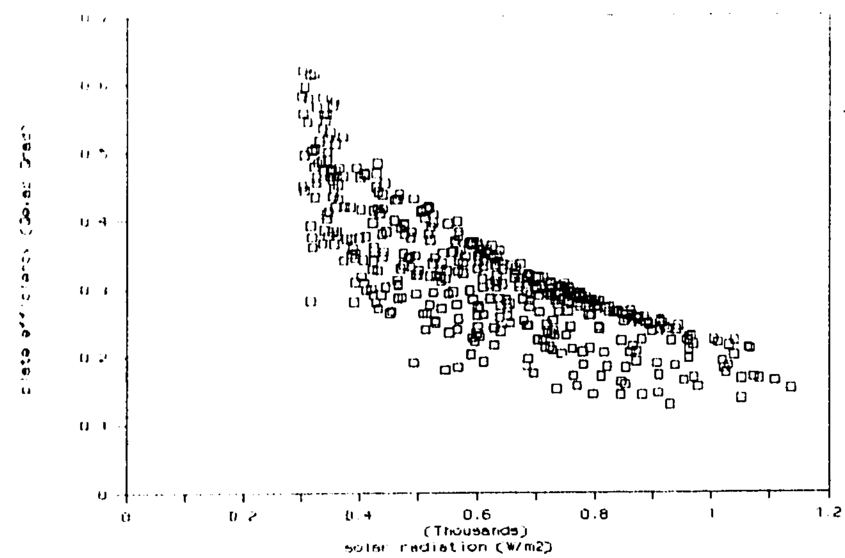

Figure 5.21 - Plate Efficiency as a function of Solar Radiation (Solar Radiation > 300 W/m^2). Data are for the '86-'87 heating season.

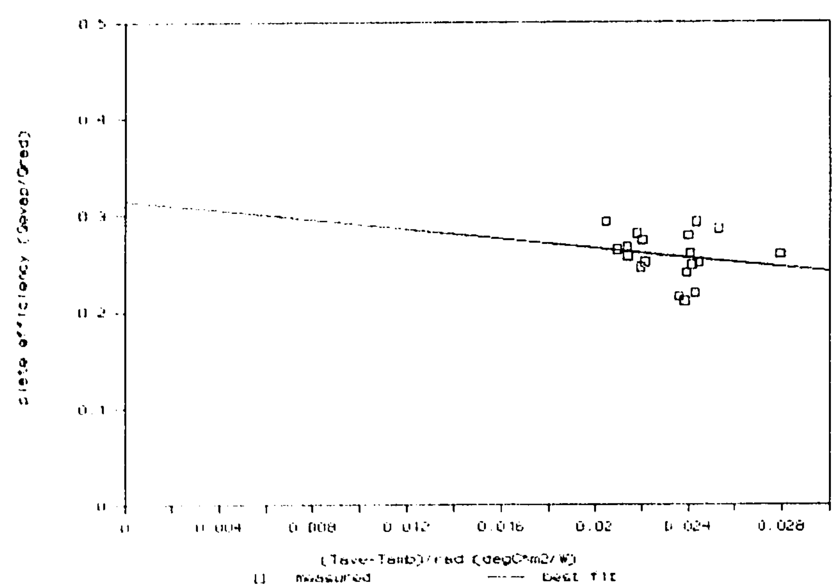

Figure 5.22 - Steady state plate Efficiency for bright sunshine/low cloud cover days in the '86-'87 heating season. Solar radiation of 700-1000 W/m^2.

collectors which are usually around 0.6 to 0.7 (Gupta and Garg (1967)). $F'U_l/L'$ is found to be 2.5 W/m^2-°C and is typical of the average solar collector plate. Fung (1983) determined these parameters for the existing collector array (on which the solar-assisted evaporator is mounted) used for the active air solar heating system operated at module six prior to the heat pump system to be 2.61 W/m^2-°C for the F_RU_L and 0.56 for the $F_R(\tau,\alpha)$ (these parameters are equivalent to the $F'U_L/L'$ and $F'(\tau,\alpha)/L'$ parameters, respectively, but are for collectors with fluids which experience no phase change). However, the correlation factor for Figure 5.22 was very poor. Because of this, the values of these parameters are not valid.

Figure 5.23 shows the temperature change at the evaporator (the amount of superheating) due to the effect of solar radiation for the '86-'87 heating season. It should be noticed that superheating occurs even at night (zero solar radiation) to a point of around 4 °C. This amount increases to almost 11 °C in the day at a radiation level of 900 W/m^2.

Figure 5.24 presents a graph of the enthalpy gain in the evaporator for the '86-'87 heating season. It appears that an increase in the solar radiation does not cause any significant changes in the net enthalpy gain. However since the refrigerant flowrate increases with the increased solar radiation, as determined previously in Figure 5.5, the amount of energy absorbed by the evaporator ultimately increases.

5.3.2 INFLUENCE OF THE COLLECTOR PLATE ON THE PERFORMANCE OF THE HEAT PUMP

In this investigation of a solar-assisted heat pump unit, it is of course logical to examine and evaluate the efficiency of the solar-assisted evaporator in improving the system performance. Was the implementation of the device to take advantage of the energy of solar radiation for heating purposes a feasible proposal? Does the increase in efficiency (if any) of the heating system justify the costs involved in the purchase and implementation of the solar-assisted evaporator? In the preceding section the collector efficiency was found. This section explores the effect of solar energy on the heating performance of the heat pump unit.

To investigate the benefits gained (if any) from the addition of solar radiation as a heat source a comparison was made between the situations where both ambient air and solar radiation are used and where only ambient air is used. For the first case, day-time data with solar radiation higher than 100 W/m^2 were employed. For the latter, night-time data were utilized. To reduce possible errors incurred by the varying ambient temperature throughout the day and night, the comparison study will only be concerned with a narrow range of ambient temperatures at a time. As before, data from the 1986-'87 heating season will be used in the analysis. Tables 5.3, 5.4, and 5.5 show the monthly averages for both day and night of the hourly heating output (**Qout**), the hourly total electrical energy consumption of module six (**Qtot**), the hourly energy consumption of the heat pump compressor (**Win**), and the resulting hourly coefficient of performance (**COP**). These parameters, with the exception of COP, are obtained by averaging the

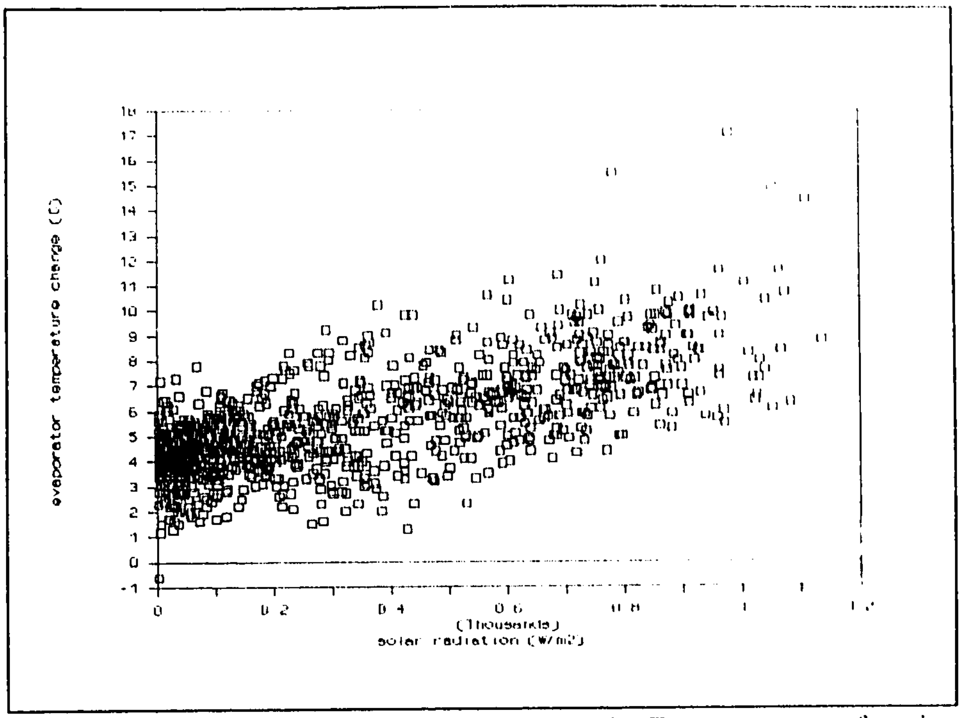

Figure 5.23 - Temperature Rise (superheat) at the Evaporator as a function of Solar Radiation for the '86-'87 heating season.

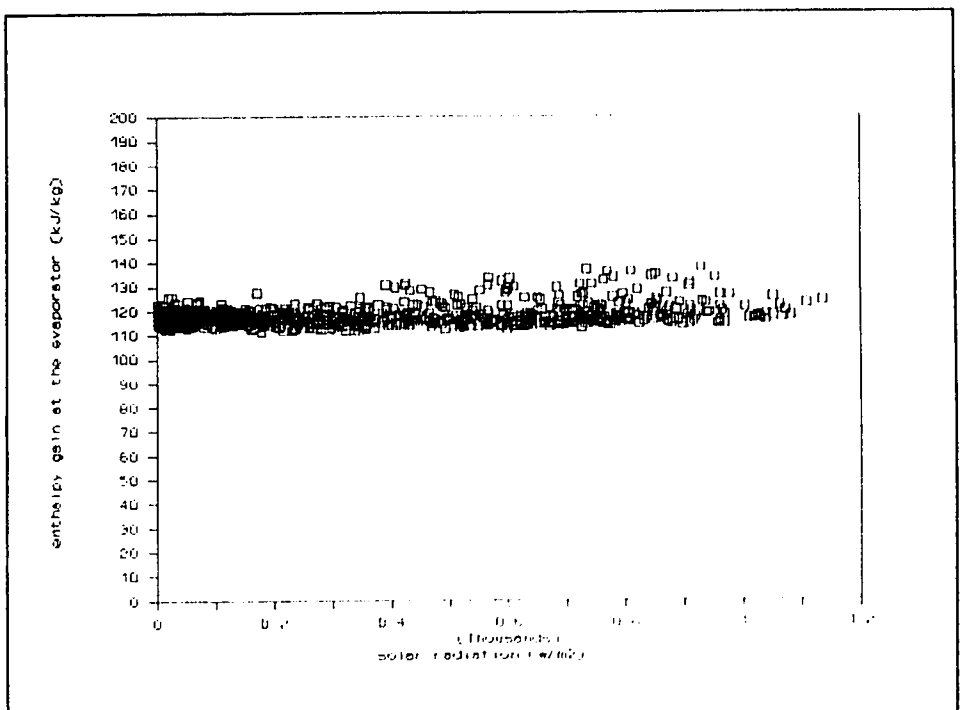

Figure 5.24 - Enthalpy Change at the Evaporator with Solar Radiation for the '86-'87 heating season.

hourly data for either day-time or night-time, of each month. For example, the day-time heating output is the average of all the hourly heating output during the day-time only, of each month. The COP is determined directly from the monthly average heating output and the monthly average energy consumption of the compressor. The parameters are listed for the ambient temperature ranges of -5 to 0 °C, -10 to -5.1 °C, and -15 to -10.1 °C respectively. All parameters (except the COP) are given in MJ per hour.

The night-time heating output of the heat pump averages lower than the day-time values. The night-time heating output appears to decrease with decreasing ambient temperature (e.g. from 5.7 MJ in Table 5.3 to 4.6 MJ in Table 5.4 to 3.7 MJ in Table 5.5 for the month of December). This indicates that during the night, as the conditions get colder the heat pump supports less and less of the heating load. This would be expected since the lower ambient temperature provides the heat pump with a lower heat energy and consequently induces the lower heating output of the heat pump. The daytime Qout on the other hand remains fairly constant with temperature since, as determined earlier, the daytime operations are mainly dictated by solar radiation.

Table 5.3 - Comparison of Day and Night Performance for -5 to 0 °C

Monthly Averages for the Ambient Temperature Range of -5 to 0 °C								
	day				night			
month	**Qout**	**Qtot**	**Win**	**COP**	**Qout**	**Qtot**	**Win**	**COP**
	MJ	MJ	MJ		MJ	MJ	MJ	
Sep'86	5.0	5.2	3.0	1.6	4.4	5.0	2.8	1.5
Oct'86	5.9	5.4	3.0	1.9	5.5	5.8	2.8	1.9
Nov'86	7.9	5.0	3.4	2.3	6.0	6.0	2.8	2.1
Dec'86	7.9	5.8	3.4	2.3	5.7	7.2	2.8	2.0
Jan'87	8.0	5.7	3.4	2.3	5.7	7.2	2.8	2.0
Feb'87	7.3	5.4	3.3	2.2	5.5	7.3	2.8	1.9
Mar'87	6.1	5.4	3.1	1.9	5.4	7.5	2.8	1.9

In the three tables shown the night-time total electrical energy consumption (Qtot) is seen to increase with decreasing ambient temperature (e.g. from 7.2 MJ in Table 5.3 to 8.9 MJ in Table 5.4 to 10.5 MJ in Table 5.5). This is also true of the day-time module energy consumption. However, it should be noted that the daytime energy consumption is consistently lower than the night-time value. It would appear that the lower day-time heating output is a consequence of the electric resistance heater being on more often during the night than in the day.

The night-time hourly compressor electrical energy required (Win) on average decreases with decreasing ambient temperature as expected (e.g. from 2.8 MJ in Table 5.3 to 2.5 MJ in Table 5.4 to 2.2 MJ in Table 5.5 for the month of December). It is also evident that the night-time Win is less than the daytime Win (e.g. 2.5 MJ compared to 3.5 MJ for December in Table 5.4). As found earlier in Figures 5.3 and 5.10, the increased amount of heat energy available in the heat sources (i.e. higher ambient temperature and solar radiation) causes the thermal expansion valve to increase the flowrate forcing the compressor to work harder and demand more electricity.

The coefficient of performance for the day-time values are consistently higher than the night-time values. The difference between the day and night COPs increase with decreasing ambient temperature from about 0.2 in Table 5.3 to 0.6 in Table 5.5 for the month of December.

Table 5.4 - Comparison of Day and Night Performance for -10 to -5.1 °C

Monthly Averages for the Ambient Temperature Range of -10 to -5.1 °C								
	day				night			
month	Qout	Qtot	Win	COP	Qout	Qtot	Win	COP
	MJ	MJ	MJ		MJ	MJ	MJ	
Sep'86	na	na	na	na	na	na	na	na
Oct'86	6.1	6.8	3.1	1.9	4.5	9.1	2.4	1.8
Nov'86	8.2	5.7	3.3	2.4	4.7	8.9	2.5	1.9
Dec'86	8.6	6.2	3.5	2.4	4.6	8.9	2.5	1.8
Jan'87	8.2	6.4	4.9	1.6	4.6	9.4	2.4	1.8
Feb'87	7.3	6.9	3.1	2.3	4.5	9.6	2.5	1.8
Mar'87	6.5	6.8	3.0	2.1	4.9	9.7	2.5	1.9

Table 5.5 Comparison of Day and Night Performance for -15 to -10.1 °C

Monthly Averages for the Ambient Temperature Range of -15 to -10.1 °C								
	day				night			
month	Qout	Qtot	Win	COP	Qout	Qtot	Win	COP
	MJ	MJ	MJ		MJ	MJ	MJ	
Sep'86	na	na	na	na	na	na	na	na
Oct'86	na	na	na	na	na	na	na	na
Nov'86	6.8	7.0	2.9	2.3	3.8	10.9	2.2	1.7
Dec'86	7.3	7.5	3.1	2.3	3.7	10.5	2.2	1.6
Jan'87	6.5	9.0	2.8	2.2	3.6	11.3	2.2	1.6
Feb'87	6.4	9.1	2.8	2.2	4.0	11.6	2.2	1.7
Mar'87	7.9	7.6	3.3	2.3	3.8	12.0	2.2	1.7

From these comparisons it can be seen that during the day, the heating output is improved, the total module electrical energy consumption is lower, and the coefficient of performance is consistently higher. Assuming that the heating load will be about the same within each temperature range and that the load will not vary significantly between the day and night it can be concluded that the combination of solar radiation and air as heat sources makes the heat pump in question operate more efficiently and effectively than if only air was available as the heat source.

Figure 5.25 shows the effect of the evaporator outlet temperature on the heating output of the heat pump. As can be expected the heating output increases with increasing evaporator outlet temperature which indicates an increase in the energy absorbed by the evaporator (due to an increased flowrate). It appears that the heating output achieves a maximum of 10 MJ per hour (which is the value found in section 5.2.1 to be the maximum output of the heat pump) at an evaporator outlet temperature of approximately 10 °C. This levelling off characteristic indicates the limiting of the flowrate by the suction pressure regulator.

