AN ABSTRACT OF THE THESIS OF

Bruce Walter Griswold for the degree of Master of Science

in Agricultural Engineering presented on November 2, 1982

Title: Development of a Water-to-Water Heat Pump for Tempering Water in Milking Parlors

Abstract approved: Redacted for privacy

Martin L. Hellickson

A water-to-water heat pump was specifically designed and operated to temper process water for cleansing and sanitizing milking parlor equipment using recovered energy from the milk refrigeration system as its heat source. Operation of the water-to-water heat pump in series with the energy recovery system was found to be both energy conserving and economically feasible. The study compared water usage and electrical energy consumption for the period before and after installation of the complete energy recycling system. A 53 percent reduction in electrical energy consumption for tempering low ($43.3^{\circ}C$) and high ($71.1^{\circ}C$) grade hot water resulted. An average coefficient of performance of 4.05 was realized for the heat pump, indicating that with rigid water (energy) conservation practices improved performance would result. Several developments were suggested by the study which may improve overall system performance and further reduce energy consumption. The use of a water-to-water heat pump for tempering water appears to have a promising future for dairy farm applications.

DEVELOPMENT OF A WATER-TO-WATER HEAT PUMP
FOR TEMPERING WATER IN MILKING PARLORS

by

Bruce Walter Griswold

A THESIS

submitted to

Oregon State University

in partial fulfillment of
the requirements for the
degree of

MASTER OF SCIENCE

Completed November 2, 1982

Commencement June 1983

APPROVED:

Redacted for privacy

Associate Professor of Agricultural Engineering in charge of major

Redacted for privacy

Head of Department of Agricultural Engineering

Redacted for privacy

Dean of Graduate School

Date thesis is presented: November 2, 1982

Typed by Cheryl A. Graham for Bruce Walter Griswold

ACKNOWLEDGEMENTS

The author wishes to express his appreciation to the many individuals and organizations who have contributed support and advice toward completion of this project.

The design and development of the water-to-water heat pump would not have been possible without the financial support of Pacific Power and Light Company. In addition, they provided information and material whenever requested.

Special thanks go to my major professor, Dr. Martin L. Hellickson, for having the answers and the questions, when I needed them. His encouragement and constructive suggestions are a credit to his profession.

Recognition must also be made to the Oregon State University dairy center for allowing this research project to be maintained at their milking facilities and the Agricultural Engineering Department shop for their guidance and fabrication of the water-to-water heat pump. I would also like to thank Ms. Cheryl Graham whose flying fingers completed the final copy with moments to spare. Her professionalism and cheerful advice made the typing process less painful.

Saving the best for last, I would like to thank my wife, Syl, and daughter, Linzy Leilani. Their support and companionship made these past two years both a fulfilling and enjoyable experience.

CONTENTS

LIST OF FIGURES

LIST OF TABLES

NOMENCLATURE

A	area
C_p	specific heat
E	Annual energy savings
f	fixed percentage of initial investment
h	enthalpy
I	initial investment
L	rate of removal of energy
M	mass of fluid in control volume
m	system life
$\dot{m}$	mass flow rate
N	total system life
Q	rate of addition of energy
T	temperature
t	time
U	overall heat transfer coefficient
W	capacity coefficient
X	economic inflation rate
Y	energy escalation rate
Z	money interest

<u>Subscripts</u>

a	ambient
B	compressor inlet
C	compressor outlet
c	city water

C_{in}	water entering heat exchanger section
C_{out}	water exiting heat exchanger section
cv	control volume
E	heat pump evaporator
e	exit state conditions
H	high temperature water
H_{in}	refrigerant entering heat exchanger section
H_{out}	refrigerant exiting heat exchanger section
i	entrance state conditions
L	low temperature water
o	initial value
P	heat pump condenser
R	refrigerant
s	water storage tank section
W	water-cooled condenser

DEVELOPMENT OF A WATER-TO-WATER HEAT PUMP FOR TEMPERING WATER IN MILKING PARLORS

I. INTRODUCTION

Energy utilized for heating and cooling accounts for a major portion of the total energy input in many agricultural production operations. Presently, the most effective means of offsetting rising energy costs are improvement in energy use efficiencies and design of energy recycling systems. If and when alternative energy sources become technologically and economically feasible for the farmer, these may also be accepted and adopted on a large scale.

A sizable percentage of the total energy input to dairy farms is that used for heating in the milking parlor. Approximately 16 percent of the purchased energy on dairy farms in the United States is used for water heating (USDA, 1977). Bickert (1979) reported the hot water requirements in milking parlors as shown in Table 1.

Commercial electric and gas water heaters provide conventional means of supplying low (40.5^{o} to $43.3^{o}C$) and high (71.1^{o} to $76.6^{o}C$) temperature water needs in milking parlors. End use water temperatures are met by drawing hot water from the water heater and mixing with cold water. At the same time that energy is being used to heat water, it must also be utilized to cool the milk. Studies have shown that approximately the same quantity of energy required in heating water for milking parlor needs is available during the milk refrigeration process. Elwell, Roller and Keener (1980) reported that the management strategy of matching supply quality (temperature) and quantity to demand quality and quantity is appropriate in milking operations.

Table 1. Hot water requirements in milking parlors[a].

Water Temperature	Description of Water Use	Quantity
40.5^{o} to $43.3^{o}C$	Cow preparation	
	automatic	11.4 - 34.1 liter/cow-day
	manual	1.9 - 3.8 liter/cow-day
	Parlor and milk-house floor	151.4 - 283.9 liter/day
71.1^{o} to $76.6^{o}C$	Milk storage tank	
	automatic wash	189.3 - 227.1 liter/wash
	manual wash	113.6 - 151.4 liter/wash
	Milk pipeline wash	283.9 - 473.2 liter/wash
	Milkers and misc. equipment	189.3 liter/day

[a]Source: Bickert (1979).

Study results indicated that the energy available in cooling milk from $40^{o}C$ to nearly $0^{o}C$ at 20 liters of milk per cow closely matched the energy required to heat process water from approximately $10^{o}C$ to $50^{o}C$ at up to 20 liters of water per cow. This was provided by the heat recovered from the milk refrigeration system with a desuperheater heat exchanger. At water quantities less than 20 liters per cow, excess energy in the form of hot water was available. At water quantities greater than 20 liters per cow, additional energy was required to maintain the desired quality of hot water.

The reclamation of rejected thermal energy from milk refrigeration systems is thus an excellent alternative to conventional methods of heating water. Significant reductions in energy requirements for low temperature (40^{o} to 50^{o}C) water needs have been demonstrated with energy recovery units by Thompson and Fairbanks (1979), Hellickson and Kirby (1979), Koelsch (1979), and Stipanuk et al. (1979).

A further extension of this energy recycling concept is application of the heat pump principle to boost low grade water temperature (40^{o} to 50^{o}C) to a higher level where it can be utilized for high temperature (71^{o} to 76^{o}C) washing operations in milking parlors. This project and resulting thesis were initiated to determine the energy conservation potential and economic feasibility, under actual production conditions, of a water-to-water heat pump specifically designed and operated to temper water for cleansing and sanitizing milking parlor equipment. Presently, commercially designed and produced water-to-water heat pumps of the size and specific capabilities required for use in milking parlors are not available. The main thrust of this project was the utilization of off-the-shelf refrigeration components specifically selected to provide the hot water requirements in milking parlors. The following specific objectives were established for this study:

(1) Design and assemble a specialized water-to-water heat pump using off-the-shelf refrigeration components based on the heating load requirements and reject thermal energy available from the milk refrigeration system in the Oregon State University dairy center.

(2) Install and monitor this specialized water-to-water heat pump under actual production conditions.

(3) Determine the energy conservation potential and economic feasibility of this heat pump.

(4) Use the data collected as a base for verification of a computer model to simulate operation and performance of this total energy recycling system.

II. LITERATURE REVIEW

Heat Recovery from Milk Refrigeration Systems

Reclamation of waste heat from milk refrigeration systems is not a new concept but an old one rediscovered. Studies as early as the late 1940s were performed to reclaim refrigeration system waste heat for water heating (Zastrow, 1948). Turner (1959 and 1960), through actual performance studies done at Cornell University, indicated a reduction in water heating costs of 52 percent by adding a water-cooled condenser to a milk refrigeration system. Although these results were case specific, they led to the following further studies by Cromarty (1968):

(1) Optimization of the water-cooled condenser design

(2) Determination of the hot water draw-off patterns on the energy recovery efficiency of the water-cooled condenser

(3) Determination of the water-cooled condenser's effect on the supplemental heat added to the milking area to prevent freezing

Presently, there are several manufacturers of heat exchangers and desuperheaters for the specific purpose of reclaiming waste heat from milk refrigeration systems. Field studies by Koelsch (1979) indicate two types of heat reclamation systems were being supplied by manufacturers. The first type replaces the traditional air-cooled condenser with a complete water-cooled condenser which has the potential of removing 100 percent of the energy of compression. In reality, 50 to 75 percent of this energy is removed. Water temperatures leaving this type of heat exchanger were reported to range from 52° to 71°C depending upon the particular brand name condenser installed and the dairy farmer's management practices. Literature sources for this type of heat exchanger did not investigate the effects of these heat recovery units on compressor discharge pressures. At these exit water temperatures, compressor discharge pressures for Freon 22, the refrigerant commonly utilized in milk refrigeration systems, range from 2027 to 3113 kPa (294 to 452 psia). These energy recovery systems rely on a compressor head pressure sensor to control excess heat recovery. Water is released from the high temperature storage tank at a specified compressor discharge pressure allowing cooler water to enter the storage tank. This precautionary measure reduces water temperatures through the condenser and thus, reduces compressor discharge pressures. Koelsch (1979) reported compressor sensor settings of 1550 to 2400 kPa (225 to 348 psi) for discharging hot water from storage or engaging an auxillary air-cooled condenser. Freon 22 may approach its critical temperature and pressure of 96.0°C and 4988 kPa during operation of the system with sensor settings in the range specified. Breakdown of the lubricating fluid in the compressor may result at these high temperatures and pressures. This will eventually lead to compressor failure. Another factor affecting the life of the compressor is the temperature differential of the cooling fluid through the condenser. Unless a moderate temperature differential is maintained through the heat exchanger, the compressor will require a longer period of operation to reject the same amount of energy. This, in turn, will slow the milk refrigeration process which is a crucial factor in reducing bacterial growth in milk. The

American Society of Agricultural Engineers Engineering Practice: ASAE EP256.2 (Agricultural Engineers Yearbook, 1980 - 81) reported the milk refrigeration system must have the capacity to cool 25 percent of the rated volume of the milk storage tank from 32° to 10°C within one hour after tank has been filled to 25 percent of its rated capacity. The cooling system is in operation during the filling period. The system must further cool the milk from 10° to 4°C during the next hour.