It was found that a rise in solar radiation generally causes a rise in the amount of energy absorbed by the evaporator (**Qevap**). Consequently Figure 5.26 shows that the coefficient of performance of the system increases with the energy absorbed. The COP rises from about 1.3 at low Qevap to about 2.6 at 7 MJ per hour of Qevap. It would seem that the solar-assistance employed by the heat pump improves the performance of the system.

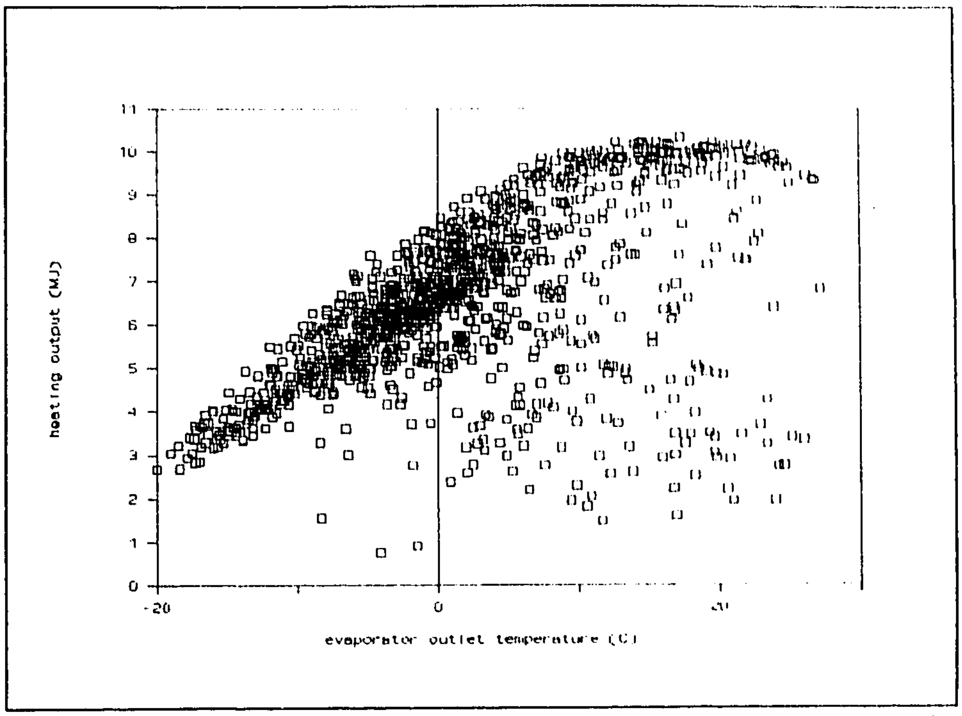

Figure 5.25 - Daytime effect of the Evaporator Outlet Temperature on the Heat Pump Heating Output for the '86-'87 heating season.

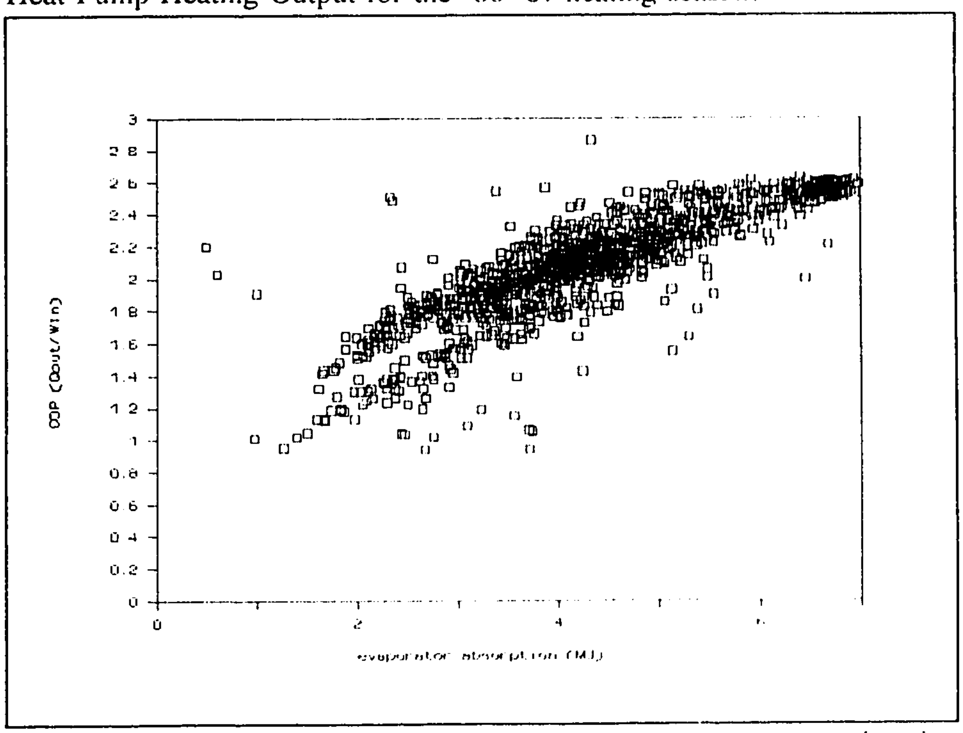

Figure 5.26 - Coefficient of Performance of the Heat Pump as a function of the Amount of Energy Absorbed by the Evaporator for the '86-'87 heating season during the day time.

5.3.3 COLLECTOR PLATE SURFACE AREA

In section 5.2.5 it was determined that the evaporator collector plate was oversized for the compressor. A reduction in the excess surface area to better suit the maximum 0.016 kg/s flowrate would reduce costs and decrease heat losses from the evaporator plate. However, decreasing the plate size might also decrease the night-time energy absorption. The determination of the optimum surface area for the system is beyond the scope of this study.

5.4 SYSTEM ENERGY CONSUMPTION

Ultimately the most important factor in the evaluation of any heating system is the amount of energy required for the system to provide sufficient heat to the designated space. This section evaluates the heat pump performance by comparing its energy consumption to those of the following cases:

- module five which only utilizes an electric resistance heater
- its own electric resistance heater operating without the heat pump

5.4.1 COMPARISON TO MODULE 5

Comparisons can be made between the energy quantities of module six to those of module five based on the assumptions presented in Chapter 4. Table 5.6 lists the various parameters for each month of the study period from September '86 to March '87.

TABLE 5.6 - ENERGY PARAMETERS FOR MODULES FIVE AND SIX

	Qt6	Qt5	Win	Qout	%Win	Qt6/Qt5
	MJ	MJ	MJ	MJ	Win/Qt6	%
Sep'86	902	1412	520	871	57.7	63.88
Oct'86	2128	3476	1127	2176	52.9	61.23
Nov'86	4021	5781	1344	2683	33.4	69.56
Dec'86	5014	7249	1769	3595	35.2	69.17
Jan'87	5948	7899	1973	3998	33.1	75.30
Feb'87	4052	5402	1385	2754	34.1	75.01
Mar'87	4862	6451	1716	3392	35.3	75.36
summary	26930	37674	9836	19471	36.5	71.48

In Table 5.6 **Qt5** and **Qt6** are the total electrical energies consumed by modules five and six, respectively for each month, **Win** is the amount of energy used by the compressor, **Qout** is the heating output of the heat pump, and **%Win** is the percent of the total electical energy of module six used by the heat pump for each month.

$$\% Win = \frac{Win}{Qt6} \times 100 \qquad (5.2)$$

Finally, **Qt6/Qt5** is the portion of the total electrical energy consumption of module five that was needed by module six for the corresponding month:

$$(Qt6/Qt5) = \frac{Qt6}{Qt5} \times 100(\%) \qquad (5.3)$$

The last row shows the corresponding totals and averages for each parameter for the entire period. Note that the total electrical energy consumption for module five is much higher than that of module six for the entire period and also for each month. Figure 5.27 illustrates the cumulative energy consumptions of the two modules for the '86-'87 heating

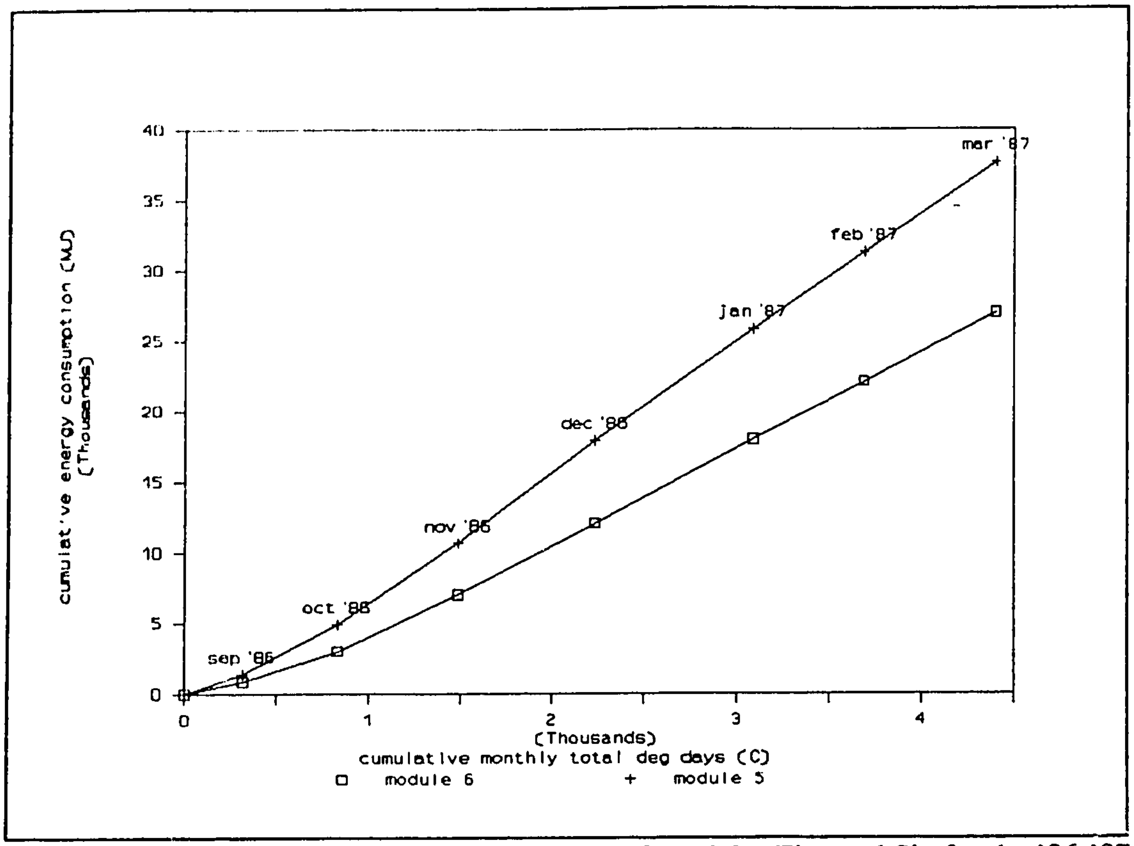

Figure 5.27 - Cumulative Energy Consumptions of Modules Five and Six for the '86-'87 heating season

season as a function of the cumulative total monthly degree days (calculated with a 21 °C base). With the aid of the heat pump, module six required about 28 percent less electrical energy (26,930 MJ) for the seven months than module five (37,674 MJ). However, of the total electrical energy consumed by module six, only about 37 percent (9,836 MJ) of it was used by the heat pump. As mentioned, the amount of heating required by the module **(Qload)** can be expressed as:

$$\begin{aligned} Qload &= Qout + Qloss + Qaux \\ &= (COP \times Win) + Qloss + Qaux \end{aligned} \tag{5.4}$$

where **Qloss** is the heat energy lost to the surroundings at the compressor and **Qaux** is the electrical energy required by the auxiliary heater and the other electrical equipment in the module (the other parameters are as defined earlier). It is evident that for a given heating load, energy can be saved (and therefore costs would be less) if more of the electrical energy was directed to the heat pump (ideally all of it - this would only be possible with a larger system). Note that the months of September and October have the highest percentage of the total module electrical energy used by the heat pump (at 58% and 53% respectively) and show the lowest values of **Qt6/Qt5** in the table. However, the costs associated with a compressor large enough to handle the entire heating load make this an unattractive alternative.

Table 5.7 shows the total monthly energy consumptions of module six for the entire three years. It can be noted that certain months such as May, June, and July of 1986 do not contain values. Data for these months are not available due to equipment failure, maintenance, or systems shutdown which all produce erroneous recordings. This is also true of a lot of the days in 1988 and is the reason for the low values obtained for that year.

TABLE 5.7 - MONTHLY ENERGY USE FOR MODULE 6

	'85		'86		'87		'88	
	Qtot	Win	Qtot	Win	Qtot	Win	Qtot	Win
JAN			4378.4	1405.8	5948.1	1973.0	15.1	5.9
FEB			3598.9	934.7	4052.4	1385.3	109.1	57.4
MAR			2415.8	1080.4	4862.1	1716.6	500.6	280.0
APR			3112.0	1576.3	2114.3	1354.4	775.5	429.8
MAY					913.2	550.5	605.9	352.2
JUN					231.9	68.5	357.7	184.0
JUL					74.5	48.4	393.3	196.2
AUG			214.3	123.4			171.1	93.9
SEP			902.2	520.5	218.6	132.8		
OCT			2128.8	1127.5	44.4	28.2		
NOV	3053.9	749.7	4021.9	1344.1				
DEC	4044.8	1402.5	5014.7	1769.5				

* all values in MJ

5.4.2 COMPARISON TO ITS OWN ELECTRIC RESISTANCE HEATER

An even better comparison can be made to illustrate the effect of the addition of the heat pump to the module. From the 24th of January till the 5th of February, 1986, the heat pump was taken off line. The entire heating load for this period was assumed by the module's auxiliary heating system. The data collected can therefore be studied to provide information regarding the electrical energy consumption of the auxiliary heater acting alone. By comparing this to the heat pump system's (including the electrical resistance heater) electrical energy consumption the amount of electricity saved (if any) with the addition of the heat pump unit can be determined.

Table 5.8 shows the electrical energy consumptions for the two aforementioned cases. For consistency the values used for the heat pump system's case are taken from conditions which are somewhat similar to the electric resistance heater's case (in terms of ambient temperatures). It is assumed that due to the sun's low elevation during the winter months, solar radiation will not have a significant effect on building heating and will therefore not be a factor in this comparison. For the periods of study, both cases experience (on a 21 °C base) 367.2 heating degree days.

Table 5.8 - Comparison of the Heat Pump System's Energy Consumption to that of the Electric Resistance Heater

	auxiliary only ('86)		hp and aux heat		both/ aux
Tamb	**date**	**Qta**	**date**	**Qtb**	**Qtb/ Qta**
°C		MJ		MJ	%
-3.4	Jan 24	258.3	Dec 18'86	168.9	65.4
-6.9	25	275.0	Feb 01'87	199.7	72.6
-4.3	26	265.7	Jan 07'86	185.2	69.7
0.3	27	225.0	Jan 17'87	143.7	63.9
-8.3	28	274.2	Mar 07'87	199.0	72.6
-11.6	29	306.3	Jan 03'86	235.1	76.8
-10.4	30	311.8	Jan 06'86	238.9	76.6
-6.1	31	273.5	Jan 01'87	208.8	76.3
-9.1	Feb 1	276.0	Mar 11'87	247.1	89.5
-3.8	2	260.4	Jan 02'87	169.1	64.9
-8.3	3	265.7	Mar 07'87	199.0	74.9
-11.4	4	288.9	Jan 04'86	233.7	80.9
-11.0	5	308.2	Feb 12'86	248.4	80.6
sum		3589.1		2676.5	

where:

Tamb = the average ambient temperature.

Qtb/Qta = total energy consumption of the heat pump and the auxiliary heater divided by the total energy consumption of the auxiliary heater acting alone.

The **auxiliary only** column contains the data for the period where only the electric resistance heater was in operation (no heat pump) (Qta). The **hp and aux heat** column contains the data for the period where both the heat pump unit and the auxiliary heater were in operation (**Qtb**). The final column, **both/aux**, is the total energy consumption of the heat pump and the auxiliary heater (**Qtb**) divided by the total energy consumption of the auxiliary heater acting alone (**Qta**) for the respective days. The last row of the table shows the sums of the total energy consumptions for the two cases. The heat pump's addition clearly yields a reduction in the electrical energy consumption at an average of about 75 MJ less per day. In the **both/aux** column it can be seen that, at higher ambient temperatures (around 0 °C), if the heat pump is used, only about 65 percent of the electricity required for the auxiliary heater case is needed. At lower ambient temperatures this figure rises to roughly 80 percent because the heat pump is unable to cope with the higher heating load and has to share it with the auxiliary heater.