Figure 1 is a schematic of a milk refrigeration system with a water-cooled condenser unit completely replacing the air-cooled condenser. Results by Koelsch (1979) from 18 New York State dairy farms, indicated a 50 to 75 percent recovery of the total heat rejected during milk cooling. This heat recovery efficiency was based on the availability of approximately 155 kJ of energy per kilogram of milk cooled from 34° to 4°C. Of this 155 kJ of energy, 75 percent

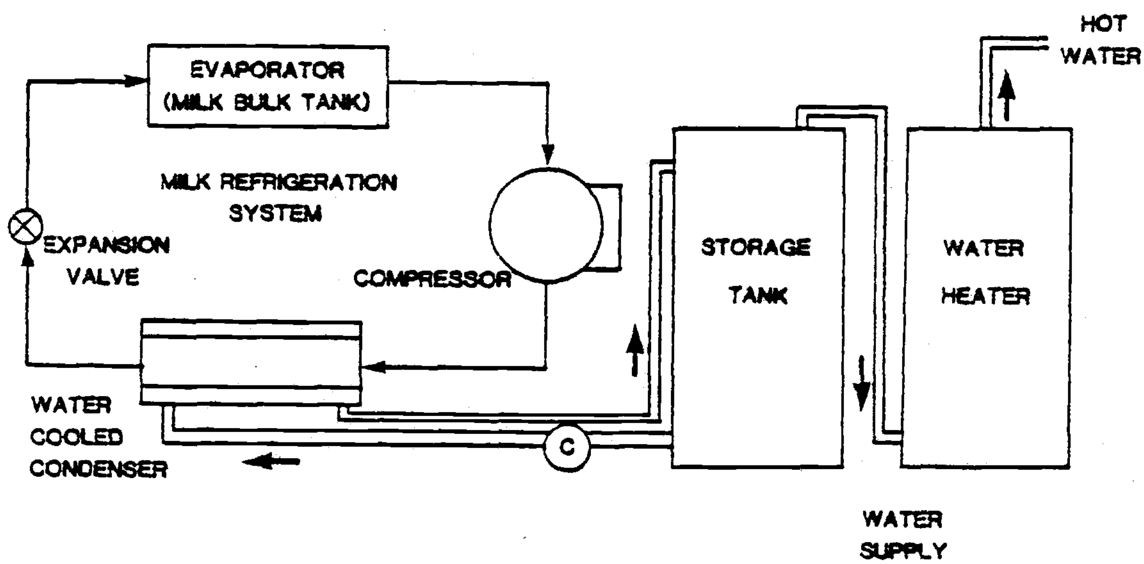

Figure 1. Water-cooled condenser unit replacing an air-cooled condenser within the milk refrigeration system.

was removed from the milk and 25 percent was from the compressor. Skinner (1980) monitored a complete water-cooled condenser unit for a 28-month period on a Tennessee dairy farm and reported an energy savings of $1.86 to $3.11 per day based on an electrical energy charge that ranged from 2.5 to 3.25 cents per kilowatt-hour over the 28-month period. Estimated total savings during this period was $2,100.00 -- enough to pay for the complete water-cooled condenser unit in approximately 18 months.

The second type of heat exchanger is known as a desuperheater which utilizes the refrigerant superheat condition to heat water. These units are an add-on feature to most refrigeration systems and are installed in the discharge line of the compressor. Figure 2 is a schematic of a milk refrigeration system with a desuperheater heat exchanger in series with the standard air-cooled condenser unit. In

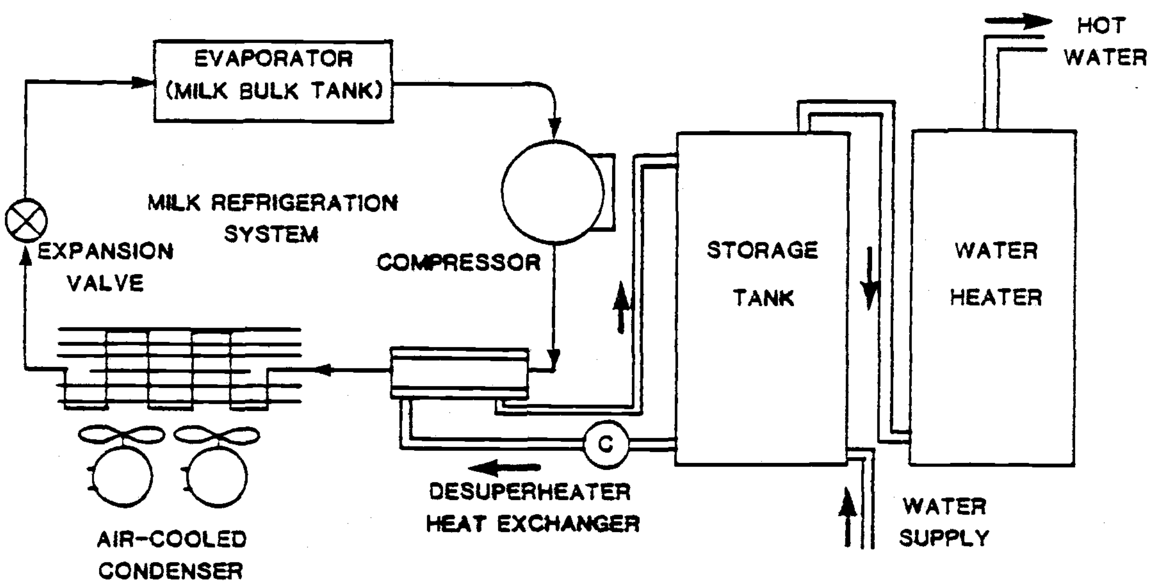

Figure 2. Desuperheater heat exchanger installed in series with air-cooled condenser unit in milk refrigeration system.

refrigeration systems using Freon 22 as the refrigerant, compressor discharge temperatures of 40.5^{o} to 60^{o}C and pressures of 1,491 to 2,527 kPa (216 to 367 psia) occur during normal operation. These conditions are more compatible with the manufacturer's recommended temperatures and pressures for Freon 22 through the compressor. Bickert (1979) and Stipanuk et al. (1979) reported water exit temperatures from the desuperheater heat exchangers ranging from 35^{o} to 43^{o}C. These desuperheaters have the capacity of recovering approximately 30 percent of the available energy or approximately 50 kJ of the estimated 155 kJ of energy per kilogram of milk available for reclamation. Koelsh (1979), in his New York State study, found the desuperheater heat exchangers reduced the energy required to heat water by approximately 50 percent. Thompson and Fairbanks (1980) reported up to 74 percent reduction in energy used to heat water after installation of a desuperheater heat exchanger in a California dairy milking 860 cows twice daily. Before installation, records showed the purchased energy required to meet the hot water needs was 2,260 MJ per day, and after installation, the energy requirement dropped to 580 MJ per day. Milk refrigeration equipment operated approximately 19 hours per day. Additional tests indicated a range from 26 to 35 percent efficiency of total waste heat recovery.

Stipanuk et al. (1979) performed a financial analysis for a 100-cow dairy utilizing a desuperheater heat exchanger providing 40.5^{o}C water and a complete water-cooled condenser unit providing 60^{o}C water. This analysis compared annual savings based on 74^{o}C hot water needs of 1500 liters per day. Results indicated an after-tax payback period (initial cost divided by annual savings) of three to four years for both types. Hellickson (1980), in a performance test of a desuperheater heat exchanger installed at the Oregon State University dairy center, realized a monthly savings of $71.79 in energy costs based on a 3.0 cents per kilowatt-hour electricity charge. The results were based on an 18-month study that monitored a herd of approximately 130 cows milked twice daily. The payback period in this case was approximately two years. Thompson and Fairbanks (1979), in their study of the two types of heat exchangers at several on-farm installations in California, realized a 12 to 18-month payback period. Their economic analysis did not include interest rates, taxes, and governmental or utility incentive payments.

Conclusions drawn from the research reports reviewed are as follows:

(1) Consistent energy savings were available for heat recovery systems, irrespective of the type of heat exchanger installed.

(2) Energy conservation potential for an individual unit was largely dependent on the dairy farm's management practices involving hot water consumption. Koelsch (1979) found a wide range of hot water consumption at the 18 New York State dairies studied. Hot water usage of 3.9 to 14.5 liters per cow-day were reported. This indicated a possible energy reduction by simply conserving the quantity of hot water used.

(3) Further studies may be required to determine the effect of the complete water-cooled condenser unit on compressor operating life. A trade-off point may

occur where the additional energy recovered is negated by the additional maintenance costs associated with reduction in compressor life.

History of the Heat Pump

The heat pump is a device for extracting thermal energy from a low temperature heat source and raising it to a higher temperature where the thermal energy can be utilized more effectively. Energy in the form of electricity, mechanical work, or high temperature thermal energy must be supplied to the heat pump to upgrade this thermal energy. In 1824 Nicholas Carnot first developed the basic heat pump principle. Thirty years later, in the 1850s, Lord Kelvin supported this theory by suggesting the use of refrigeration equipment for heating (Pietsch, 1977; Sporn et al., 1947). From the 1850s to the mid 1930s, research in the development of a feasible heat pump system for heating continued at a slow pace. During the late 1930s a small number of demonstration installations were attempted. Although few in number, these projects confirmed the principle of the heat pump as set forth by Kelvin (Pietsch, 1977; Kemler and Oglesby, 1950).

The development of a unitary heat pump in the early 1950s resulted from research work accomplished during the previous decade. Researchers concluded that public acceptance of heat pumps for heating would improve if a unitary model could be developed. Unitary heat pumps are factory-designed and factory-built units that contain the major components and corresponding controls and are shipped and installed in one or two sections. These air-to-air heat pumps were specifically designed for residential application and ranged from 5.3 to 70.3 kW of heating capacity. Units were first brought into production in the southern United States. Heat pump failures due to increased stress on components designed for cooling plus declining energy costs severely reduced the production of heat pumps in the late 1950s (Pietsch, 1977). Improvements in component design and durability were the main thrust of manufacturers in the 1960s and early 1970s. As the public became more aware of the energy shortage situation in the 1970s, interest renewed in the heat pump for heating and its efficient use of energy in certain other applications. By this time industry had corrected earlier design mistakes and further development activity increased dramatically (Pietsch, 1977; Moore, 1976).

Current heat pumps employ a wide array of heat sources and sinks. Heat pump classification generally falls under the categories listed in the ASHRAE Systems Guide and Data Book (1980) and presented in Table 2. Other heat sources being considered for the heat pump include solar energy and geothermal energy. Each of these has its own unique set of advantages and disadvantages and will not be discussed here.

Basic Heat Pump Cycle

The basic heat pump circuit consists of an evaporator, compressor, condenser, and expansion valve, connected as shown in Figure 3. The evaporator receives low grade thermal energy from a waste heat source, the heat of vaporization being absorbed by the working fluid (refrigerant) circulated through the heat pump.

Table 2. Heat pump classifications.

Category	Typical Use	Disadvantages	Advantages
Air-to-Air	residential heating and cooling	least heat available when demand greatest; coil frosting	low initial and operating cost; universal availability of heat source
Air-to-water	industrial hot water heating; large building climate control	same as air-to-air	same as air-to-air
Water-to-air	residential and industrial heating and cooling	corrosion scale forms on heat transfer surfaces	low initial and operating cost; approximately constant heat source temperature year-round
Water-to-water	industrial hot water heating	corrosion scale forms	same as water-to-air

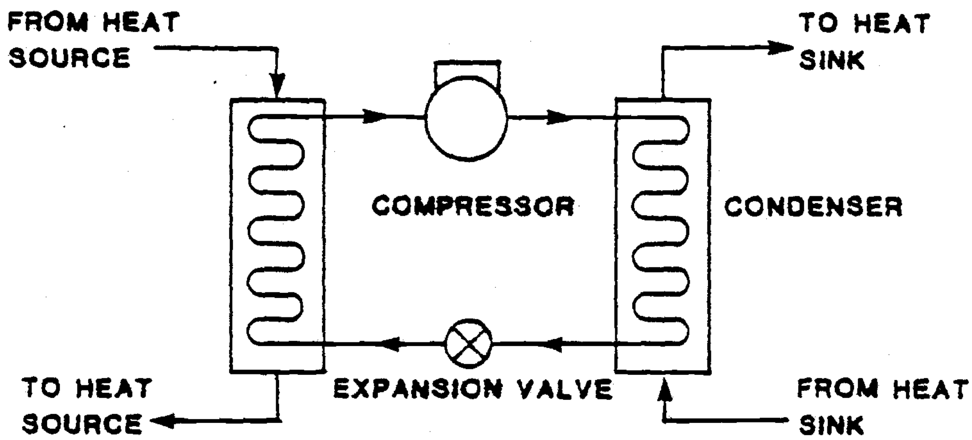

Figure 3. Basic heat pump circuit.

The refrigerant leaves the evaporator as a low pressure, low temperature, saturated vapor and enters the compressor where its temperature and pressure are increased through the addition of mechanical work of compression. From the compressor this high temperature, high pressure, superheated vapor enters the condenser where the refrigerant is first desuperheated and then condensed to a liquid by rejecting the heat equivalent of work of compression and the heat collected at the evaporator. The hot condensate then passes through a valve where expansion of the refrigerant results in a low temperature, low pressure, low quality (partial liquid/partial vapor) working fluid. Upon leaving the expansion valve, the refrigerant enters the evaporator as a mixture of approximately 75 percent liquid and 25 percent vapor, where the cycle is repeated. The working fluid, commonly a fluorinated hydrocarbon, circulates through a closed loop circuit during the cycle (Reay, 1979).

Use of the conventional pressure-enthalpy (Mollier) diagram provides a visualization of the changes of state that occur during the various thermodynamic processes in a heat pump cycle. Figure 4 depicts a pressure-enthalpy diagram for a heat pump refrigerant cycle as presented by Healy et al. (1965). Temperatures and pressures are not included in the figure because of the variability of heat cycles for various refrigerants and their application. The primary emphasis of this illustration is comprehension of the thermodynamic processes that occur during a heat pump cycle.

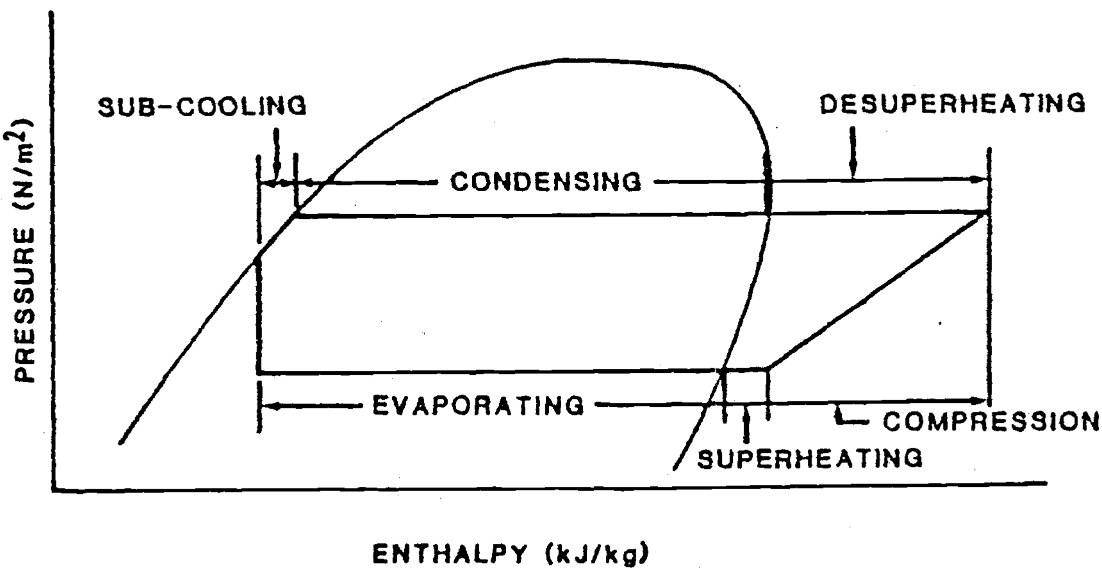

Figure 4. Pressure-enthalpy diagram for a heat pump refrigerant cycle.

An index of the performance of heat pumps is the coefficient of performance (COP). Ambrose (1966) defined the actual coefficient of performance of a heat pump, during the heating cycle, as being equal to the total instantaneous energy output at stated conditions divided by the heat equivalent of the work required to produce this effect. In equation form, this becomes:

$$\text{COP} = \frac{\text{Energy output}}{\text{Work input}} \tag{1}$$

As illustrated in Figure 4, the quantities of energy being compared in the numerator and denominator yield:

$$\text{COP} = \frac{\text{Desuperheating + condensing + subcooling}}{\text{Compression}} \tag{2}$$

A heating coefficient of performance of 4.0 indicates that 1 kW of work input to the heat pump in the form of electricity will provide 4 kW of thermal energy to the heat sink. In terms of electrical energy consumption and ultimately operating costs, one-fourth or 25 percent of that used in a standard electric resistance heating system will be required.

Several factors have considerable influence on the efficiency and hence, the coefficient of performance of a heat pump. Sporn et al. (1947) stressed the importance of operating the system at a minimum compressor discharge pressure and maximum compressor suction pressure to maintain a high coefficient of performance. Ambrose (1966) noted that the coefficient of performance varies directly as the compressor suction pressure, which in turn, is a function of the heat source temperature. Maintaining a high heat source temperature helped maintain a high compressor suction pressure, and thus, a high coefficient of performance. Gilman (1975) listed the end results of lowering the evaporating temperature, at constant condensing temperature, as:

(1) decreased coefficient of performance.

(2) increased work of compression per pound of refrigerant circulated.

(3) decreased refrigeration effect per pound of refrigerant circulated.

(4) decreased heating and cooling capacity.

(5) increased compression ratio and thus, lower volumetric efficiency.

(6) decreased pounds of refrigerant pumped per unit time.

Gilman also stated an increase in evaporator heat transfer area will increase evaporation temperature and decrease the temperature differential between the heat source and heat sink. Keeping this temperature differential to a minimum will result in a heat pump with a high coefficient of performance and decreased operating costs.

Water-to-Water Heat Pump Applications in Agriculture

The water-to-water heat pump cycle is identical to the basic heat pump cycle with the exception of the heat source and heat sink. Water is used as both the heat source and heat sink and to transfer thermal energy from the condenser and to the evaporator in the water-to-water heat pump. The temperature of the water being circulated through the evaporator (heat source) is decreased as it transfers thermal energy to the refrigerant. Conversely, the temperature of the water circulated through the condenser (heat sink) is increased as it absorbs thermal energy from the refrigerant. The water-to-water heat pump may operate with its refrigerant being circulated in the heating-only cycle (i.e., heat absorbed in the evaporator and rejected in the condenser), or in a cooling mode. The cooling mode may be established by changing the direction of water flow through the heat exchangers to make the heat source a sink and the sink a source while maintaining a fixed refrigerant circuit.

The ability to amplify energy output gives the heat pump outstanding potential for on-farm use, particularly in energy recycling systems and for residential water and space heating. Braude (1979) used computer models to predict and compare annual energy consumption for both air and water source heat pumps at several locations in the northern and southern United States. Results indicated the hydronic heat pump to be more favorable in northern climates based upon consistent well water temperatures used for the heat source. At all locations water source heat pumps required less supplemental heat than air source heat pumps. Hustrulid and Cloud (1952), using a home refrigerator unit as the heat source for tempering water, found a system of this configuration to be practical if a "performance factor" (COP) of two or greater was achieved.

The application of a water-to-water heat pump for on-farm use requires careful consideration to provide the farmer with substantial energy savings through waste heat recovery and maximum efficiency in applying energy conservation techniques. The following considerations, established by Westinghouse (1979) in a pamphlet developed for industrial applications of their large commercial water-to-water heat pump units, are also applicable for small agricultural applications.

(1) There must be a "free" source of waste thermal energy available in the form of water or a similar fluid, the higher the temperature the better.

(2) There must be a requirement for process hot water at a usable temperature below 100°C.

(3) There must be the capacity for thermal storage at either the heat source or sink when the process and source energy usage are not simultaneous.

(4) Keeping the temperature differential between the heat source and delivery hot water to a minimum will increase performance characteristics of the heat pump. This will reduce the initial and operating costs of the heat pump.

(5) A high annual hot water requirement that produces the most hours of operation per year will increase the energy savings derived from the heat pump to offset the heat pump's higher installed initial cost. The results of this are a shorter payback period and a greater return on the capital investment.

No documented research was available on the application of a water-to-water heat pump to boost water temperatures in milking parlors using reclaimed thermal energy from the milk refrigeration process as the heat source. Lang (1979) reported on the use of an air-to-water heat pump installed in a dairy in Denmark that recovered heat exhausted in the barn ventilation system to heat water. Preliminary results indicated a substantial savings in energy costs for the 24 months it was installed and monitored.

III. MATERIALS AND METHODS

Operating Conditions at Research Site

The Oregon State University dairy research center has a herd capacity of 150 cows with an average of approximately 130 cows in various stages of lactation. The dairy center utilizes a double-four herringbone milking parlor for its milking operation. Present management practices maintain a schedule of two milkings per day from 6:00 to 11:00 am and pm. Average daily milk production is 24 kg per cow-day (for milk: 1.04 kg $\cong$ 1.0 liter).

During each milking operation low grade hot water (43.3^{o}C) is supplied to the milking parlor for manual cleansing and stimulation of each cow's udder. This water is also used to rinse down the milking parlor and washroom floors at the conclusion of each milking period. Records kept by Hellickson (1980) for the period from March 1979 through June 1980 indicated an average of 1,423 liters per day of low grade hot water was used in the milking operation. This amounted to 11.0 liters per cow-day including floor washdown. A new metering system for the low grade hot water was installed in early 1981 to reduce usage. This system maintains a high pressure required for cleansing and udder stimulation but reduces the quantity of water used. Records maintained from January 1981 through October 1981 indicated an average of 490 liters per day of low grade hot water was used. This amounted to 3.77 liters per cow-day, a 66 percent reduction over previous low grade hot water consumption.

The low grade hot water supply is preheated using energy recovered from the primary milk refrigeration system and stored in an insulated tank. Water drawn for milking parlor use from the storage tank passes through a 189.3 liter commercial electric water heater before entering the water metering system. The water heater thermostats are set at 43.3^{o}C to increase water temperature if below the thermostat setting.