Figure 5.28 illustrates the effect of ambient temperature on the total electrical energy consumption of the auxiliary heater acting alone and the heat pump system. The graph also gives a clearer view of the difference between the two cases and shows that the difference decreases with lower temperatures. This is due to the increase in the auxiliary heater activity as the heat pump is unable to support the increasing heating load with decreasing ambient temperatures. The non-linearity of the graphs is probably due to wind and weather conditions which were not measured and not taken into account in this study.

5.5 SYSTEM IRREVERSIBILITIES

In any thermodynamic system irreversibilities are present. As mentioned in

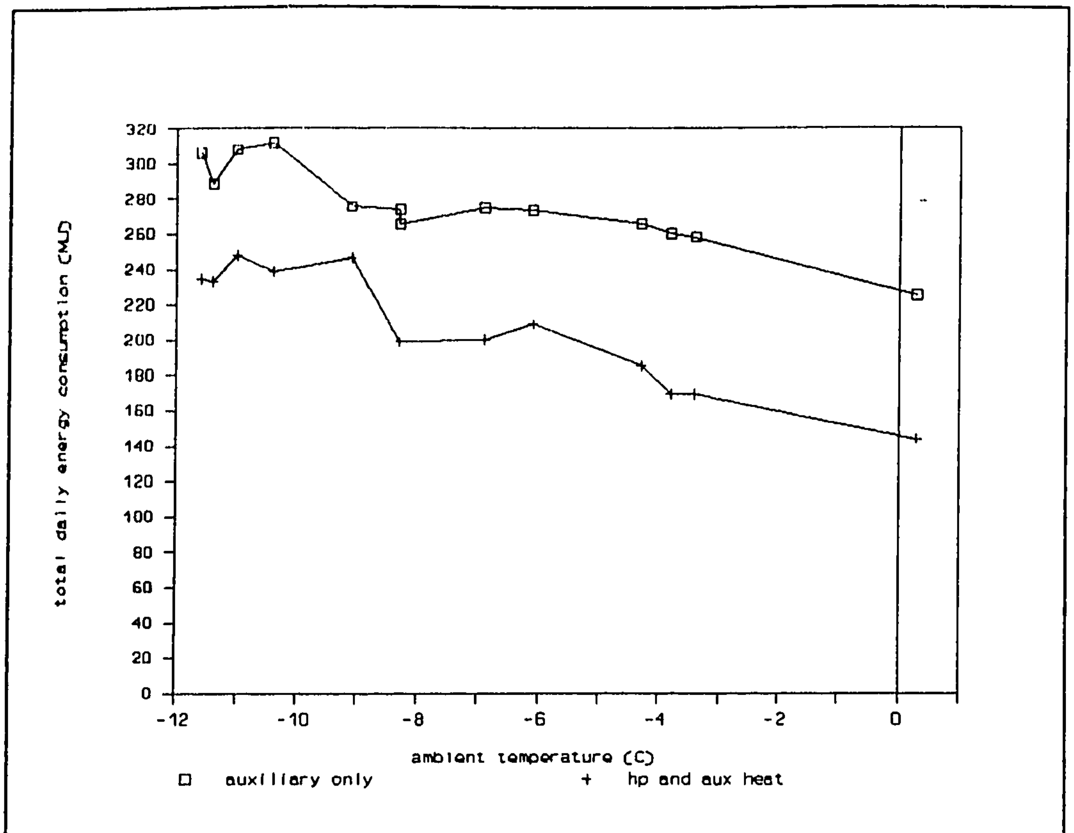

Figure 5.28 - Effect of Ambient Temperature on the Total Daily Electrical Energy Consumption of Module Six for the Auxiliary Heater Only and for the Heat Pump and Auxiliary Heater for selected days in '86 and '87

chapter 3, irreversibilities in a vapor compression cycle result from non-isentropic processes caused by friction of the flowing fluid along the pipe walls and heat transfer through a finite temperature difference. Determination of the system irreversibilities provides a method of evaluating the various components of the heat pump unit.

The program developed computes the irreversibility of each of the four main components and the lines connecting them. For the analysis, average irreversibility values for the month of January '87 were used. Figure 5.29 shows the average hourly values for each component and pipe along with their respective percentages of the total

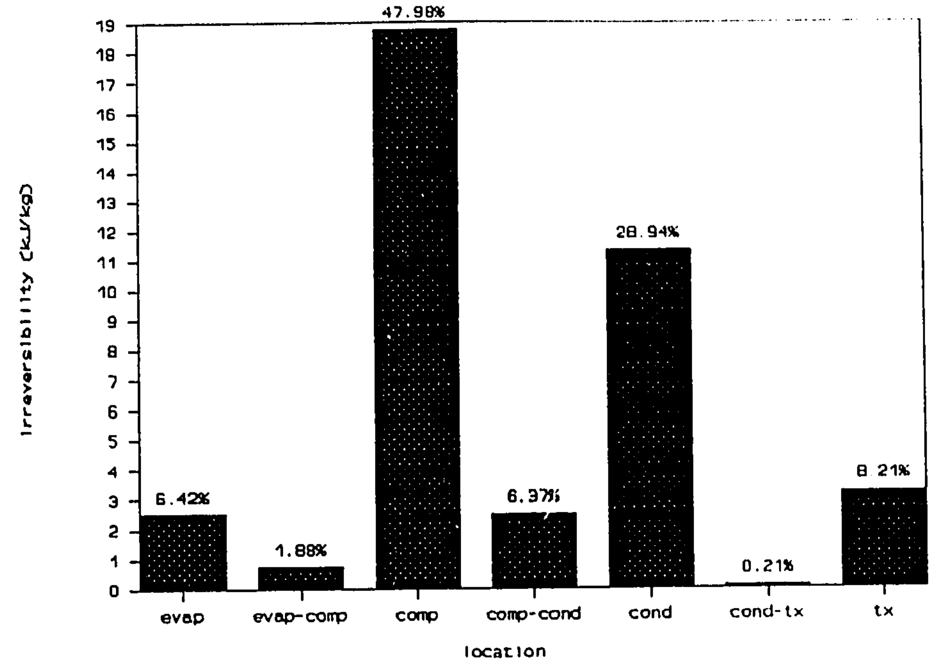

Figure 5.29 - Average Hourly System Irreversibilities in kJ/kg and percentages for January '87

irreversibility. Note that the largest irreversibility (48%) occurs at the compressor. This is probably due to the fact that the refrigerant fluid was passed over the coils as a coolant.

The next highest irreversibility occurs at the condenser at 29%. As would be expected, the irreversibility here is mainly caused by the heat transfer from the refrigerant to the air although some wall friction may also be present.

The thermal expansion valve (8%) and the evaporator (6%) are the other main sources of system irreversibilities. Irreversibility is produced in the thermal expansion valve as a result of unrestrained expansion and in the evaporator due to heat transfer across a finite temperature difference. Note that the line from the thermal expansion valve to the evaporator is considered as part of the evaporator due to lack of information at the evaporator inlet. The lines from the compressor to the condenser and from the evaporator to the compressor also have some irreversibilities at 6% and 2%, respectively.

5.6 ECONOMIC ANALYSIS OF THE SYSTEM

In any engineering study, it is of course vitally important to investigate the economics of the matter (Bessler and Hwang (1980)). For the analysis the September '86 to March '87 period is used. Fuel prices are for 1993 and are:

- price of electricity = \$0.0636 per kWh = \$17.67 per GJ
- price of natural gas = \$2.639 per GJ

The electric resistance heater and the solar-assisted heat pump are powered by electricity. However, the price of natural gas is required so that a comparison between the heat pump and the natural gas furnace can be performed. For the natural gas furnace a midsize model with an efficiency of 75% is assumed. Installation costs are not included.

A comparison between the solar-assisted heat pump and a heat pump which is not solar-assisted (i.e. does not use solar radiation as a heat source, only uses ambient air) was also performed. The COP of the latter was found to average at about 1.8 (see section 5.2.3).

From the months of September to March the total number of hours is 5088. According to Table 2.1 these 7 months provide 4793 heating degree Celsius days in Edmonton. Assuming the UA factor (heat loss coefficient) of module six as 143 W/°C

(Gilpin et al. (1980)) the total amount of heating required for the module would be 59.2 GJ. At current electricity prices the cost of heating for these months with an electric resistance heater is $1,046.20. In contrast only 3426.2 hours of measured data are available for the period of September '86 to March '87. As a result only 40.4 GJ of heating is required for the period. Table 5.9 shows the economic comparison of five different heating systems: the solar-assisted heat pump system (including auxiliary heater), the electric resistance heater, the natural gas furnace (efficiency 75%), the solar-assisted heat pump only, and a heat pump without solar-assistance.

Table 5.9 - Heating Costs of Five Different Systems for the Sept'86 - Mar'87 period

type of heating	total cost for period	cost per GJ
solar-assisted heat pump system	$384.96	
electric resistance	$714.53	$17.67 /GJ
natural gas furnace	$142.31	$3.52 /GJ
solar-assisted heat pump only	-	$8.93 /GJ
heat pump w/o solar-assistance	-	$9.82 /GJ

The solar-assisted heat pump system results in quite a significant saving when compared to the electric resistance heater. It costs just over half as much to heat with the heat pump system for the period. Current gas prices in Alberta however make the natural gas furnace the best choice by far at only $3.52 /GJ, assuming a mid-size gas furnace with a 75% efficiency, a saving of $242.65 compared to the heat pump system and $572.22 compared to the electric resistance heater for the entire period. It is evident that the addition of solar-assistance to the heat pump improves savings (by 89 cents per GJ - comparing the heat pump with solar-assistance to the heat pump w/o solar-assistance) and COP (by 0.18). The cost of the solar-assisted evaporator is approximately $2740. Based on expected yearly heating requirements (5704 heating degree days for Edmonton (see section 2.2.1)) and assuming an initial cost of a normal evaporator with fan as $400 and a 1.5% inflation rate, it would take approximately 30 years for the solar evaporator to pay for itself. Therefore although the COP is improved with the addition of the solar evaporator the long payback period of the solar evaporator does not make the implementation a feasible one.

If the annual amount of heating required for module six is calculated assuming 5704 degree days (see Table 2.1) per year, a heating load of 70.474 GJ per year would be expected. For four heating systems the per annum heating costs at current fuel prices would be:

- electric resistance heater = $1,245 per year
- solar-assisted heat pump unit = $629 per year
- natural gas furnace = $247 per year
- heat pump unit w/o solar-assist = $692 per year

Again, if one assumes a 1.5% inflation rate and an initial cost of $4000 for the heat pump, it would take 6 years for the heat pump system to pay for itself as a result of the savings in electricity. This makes the solar-assisted heat pump system a more feasible idea compared to the electric resistance heater. However, the heat pump's inability to operate at ambient temperatures of -23 °C and below, a condition which is not unusual in Alberta, puts it at a disadvantage against the natural gas furnace. Therefore, until the price of natural gas in Alberta increases substantially, the heat pump system will not be a practical alternative to the natural gas furnace for heating purposes in the region.

5.7 PRESENTATION OF DATA

Data for the heat pump's operation are tabulated and shown in the appendix for the '86-'87 heating season (for the months of September '86 to Mar '87). The data are presented in a daily format for each month. These values are derived from the hourly format which was used in the majority of the previous sections. The various parameters are as follows:

(a) **DATE** - the date (mm-dd-yy)

(b) **RAD** - the sum of the solar radiation for the day (W/m^2)

(c) **TA** - the average ambient temperature for the day (°C)

(d) **FR12** - the average refrigerant fluid mass flowrate for the day (kg/s)

(e) **THP** - the total number of hours the heat pump was in operation for the day (hrs)

(f) **QOUT** - the total heat pump heating output for the day (MJ)

(g) **QIN** - the total amount of energy gained from the compressor during the day (MJ)

(h) **QTOT** - the total electrical energy consumed by the module (module six) during the day (MJ)

(i) **WIN** - the total electrical energy required by the heat pump compressor during the day (MJ)

(j) **QEVAP** - the total amount of energy absorbed by the heat pump evaporator during the day (MJ)

(k) **QAUX** - the total electrical energy required by the auxiliary heater and the other electrical appliances (such as the data acquisition equipment, measuring devices, lights, etc.) during the day (MJ)

(l) **QLOAD** - the total amount of heat energy required for heating the module per day (module six)

(m) **COP** - the average coefficient of performance of the heat pump for the day **(QOUT/WIN)**

Zero values for all the parameters (except the date) on any given day means data for the heat pump was either missing or the heat pump was off-line. A monthly total of the daily values is given in the **SUMMARY** row at the bottom of the table. The daily format was chosen to allow for a clear and concise style of data presentation. Although the hourly format would convey more information (especially in regards to the solar-assisted evaporator), it would be unsuitable for presentation purposes. It would require an immense amount of space and would ultimately be overly disarrayed and confusing. For clarity and ease in presentation and reference the daily values are grouped into months.

5.8 SUMMARY OF RESULTS

Based on the '86-'87 heating season hourly data (Sept'86 - March'87), the performance of the solar-assisted heat pump which used ambient air and solar radiation as heat sources was evaluated. Solar radiation dictated the performance of the heat pump during the day and ambient temperature dictated the performance during the night. Increasing solar radiation improved the amount of absorption at the evaporator, the heating output of the heat pump, and the coefficient of performance of the unit while decreasing the total electrical energy consumption of module six. For the period of study, the total electrical energy consumption of module six averaged at 5.5 MJ per hour at high solar radiation values and about 10 MJ per hour at low or no solar radiation. Increasing the ambient temperature has the same effect as solar radiation.

Maximum performance (highest heating output and coefficient of performance) occurs during the day-time when solar radiation is 600 W/m^2 and above. At this condition the refrigerant flowrate is at a maximum of 0.016 kg/s and led to the maximum heating output of 10 MJ per hour, the maximum absorbed energy at the evaporator of 6.8 MJ per hour, the maximum coefficient of performance of 2.6, and an energy consumption of the compressor of 4 MJ. The night-time maximum hourly performance occurred at an ambient temperature of 4 °C when the refrigerant flowrate was 0.012 kg/s and were: heating output of 8 MJ, absorbed energy at the evaporator of 5 MJ, coefficient of performance of 2.3, and an energy consumption of the compressor of 3.5 MJ. The minimum hourly values recorded for the period of study occurred at an ambient temperature of -19 °C and were 2.5 MJ of heating output, 1.7 MJ of energy absorption at the evaporator, and a coefficient of performance of 1.4.

The lower operating limit of the heat pump was found to be when the compressor inlet pressure dropped below approximately 110 kPa. This corresponded to an ambient temperature of -23 °C. The upper operating limit could not be determined but the compressor was safeguarded from inlet pressures exceeding 300 kPa by the suction pressure regulator. System cycling (the process of the heat pump turning on and off) occurred at an ambient temperature of about 4 °C. Between ambient temperatures of 0 and 4 °C the heat pump could handle the full heating load. Below 0 °C the heat pump had to operate continuously and required the intermittent aid of the auxiliary heater. From 0 to -8 °C the auxiliary heater only had to operate for no more than 12 minutes per hour and the heat pump could still support part of the load. Below -23 °C however, the heat pump's operating limit was reached and the heating load had to be handled solely by the auxiliary heater.

For the '86-'87 heating season, the average seasonal coefficient of performance for the heat pump was found to be 2.0. The average coefficient of performance of the heat pump unit during the day-time for the same period was found to be 2.1 (calculated only with the day-time values). Similarly the average coefficient of performance of the heat pump unit during the night-time for the same period was 1.8. This value was used to approximate the coefficient of performance of a heat pump unit which does not have solar-assistance. This hypothetical heat pump without solar-assistance is assumed to have an evaporator which exhibits a similar characteristic to the solar collector in the absence of solar radiation.

An analysis showed that the main irreversibilities found in the heat pump unit were at the compressor and the condenser. These values were 48% and 29%, respectively.

A performance evaluation of the solar evaporator collector showed that there were three conditions experienced by the component:

:night - heat energy absorbed by convection from ambient air only

:day (solar radiation below 200 W/m^2) - heat energy absorbed by convection and radiation from solar radiation and ambient air.

:day (solar radiation above 200 W/m^2) - heat energy absorbed by radiation from solar radiation only. Heat loss from collector mainly by convection to ambient air.