The milking equipment and conveyance lines are cleaned at the conclusion of each milking period. Cleaning operations consisted of a rinsing cycle with warm (43.3^{o}C) water, a wash cycle with hot (71.1^{o}C) water, and a rinsing and sanitizing cycle with warm water (43.3^{o}C). Warm water is obtained by blending hot and cold water. High grade hot water is also utilized in the wash and sanitize cycle for the milk storage tank. Milk pick-up occurs every other day at approximately 12:00 pm. Additional miscellaneous uses for high grade hot water include washing calf bottles and other small milking utensils at approximately 10:00 am and 5:00 pm. Records kept from March 1979 through June 1980 indicated an average use of 863 liters per day. Records kept from September 1981 through May 1982 indicated an average use of 775 liters per day. This 18 percent reduction was a result of management practices to reduce consumption of high temperature water for miscellaneous usage. The cleansing and sanitizing operations are automated and thus, unaffected by management practices. The high temperature (71.1^{o}C) water is drawn from a 378.5 liter commercial electric resistance water heater that has both upper and lower thermostats set at 71.1^{o}C.

Waste Heat Recovery System

A waste heat recovery system was installed at the Oregon State University dairy center in 1978 for the dual purpose of determining the quantity and temperature of water that could be accumulated from a desuperheater heat exchanger installed in a milk refrigeration system and evaluating the system cost and length of time required to recover the capital investment (Hellickson and Kirby, 1979). The system consisted of a 10.6 kW counterflow tube-in-tube heat exchanger installed in the compressor discharge line on the primary milk refrigeration system. The water-cooled condenser unit was specifically designed for water flow through the inner tube and refrigerant flow through the annular space. The inner tubes were finned in the desuperheating section of the heat exchanger to increase heat transfer area and thus, improve efficiency. All tubing was formed using seamless copper with a bronze header at each end. Condenser specifications are included in Appendix A.

The primary milk refrigeration system, which consisted of a flat plate heat exchanger cooled by a sweet (fresh) water ice builder storage unit, operated approximately 18 hours per day. The compressor started approximately one hour after milking commenced and ran for approximately nine hours. Milk passed through the plate heat exchanger and was cooled from 35°C to 4.4°C before entering the milk storage tank. The rejected heat from this sweet water refrigeration system was the heat recovered by the water-cooled condenser. Additional cooling of the milk in storage from 4.4°C to 3.3°C was provided by a separate refrigeration system. No attempt was made to recover waste heat from this system as it represented only a small fraction of the total energy available.

Low Temperature Water Storage Tank

A new low temperature water storage tank was selected to replace the existing water storage tank used by Hellickson and Kirby (1979). The existing water storage tank consisted of a 4,700 liter insulated fiberglass tank open to the atmosphere. This tank was replaced to convert the system to city water pressure and re-evaluate the water storage capacity required. Sizing and selection of the low grade hot water storage tank were based on the following criteria:

(1) The quantity and quality of low temperature (40.5°C) water stored must meet the minimum daily requirements in the milking parlor with the least amount of additional heating by a commercial electric water heater.

(2) There must be sufficient quantity and quality hot water to provide the water-to-water heat pump with the highest temperature heat source possible.

(3) The water storage tank must operate at a significant degree of stratification. This would allow hotter water at the top of the tank for milking parlor use and the heat source of the heat pump. Cooler water from the bottom of the tank could be circulated through the water-cooled condenser for maximum energy transfer between the refrigerant and water.

(4) The storage tank construction must meet safety and sanitary regulations as specified by local and USDA codes.

Based on the analysis of energy recovered from the primary milk refrigeration system and energy required to meet the low grade hot water needs, an 833 liter galvanized steel tank was selected. Calculations for sizing and selection of the water storage tank are included in Appendix B. Additional taps were installed in the tank to provide the plumbing arrangement as shown in Figure 5. The tank was insulated

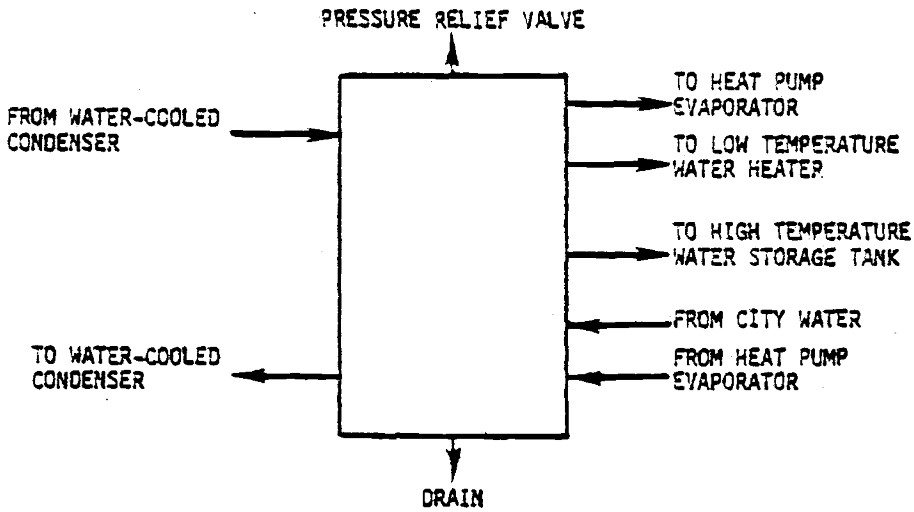

Figure 5. Low temperature water storage tank plumbing arrangement.

with 15.2 cm of fiberglass batting material and installed outside adjacent to the milking parlor in August 1981. A corrugated fiberglass shelter was constructed to protect the tank from the elements.

Water-to-Water Heat Pump Component Selection

The water-to-water heat pump system included the heat pump, the heat sink thermal storage tank (71.1^{o}C), and the heat source thermal storage tank (40.5^{o}C) previously described. Availability of components and ease of replacement are important factors to be considered by the dairy farmer where an extended down-time due to equipment failure can be extremely detrimental. Therefore, serious consideration was taken to assemble a water-to-water heat pump that utilized standard off-the-shelf refrigeration and air conditioning components specifically selected to provide the hot water requirements in milking parlors. Figure 6 is a schematic showing heat pump components (refrigerant and water) plus flow paths.

Refrigerant Selection

Sizing and selection of the heat pump components were based on the underlying decision to use Freon 114, a refrigerant for high temperature applications. Other common refrigerants such as Freon 12 or 22 have excessive condensing pressures at the temperatures involved in this application. ASHRAE Handbook of Fundamentals (1981) recommends pressure in an evaporator be as high as possible and, at the same time, a low condenser pressure in the design of a heat pump is desirable. Evaporator and condenser pressures at the desired evaporating and condensing temperatures of 26.7^{o} and 82.2^{o}C, respectively for Freon 12, 22, and 114 are shown in Table 3. Both Freon 12 and 22 have pressures that are detrimental to a compressor's service life at this condenser temperature. Personal communication with a manufacturer indicated a 26.7^{o}C to 32.2^{o}C suction gas temperature for Freon 114 into a

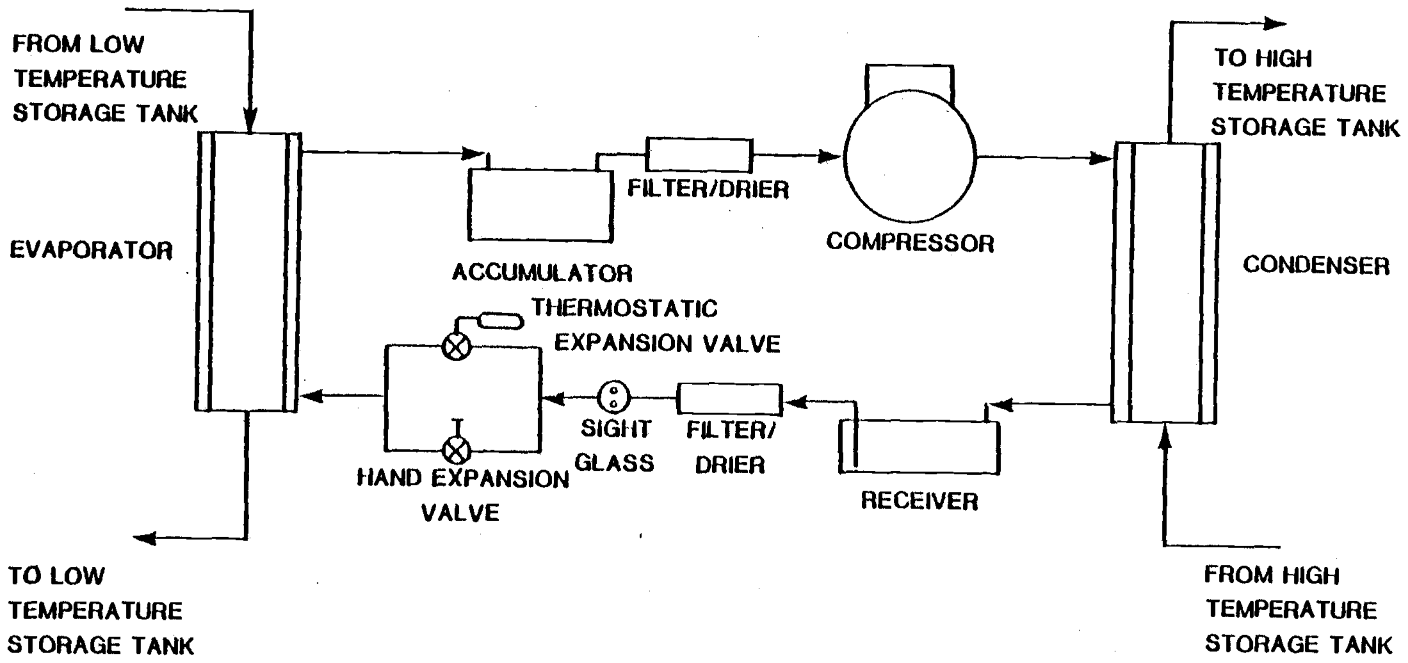

Figure 6. Experimental water-to-water heat pump component arrangement.

Table 3. Evaporator and condenser pressures at 26.7° and 82.2°C for Freon 12, 22, and 114.

Refrigerant	Evaporator Pressure at 26.7°C	Condenser Pressure at 82.2°C
Freon 12	689.5 kPa	2,413.2 kPa
Freon 22	1,091.7 kPa	3,825.1 kPa
Freon 114	225.2 kPa	971.8 kPa

hermetic compressor was acceptable while the maximum acceptable temperature for Freon 12 is approximately 18.3° to 21.1°C (E. I. DuPont, 1981). Therefore, the condensing pressure and temperature were the limiting conditions in selecting the components. Assembly of the components was in accordance with the specific properties of Freon 114.

Figure 7 illustrates the design heat pump cycle established for a condensing temperature of 82.2°C and an evaporating temperature of 26.7°C for Freon 114 on a pressure-enthalpy diagram as taken from ASHRAE Handbook of Fundamentals (1981). Assumptions made in the development of this cycle included:

(1) Evaporating temperature included an 11.1°C increase from superheating (Point B of Figure 7).