The ficiency of the plate was evaluated only for the third case (absorption from solar radiation only). The collector efficiency ranged from 0.2 at 1000 W/m^2 to 0.6 at 300 W/m^2. Evaluation of the efficiency based on the methods suggested by the ASHRAE STANDARD (1977) was performed for solar radiation levels between 700 and 1000 W/m^2 from days with bright sunshine and low cloud cover. The $F'(\tau,\alpha)/L'$ and $F'U_l/L'$ (collector loss) terms were determined from the efficiency curve as 0.3 and 2.5 $W/m^{2\circ}C$, respectively. However, due to poor correlation these parameters were not valid. The collector efficiency for the case where the solar radiation was below 200 W/m^2 was not determined because the plate temperature was not measured and any results would be suspect at best. For the night-time case the collector efficiency is assumed to be approximately 1. The plate heat transfer coefficient was determined and found to average around 230 W/°C. During the night, superheating of the refrigerant fluid was about 4 °C. During the day the maximum superheating recorded for the period of study was 11 °C (corresponding to 900 W/m^2).

The heat pump electrical energy consumption was evaluated with two comparisons. The first showed that with the heat pump, module six needed 28 percent less electricity than the module heated by an electric resistance heater similar to the heat pump's auxiliary heater. In addition only 37 percent of the total module's electrical consumption was used by the heat pump unit. The second comparison was based on the heat pump system's own electric resistance heater and showed a saving of 75 MJ less electricity per day with the use of the heat pump unit. At high ambient temperatures the heat pump needed only 65 percent of the total electricity used by the resistance heater to provide the same amount of heating and 80 percent at low ambient temperatures. Certainly the results showed that the heat pump has a significant advantage over the resistance heater.

CHAPTER 6

CONCLUSIONS

The limiting factor for the performance of the heat pump unit was found to be the suction pressure regulator. The compressor was the component that was ultimately responsible for the performance limit of the heat pump. It can also be said instead that the evaporator was oversized for the compressor. The thermal expansion valve controlled the flowrate of the refrigerant below an evaporator outlet pressure of 0.3 MPa.

An economic analysis of the heat pump system was performed based on current fuel prices in the Alberta region of $17.67 per GJ of electricity and $2.64 per GJ of natural gas. For the period of study the total cost of heating with the heat pump system (which includes the auxiliary heater) was found to be approximately $3855. Compared to the electric resistance heater this yielded a saving of $330. However compared to the natural gas furnace the heat pump system costs $243 more to operate.

Comparing the per GJ costs of the solar-assisted heat pump unit ($8.93 /GJ) to those of a heat pump unit without solar-assistance ($9.82 /GJ) showed a saving of 89 cents per GJ of heat. This is to be expected since solar-assistance was found to improve the system's performance by increasing absorbed energy and therefore the heating output and the coefficient of performance of the heat pump. However, based on these prices and a 1.5% inflation rate, the solar collector would take 30 years to pay for itself in comparison to a normal evaporator with fan (which exhibits a similar characteristic to the solar-assisted collector in the absence of solar radiation). This makes solar-assistance for this particular unit (and in this particular region) unfeasible. Excluding installation costs, the solar-assisted heat pump compared to the electric resistance heater (also assuming a 1.5% inflation rate) would take 6 years to pay for itself and is therefore feasible. But until the price of natural gas in Alberta becomes much higher, the heat pump will not be a practical alternative to the natural gas furnace.

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APPENDIX A

A.1 HEAT PUMP COMPONENTS PRICE LIST

The following is a list of the approximate prices of the main components in the solar-assisted heat pump under investigation and does not include labor and piping. This list was obtained from a student report done in 1986.

component		price
solar collectors (0.9 m x 2.4 m) (Amteck)		4 x $684.58
multistage thermostat (Honeywell T872)		$80.79
condenser		$100.00
receiver dryer (Dayton)		$29.36
sight glass and moisture indicator (Parker)		$17.80
compressor (3500 RPM, hermetically sealed)		$360.00
rotolock valve		$37.17
commercial dual pressure controls (Dayton)		$105.90
temperature control (Dayton)		$64.80
thermostatic expansion valve (Parker)		$77.50
liquid line filter/dryer (Parker)		$10.90
suction line filter/dryer (Parker)		$36.40
refrigeration solenoid valve (Parker)		$44.90
40 VA control transformer		$8.01
contactor		$36.25
accumulator		$100.00
toggle switch		$4.50
	total:	**$3852.60**

A.2 COEFFICIENT OF PERFORMANCE

- the coefficient of performance is calculated as:

COP = heating output / work input to compressor

- the COP can be determined for the heat pump and a theoretical heat pump without solar-assistance.

<u>(a) COP of the solar-assisted heat pump</u>

- the COP of the solar-assisted heat pump will be determined based on the measured data for the September '86 to March '87 period.

COP solar heat pump = (total heating output of heat pump) / (total work input to compressor)

- for the period the total heating output of the heat pump was found to be = 19471.8 MJ and the total work input to the compressor was = 9836.5 MJ

therefore,

COP solar heat pump = **1.98**

<u>(b) COP of a heat pump without solar-assistance</u>

- the COP of a heat pump without solar-assistance can be approximated as the COP of the unit during the night. From Figure 5.7 in Chapter 5, the night-time COP (solar radiation = 0 W/m^2) appears to range from 1.3 to 2.3. The COP can be approximated as the average of these values:

COP heat pump without solar-assist = **1.8**

A.3 ECONOMIC ANALYSIS

The economic analysis performed here is based on fuel prices of 1993. The period from September '86 to March '87 is used for the analysis except for the determination of the payback period which is calculated using annual costs. In calculating the theoretical heating cost the values for the heating degree days in Edmonton for the corresponding heating months are obtained from Duffie and Beckman (1980) (see section 2.2.1).

<u>(a) Module Six Heating Cost - with Electric Resistance Heater only (using UA factor)</u>

from Table 2.1 in Chapter 2:

- total theoretical heating degree days (Celcius) for the months of Sept-Mar

= 4793 degree days

from Gilpin et al (1980):

- measured heat loss coefficient of module six (UA) = 143 W/°C

{total amount of heating required for period of September to March}

= UA x heating degree days x 24 hrs

= 143 x 4793 x 24 = 16449576 Wh

= 16449.58 kWh

- cost of electricity at the time of writing = $0.0636 /kWh

= $17.67 /GJ

(obtained from Edmonton Power)

{total cost of heating for period of September to March}

= 16449.58 kWh x \$0.0636 /kWh

= **\$1,046.20**

<u>(b) Module Six Heating Cost - with Electric Resistance Heater only (using measured data)</u>

- heating load = heat output of heat pump + heat output of electric resistance heater

  + heat loss from compressor

- the actual heating load of module six for the period of September '86 to March '87

  = 40445 MJ = 40.445 GJ

- cost of heating = 40.445 GJ x \$17.67 /GJ

  = **\$714.53**

- cost per GJ = **\$17.67 /GJ**

<u>(c) Module Six Heating Cost - with the Solar-Assisted Heat Pump System</u>

(note: the solar-assisted heat pump system includes the electric resistance heater as the auxiliary heater)

- actual total electricity consumption of the module for the Sept'86 to Mar'87 period

  = 26.93 GJ

- approximate electricity consumption of the other equipment (computer, lights, etc) in the module not for heating purposes per hour (see section 5.2.1)

  = 1.5 MJ/hr

- for total period (3426.2 hours of data measured = 142.76 days)

= 1.5 MJ/hr x 3426.2 hrs

= 5139 MJ = 5.14 GJ

- actual electricity consumption of the heating system

= (total electricity consumption) - (electricity consumption of non-heating equipment)

= 26.93 GJ - 5.14 GJ

= 21.79 GJ

- cost of heating for the system = 21.79 GJ x $17.67 /GJ

= $384.96

<u>(d) Module Six Heating Cost (per GJ) - with the Solar-Assisted Heat Pump only</u>

- the heat pump's average COP for the period = 1.98

- cost per GJ for the heat pump only

= (cost of electricity per GJ) / COP

= ($17.67 /GJ) / 1.98

= $8.93 /GJ

<u>(e) Cost of Heating Module Six with a Natural Gas Furnace only</u>

- cost of natural gas at the time of writing = $2.639 /GJ

(obtained from Northwestern Utilities)

- using the actual heating load for the September '86 to March '87 period (= 40.445 GJ) and assuming a midsized Natural Gas Furnace with an efficiency of 75%

{total cost of heating module six for the period with a natural gas furnace}

= (40.445 GJ x \$2.639/GJ) / 0.75

= **\$142.31**

- cost per GJ = **\$3.52 /GJ**

(f) Cost (per GJ) of Heating Module Six with a Heat Pump without Solar-Assistance

- the COP of a heat pump without solar-assistance = 1.8

- the cost per GJ of using the heat pump without solar-assistance would be:

= (cost of electricity per GJ) / COP

= (\$17.67 /GJ) / 1.8

= **\$9.82 /GJ**

(g) Cost of Heating Module Six per year (using UA factor)

from Table 2.1 in Chapter 2:

- total number of heating degree days (Celsius) per year

= 5704 degree days

- total heating required per year

= 143 W/°C x 5704 degree days x 24 hrs

= 19576128 Wh

= **19576.128 kWh = 70.474 GJ**

(i) for the **solar-assisted heat pump:**

- cost per year

= heating required per year x cost of heating with heat pump

= 70.474 GJ x $8.93 /GJ

= **$629.33 per year**

(ii) for the **electric resistance heater:**

- cost per year = heating required per year x cost of electricity

= **$1,245.00 per year**

(iii) for the **natural gas furnace:**

- cost per year

= (heating required per year x cost of natural gas per GJ) / furnace efficiency

= (70.474 GJ x $2.639 /GJ) / 0.75

= **$247.97 per year**

(iv) for a **heat pump with no solar-assistance**

- cost per GJ for the heat pump w/o solar-assistance = $9.82 /GJ

- cost per year

= (heating required per year) x (cost of heating with heat pump w/o solar)

= 70.474 GJ x $9.82 /GJ

= **$692.05 per year**

<u>(h) Calculation of the Payback Period of the Solar Evaporator</u>

- assuming: - 1.5% inflation rate:

- initial cost of solar evaporator = $2738.32

- average initial cost of a normal evaporator and fan = $400

- therefore difference in initial cost = $2338.32

TABLE A.1 - Payback period of the solar evaporator

year #	cost per year		difference per year	cumulative difference
	hp (w/o solar)	hp (solar)		
1	$692.05	$629.33	$62.72	$62.72
2	$702.43	$638.77	$63.66	$126.38
3	$712.97	$648.35	$64.62	$191.00
4	$723.66	$658.08	$65.58	$256.58
5	$734.52	$667.95	$66.57	$323.15
6	$745.53	$677.97	$67.57	$390.72

at this rate the savings incurred with the use of a solar evaporator will be more than the difference between the initial costs of the two types of evaporators ($2338.32) in 30 years (ie the solar evaporator pays for itself in 30 years).

(i) Calculation of the Payback Period of the Heat Pump

- the payback period for the heat pump in comparison to an electric resistance heater system can be calculated with the following assumptions:

  - the initial cost of the heat pump unit (from appendix A.1) = $3852.60

  - the initial cost of an electric resistance heater = $500

    (therefore difference = $3852.60 - $500 = $3352.60)

  - an annual inflation rate of 1.5%

TABLE A.2 - Payback period of the heat pump

year #	cost per year		difference per year	cumulative difference
	resistance heater	solar heat pump		
1	$1,245.00	$629.33	$615.67	$615.67
2	$1,263.68	$638.77	$624.91	$1,240.58
3	$1,282.63	$648.35	$634.28	$1,874.86
4	$1,301.87	$658.08	$643.79	$2,518.65
5	$1,321.40	$667.95	$653.45	$3,172.10
6	$1,341.22	$677.97	$663.25	$3,835.35
7	$1,361.34	$688.14	$673.20	$4,508.55

The energy savings of the solar-assisted heat pump compared to the electric resistance heater results in the surpassing of the difference in the initial costs of the heat pump unit and the electric resistance heater ($3352.60) by the sixth year. Therefore, the heat pump pays for itself within 6 years.

A.4 PROGRAM FUNCTIONS

In Chapter 3 some of the program's calculation procedures were discussed. The main functions (ie the equations) used in the determination of the state point properties of the refrigerant in the heat pump circuit are obtained from Downing (1974). There are 6 main subroutines in the program:

- VAPOR - calculates the thermodynamic properties of superheated refrigerant given temperature and pressure
- SCOL - calculates the thermodynamic properties of subcooled liquid refrigerant given temperature and pressure
- SATPRP - calculates the saturation thermodynamic properties of a refrigerant given the saturation temperature and pressure
- SPVOL - a function that determines the specific volume of superheated refrigerant given temperature and pressure
- PSAT - a function that determines the saturation pressure of a refrigerant given temperature
- TSAT - a function that determines the saturation temperature of a refrigerant given saturation pressure

Four basic properties of refrigerants are calculated to determine the thermodynamics of the refrigerant:

<u>(a) Liquid Density</u>

d_L = liquid density (lbs/cubic foot)

T = temperature (°R)

$$d_L = AL + BL\left(1-\frac{T}{T_c}\right)^{1/3} + CL\left(1-\frac{T}{T_c}\right)^{2/3} + DL\left(1-\frac{T}{T_c}\right) + EL\left(1-\frac{T}{T_c}\right)^{4/3} + FL\left(1-\frac{T}{T_c}\right)^{1/2} + GL\left(1-\frac{T}{T_c}\right)^{2} \tag{A.1}$$

T_c = critical temperature (°R)

(b) Vapor Pressure

$$\log_{10} P = AVP + \frac{BVP}{T} + CVP\log_{10} T + (DVP \times T) + EVP\left(\frac{FVP-T}{T}\right)\log_{10}(FVP-T) \tag{A.2}$$

P = pressure (psia)

T = temperature (°R)

According to Martin (1959), who is largely responsible for the establishment of these refrigerant equations, no equation has been proposed which is superior to that of equation A.2 for the vapor pressure.

(c) Equation of State

$$P = \frac{RT}{v-B} + \frac{A2+B2(T)+C2\exp(-XK(T/T_c))}{(v-B)^2} + \frac{A3+B3(T)+C3exp(-XK(T/T_c))}{(v-B)^3} + \frac{A4+B4(T)+C4exp(-XK(T/T_c))}{(v-B)^4} + \frac{A5+B5(T)+C5exp(-XK(T/T_c))}{(v-B)^5} + \frac{A6+B6(T)+C6exp(-XK(T/T_c))}{\exp(\alpha \times v)(1+CPRexp(\alpha \times v)} \tag{A.3}$$

P = pressure (psia)

v = specific volume (cubic feet/lb)

T = temperature (°R)

(d) Heat Capacity of the Vapor

$$C_V = ACV + BCV(T) + CCV(T^2) + DCV(T^3) + \frac{FCV}{T^2} - \frac{J(XK)^2\, Texp(-XK(T/T_c))}{T_c^2} \times [AAA] \qquad (A.4)$$

where

$$[AAA] = \frac{C2}{v-B} + \frac{C3}{2(v-B)^2} + \frac{C4}{3(v-B)^3} + \frac{C5}{4(v-B)^4} + \frac{C6}{alpha(\exp(alpha(v)))} - \frac{C6(CPR)}{alpha}(\ln 10)\log_{10}\left(1 + \frac{1}{CPR(\exp(alpha(v)))}\right) \qquad (A.5)$$

C_v = heat capacity at constant volume (Btu/lb R)

J = 0.185053

T = temperature (°R)

CPR = 0 for R-12

Using equation A.2 the saturation pressure can be determined for the given temperature. The saturation temperature is calculated given pressure using newton iteration:

$$Tr = Tro - \frac{F}{FP} \qquad (A.6)$$

Tr = temperature (°R)

Tro = initial estimate of temperature (°R)

The initial saturation temperature estimate is computed as:

P = pressure (psia)

$$Tro = AA \times \log_{10} P + BB \tag{A.7}$$

The four basic equations (equations A.1 - A.5) are combined by exact thermodynamic relationships to calculate properties essential for a proper analysis of heat pump systems. In the developed program the following state point parameters are determined with the aforementioned subroutines:

(1) **saturation thermodynamic properties**

(a) <u>specific volume of saturated vapor (v_g)</u> - may be obtained from the P-v-T relation (equation of state) (equation A.3) by substituting values of pressure and temperature (by iteration) which satisfy equation A.2 (vapor pressure equation)

(b) <u>specific volume of saturated liquid (v_f)</u> - this is simply the reciprocal of the saturated liquid density (equation A.1)

(c) <u>latent heat of vaporization (h_{fg})</u>

$$h_{fg} = JT(v_g - v_f)\left[P(\ln 10)\left(-\frac{BVP}{T^2} + \frac{CVP}{T(\ln 10)} + DVP - EVP\left(\frac{\log_{10} e}{T} + \frac{FVP \log_{10}(FVP-T)}{T^2}\right)\right)\right] \tag{A.8}$$

J = 0.185053

v_g, v_f - saturated volume of vapor, liquid (cubic ft/lb)

P = vapor pressure (psia)

T = temperature (°R)

(d) enthalpy of saturated vapor (h_g)

$$h_g = ACV(T) + \frac{BCV(T^2)}{2} + \frac{CCV(T^3)}{3} + \frac{DCV(T^4)}{4} - \frac{FCV}{T} + JPv \\ + J\,[BBB] + Je^{(-XK(T)/T_c)}(1 + \frac{XKT}{T_c})[AAA] + X \tag{A.9}$$

where

$$[BBB] = \frac{A2}{v-B} + \frac{A3}{2(v-B)^2} + \frac{A4}{3(v-B)^3} + \frac{A5}{4(v-B)^4} \\ + \frac{A6}{alpha}[\frac{1}{e^{alpha(v)}} - CPR(ln10)\log_{10}(1 + \frac{1}{CPR(e^{alpha(v)})})] \tag{A.10}$$

J = 0.185053

v = specific volume (cubic ft/lb)

P = pressure (psia)

[AAA] is as defined in equation A.5. In the program developed only the first four terms of equations A.5 and A.10 are used.

(e) enthalpy of saturated liquid (h_f) - determined by subtracting the latent heat of vaporization from the enthalpy of saturated vapor (h_g - h_{fg})

(f) entropy of vaporization (s_{fg}) - calculated as the enthalpy divided by temperature

(g) entropy of saturated vapor (s_g)

$$s_g = ACV(ln10)\log_{10}T + BCV(T) + \frac{CCV(T^2)}{2} + \frac{DCV(T^3)}{3} - \frac{FCV}{2T^2} \\ + JR(ln10)\log_{10}(v-B) - J[CCC] + \frac{J(XK)e^{-XKT/T_c}}{T_c}[AAA] + Y \tag{A.11}$$

J = 0.185053

v = specific volume (cubic ft/lb)

P = pressure (psia)

[AAA] is as defined in equation A.5. To obtain [CCC] the C2,C3,C4... terms in equation A.5 are replaced by B2,B3,B4... respectively.

(h) entropy of saturated liquid (s_f) - determined by subtracting the entropy of vaporization from the entropy of saturated vapor (s_g - s_{fg}).

(2) **superheated thermodynamic properties**

(a) specific volume - calculated the same way as for the specific volume of saturated vapor (through iteration)

(b) enthalpy, entropy - calculated with the same equations as those of saturated vapor

(3) **compressed liquid thermodynamic properties**

(a) specific volume - since pressure has little effect on the volume of the liquid in this region, the specific volume is taken to be the same as the volume of saturated liquid at the same temperature.

(b) enthalpy - expressed as a linear interpolation between the reference conditions and the saturation temperature

(c) entropy - the change is calculated from the difference between the saturation temperature and the desired state point using the specific heat capacity Cp at the given pressure. The enthalpy change with temperature between the desired and the reference points gives the value of the heat capacity.

A.5 LIST OF CONSTANTS FOR R-12

To commence computations, the developed program must first read data regarding the specified refrigerant. In the investigation the refrigerant is R-12. The constants used by the program for this refrigerant are obtained from Downing (1974) and are listed below.

12	refrigerant number	
0.88734000000E-01	R	
0.65093886000E-02	B	
-0.34097271300E 01	A2	
0.15943484800E-02	B2	
-0.56762767100E 02	C2	
0.60239446500E-01	A3	
-0.18796184300E-04	B3	
0.13113990800E 01	C3	
-0.54873701000E-03	A4	
0.00000000000E 00	B4	
0.00000000000E 00	C4	
0.00000000000E 00	A5	Equation of State Constants
0.34688340000E-08	B5	
-0.25439067800E-04	C5	
0.00000000000E 00	A6	
0.00000000000E 00	B6	
0.00000000000E 00	C6	
0.54750000000E 01	XK	
0.00000000000E 00	ALPHA	
0.00000000000E 00	CPR	
0.69330000000E 03	TC	
0.45970000000E 03	TFR	
0.34840000000E 02	AL	
0.53341187000E 02	BL	
0.00000000000E 00	CL	
0.18691370000E 02	DL	Liquid Density Constants
0.00000000000E 00	EL	
0.21983960000E 02	FL	
-0.31509940000E 01	GL	

0.39883817270E 02	AVP	
-0.34366322280E 04	BVP	
-0.12471522280E 02	CVP	Vapor Pressure Constants
0.47304424400E-02	DVP	
0.00000000000E 00	EVP	
0.70000000000E 03	FVP	
0.80945000000E-02	ACV	
0.33266200000E-03	BCV	
-0.24138960000E-06	CCV	Specific Heat Constants
0.67236300000E-10	DCV	(at constant volume)
0.00000000000E 00	ECV	
0.00000000000E 00	FCV	
0.39556551336E 02	X	Vapor Enthalpy and Entropy
-0.16537936091E-01	Y	Constants
0.59690000000E 03	Pcrit	Critical Pressure
0.12000000000E 03	AA	Coefficients for Initial Estimate
0.31200000000E 03	BB	of 'TSAT'
		(TSAT=AA*ALOG10(P)+BB)

A.6 CALCULATION OF THE REFRIGERANT FLOWRATE

To calculate the rate of refrigerant mass flow in the heat pump circuit a heat balance was conducted across the condenser. It was assumed that the volume flowrate of air across the condenser was constant at 0.2124 m^3/s (a value found to be the best estimate in a previous test). The room temperature is also assumed to be constant at 21 °C.

CPA = the heat capacity of air (kJ/(kg °K)) = 1.0035 kJ/(kg °K)

SPA = volume air-flowrate across the condenser (m^3/s) = 0.2124 m^3/s (450 cfm)

Proom = room pressure (kPa) = 92.366 kPa

Troom = room temperature (°K) = 294.15

Tcoil1,2 = upstream and downstream condenser coil temperatures (°K)

mair, mfreon = air and refrigerant mass flowrates (kg/s)

R = ideal gas constant (0.287 kJ/kg °K)

hin, hout = condenser inlet and outlet enthalpies (kJ/kg)

RHO = density of the room air (kg/m^3)

Since all the heat exiting the condenser goes into the air and assuming ideal gas for air:

$$Q_{air} = Q_{condenser} \tag{A.12}$$

$$Q_{air} = mair \times CPA \times (Tcoil2 - Tcoil1) \tag{A.13}$$

$$= RHO \times SPA \times CPA \times (Tcoil2 - Tcoil1)$$

to calculate the air density:

$$RHO = \frac{Proom}{(R \times Troom)} = \frac{92.366}{(0.287 \times 294.15)} = 1.0941 kg/m^3 \qquad (A.14)$$

for the condenser:

$$Q_{condenser} = mfreon \times (hin - hout)\ (kW) \qquad (A.15)$$

Rearranging to solve for the refrigerant flowrate:

$$mfreon = \frac{RHO \times SPA \times CPA \times (Tcoil2 - Tcoil1)}{hin - hout} \qquad (A.16)$$

with the variables being the condenser inlet and outlet enthalpies and the upstream and downstream condenser coil temperatures.

A.7 P-H DIAGRAM OF THE HEAT PUMP PROCESS

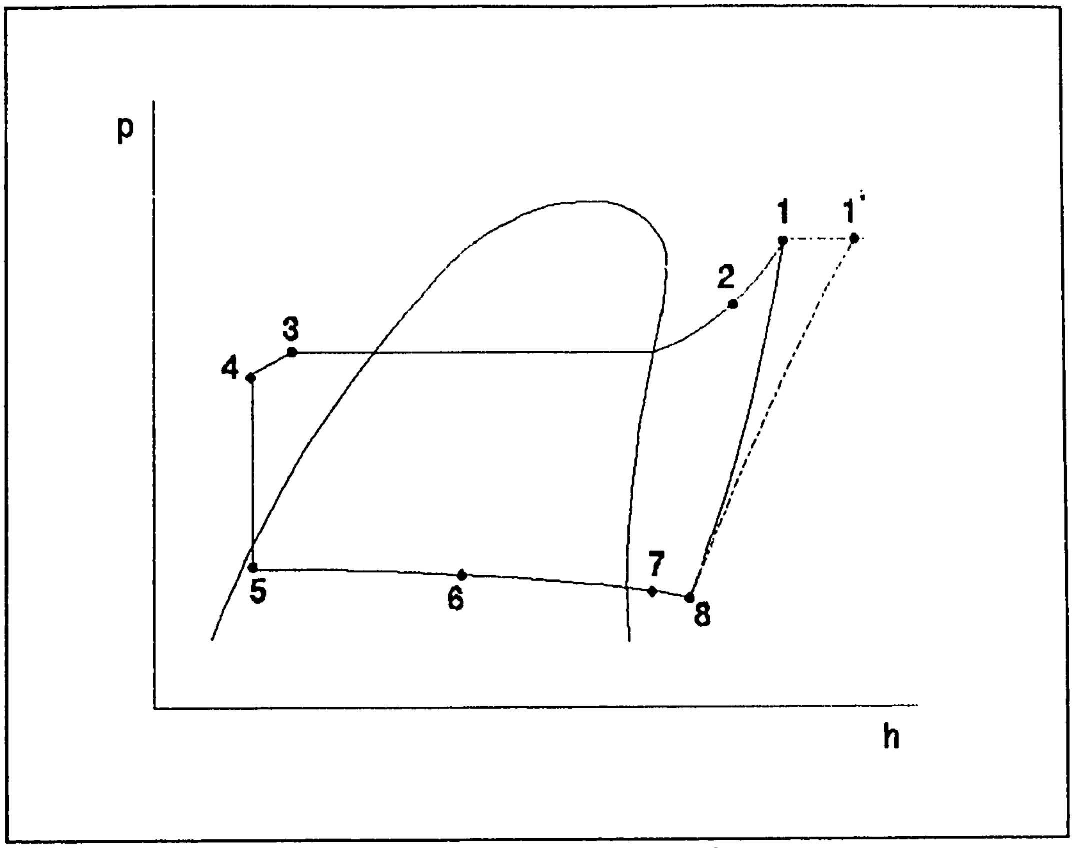

Figure A.1 - P-H Diagram for the Heat Pump's Actual Process

1-2 heat rejection with pressure loss due to friction

2-3 heat rejection by condensation with pressure loss

3-4 heat rejection with subcooling

4-5 isenthalpic expansion

5-8 heat absorption with pressure loss and superheating

8-1 compression with a variable polytropic exponent

note: see section 2.3.1 for details about 1'

A.8 ERROR ANALYSIS

The following table lists the errors in the calculation of the various parameters. Errors are due to equipment inaccuracies.

TABLE A.3 - ERRORS IN THE VARIOUS PARAMETERS

	% error
refrigerant flowrate (FR12)	6 %
heat output (Qout)	6 %
heat input (Qin)	6 %
COP	6 %
auxiliary energy (Qaux)	1.2 %
heat load (Qload)	7.7 %
evap absorption (Qevap)	6 %
collector efficiency	6 %