(2) Pressure losses through the refrigerant lines were considered negligible. Losses through the heat exchangers and other components were estimated.

(3) Isentropic compression occurred in the superheat region (Point B to Point C of Figure 7).

(4) No subcooling of the refrigerant occurred in the condenser (Point D of Figure 7).

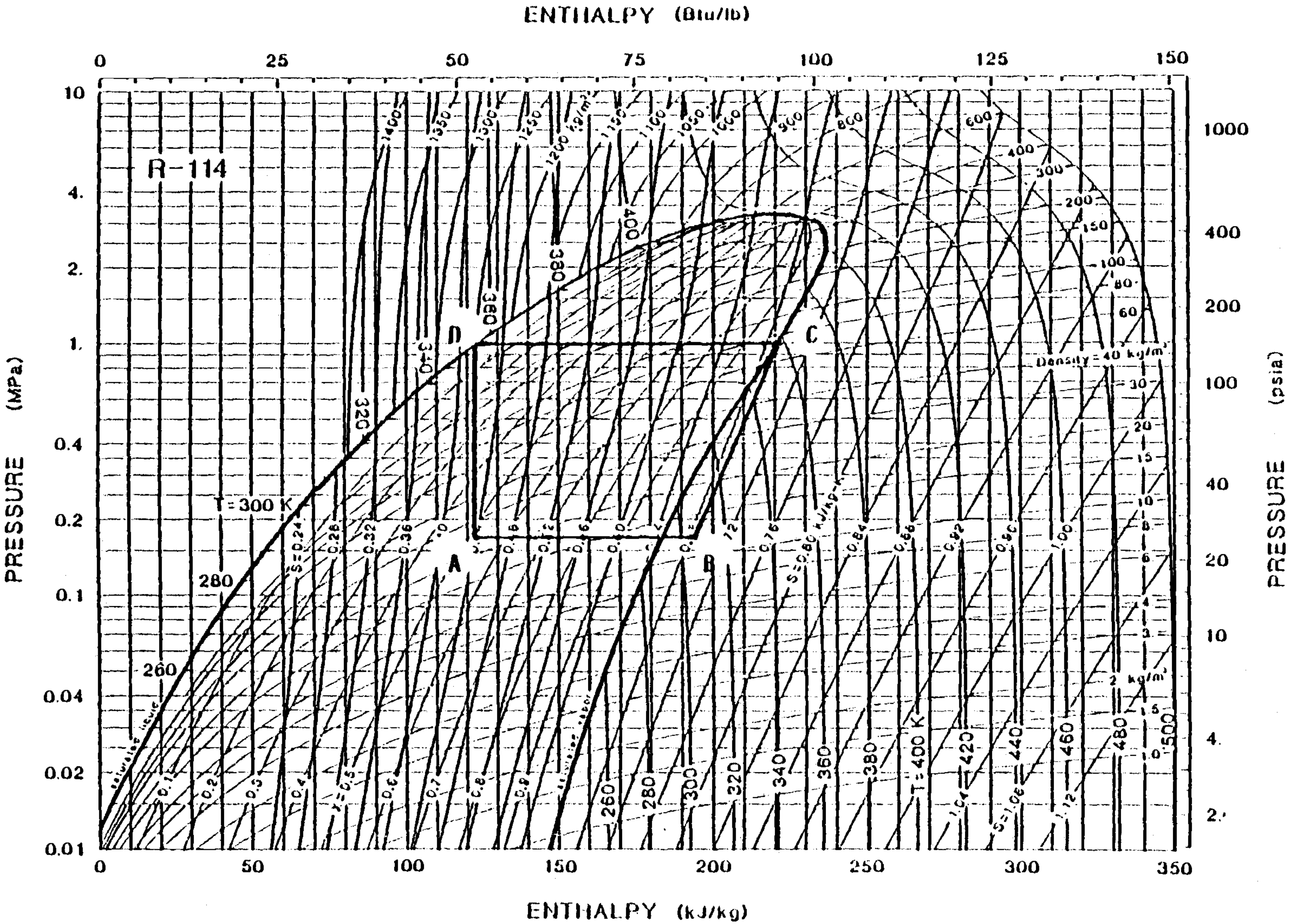

Figure 7. Design heat pump cycle using Freon 114 as the refrigerant.

Condenser Selection

Heat transfer in the condenser was analyzed by considering the condenser as two separate control volumes, one which had Freon 114 flow across its control surface and the other had water flow across its control surface. Additional heat transfer from one control surface to the other was possible. A steady-state, steady-flow energy transfer equation of the form:

$$Q_{cv} = \dot{m}\,(h_e - h_i) \tag{3}$$

was used for each control volume, where:

Q_{cv} = rate of energy transfer, kJ/hr

$\dot{m}$ = mass flow rate, Kg/hr

h_e = enthalpy at exit state conditions, kJ/Kg

h_i = enthalpy at entrance state conditions, kJ/Kg

Figure 8 represents the control volume arrangement for the condenser.

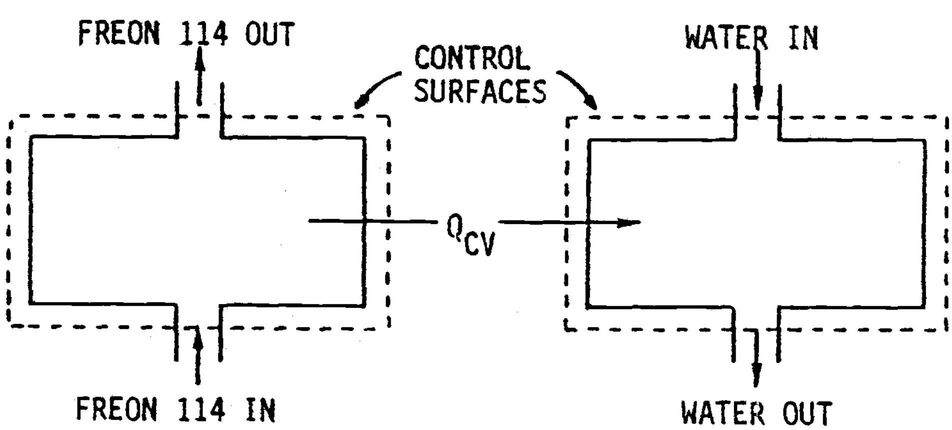

Figure 8. Heat pump condenser considered as two separate control volumes.

Heat transfer in the water side control volume resulted in a peak heating load of approximately 67,300 kJ/hr or 20.8 kW. Sizing the condenser to meet peak heating loads would not have been economical, thus, selection of the condenser was limited to either a 10.6 or 17.7 kW nominal size unit that would meet 50 or 85 percent, respectively, of peak heating load in the same time period and still remain cost competitive. This was justifiable because the recovery time to heat water in the high temperature water storage tank was not critical. This assumed a water temperature increase of approximately 11.1^{o} to 16.7^{o}C and a water mass flow rate of 570 Kg/hr through the water-cooled condenser. Refrigerant mass flow rates required for these two units were determined by using the steady-state, steady-flow energy equation and solving for $\dot{m}$. Refrigerant entrance and exit conditions were based on the following assumptions:

(1) Saturated vapor enters condenser at 82.2^{o}C and 971.4 kPa.

(2) Refrigerant pressure drop through the condenser was 34.5 kPa.

(3) No subcooling of refrigerant occurred in condenser.

(4) Saturated liquid exits condenser at 80.0^{o}C and 937.0 kPa.

Refrigerant mass flow rate through the condenser for the 10.6 and 17.7 kW units was 356.3 and 586.4 Kg/hr, respectively.

Evaporator Selection

The same procedure was followed for sizing and selection of the evaporator. Figure 9 represents a schematic of the control volume arrangement for the evaporator. Cost constraints resulted in consideration of only a 10.6 or 17.7 kW nominal size commercial unit.

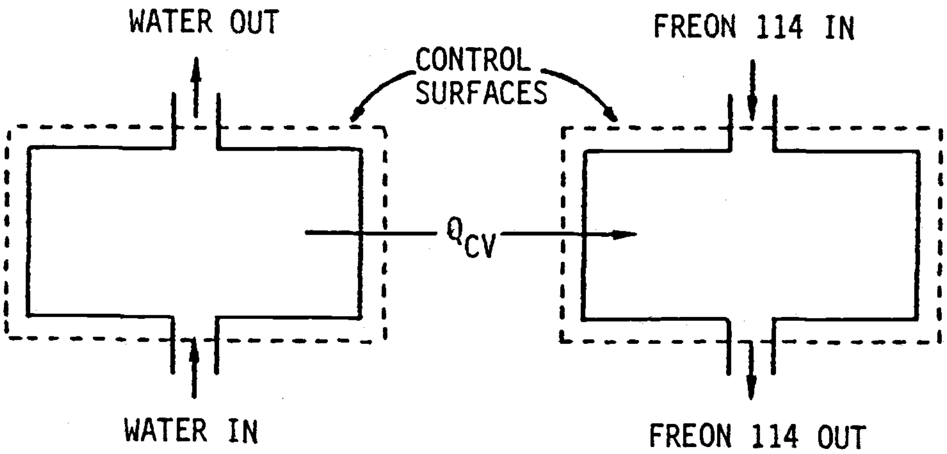

Figure 9. Heat pump evaporator considered as two separate control volumes.

Analysis of the water side control volume indicated an energy removal rate of 26,760 kJ/hr or 8.0 kW. This assumed an 11.1°C decrease in water temperature and a water mass flow rate of 570 kg/hr through the evaporator. Refrigerant entrance and exit conditions were based on the following assumptions:

(1) Refrigerant entered evaporator as a liquid-vapor mixture at a temperature of 15.5°C and a pressure of 166.8 kPa.

(2) Refrigerant pressure drop through the evaporator was 34.5 kPa.

(3) Superheat of 11.1°C occurred in evaporator.

(4) Refrigerant exited evaporator as a superheated vapor at a temperature of 26.6°C and a pressure of 132.2 kPa.

Refrigerant mass flow rates required to meet this heat transfer rate for the 10.6 and 17.7 kW evaporator units were 490 and 816 kg/hr, respectively.

A chiller barrel evaporator was first considered. Advantages of this type of heat exchanger included specific design for optimum turbulence and uniform flow patterns throughout the barrel to assure more and faster heat transfer to refrigerant tubes. Also the unit was completely self-contained, including insulation. Disadvantages included weight, length, and excessive cost. A 17.7 kW flooded shell-and-tube type chiller barrel had an estimated weight, length, and cost of 109 kg, 147.3 cm, and $1,100, respectively. These factors made using a chiller barrel prohibitive even with the higher heat transfer capacity. The decision to install a water-cooled condenser and operate it as an evaporator was influenced by cost, availability, and results of previous work studying heating performance of water-cooled condensers used as evaporators. Means (1980), in a series of experiments testing the performance of water-cooled condensers used as evaporators, found a strong correlation between refrigerant pressure drop and refrigerant flow rates. Sizable refrigerant pressure drops were recorded through the evaporators. Results showed overall heat transfer coefficients to be a function of the water and refrigerant flow rates and direction of flow (parallel versus counter-flow). Personal communication with the manufacturers indicated acceptable performance for the particular condenser being selected when used as an evaporator although no actual testing had been documented. Recommendations by the manufacturer included oversizing the evaporator by approximately 50 percent (i.e., use a condenser rated at 17.7 kW versus one rated at 10.6 kW) (Halstead and Mitchell, 1981).