A.9 LISTING OF THE PROGRAM SUBROUTINES

The following is a listing of the original program and subroutines used to calculate the refrigerant properties in the heat pump cycle.

```
      MAIN PROGRAM
C     PROGRAM TO CALCULATE THE STATE POINT PROPERTIES OF R12, THE
C     CYCLE ENERGY QUANTITIES OF THE SOLAR SOURCE HEAT PUMP INSTALLED
C     ON MODULE 6 AT THE AHHRF AND ITS COEFFICIENT OF PERFORMANCE.
C     CALCULATIONS OF REFRIGERANT PROPERTIES ARE DONE IN IMPERIAL AND
C     OUTPUTS ARE CONVERTED TO S.I. UNITS (VAN WYLEN, SONNTAG).
C        1, 5  = INPUT        ( 5 = rprop.dat )
C           6  = OUTPUT
         IMPLICIT REAL*8(A-H,O-Z)
         DIMENSION P(10),T(10),V(10),H(10),S(10),X(10),DELH(10),E(10)
         NR=12
         CALL RPDATA(NR)
         PATM=13.5
         CF=1
       1 READ(1,100,END=99) IDATE,ITIME,TA,TR,TST,TSB,TCOIL1,TCOIL2,
        *T(2),T(3),T(4),T(5),T(7),T(6),T(8),T(1),IQT,IWHP,ITHP,P(1),
        *P(6),P(2),P(7),P(3),P(8),P(4),RAD
     100 FORMAT(I7,I6,14F6.1,3I6,8F6.1)
         IF(P(3).GT.P(2)) P(2)=P(3)
         IF(P(2).GT.P(1)) P(1)=P(2)
         IF (RAD.LT.0.0) RAD=0.0
         ITHP=ITHP*CF
         WRITE(6,300) IDATE,ITIME,RAD,TA,TR,ITHP,IQT,IWHP,TST,TSB,
        *TCOIL1,TCOIL2
     300 FORMAT(I7,I4,F8.1,2F6.1,3I6,4F6.1)
         IF(IQT.LE.0) GO TO 5
         DO 10 J=1,8
         T(J)=T(J)*1.8+32.0
         IF(J.EQ.5) GO TO 2
         P(J)=P(J)+PATM
         IF (J.EQ.6) GO TO 10
         I=1
         GO TO 3
       2 P(J)=PSAT(NR,T(J))
         H(J)=H(J-1)
         I=3
       3 CALL REFPRP(NR,T(J),P(J),V(J),H(J),S(J),X(J),I)
         IF(J.EQ.1.OR.J.EQ.7) GO TO 10
         TT=TR+273.15
         IF(J.EQ.8) TT=TA+273.15
         DELH(J)=(H(J)-H(J-1))*2.326
         DELS=(S(J)-S(J-1))*4.1868
         E(J)=(TA+273.15)*(DELS-DELH(J)/TT)
      10 CONTINUE
         DELH(1)=(H(1)-H(8))*2.326
         E(1)=(TA+273.15)*(S(1)-S(8))*4.1868
         DELH(7)=(H(7)-H(5))*2.326
         E(7)=(TA+273.15)*((S(7)-S(5))*4.1868-DELH(7)/TT)
         SE=0.0
         DO 20 J=1,8
         T(J)=(T(J)-32.0)/1.8
         P(J)=P(J)/145.0
         IF(J.EQ.6) GO TO 4
         V(J)=V(J)*0.0624219
         H(J)=H(J)*2.326
         S(J)=S(J)*4.1868
         SE=SE+E(J)
         WRITE(6,400) J,P(J),T(J),H(J),S(J),V(J),X(J),DELH(J),E(J)
     400 FORMAT(I2,F10.5,F7.2,6F11.5)
```

```
      GO TO 20
    4 WRITE(6,500) J,P(J),T(J)
  500 FORMAT(I2,F10.5,F7.2)
   20 CONTINUE
      QO=H(1)-H(4)
      QI=DELH(1)
      COPH=QO/QI
      COPID=(TR+273.15)/(TR-TA)
      GO TO 25
    5 DO 30 J=1,8
      P(J)=0.
      H(J)=0.
      S(J)=0.
      V(J)=0.
      X(J)=0.
      DELH(J)=0.
      E(J)=0.
      WRITE(6,401) J,P(J),T(J),H(J),S(J),V(J),X(J),DELH(J),E(J)
  401 FORMAT(I2,F10.5,F7.2,6F11.5)
   30 CONTINUE
      QO=0.
      QI=0.
      COPH=0.
      COPID=0.
      SE=0.
   25 WRITE(6,600) QO,QI,COPH,COPID,SE
  600 FORMAT(2F11.5,3F9.3)
      GO TO 1
   99 STOP
      END
      SUBROUTINE RPDATA(NR)
C  This subroutine reads in the constants and coefficients
C  used for calculating the thermodynamic properties of a refrigerant.
      IMPLICIT REAL*8(A-H,O-Z)
      COMMON/CVHSJ/ACV,BCV,CCV,DCV,ECV,FCV,X,Y,XJ,XLE10,XL10E
      COMMON/STATEQ/R,B,A2,B2,C2,A3,B3,C3,A4,B4,C4,A5,B5,C5,A6,B6,C6,
     *XK,ALPHA,CPR
      COMMON/TMP/TC,TFR
      COMMON/LIQ/AL,BL,CL,DL,EL,FL,GL
      COMMON/SAT/AVP,BVP,CVP,DVP,EVP,FVP
      COMMON/TST/PCRIT,AA,BB
      XJ=0.185053
      XLE10=2.302585093
      XL10E=0.4342944819
      NRR=NR
C  Refrigerant Number
    1 CALL FREAD(5,'I:',NR)
C  Equation of State Constants
      READ(5,100)R,B,A2,B2,C2,A3,B3,C3,A4,B4,C4,A5,B5,C5,
     *A6,B6,C6,XK,ALPHA,CPR,TC,TFR
C  Liquid Density Constants
      READ(5,100)AL,BL,CL,DL,EL,FL,GL
C  Vapour Pressure Constants
      READ(5,100)AVP,BVP,CVP,DVP,EVP,FVP
C  Specific Heat at Constant Volume Constants
      READ(5,100)ACV,BCV,CCV,DCV,ECV,FCV
C  Vapour Enthalpy and Entropy Constants
      READ(5,100)X,Y
C  Critical Pressure
      READ(5,100)PCRIT
C  Coefficients for Initial Estimate of 'TSAT'
C  ( TSAT=AA*DLOG10(P)+BB )
      READ(5,100)AA,BB
      IF(NR.NE.NRR)GO TO 1
      RETURN
  100 FORMAT(1X,E18.11)
      END
      SUBROUTINE REFPRP(NR,T,P,V,H,S,X,I)
C  Determines the thermodynamic properties of a refrigerant,
```

```
C  given pressure and one other parameter.
C  The phase of the refrigerant is determined automatically.
C           I = 1 : SECOND PARAMETER IS TEMPERATURE
C               2 :                     SPECIFIC VOLUME
C               3 :                     SPECIFIC ENTHALPY
C               4 :                     SPECIFIC ENTROPY
C               5 :                     THERMODYNAMIC QUALITY
      IMPLICIT REAL*8(A-H,O-Z)
      TOL=0.0001
      TSA=TSAT(NR,P)
      CALL SATPRP(NR,TSA,P,VF,VG,HF,HFG,HG,SF,SG)
      J=2
      GO TO(10,20,30,40,50),I
      GO TO 600
   10 IF((TSA-T).GT.TOL)GO TO 12
      IF((T-TSA).GT.TOL)GO TO 14
      GO TO 700
   12 CALL SCOL(NR,T,P,V,H,S)
      GO TO 16
   14 CALL VAPOR(NR,T,P,V,H,S)
   16 X=(H-HF)/HFG
      GO TO 500
   20 Y=V
      IF(((VF-V)/V).GT.TOL)J=1
      IF(((V-VG)/V).GT.TOL)J=3
      GO TO 100
   30 Y=H
      IF(((HF-H)/H).GT.TOL)J=1
      IF(((H-HG)/H).GT.TOL)J=3
      GO TO 100
   40 Y=S
      IF(((SF-S)/S).GT.TOL)J=1
      IF(((S-SG)/S).GT.TOL)J=3
      GO TO 100
   50 Y=X
      IF(X.LT.0.0)J=1
      IF(X.GT.1.0)J=3
C  Calculations for Two-phase Case
  100 IF(J.NE.2)GO TO 200
      T=TSA
      GO TO(10,120,130,140,150),I
  120 X=(V-VF)/(VG-VF)
      H=HF+HFG*X
      S=SF+(SG-SF)*X
      GO TO 500
  130 X=(H-HF)/HFG
      V=VF+(VG-VF)*X
      S=SF+(SG-SF)*X
      GO TO 500
  140 X=(S-SF)/(SG-SF)
      V=VF+(VG-VF)*X
      H=HF+HFG*X
      GO TO 500
  150 V=VF+(VG-VF)*X
      H=HF+HFG*X
      S=SF+(SG-SF)*X
      GO TO 500
C  Iteration Loop for Subcooled and Superheated Cases
  200 T1=TSA
      DT=0.000001
      DO 300 L=1,20
      T11=T1+DT
      GO TO(210,100,220),J
  210 CALL SCOL(NR,T1,P,V1,H1,S1)
      CALL SCOL(NR,T11,P,V11,H11,S11)
      GO TO 230
  220 CALL VAPOR(NR,T1,P,V1,H1,S1)
      CALL VAPOR(NR,T11,P,V11,H11,S11)
  230 GO TO(10,232,233,234,235),I
```

```
  232 Y1=V1
      Y11=V11
      GO TO 240
  233 Y1=H1
      Y11=H11
      GO TO 240
  234 Y1=S1
      Y11=S11
      GO TO 240
  235 Y1=(H1-HF)/HFG
      Y11=(H11-HF)/HFG
  240 IF((DABS(Y-Y1)/Y).LE.TOL)GO TO 400
      T2=T1+DT*(Y-Y1)/(Y11-Y1)
  300 T1=T2
      WRITE(6,6001)
  400 T=T1
      IF(I.NE.2)V=V1
      IF(I.NE.3)H=H1
      IF(I.NE.4)S=S1
      IF(I.NE.5)X=(H1-HF)/HFG
  500 RETURN
  600 WRITE(6,6002)I
  700 WRITE(6,6003)
      PAUSE 'REFPRP'
 6001 FORMAT(/' *** WARNING *** REFPRP : NOT CONVERGED'/)
 6002 FORMAT(/' ***  ERROR  *** REFPRP : I =',I3/)
 6003 FORMAT(/' ***  ERROR  *** REFPRP : TWO-PHASE CASE.',/,
     *'    "I" COULD NOT BE 1. CHOOSE OTHER PARAMETERS'/)
      RETURN
      END
      SUBROUTINE VAPOR(NR,TF,PPSIA,VVAP,HVAP,SVAP)
C  Determines the thermodynamic properties of superheated
C  refrigerant, given temperature and pressure
C          TF   = TEMPERATURE           (DEG F)
C          PPSIA= ABSOLUTE PRESSURE     (P.S.I.A.)
C          VVAP = SPECIFIC VOLUME       (CU FT/LB)
C          HVAP = SPECIFIC ENTHALPY     (BTU/LB)
C          SVAP = SPECIFIC ENTROPY      (BTU/LB DEG F)
      IMPLICIT REAL*8(A-H,O-Z)
      COMMON/CVHSJ/ACV,BCV,CCV,DCV,ECV,FCV,X,Y,XJ,XLE10,XL10E
      COMMON/STATEQ/R,B,A2,B2,C2,A3,B3,C3,A4,B4,C4,A5,B5,C5,A6,B6,C6,
     *XK,ALPHA,CPR
      COMMON/TMP/TC,TFR
C  Convert 'TF' to absolute temp (deg r) and check if above zero
      T=TF+TFR
      IF(T.LE.0.0) GO TO 902
C  Check if TEMP.GE.SATURATION TEMP
      TFSAT=TSAT(NR,PPSIA)
      IF(TF.LT.(TFSAT-0.0001)) GO TO 903
C  Check if pressure above zero
      IF(PPSIA.LE.0.0) GO TO 904
C  Calculate specific volume 'VVAP'
      VVAP=SPVOL(NR,TF,PPSIA)
C  Calculate spec enthalpy 'HVAP' and spec entropy 'SVAP'
      T2=T*T
      T3=T2*T
      T4=T3*T
      VR=VVAP-B
      VR2=2.0*VR*VR
      VR3=1.5*VR2*VR
      VR4=VR2*VR2
      XKTDTC=XK*T/TC
      EKTDTC=DEXP(-XKTDTC)
      EMAV=DEXP(-ALPHA*VVAP)
      H1=ACV*T+BCV*T2/2.+CCV*T3/3.+DCV*T4/4.-FCV/T
      H2=XJ*PPSIA*VVAP
      H3=A2/VR+A3/VR2+A4/VR3+A5/VR4
      H4=C2/VR+C3/VR2+C4/VR3+C5/VR4
      S1=ACV*DLOG(T)+BCV*T+CCV*T2/2.+DCV*T3/3.-FCV/
```

```
     1(2.*T2)
      S2=XJ*R*DLOG(VR)
      S3=B2/VR+B3/VR2+B4/VR3+B5/VR4
      S4=H4
      IF(DABS(ALPHA).LE.1.0E-20) GO TO 6
      IF(DABS(CPR).GT.1.0E-20) GO TO 5
    4 H3=H3+A6/ALPHA*EMAV
      S3=S3+B6/ALPHA*EMAV
      GO TO 6
    5 HO=1./ALPHA*(EMAV-CPR*DLOG(1.+EMAV/CPR))
      H3=H3+A6*HO
      H4=H4+C6*HO
      S3=S3+B6*HO
      S4=S4+C6*HO
    6 HVAP=H1+H2+XJ*H3+XJ*EKTDTC*(1.+XKTDTC)*H4+X
      SVAP=S1+S2-XJ*S3+XJ*EKTDTC*XK/TC*S4+Y
      RETURN
  902 WRITE(6,6002)
      GO TO 999
  903 WRITE(6,6003)
      GO TO 999
  904 WRITE(6,6004)
  999 PAUSE 'VAPOR'
 6002 FORMAT(/' *** ERROR *** VAPOR : TEMP.LE.ZERO'/)
 6003 FORMAT(/' *** ERROR *** VAPOR : TEMP.LT.SAT TEMP'/)
 6004 FORMAT(/' *** ERROR *** VAPOR : PRESSURE.LE.ZERO'/)
      RETURN
      END
      SUBROUTINE SCOL(NR,TF,PPSIA,VA,HA,SA)
C  Determines the thermodynamic properties of subcooled liquid
C  refrigerant,  given temperature and pressure
C  The approach to calculate the enthalpy and entropy of liquid
C  refrigerants can be one of the followings:
C    IAP=1:  Enthalpy is expressed as a linear interpolation
C            between that at the reference conditions and that
C            at the saturation temperature.
C            Entropy change is calculated from the difference
C            between the saturation temperature and the desired
C            state point using the specific heat capacity Cp at
C            the given pressure.  The Cp is expressed as the
C            enthalpy change with temperature between the desired
C            point and the reference point.
C    IAP=2:  For a given temperature, both the enthalpy and the
C            entropy of the liquid refrigerant are taken as
C            being equal to the saturation values at the same
C            temperature.
      IMPLICIT REAL*8(A-H,O-Z)
      IAP=1
      T=TSAT(NR,PPSIA)
      PSA=PSAT(NR,TF)
      DELT=T-TF
C  Check if TEMP.LE.SATURATION TEMP
      IF(DELT.LT.-0.0001)GOTO901
      CALL SATPRP(NR,T,PPSIA,VL,VG,HL,HLG,HG,SL,SG)
      CALL SATPRP(NR,TF,PSA,VL1,VG1,HL1,HLG1,HG1,SL1,SG1)
      VA=VL1
      GO TO (10,20), IAP
      WRITE (6,6002) IAP
   10 IF(DABS(T+40.0).LE.1.0E-06)GOTO130
      HA=HL*(1.0-DELT/(T+40.))
      GOTO 140
  130 HA=HL1
  140 IF(DABS(TF+40.0).LE.1.0E-06)GOTO150
      CP=HA/(TF+40.0)
      SA=SL+CP*DLOG((TF+459.7)/(T+459.7))
      GOTO160
   20 HA=HL1
  150 SA=SL1
  160 CONTINUE
```

```
      RETURN
  901 WRITE(6,6001)
  999 PAUSE 'SCOL'
 6001 FORMAT(/' ***  ERROR  *** SCOL : TEMP.GT.SAT TEMP'/)
 6002 FORMAT(/' *** WARNING *** SCOL : IAP =',I3,
     *', THE PROGRAM ASSUMES THE FIRST APPROACH IS USED TO
     * CALCULATE THE PROPERTIES OF LIQUID REFRIGERANTS',/)
      RETURN
      END
      SUBROUTINE SATPRP(NR,TF,PSA,VF,VG,HF,HFG,HG,SF,SG)
C  Determines the saturation thermodynamic properties of a
C  refrigerant given the saturation temperature and pressure
C         VF  = SPEC VOL OF SAT LIQUID        (CU FT/LB)
C         VG  = SPEC VOL OF SAT VAPOUR        (CU FT/LB)
C         HF  = SPEC ENTH OF SAT LIQ          (BTU/LB)
C         HFG = LATENT HEAT OF VAPORIZATION   (BTU/LB)
C         HG  = SPEC ENTH OF SAT VAPOUR       (BTU/LB)
C         SF  = SPEC ENTROPY SAT LIQUID       (BTU/LB DEG F)
C         SG  = SPEC ENTROPY SAT VAPOUR       (BTU/LB DEG F)
      IMPLICIT REAL*8(A-H,O-Z)
      COMMON/LIQ/AL,BL,CL,DL,EL,FL,GL
      COMMON/CVHSJ/ACV,BCV,CCV,DCV,ECV,FCV,X,Y,XJ,XLE10,XL10E
      COMMON/STATEQ/R,B,A2,B2,C2,A3,B3,C3,A4,B4,C4,A5,B5,C5,A6,B6,C6,
     *XK,ALPHA,CPR
      COMMON/TMP/TC,TFR
      COMMON/SAT/AVP,BVP,CVP,DVP,EVP,FVP
C  convert 'TF' to absolute temp (deg r) and check if above zero
      T=TF+TFR
      IF(T.LE.0.0)GO TO 902
C  Check if TEMP.LE.CRITICAL TEMP
      IF(T.GT.TC) GO TO 903
C  Calculate specific volume of sat vapour 'VG'
   12 VG=SPVOL(NR,TF,PSA)
C  Calculate specific volume of sat liquid 'VF'
    2 IF(NR.EQ.21.OR.NR.EQ.113)GOTO20
      TR1=1.-T/TC
      VF=1./(AL+BL*TR1**(1./3.)+CL*TR1**(2./3.)+DL*TR1+
     *EL*TR1**(4./3.)+FL*TR1**0.5+GL*TR1*TR1)
      GOTO21
   20 VF=1.0/(AL+BL*T+CL*T*T)
   21 CONTINUE
C   Calculate latent heat 'HFG' by clausius clapeyron equation
      IF(DABS(EVP).LE.1.0E-20)GOTO30
      HFG=(VG-VF)*PSA*XLE10*(-BVP/T+CVP/XLE10+DVP*T-
     1EVP*(XL10E+FVP*DLOG10(FVP-T)/T))*XJ
      GOTO31
   30 HFG=(VG-VF)*PSA*XLE10*(-BVP/T+CVP/XLE10+DVP*T)*XJ
   31 CONTINUE
      SFG=HFG/T
C   Calculate spec enthalpy 'HG' and spec entropy 'SG' (sat vapour)
      T2=T*T
      T3=T2*T
      T4=T3*T
      VR=VG-B
      VR2=2.0*VR*VR
      VR3=1.5*VR2*VR
      VR4=VR2*VR2
      XKTDTC=XK*T/TC
      EKTDTC=DEXP(-XKTDTC)
      EMAV=DEXP(-ALPHA*VG)
      H1=ACV*T+BCV*T2/2.+CCV*T3/3.+DCV*T4/4.-FCV/T
      H2=XJ*PSA*VG
      H3=A2/VR+A3/VR2+A4/VR3+A5/VR4
      H4=C2/VR+C3/VR2+C4/VR3+C5/VR4
      S1=ACV*DLOG(T)+BCV*T+CCV*T2/2.+DCV*T3/3.-FCV/
     1(2.*T2)
      S2=XJ*R*DLOG(VR)
      S3=B2/VR+B3/VR2+B4/VR3+B5/VR4
      S4=H4
```

```
      IF(DABS(ALPHA).LE.1.0E-20)GOTO6
      IF(DABS(CPR).GT.1.0E-20)GOTO5
    4 H3=H3+A6/ALPHA*EMAV
      S3=S3+B6/ALPHA*EMAV
      GO TO 6
    5 HO=1./ALPHA*(EMAV-CPR*DLOG(1.+EMAV/CPR))
      H3=H3+A6*HO
      H4=H4+C6*HO
      S3=S3+B6*HO
      S4=S4+C6*HO
    6 HG=H1+H2+XJ*H3+XJ*EKTDTC*(1.+XKTDTC)*H4+X
      SG=S1+S2-XJ*S3+XJ*EKTDTC*XK/TC*S4+Y
C  Calculate spec enthalpy 'HF' and spec entropy 'SF' (sat liquid)
      HF=HG-HFG
      SF=SG-SFG
      RETURN
  902 WRITE(6,6002)
      GOTO999
  903 WRITE(6,6003)
  999 PAUSE 'SATPRP'
 6002 FORMAT(/' ***  ERROR  *** SATPRP : TEMP.LE.ZERO'/)
 6003 FORMAT(/' ***  ERROR  *** SATPRP : TEMP.GT.CRIT TEMP'/)
      RETURN
      END
      FUNCTION SPVOL(NR,TF,PPSIA)
C  Determines the specific volume of superheated refrigerant
C  given the temperature and pressure
C          SPVOL= SPECIFIC VOLUME       (CU FT/LB)
      IMPLICIT REAL*8(A-H,O-Z)
      COMMON/STATEQ/R,B,A2,B2,C2,A3,B3,C3,A4,B4,C4,A5,B5,C5,A6,B6,C6,
     *XK,ALPHA,CPR
      COMMON/TMP/TC,TFR
C   Convert 'TF' to absolute temp (deg r) and check if above zero
      T=TF+TFR
      IF(T.LE.0.0) GO TO 902
C   Check if TEMP.GE.SATURATION TEMP
      TFSAT=TSAT(NR,PPSIA)
      IF(TF.LT.(TFSAT-0.0001)) GO TO 903
C   Check if pressure above zero
      IF(PPSIA.LE.1.0E-20) GO TO 904
C   Calculate constants
      ES0=DEXP(-XK*T/TC)
      ES1=PPSIA
      ES2=R*T
      ES3=A2+B2*T+C2*ES0
      ES4=A3+B3*T+C3*ES0
      ES5=A4+B4*T+C4*ES0
      ES6=A5+B5*T+C5*ES0
      ES7=A6+B6*T+C6*ES0
      ES32=2.*ES3
      ES43=3.*ES4
      ES54=4.*ES5
      ES65=5.*ES6
C   Compute initial value of 'V' from gas law
      VN=R*T/PPSIA
C   Compute 'V' to within (1.E-08*V) by iteration
      DO 10 L=1,50
      V=VN
      V2=V*V
      V3=V2*V
      V4=V3*V
      V5=V4*V
      V6=V5*V
      EMAV=DEXP(-ALPHA*(V+B))
      IF(DABS(CPR).GT.1.0E-20)GOTO3
    2 F=ES1-ES2/V-ES3/V2-ES4/V3-ES5/V4-ES6/V5-ES7*EMAV
      FV=ES2/V2+ES32/V3+ES43/V4+ES54/V5+ES65/V6+ES7*ALPHA*EMAV
      GO TO 4
    3 EM2AV=EMAV*EMAV
```

```
      F=ES1-ES2/V-ES3/V2-ES4/V3-ES5/V4-ES6/V5-ES7*EM2AV/(EMAV+CPR)
      FV=ES2/V2+ES32/V3+ES43/V4+ES54/V5+ES65/V6+ES7*EM2AV*ALPHA
     1*(EMAV+2.*CPR)/(EMAV+CPR)**2
    4 FFV=F/FV
      IF(DABS(FFV).GE.VN)  FFV=FFV/5.
      VN=V-FFV
      IF(DABS((VN-V)/V).LE.1.E-08) GO TO 20
   10 CONTINUE
      WRITE(6,6000) VN,V,B
   20 SPVOL=VN+B
      RETURN
  902 WRITE(6,6002)
      GOTO999
  903 WRITE(6,6003) TF,TFSAT,PPSIA
      GOTO999
  904 WRITE(6,6004)
  999 PAUSE 'SPVOL'
 6000 FORMAT(/' *** WARNING *** SPVOL : NOT CONVERGED, VN =',F10.5,
     *',  V =',F10.5,',  B =',F10.7,' CU FT/LB',/)
 6002 FORMAT(/' ***  ERROR  *** SPVOL : TEMP.LE.ZERO'/)
 6003 FORMAT(/' ***  ERROR  *** SPVOL : TEMP.LT.SAT TEMP, TF =',
     *F9.4,' F, TFSAT =',F9.4,' F, PPSIA =',F10.5,' P.S.I.A.',/)
 6004 FORMAT(/' ***  ERROR  *** SPVOL : PRESSURE.LE.ZERO'/)
      RETURN
      END
      FUNCTION TSAT(NR,PSA)
C   Determines saturation temperature of a refrigerant
C   given saturation pressure
C           PSA  = ABSOLUTE PRESSURE      (P.S.I.A.)
C           TSAT = TEMPERATURE            (DEG F)
      IMPLICIT REAL*8(A-H,O-Z)
      COMMON/SAT/AVP,BVP,CVP,DVP,EVP,FVP
      COMMON/TST/PCRIT,A,B
      COMMON/TMP/TC,TFR
      XLE10=2.302585093
      IF(PSA.GT.PCRIT) GO TO 902
C   Compute initial estimate of 'TSAT'
      PLOG=DLOG10(PSA)
      TR=A*PLOG+B
C   Iterate to within .0001 deg f by newton iteration
      DO 10 L=1,50
      TRO=TR
      C=DLOG10(DABS(FVP-TRO))
      F=AVP+BVP/TRO+CVP*DLOG10(TRO)+DVP*TRO
     1+EVP*((FVP-TRO)/TRO)*C-PLOG
      FP=-BVP/TRO**2+CVP/(XLE10*TRO)+DVP-EVP
     1*(1./(XLE10*TRO)+FVP*C/TRO**2)
      TR=TRO-F/FP
      IF(DABS(TR-TRO).LE.0.0001) GO TO 20
   10 CONTINUE
      WRITE(6,6000) TR,TRO
   20 TSAT=TR-TFR
      RETURN
  902 WRITE(6,6002)
  999 PAUSE 'TSAT'
 6000 FORMAT(/' *** WARNING *** TSAT : NOT CONVERGED, TR =',F9.4,
     *' R, TRO =',F9.4,' R',/)
 6002 FORMAT(/' ***  ERROR  *** TSAT : PSA.GT.CRIT PRESSURE'/)
      RETURN
      END
      FUNCTION PSAT(NR,TF)
C   Determines the saturation pressure of a refrigerant
C   given temperature
C         TF   = TEMPERATURE          (DEG F)
C         PSAT = ABSOLUTE PRESSURE    (P.S.I.A.)
      IMPLICIT REAL*8(A-H,O-Z)
      COMMON/SAT/AVP,BVP,CVP,DVP,EVP,FVP
      COMMON/TMP/TC,TFR
      T=TF+TFR
```

```
      IF(T.LE.0.0) GO TO 902
      IF(T.GT.TC) GO TO 903
   11 PSAT=10.**(AVP+BVP/T+CVP*DLOG10(T)+DVP*T+EVP*
     1((FVP-T)/T)*DLOG10(DABS(FVP-T)))
      RETURN
  902 WRITE(6,6002)
      GOTO999
  903 WRITE(6,6003)
  999 PAUSE 'PSAT'
 6002 FORMAT(/' ***  ERROR  *** PSAT : TEMP.LE.ZERO'/)
 6003 FORMAT(/' ***  ERROR  *** PSAT : TEMP .GT.CRIT TEMP'/)
      RETURN
      END
```