Compressor Selection

Sizing the compressor was dependent upon the work of compression required to increase the refrigerant temperature and pressure from the evaporator exit conditions to the condenser entrance conditions. A steady-state, steady-flow energy transfer equation of the form:

$$Q_C = \dot{m}_R (h_C - h_B)$$

was used to compute the compressor capacity required where

Q_C = compressor capacity, kJ/hr

$\dot{m}_R$ = refrigerant mass flow rate, kg/hr

h_C = compressor discharge enthalpy, kJ/kg

h_B = compressor inlet enthalpy, kJ/kg

A compressor capacity of 13,500 kJ/hr was required to boost the refrigerant from evaporator exit conditions to condenser entrance conditions using a refrigerant mass flow rate of 490 kg/hr for the 10.6 kW evaporator unit. This resulted in a power input to the compressor of 6.0 kW assuming a compressor efficiency of 75 percent. Using a 17.7 kW evaporator unit with its higher refrigerant mass flow rate increased the compressor size to the point that compressor cost and availability became the deciding factors.

The compressor selected was a hermetically-sealed unit designed for refrigeration applications. The unit was oversized to accomodate the high mass flow rates required for use with the refrigerant, Freon 114. This unit was specifically designed for use with Freon 22 but due to the unavailability of a compressor for use with Freon 114 in the desired size, this model was selected. Personal communication with manufacturers concerning application of this particular model in a water-to-water heat pump using Freon 114 resulted in selection of a compressor rated at 26.8 kW nominal capacity (Coplaweld, 1981). This particular model was also selected because of its high efficiency rating.

Final selection of the major components in the heat pump were as follows, based on the ability to provide adequate heat transfer, component cost, and availability.

(1) Water-cooled condenser
- 10.6 kW refrigeration capacity
- Tube-in-tube construction

(2) Evaporator
- 17.7 kW refrigeration capacity
- Tube-in-tube construction

(3) Compressor
- 26.8 kW refrigeration capacity (rated for Freon 22 refrigerant)
- hermetically-sealed

The complete calculations for sizing each major component of the heat pump are included in Appendix B. Manufacturer's specification for the major components are included in Appendix A.

Auxillary Component Selection

While the compressor, condenser, and evaporator constitute the major components of a heat pump, auxillary components also play an important role in maintenance-free operation. Both hand operated and thermostatic expansion valves were installed in parallel in the heat pump circuit for comparison of system performance. A thermostatic expansion valve was selected to provide a refrigerant superheat condition of $6.7^{o}C$ at the evaporator outlet for a 10.6 kW capacity heat pump using Freon 114 as its refrigerant. The hand expansion valve allowed any desired superheat condition to be achieved in order to evaluate the performance of the heat pump under various conditions.

The liquid receiver, suction line accumulator, and filter/driers were selected to provide sufficient refrigerant holding capacity, positive oil return, and minimal pressure drop. A combination sight-glass and moisture-indicator was also selected to provide visual inspection of the liquid refrigerant entering the expansion valves. Type L copper tubing was sized to provide positive oil return and minimal pressure drop. Manufacturer's specifications for the auxillary components are included in Appendix A.

High Temperature Water Storage Tank

A high temperature ($71.1^{o}C$) water storage tank was required to provide a heat sink for the water-to-water heat pump system. Sizing and selection of a high temperature water storage tank were based on investigation of the potential of providing an adequate quantity and quality of hot water to eventually eliminate the need for a high temperature electric water heater. Based on the high grade hot water requirements as shown in Appendix B, the high temperature water storage tank was sized at 454 liters and was the same construction as the low temperature water storage tank. Figure 10 indicates the plumbing arrangement for the high temperature water storage tank. The heat sink thermal storage tank was insulated with a 7.6 cm water heater

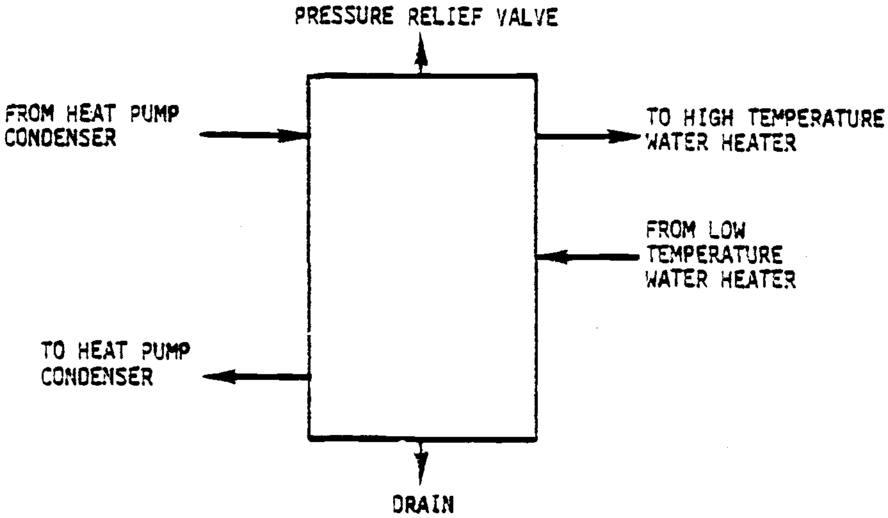

Figure 10. High temperature water storage tank plumbing arrangement.

jacket. This blanket wrap served a dual purpose by insulating the tank to reduce heat transfer to ambient surroundings and protecting the tank from water spilled in the area. The tank was installed in the milk washroom adjacent to the high temperature electric water heater in August 1981.

Installation of Heat Pump

The entire heat pump was assembled on an 89 by 61 cm square tube, metal frame in the Agricultural Engineering Department shop. Overall unit height was 46 cm. Figure 11 shows the heat pump installed at the Oregon State University dairy research center. The unit was located in the dairy's washroom to keep water conveyance lines as short as possible and minimize friction loss in the pipes.

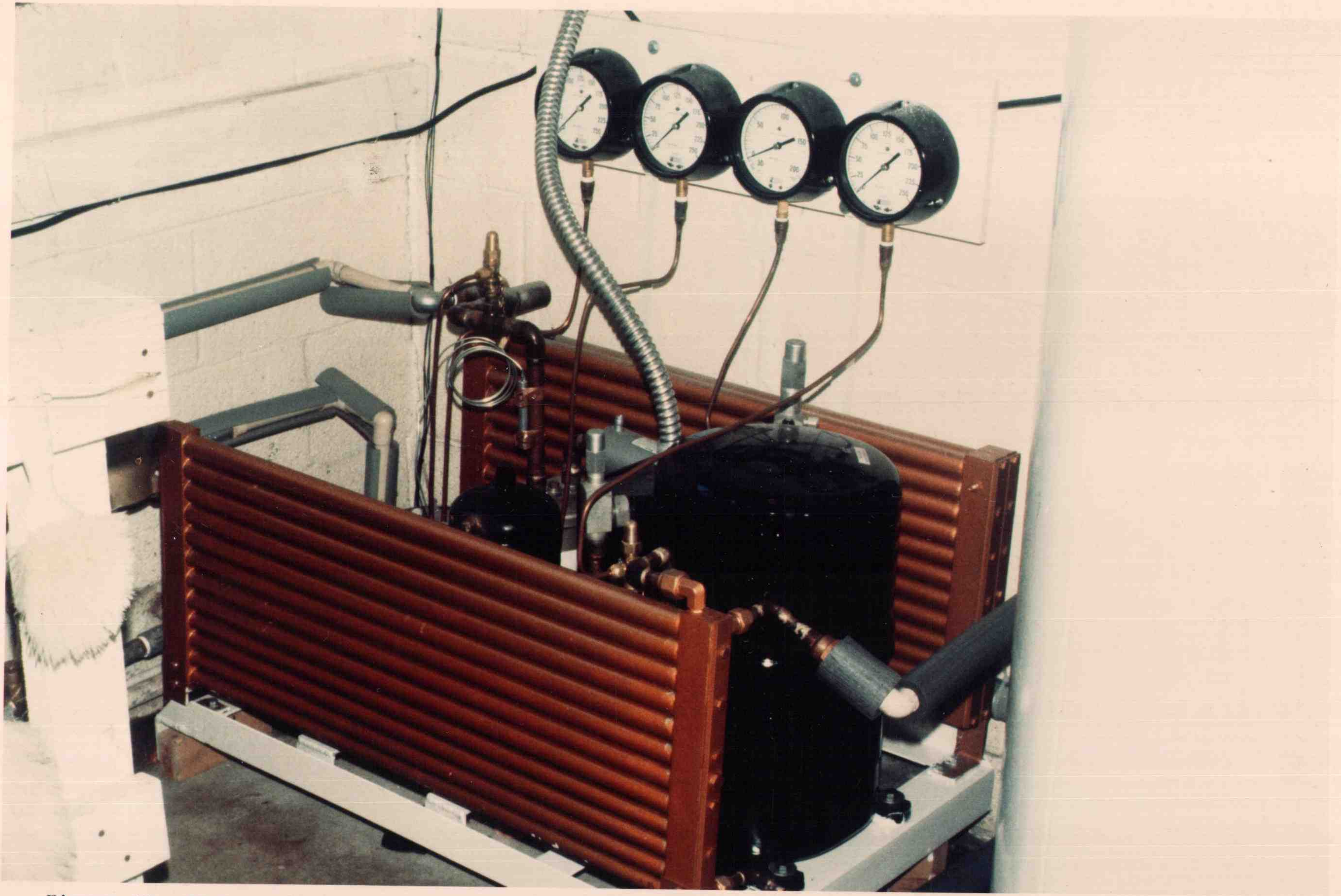

Figure 11. Water-to-water heat pump installed at OSU dairy.

Standard flare fittings were soldered into the evaporator and condenser water inlets and outlets to facilitate easy connection and disconnection of the water lines in the event the heat pump required removal.

Two 0.062 kW centrifugal pumps provided water circulation through the evaporator and condenser circuits. Various gate, check, and flow control valves were located throughout the water conveyance system to isolate an area in case of maintenance or to control flow rates through the evaporator and condenser. The entire water conveyance system including the waste heat recovery system operated at city water system pressure of 413.6 kPa. Safety high pressure relief valves were installed on the water storage tanks.

Installation of the heat pump at the dairy required connection of the compressor to a three-phase 240-volt power source from a breaker box. The electrical wiring and charging of the heat pump with Freon 114 refrigerant were performed by outside contractors to comply with local code specifications. Activation of the heat pump was controlled by a remote thermostat located in the hot water line between the high temperature storage tank and electrical resistance water heater. This remote thermostat was an immersion-type hot water controller with an adjustable high temperature cut-out ranging from 37.8^{o} to $115.6^{o}C$ and a manual temperature differential of $3.3^{o}C$, $6.6^{o}C$ and $9.9^{o}C$. The controller activated and deactivated the heat pump and water circulation pumps simultaneously.