APPENDIX B : MONTHLY DATA FOR THE 1986-87 HEATING SEASON (SEPTEMBER '86 TO MARCH '87)

SEPTEMBER '86

DATE	RAD	TA	FR12	THP	QOUT	QIN	QTOT	WIN	QEVAP	QAUX	QLOAD	COP
	W/m^2	DegC	kg/s	Hr	MJ	MJ	MJ	MJ	MJ	MJ	MJ	
90186	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
90286	.0	9.8	.0075	1.18	5.20	.92	8.54	4.19	4.00	4.35	12.83	1.24
90386	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
90486	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
90586	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
90686	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
90786	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
90886	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
90986	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
91086	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
91186	36.1	4.4	.0120	5.98	44.86	14.14	34.14	20.79	29.28	13.35	64.85	2.16
91286	.0	3.9	.0106	4.00	26.92	8.44	21.99	13.19	17.56	8.80	40.47	2.04
91386	156.8	5.4	.0102	3.90	24.57	6.94	24.18	11.91	16.70	12.28	41.81	2.06
91486	305.7	4.9	.0112	12.15	84.18	24.75	70.41	40.64	56.42	29.77	129.85	2.07
91586	74.8	6.0	.0106	7.45	48.04	12.54	45.14	26.61	33.47	18.53	80.64	1.81
91686	56.1	6.7	.0101	6.99	43.25	11.78	42.04	24.53	29.84	17.52	73.52	1.76
91786	371.5	5.0	.0098	10.17	62.97	18.36	58.31	34.02	42.26	24.29	102.93	1.85
91886	409.2	1.4	.0085	11.90	65.25	19.89	63.39	36.59	42.87	26.79	108.75	1.78
91986	1160.7	1.2	.0074	12.50	59.30	17.37	67.72	38.82	39.73	28.90	109.65	1.53
92086	14.4	3.5	.0077	10.49	51.34	15.18	58.42	33.75	34.28	24.67	94.58	1.52
92186	549.1	.8	.0071	11.79	54.20	16.20	63.53	35.88	35.94	27.66	101.53	1.51
92286	.0	5.4	.0081	5.78	29.48	8.64	33.63	18.85	19.80	14.78	54.47	1.56
92386	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
92486	682.3	6.8	.0080	3.84	18.66	4.57	26.14	13.36	13.35	12.78	40.22	1.40
92586	44.6	6.4	.0078	3.11	14.70	3.53	19.87	10.74	10.54	9.13	31.04	1.37
92686	288.0	3.8	.0091	13.69	77.42	22.54	80.92	46.90	52.09	34.02	135.80	1.65
92786	105.9	3.8	.0084	9.64	50.30	13.77	57.66	35.74	34.46	21.92	94.19	1.41
92886	68.7	4.0	.0085	7.55	40.01	11.21	43.15	24.71	27.21	18.44	71.95	1.62
92986	2.6	4.0	.0082	6.87	35.69	10.72	38.57	22.34	23.74	16.23	63.54	1.66
93086	39.7	6.0	.0079	7.27	35.04	8.91	44.40	26.96	24.68	17.43	70.53	1.30
SUMMARY	4366.2	4.1	.0089	156.2	871.4	250.4	902.2	520.5	588.2	351.6	1523.1	1.67

OCTOBER '86

DATE	RAD	TA	FR12	THP	QOUT	QIN	QTOT	WIN	QEVAP	QAUX	QLOAD	COP
	W/m^2	°C	kg/s	Hr	MJ	MJ	MJ	MJ	MJ	MJ	MJ	
100186	388.8	2.5	.0081	18.75	97.39	29.53	101.89	60.39	64.40	41.50	169.75	1.61
100286	2745.6	.0	.0082	19.23	100.32	30.17	120.93	60.55	66.53	60.38	191.08	1.66
100386	543.4	4.2	.0088	7.49	41.51	12.22	42.96	25.10	27.91	17.85	72.25	1.65
100486	18.9	6.6	.0102	3.27	20.35	5.33	20.98	11.44	14.21	9.54	36.00	1.78
100586	29.8	7.1	.0114	4.70	32.52	8.78	27.45	15.96	22.49	11.49	51.18	2.04
100686	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
100786	206.9	4.6	.0109	18.78	127.66	38.13	108.62	65.69	84.85	42.93	198.14	1.94
100886	393.7	4.4	.0116	16.24	118.13	35.98	94.15	56.06	78.01	38.09	176.29	2.11
100986	193.2	5.2	.0108	15.51	106.16	32.28	87.66	52.25	70.13	35.42	161.54	2.03
101086	3061.7	1.0	.0101	17.16	110.59	33.69	98.54	54.07	73.06	44.47	175.45	2.05
101186	1162.9	-1.1	.0091	13.69	80.61	24.34	93.32	41.94	53.01	51.38	149.59	1.92
101286	1198.5	2.2	.0097	13.24	81.97	24.45	75.50	41.86	54.48	33.63	133.02	1.96
101386	545.1	1.9	.0101	8.70	55.79	17.02	49.49	27.72	36.70	21.77	88.26	2.01
101486	106.6	3.5	.0100	6.91	43.93	13.44	39.66	22.29	28.83	17.37	70.15	1.97
101586	507.1	7.0	.0098	5.48	32.78	8.61	36.06	18.84	22.87	17.22	60.24	1.74
101686	114.7	4.0	.0099	7.23	44.57	12.73	44.01	23.94	30.07	20.07	75.86	1.86
101786	1007.8	3.2	.0101	7.33	46.60	13.69	43.22	24.10	31.17	19.12	76.12	1.93
101886	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
101986	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
102086	.0	7.0	.0104	1.00	6.47	1.85	5.88	5.33	4.41	2.56	10.50	1.94
102186	80.5	4.6	.0107	6.73	45.23	13.65	39.88	22.37	29.96	17.51	71.46	2.02
102286	406.9	3.3	.0103	11.05	71.59	21.10	65.38	36.93	47.83	28.45	115.86	1.94
102386	506.6	2.5	.0101	11.17	71.09	21.09	64.53	40.49	47.31	24.04	114.53	1.76
102486	1012.1	1.4	.0102	12.73	82.60	25.44	70.81	40.73	54.23	30.08	127.97	2.03
102586	594.5	1.6	.0095	12.49	[illegible]	22.81	69.53	39.09	50.16	30.45	122.63	1.94
102686	316.9	3.6	.0102	10.31	[illegible]	19.69	59.32	33.68	43.94	25.64	105.78	1.96
102786	1118.1	5.2	.0101	9.12	[illegible]	15.48	56.38	31.58	39.15	24.79	97.81	1.80
102886	296.3	.0	.0101	23.88	153.83	47.87	129.25	74.33	100.11	54.93	235.21	2.07
102986	428.1	-.7	.0102	23.91	155.49	48.26	136.75	[illegible]	101.29	63.63	243.98	2.13
103086	746.6	-3.2	.0086	22.76	127.89	39.26	166.38	64.57	83.11	101.81	255.01	1.98
103186	5121.1	-6.4	.0081	23.23	121.98	36.43	180.26	65.10	80.06	115.16	265.80	1.87
SUMMARY	22852.4	1.6	.0097	352.1	2176.0	653.3	2128.8	1127.5	2028.5	1001.3	3651.5	1.93

NOVEMBER '86

DATE	RAD	TA	FR12	THP	QOUT	QIN	QTOT	WIN	QEVAP	QAUX	QLOAD	COP
	W/m^2	°C	kg/s	Hr	MJ	MJ	MJ	MJ	MJ	MJ	MJ	
110186	3828.7	-3.8	.0093	21.16	126.23	39.06	152.41	61.81	82.26	90.60	239.57	2.04
110286	152.1	5.1	.0105	15.26	100.11	29.01	77.12	53.53	67.35	23.59	148.22	1.87
110386	313.3	5.9	.0106	11.99	78.55	22.38	57.95	39.38	53.25	18.56	114.11	1.99
110486	2148.1	5.1	.0099	12.66	78.34	22.26	60.27	40.71	53.26	19.56	116.35	1.92
110586	2528.8	1.6	.0096	20.15	124.47	37.55	96.15	62.37	82.41	33.78	183.07	2.00
110686	816.7	-.2	.0099	9.95	62.90	19.41	49.31	30.50	41.19	18.80	92.80	2.06
110786	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
110886	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
110986	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
111086	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
111186	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
111286	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
111386	2367.5	-8.5	.0086	11.58	64.20	19.09	94.85	32.16	42.08	62.69	139.96	2.00
111486	5285.3	-16.2	.0078	20.93	106.44	32.91	214.30	52.50	68.29	161.80	287.82	2.03
111586	745.4	-4.8	.0090	23.93	140.79	43.88	177.97	66.55	90.49	111.41	274.88	2.12
111686	2543.7	-11.7	.0075	21.91	108.96	33.60	195.17	55.95	69 74	139.21	270.53	1.95
111786	664.6	-16.3	.0045	23.53	73.69	21.90	288.38	48.24	45.97	240.14	340.17	1.53
111886	381.5	-17.7	.0045	22.85	71.23	21.27	302.65	45.44	43.92	257.21	352.60	1.57
111986	1879.2	-17.4	.0047	20.40	65.90	19.47	255.11	42.39	41.39	212.72	301.55	1.55
112086	1918.8	-13.5	.0066	23.23	104.01	31.82	239.85	55.62	65.79	184.23	312.05	1.87
112186	2418.7	-10.0	.0079	22.96	119.59	37.21	202.88	59.34	76.39	143.54	285.26	2.02
112286	2506.1	-12.8	.0073	22.69	110.03	34.05	226.92	55.94	69.91	170.99	302.91	1.97
112386	1020.0	-2.2	.0103	22.89	152.54	47.98	158.44	69.11	98.48	89.32	262.99	2.21
112486	1469.5	1.0	.0115	22.93	166.51	52.62	115.22	74.26	108.19	40.95	229.11	2.24
112586	1743.9	-4.4	.0094	23.94	146.24	45.75	168.60	69.11	94.31	99.49	269.09	2.12
112686	1703.8	1.7	.0122	23.54	180.56	57.30	117.31	79.73	117.27	37.58	240.58	2.26
112786	2280.1	-6.2	.0087	22.71	129.60	40.35	166.53	63.62	83.71	102.91	255.77	2.04
112886	2468.4	-10.2	.0078	23.96	123.36	38.43	207.78	61.85	78.8[illegible]	145.93	292.71	1.99
112986	3751.3	-10.8	.0080	22.70	119.69	37.18	205.21	59.42	76.81	145.79	287.71	2.01
113086	1679.8	-7.2	.0082	23.93	129.82	40.34	191.54	64.59	83.19	126.95	281.02	2.01
SUMMARY	46615.3	-7.1	.0084	491.8	2683.8	824.8	4021.9	1344.1	3763.0	2677.8	5880.8	2.00