The entire water conveyance system was constructed of 1.9 cm I.D. Schedule 4120 thermoplastic pipe insulated with 1.3 cm of expanded polyurethane foam to minimize heat loss to the environment. Heat loss to the environment was also minimized at the heat pump by insulating the condenser, the evaporator, and the suction line with 7.6 cm of fiberglass batting material. The heat pump was installed in October 1981 and began operation under actual production conditions in November 1981.

Data Collection and Instrumentation

Data were collected for combined heat pump and waste heat recovery system operation from November 1, 1981 to May 31, 1982. System temperatures were monitored and recorded on an hourly basis. On selected days, additional data on water temperatures, heat pump refrigerant temperatures, and heat pump refrigerant pressures were recorded at five-minute intervals during the heat pump operating cycle.

System temperatures were measured using 0.39 mm diameter (24 gauge, Browne and Sharp) copper-constantan thermocouples. Thermocouples were attached to the fluid-carrying tubing at selected locations and insulated with 7.6 cm of fiberglass batting material to minimize temperature variations and the temperature profile across the tube wall. Temperatures were recorded by an Esterline-Angus multipoint potentiometer equipped with both a digital printout and a paper tape perforation unit. The perforated paper tape was used to enter data into the university computer system for reduction purposes. Location of temperature sensors are identified in Table 4.

Four heat pump pressure sensors were located in the compressor discharge line, before entering the expansion valve, after leaving the expansion valve, and in the compressor suction line. Pressures

Table 4. Location of temperature sensors in heat pump and energy recovery system.

Sensor Number[a]	Location
2	Water entering water-cooled condenser
3	Water leaving water-cooled condenser
4	Water entering low temperature (40.5^{o}C) water heater
5	Top third of low temperature water storage tank
6	Middle third of low temperature water storage tank
7	Bottom third of low temperature water storage tank
8	Water entering heat pump evaporator
9	Water leaving heat pump evaporator
10	Water entering heat pump condenser
11	Water leaving heat pump condenser
12	Top third of high temperature water storage tank
13	Middle third of high temperature water storage tank
14	Bottom third of high temperature water storage tank
15	Water entering high temperature water heater
16	Water from low temperature storage tank entering high temperature storage tank
17	Water entering from city water supply
18	Heat pump -- discharge line between compressor and condenser
19	Heat pump -- between condenser and expansion valve
20	Heat pump -- suction line before entering compressor
21	Heat pump -- after expansion valve entering evaporator
22	Heat pump -- leaving evaporator -- inside tube rack
23	Heat pump -- leaving evaporator -- outside tube rack

[a]Sensor number corresponds to recorder channel number. Channel 1 was a reference channel.

were measured with bourbon tube pressure gauges having a range of 0 to 1,724 kPa gauge (0 to 250 psia gauge).

Water consumption in the milking parlor was measured for low temperature (43.3^{o}C), high temperature (71.1^{o}C), and total water use. Three oscillating-disk flow meters were installed to measure water utilized for each purpose. Readings were taken daily. Total water flows through the heat pump condenser and evaporator were measured using two oscillating-disk flow meters. A high temperature meter was installed in the condenser water line to eliminate binding problems with the disk mechanism. Flow rates were calculated by timing the flow through the flow meters. Readings were made daily and during selected short term observations of the heat pump cycle.

Electrical energy consumption was measured for the low temperature (43.3^{o}C) water heater, high temperature (71.1^{o}C) water heater, and heat pump. Each water heater was connected to a kilowatt-hour meter that recorded electrical energy usage plus peak demand. The heat pump was only monitored for electrical energy usage. Each meter was ready daily. Figure 12 is a schematic of the entire energy recycling system. Temperature and pressure sensor and water flow meter locations are identified as T, P and M, respectively.

Heat Pump System Analysis

During the seven-month data collection period, several test conditions were established to evaluate the heat pump system's operating characteristics. Table 5 contains the test conditions including objective of each test, parameters varied during each test, and parameters held approximately constant. The primary purpose of these tests was the evaluation of variable effects on the heat pump's

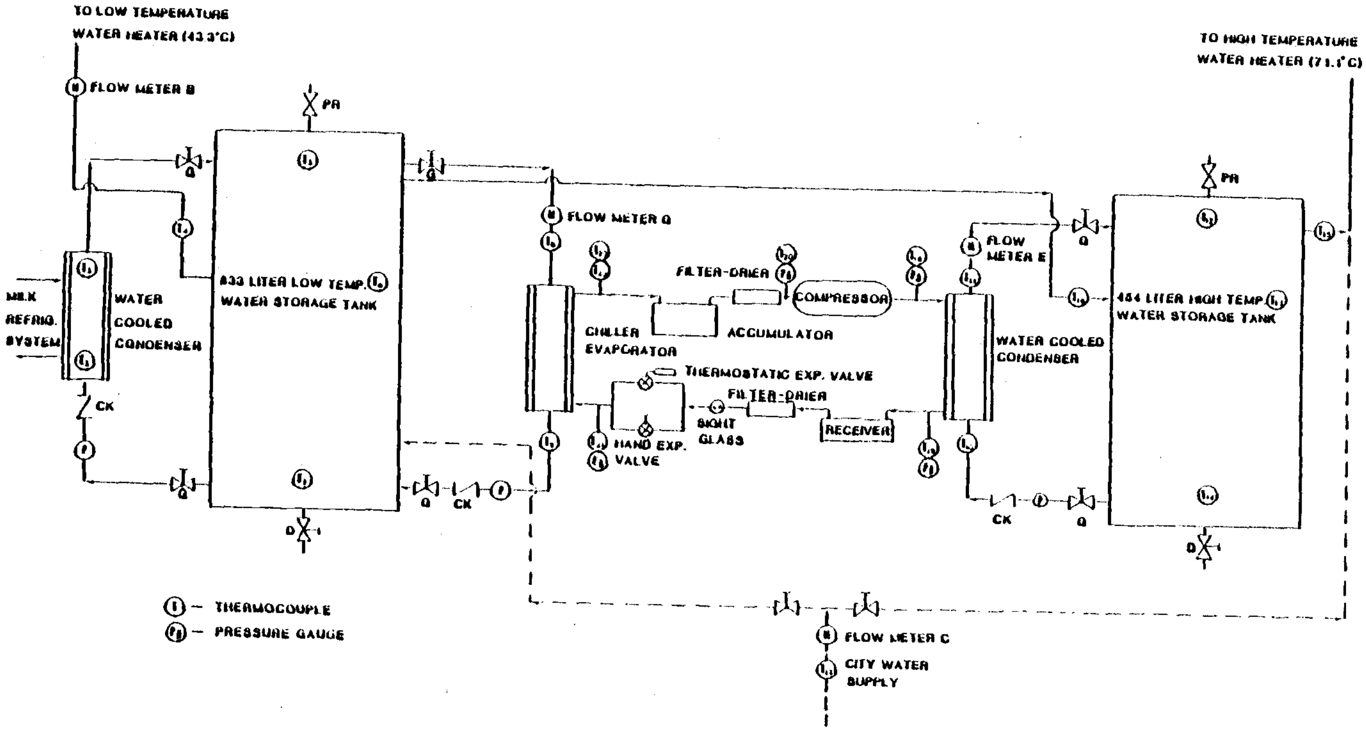

Figure 12. Schematic of energy recycling system.

Table 5. Heat pump test conditions.

Test	Objective	Parameter Being Varied	Parameter Held Constant
(1) Heat pump coefficient of performance versus water flow rate through evaporator	To determine the effect of water flowrate through the evaporator on the heat pump's coefficient of performance during the entire heat pump operating cycle.	water flow rate through the evaporator. Range: 7.6 to 15.2 liters per minute.	water flow rate through condenser (9.5 liters per minute). Thermostatic expansion with 11.1^{o}C super heat.
(2) Heat pump coefficient of performance versus water flow rate through condenser.	To determine the effect of water flow rate through the condenser on the heat pump's coefficient of performance during the entire heat pump operating cycle.	water flow rate through the condenser. Range: 7.6 to 13.2 liters per minute.	water flow rate through the evaporator (14.2 liters per minute). Thermostatic expansion with 11.1^{o}C super heat.
(3) Heat pump's operating schedule versus set point temperature and manual temperature differential control on heat pump's remote thermostat	To determine the effect of the set point temperature and manual temperature differential control for the heat pump's remote thermostat on the heat pump's operating schedule.	set point temperature of heat pump remote thermostat. Range: 65.5^{o} to 73.9^{o}C. Manual temperature differential on heat pump remote thermostat. Range: 3.3^{o} to 10.0^{o}C.	water flow rates through the condenser and evaporator (9.5 and 14.2 liters per minute, respectively). Thermostatic expansion with a 11.1^{o} C super heat.

coefficient of performance and operating cycle under actual production conditions.

Each test was run for a minimum of two weeks to study the long term effects under actual production conditions. During these two-week periods, temperatures and pressures of the heat pump system and water storage facilities were monitored and recorded. The heat pump's coefficient of performance was calculated for the heat pump operating cycle using these data. Temperature stratification in both the low and high temperature water storage tanks were also monitored and recorded.

IV. RESULTS AND DISCUSSION

The energy recovery unit and water-to-water heat pump were evaluated from November 1, 1981 to May 31, 1982. Performance comparisons were made between these data and data collected during two years of previous energy conservation research. This section includes results of water use and electrical energy consumption for both previous and present water heating schemes. Performance characteristics have also been documented for the heat pump operation at various test conditions.

System Performance

Water Use

Water use was monitored at the Oregon State University dairy research center from March 1978 to May 31, 1982. Comparison of water use in the milking parlor was made between the previous water heating scheme using commercial electric water heaters and the current method involving the energy recovery unit and water-to-water heat pump. Water quantities monitored included daily low temperature, high temperature, and city water supplied to the milking parlor. Table 6 summarizes the average daily water use.

Several external factors affected the data reported during the span of this research effort. A metering system was installed in the low temperature water supply line in early January 1981. The system maintained the pressure required for efficient udder cleansing and stimulation, yet substantially reduced low temperature water use. A 71 percent reduction in low grade tempered water usage was recorded as compared to pre-1981 records. A second external factor was the dairy's water management practices, particularly those involving low temperature water usage. Although the dairy's management practices were beyond the author's control, they must be included in developing an overall energy conservation program for the dairy. Water usage at a privately-owned and operated dairy tends to be more conservative than at a university-owned and operated dairy where research is a major emphasis. During several periods of subfreezing weather in December 1981 and January 1982, large quantities of low temperature water were used for non-milking parlor related applications. During these two periods, low temperature water use increased approximately 90 percent. Usage of these large quantities of warm water resulted in the low temperature water storage tank being completely depleted of its normally 40.5^{o}C water. Recovery of the water temperature to the 40.5^{o}C operating level required approximately 24 hours during which time the milking operation ran short of low grade tempered water. The heat source temperature for the heat pump was also reduced, thereby decreasing the heat pump's performance and increasing electrical energy consumption by the heat pump and electric resistance water heaters.