DECEMBER '86

DATE	RAD	TA	FR12	THP	QOUT	QIN	QTOT	WIN	QEVAP	QAUX	QLOAD	COP
	W/m²	°C	kg/s	Hr	MJ	MJ	MJ	MJ		MJ	MJ	
120186	379.1	-5.3	.0077	24.00	121.50	37.27	191.03	63.52	78.22	127.50	275.26	1.91
120286	5679.2	-11.0	.0070	23.89	111.33	34.32	219.70	60.87	71.67	158.83	296.70	1.83
120386	4494.5	-15.8	.0072	21.30	101.61	31.58	216.09	52.27	65.03	163.82	286.12	1.94
120486	3042.8	-12.8	.0083	20.98	113.73	35.37	195.96	54.68	73.23	141.28	274.33	2.08
120586	444.6	-8.2	.0074	23.96	117.19	36.05	209.33	64.80	74.77	144.53	290.46	1.81
120686	3494.4	-14.1	.0081	14.89	78.72	24.63	157.26	38.66	50.97	118.59	211.35	2.04
120786	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
120886	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
120986	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
121086	1325.2	2.7	.0110	10.44	73.09	22.82	55.54	34.28	48.04	21.27	105.81	2.13
121186	1455.1	-2.1	.0099	23.95	152.06	47.54	137.81	72.13	98.60	65.67	242.33	2.11
121286	738.7	-4.8	.0087	23.93	136.03	42.18	171.99	66.18	87.52	105.80	265.84	2.06
121386	3015.8	-4.1	.0086	21.57	120.32	36.44	158.30	61.69	78.88	96.61	242.19	1.95
121486	3504.2	-8.9	.0079	23.76	123.26	37.75	201.98	62.10	79.70	139.89	287.49	1.98
121586	3599.6	-8.1	.0085	22.96	127.41	39.85	190.11	62.19	82.22	127.92	277.67	2.05
121686	3192.1	-2.6	.0094	22.45	135.64	40.81	147.93	66.92	89.40	81.01	242.76	2.03
121786	781.7	-5.9	.0078	23.90	123.59	38.10	195.25	63.07	79.24	132.18	280.74	1.96
121886	554.1	-3.4	.0086	23.95	134.67	41.55	168.89	66.89	86.95	102.00	262.02	2.01
121986	3489.1	-9.9	.0082	23.94	128.81	40.22	207.82	63.85	82.80	143.97	296.41	2.02
122086	3556.5	-8.6	.0086	23.85	133.13	41.58	205.26	64.89	85.81	140.37	296.81	2.05
122186	260.0	-2.3	.0087	23.07	131.13	40.37	151.15	65.25	84.91	85.90	241.90	2.01
122286	1191.9	-5.6	.0089	23.95	138.96	43.44	188.40	66.41	89.16	121.99	283.93	2.09
122386	3501.0	-3.2	.0097	23.49	146.39	45.37	161.91	69.42	95.27	92.50	262.93	2.11
122486	262.9	-4.4	.0085	23.90	132.82	40.99	191.62	65.28	85.38	126.34	283.45	2.03
122586	1079.7	-6.4	.0080	23.89	126.38	39.01	219.54	63.65	80.99	155.89	306.91	1.99
122686	1036.1	-.9	.0105	23.93	161.74	50.74	161.76	73.82	104.81	87.94	272.75	2.19
122786	2866.8	-3.2	.0099	23.50	150.68	46.89	167.37	70.32	97.98	97.05	271.16	2.14
122886	1612.4	-3.1	.0098	23.[illegible]	151.23	47.34	183.41	70.53	97.64	112.89	287.30	2.14
122986	524.6	-.8	.0097	23.95	149.15	46.34	176.72	71.95	96.82	104.77	279.53	2.07
123086	2802.9	-1.3	.0097	21.90	136.39	41.62	156.82	69.02	89.55	87.79	251.58	1.98
123186	4005.9	-8.8	.0088	23.90	138.05	43.22	225.74	64.85	88.46	160.89	320.57	2.13
SUMMARY	61890.9	-5.8	.0087	633.1	3595.0	1113.4	5014.7	1769.5	6087.1	3245.2	7496.3	2.03

JANUARY '87

DATE	RAD	TA	FR12	THP	QOUT	QIN	QTOT	WIN	QEVAP	QAUX	QLOAD	COP
	W/m^2	°C	kg/s	Hr	MJ	MJ	MJ	MJ	MJ	MJ	MJ	
10187	1549.0	-6.0	.0087	23.95	136.30	42.33	208.79	64.96	87.37	143.83	302.75	2.10
10287	3536.0	-3.7	.0098	23.96	150.66	46.37	169.13	70.04	98.48	99.09	273.42	2.15
10387	740.8	-5.8	.0088	23.88	138.74	43.17	213.05	64.97	88.67	148.08	308.61	2.14
10487	2793.9	-3.0	.0095	22.92	141.52	44.07	160.98	67.62	92.08	93.37	258.43	2.09
10587	3706.0	-10.8	.0081	23.96	127.14	39.66	226.59	62.69	81.84	163.90	314.06	2.03
10687	4248.2	-11.2	.0080	23.89	125.65	39.28	237.15	61.88	80.70	175.27	323.53	2.03
10787	4068.9	-8.7	.0084	22.79	125.12	38.96	210.89	62.49	80.86	148.40	297.05	2.00
10887	3476.1	-9.1	.0082	21.99	117.97	36.75	199.90	58.38	76.09	141.52	281.12	2.02
10987	2353.8	-8.8	.0080	21.91	114.89	35.68	210.77	57.46	73.99	153.30	289.97	2.00
11087	2022.6	-1.5	.0101	23.96	156.43	48.87	161.02	73.10	101.86	87.92	268.57	2.14
11187	2710.0	-.4	.0103	21.75	143.56	43.67	130.11	67.23	95.14	62.88	229.99	2.14
11287	2031.1	-1.3	.0103	20.89	137.81	42.66	132.23	65.87	90.54	66.36	227.38	2.09
11387	392.7	-5.5	.0078	23.25	119.41	36.45	187.84	61.62	77.22	126.22	270.81	1.94
11487	699.1	-14.7	.0046	22.59	71.34	21.23	279.67	46.85	44.65	232.82	329.78	1.52
11587	2522.0	-11.0	.0076	16.16	81.12	24.62	175.67	41.37	52.38	134.30	232.17	1.96
11687	2759.0	-8.2	.0076	19.63	99.55	30.70	179.73	51.65	64.37	128.07	248.58	1.93
11787	531.8	.2	.0102	23.84	156.34	48.66	143.74	76.02	101.89	67.72	251.42	2.06
11887	808.6	-.8	.0102	23.86	156.66	48.82	168.32	74.36	101.95	93.96	276.17	2.11
11987	4339.6	-5.4	.0091	23.51	139.71	42.78	187.75	69.21	91.53	118.55	284.68	2.02
12087	3387.0	.7	.0119	23.37	176.29	55.49	159.54	78.06	115.27	81.49	280.34	2.26
12187	4070.0	-5.7	.0085	21.56	119.22	36.37	169.19	60.99	78.46	108.20	252.03	1.95
12287	3660.1	-11.8	.0074	21.87	107.63	33.25	221.23	56.02	69.36	165.21	295.61	1.92
12387	3197.4	-8.8	.0084	23.39	129.02	39.85	210.37	63.96	83.20	146.41	299.54	2.02
12487	1722.1	-10.0	.0075	23.92	119.44	36.94	244.05	60.14	75.70	183.91	326.55	1.99
12587	3771.6	-8.9	.0075	22.67	111.86	33.62	214.17	60.62	73.05	153.55	292.41	1.85
12687	5111.9	-5.9	.0084	22.04	120.61	36.02	192.34	63.66	79.45	128.69	276.94	1.89
12787	4945.4	-10.6	.0080	23.32	122.43	37.10	219.66	62.59	79.87	157.06	304.99	1.96
12887	2817.4	-5.5	.0092	23.68	143.29	44.34	188.77	68.04	92.99	120.73	287.73	2.11
12987	3196.3	-6.5	.0092	23.86	142.65	44.70	194.27	68.10	91.92	126.17	292.22	2.09
13087	1374.1	-5.0	.0086	23.88	134.62	41.74	180.71	67.52	86.89	113.18	273.58	1.99
13187	3256.5	-5.3	.0089	22.74	131.45	40.13	170.52	65.55	85.65	104.98	261.84	2.01
SUMMARY	85799.0	-6.4	.0087	705.0	3998.4	1234.3	5948.[illegible]	1973.0	11273.9	3975.1	8712.3	2.03

FEBRUARY '87

DATE	RAD	TA	FR12	THP	QOUT	QIN	QTOT	WIN	QEVAP	QAUX	QLOAD	COP
	W/m^2	°C	kg/s	Hr	MJ	MJ	MJ	MJ	MJ	MJ	MJ	
20187	564.6	-6.8	.0073	23.94	115.91	35.63	199.68	62.20	74.33	137.49	279.97	1.86
20287	3676.0	-7.8	.0081	21.47	113.96	34.95	191.91	58.89	74.71	133.02	270.92	1.94
20387	2814.2	-10.0	.0077	23.93	121.21	37.56	242.40	61.32	77.42	181.08	326.05	1.98
20487	4913.4	-5.4	.0095	23.19	141.35	42.97	187.52	68.99	92.99	118.54	285.91	2.05
20587	2393.0	-1.6	.0100	21.88	141.16	43.89	138.82	67.60	92.17	71.23	236.09	2.09
20687	926.1	-.2	.0096	23.88	147.34	45.34	154.43	73.37	96.46	81.06	256.44	2.01
20787	41.7	3.6	.0107	9.98	68.03	21.07	52.10	32.92	44.73	19.18	99.05	2.07
20887	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
20987	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
21087	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
21187	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
21287	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
21387	56.5	-2.9	.0097	9.09	57.30	17.78	65.28	27.02	37.10	38.26	104.79	2.12
21487	721.3	-5.5	.0074	23.96	117.96	35.72	193.99	64.29	76.53	129.70	276.23	1.84
21587	466.0	-5.4	.0079	18.89	97.63	29.63	186.48	51.34	63.29	135.13	254.47	1.90
21687	788.8	-2.7	.0090	23.92	139.34	42.65	190.66	69.74	90.84	120.92	287.35	2.00
21787	2902.0	-2.5	.0085	20.35	112.85	33.67	146.09	60.12	74.73	85.97	225.28	1.88
21887	4296.0	-3.9	.0090	20.23	118.14	35.38	156.96	59.94	78.14	97.02	239.72	1.97
21987	3006.0	-1.2	.0101	19.43	126.08	38.55	124.17	60.83	83.28	63.33	211.70	2.07
22087	4120.8	1.1	.0102	19.62	126.75	37.80	115.07	63.20	84.72	51.87	204.02	2.01
22187	2181.7	-3.1	.0092	22.96	136.77	42.46	160.56	68.10	88.92	92.45	254.87	2.01
22287	2844.0	-2.1	.0091	19.52	115.47	34.99	121.31	58.97	76.27	62.34	201.78	1.96
22387	2098.1	-5.3	.0090	22.91	133.59	41.29	173.66	66.64	86.70	107.02	265.96	2.00
22487	2089.9	-8.7	.0069	23.96	110.78	33.64	208.46	61.90	71.43	146.56	285.61	1.79
22587	6714.5	-12.2	.0084	23.93	132.23	40.98	268.28	62.49	84.81	205.79	359.52	2.12
22687	5485.8	-12.2	.0084	23.91	133.72	41.44	308.53	61.26	85.05	247.26	400.81	2.18
22787	2645.5	-10.8	.0080	23.88	127.06	39.27	239.68	63.18	81.30	176.50	327.47	2.01
22887	4416.0	-10.8	.0076	23.96	120.01	37.14	226.37	61.05	76.83	165.33	309.25	1.97
SUMMARY	60161.9	-5.5	.0086	488.8	2754.6	843.8	4052.4	1385.3	13066.6	2667.1	5963.3	1.99

MARCH '87

DATE	RAD	TA	FR12	THP	QOUT	QIN	QTOT	WIN	QEVAP	QAUX	QLOAD	COP
	W/m^2	°C	kg/s	Hr	MJ	MJ	MJ	MJ	MJ	MJ	MJ	
30187	4454.9	-12.2	.0079	21.70	112.39	34.44	220.05	56.89	72.96	163.15	298.00	1.98
30287	1168.3	-7.5	.0083	23.94	131.55	40.60	215.63	64.07	84.09	151.56	306.58	2.05
30387	1278.0	-8.9	.0075	23.96	119.52	36.56	225.09	62.18	76.57	162.91	308.05	1.92
30487	1580.8	-2.4	.0091	19.80	117.51	35.07	148.20	58.66	76.97	89.54	229.84	2.00
30587	1927.0	-6.6	.0088	23.86	137.60	42.48	186.29	68.15	89.05	118.14	281.41	2.02
30687	2489.3	-3.3	.0091	18.47	108.86	33.13	128.62	56.33	71.58	72.28	204.35	1.93
30787	3638.6	-8.3	.0077	23.76	119.80	36.38	198.98	64.61	78.00	134.37	282.40	1.85
30887	5810.8	-12.7	.0076	23.94	121.43	37.38	285.71	59.44	77.45	226.27	369.76	2.04
30987	1538.8	-10.0	.0072	23.88	115.15	35.19	235.95	60.59	73.74	175.36	315.92	1.90
31087	5712.0	-12.5	.0077	22.26	112.01	33.51	221.10	59.00	73.56	162.10	299.60	1.90
31187	1990.6	-9.1	.0087	23.96	138.80	43.20	247.06	63.42	88.29	183.64	342.66	2.19
31287	1151.2	-5.0	.0094	23.89	146.79	45.58	190.53	68.60	94.70	121.92	291.74	2.14
31387	827.8	-3.4	.0096	23.87	148.51	46.21	172.25	70.19	96.04	102.06	274.55	2.12
31487	2446.6	-4.3	.0085	21.20	117.37	35.20	150.40	61.77	77.30	88.64	232.57	1.90
31587	2112.5	-5.7	.0079	20.05	103.76	31.23	160.16	57.48	67.71	102.67	232.68	1.80
31687	5456.0	-6.7	.0096	22.00	135.93	41.36	185.91	64.76	89.26	121.15	280.48	2.10
31787	3309.4	.9	.0117	22.61	166.70	51.58	134.61	75.72	109.59	58.89	249.73	2.20
31887	603.9	.5	.0101	23.94	154.49	47.63	139.78	75.36	101.53	64.43	246.64	2.05
31987	754.8	.2	.0095	23.89	146.48	44.91	153.19	73.71	96.45	79.48	254.76	1.99
32087	2917.4	-2.1	.0077	22.51	113.52	33.98	150.63	64.66	75.19	85.98	230.17	1.76
32187	2412.4	-2.9	.0078	23.97	122.51	36.98	177.75	68.81	80.67	108.94	263.28	1.78
32287	1730.2	-4.1	.0076	23.94	119.86	36.24	179.31	66.86	78.49	112.45	262.94	1.79
32387	4490.9	-5.0	.0078	22.06	112.78	33.32	168.10	63.01	74.82	105.09	247.57	1.79
32487	1944.0	-4.1	.0081	16.63	88.27	26.61	138.52	47.39	57.94	91.12	200.18	1.86
32587	3354.1	.0	.0096	19.29	117.40	34.80	129.47	61.08	78.10	68.39	212.01	1.92
32687	1942.6	-1.2	.0104	23.91	158.89	49.60	158.41	75.99	103.60	82.42	267.69	2.09
32787	6988.6	-14.1	.0103	15.89	104.75	33.01	160.41	47.91	68.44	112.51	232.15	2.19
32887	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
32987	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
33087	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
33187	.0	.0	.0000	.00	.00	.00	.00	.00	.00	.00	.00	.00
SUMMARY	74031.5	-5.5	.0087	599.2	3392.6	1037.0	4862.1	1716.6	[illegible]	3345.5	7217.[illegible]	1.98