Table 6. Average daily water use in OSU milking parlor.

Period	Water Use in Liters (Gallons)		
	43.3^{o}C	71.1^{o}C	City Water Supplied
3/78 - 6/80	1,423.0 (376.0)	863.1 (228.0)	5,768.0 (1,524.0)
11/81 - 5/82	405.8 (107.2)	775.3 (204.8)	3,414.8 (902.1)

On another occasion in February 1982, a water control valve was inadvertently closed to stop a major leak in the heat pump condenser's water line. This resulted in loss of both low and high temperature water from the storage tanks and necessitated disengagement of the heat pump for approximately one week until the problem was pinpointed and rectified. City water was continuously flowing through the system, allowing no storage of low and high temperature water. Use of low temperature, high temperature, and city water increased by approximately 70, 11, and 35 percent, respectively, during this period. Once the problem was recognized and corrected, system performance returned to expected levels.

Daily quantities of low grade tempered water used varied inversely with the ambient air temperature as expected. Milking personnel increased the quantity of low temperature water used in the milking and floor cleaning operations as ambient air temperature decreased. During the seven-month period of data collection from November 1981 through May 1982, the minimum low temperature water usage was 112.3 liters in May 1982 and the maximum was 837.4 liters in January 1982. The maximum included the large quantities of warm water diverted to non-milking parlor uses as previously described.

High temperature water use remained essentially constant for the two methods for heating water. The 10 percent decrease in the high grade tempered water usage can be attributed to rescheduling miscellaneous equipment cleaning to coincide with the normal milking equipment wash cycle. Wash cycle water was then used for cleaning the miscellaneous equipment.

Electrical Energy Consumption

Electrical energy consumption was monitored in the OSU milking parlor for the low temperature (43.3^{o}C) water heater, high temperature (71.1^{o}C) water heater, and water-to-water heat pump. Electrical energy consumption by the water circulation pumps in the waste heat recovery unit and water-to-water heat pump were not monitored as their electrical energy requirements were minimal in comparison to the water heaters and heat pump. Previous data presented by Hellickson (1980) showed an average energy requirement of 0.9 kW-hr per day for the same size and make of pumps. Comparison of electrical energy consumption in the milking parlor was made between the previous water heating scheme and the current one. The time period coincided with the water use analysis. Table 7 summarizes the average daily electrical energy consumption in the OSU milking parlor for heating water. Based upon water use and electrical energy consumption records prior to 1981, the average daily electrical energy requirements for the previous water heating scheme was 115.3 kW-hr. This assumed the use of separate electric

Table 7. Average daily electrical energy consumption in OSU milking parlor.

	Electrical Energy Consumption in Kw-hr			
Period	Low Temperature Water Heater	High Temperature Water Heater	Water-to-Water Heat Pump	Total
3/78 - 6/80				
$\overline{X}$	51.7 (4.0)[a]	63.6 (4.0)	--	115.3
σ	15.3	7.2		
11/81 - 5/82				
$\overline{X}$	10.9 (2.1)	6.2 (1.9)	12.7	29.8
σ	6.3	3.6	0.8	

[a]Peak demand

resistance water heaters for the low and high temperature water demands. Subsequent to installation of the energy recovery unit and water-to-water heat pump, the average electrical energy requirement decreased to 29.75 kW-hr per day, an overall reduction of 74.2 percent as compared to previous requirements. A breakdown for the separate water heaters indicated decreases of 80.5 and 90.3 percent in electrical energy consumption by the low and high temperature water heaters, respectively. Peak energy demand by the low and high temperature water heaters also decreased by 47.5 and 52.2 percent, respectively. Peak demand by the water-to-water heat pump was not monitored.

Decreased energy consumption by the low temperature water heater was the result of two major improvements. The first was installation of the low temperature water metering system. The immediate reduction in low temperature water usage from 1,423 to 406 liters per day effected an approximate electrical energy savings of 9,829 kJ or 30.4 kW-hr per day based on heating water from 12.2^{o} to 43.3^{o}C. This also caused a larger quantity of low grade tempered water to be available to the heat source coil of the heat pump. The second improvement resulted from installation of the energy recovery unit in the primary milk refrigeration system. An additional average energy reduction of 33,607 kJ or 10.3 kW-hr per day was realized. A secondary effect of installation of the energy recovery unit was improved performance of the primary milk refrigeration system. Visual observation indicated a shorter operating schedule. The increased heat transfer in the water-cooled condenser allowed the refrigeration system to maintain a greater refrigerating effect, thus, a shorter operating schedule.

During the data collection period, average daily minimum energy consumption by the low temperature water heat was 6.06 kW-hr (May 1982). The average daily maximum value was 22.86 kW-hr (January 1982). The maximum value was again in part due to low grade hot water use for other than milking parlor purposes. Minimum and maximum peak energy demand values were 1.8 and 2.4 kW-hr and occurred in February 1982 and December 1981, respectively. Energy consumption for meeting high temperature demands was primarily provided by the heat pump with the high temperature water heater used as backup. Thermostatic control of the heat pump caused it to operate during periods when low grade hot water was also being used in the milking parlor. The combined demands resulted in the low temperature water heater operating longer than expected. A disadvantage of using a separate high temperature water storage tank for the heat pump heat sink was some reheating of water as it entered the high temperature water heater. Combining the water storage tank and water heater would have reduced this reheating process by meeting the required water temperature level in one storage facility rather than two. The water heater would, however, provide backup capacity in case of heat pump failure.

The major improvement in reducing high temperature energy demand was a result of installation of the heat pump. An overall reduction of 70.2 percent (including both high temperature water heater and heat pump) was realized for the present water heating scheme as compared to the previous method. For the data collection period an average minimum daily energy requirement for the high temperature water heater and heat pump was 3.51 kW-hr (March 1982) and 11.3 kW-hr (January 1982), respectively. Maximum average daily energy consumption for the high temperature water heater and heat pump was 13.45 kW-hr (January 1982) and 13.54 kW-hr (December 1982), resspectively. Maintenance problems described earlier were a major factor in causing the maximum values to be as large as reported.

Formation of a normalized base for comparison of energy consumption by the two water heating schemes was required because of the large reduction in low temperature water use that occurred. In order to make a valid comparison, the energy require to heat 1,000 liters (264 gallons) of water was determined for both low and high temperature demand. Table 8 summarizes the adjusted average energy consumption for low and high temperature demands in the OSU milking parlor.

Table 8. Adjusted average electrical energy consumption in OSU Milking parlor.

	Electrical Energy Consumption in kW-hr per 1,000 Liters (264 Gallons)[a]		
Period	Low Temperature Consumption	High Temperature Consumption	Total
3/78 - 6/80	36.3	73.7	110.0
11/81 - 5/82	26.9	24.4[b]	51.3

[a]Based on mean values from Table 7.

[b]Includes water-to-water heat pump energy consumption.

Installation of the energy recovery unit in the milk refrigeration system resulted in a 26 percent reduction in electrical energy required to meet the low temperature demand. On a per day basis for present low temperature water usage, this reduction amounted to 3.80 kW-hr per day or 115.0 kW-hr per month. Installation of the heat pump resulted in a 67 percent reduction in electrical energy required to meet high temperature demands. Based on present high temperature water usage, the energy reduction was 38.2 kW-hr per day or 1,146.0 kW-hr per month. Appendices D and E contain the water usage and electrical energy consumption data recorded, respectively.

An energy transfer analysis was made to determine the amount of energy recovered from the milk refrigeration system and subsequently utilized for water heating purposes. An average daily energy requirement of 138,530 kJ or 42.8 kW-hr was needed to heat water from approximately 12° to $43^{\circ}C$ based on the low temperature water usage including preheating the high temperature water. The temperature differential represented city water entering the low temperature storage tank and low grade tempered water leaving the low temperature water heater. Data collected indicated an approximate enthalpy increase through the energy recovery unit of 22.0 kJ/kg (10.0 Btu/lbm) of water. For a water flow rate of 15.1 liters per minute (4.0 gpm) and operating period of approximately 16 hours per day, the energy recovered and stored in the water was approximately 303,600 kJ (89 kW-hr) per day.

The heat pump heat source also utilized a portion of the energy recovered from the milk refrigeration system. An approximate enthalpy decrease of 26 kJ/kg (11 Btu/lbm) of water occurred in the heat pump evaporator. This resulted in an energy removal rate of 110,000 kJ (34.0 kW-hr) per day based on a water flow rate of 14.2 liters per minute (3.75 gpm) and heat pump operating time of 5.5 hours per day. The analysis did not account for any system inefficiencies such as heat loss from the system to the environment, variability of the overall heat transfer coefficient in the evaporator, and non-uniform low temperature water usage.

Energy recovered from the milk refrigeration system exceeded that required for meeting the low grade hot water and heat source demands. However, timing of water-cooled condenser operation, heat pump operation, and water use was not concurrent and resulted in poor utilization of the energy recovered. Visual observation of the milk refrigeration system showed the air-cooled condenser cycling on and off late in its operating schedule which indicated water storage temperatures had reached 40.5^{o}C. Better utilization of low temperature water would have the heat pump operating during the periods of maximum temperatures and not when temperatures are depressed from water removal. Energy consumption by the low temperature water heater was required for heating water from 41^{o} to 49^{o}C (thermostat setting), maintaining the required temperature between periods of use, and tempering water as a result of excessive low temperature water removal.

The heat recovered by cooling the milk through the plate heat exchanger was approximately 353,330 kJ (110 kW-hr) per day. This assumed a plate heat exchanger efficiency of 75 percent and milk temperature differential of 32^{o}C through the heat exchanger. Of this 353,330 kJ per day, approximately 85 percent was recovered by the desuperheater heat exchanger. Past studies have indicated that the desuperheater heat exchanger captures approximately 40 to 60 percent of the energy removed from the milk. The remaining energy removed was from the work of compression in the milk refrigeration system.

Energy removal from the heat pump condenser was approximately 143,500 kJ (44 kW-hr) per day, based on a water flow rate of 9.5 liters per minute (2.5 gpm), an approximate enthalpy increase of 46 kJ/kg (22 Btu/lbm) of water, and a heat pump operating time of 5.5 hours. Present high temperature water usage indicated an average daily energy requirement of 109,980 kJ (34 kW-hr) was needed to heat 775 liters of water from 43.3^{o} to $71.1^{o}C$. This temperature differential represented water entering the high temperature storage tank from the low temperature storage tank and water entering the high temperature water heater. These energy requirements were met by the heat pump condenser. The major energy consumption by the high temperature water heater was maintaining the water temperature at the thermostat setting between periods of water use.

Heat Pump Performance

Evaluation of the heat pump performance under actual production conditions was made based on computed coefficients of performance. Preliminary analysis established the design coefficient of performance at 3.04. The actual coefficient of performance was determined from five-minute test intervals over the entire heat pump operating cycle on selected days. Figure 13 illustrates both design and actual heat pump cycle on a pressure-enthalpy diagram for Freon 114. An average coefficient of performance of 4.05 was determined for 128 observations, a 38 percent increase over the design value. The large increase was partially due to the amount of subcooling that occurred

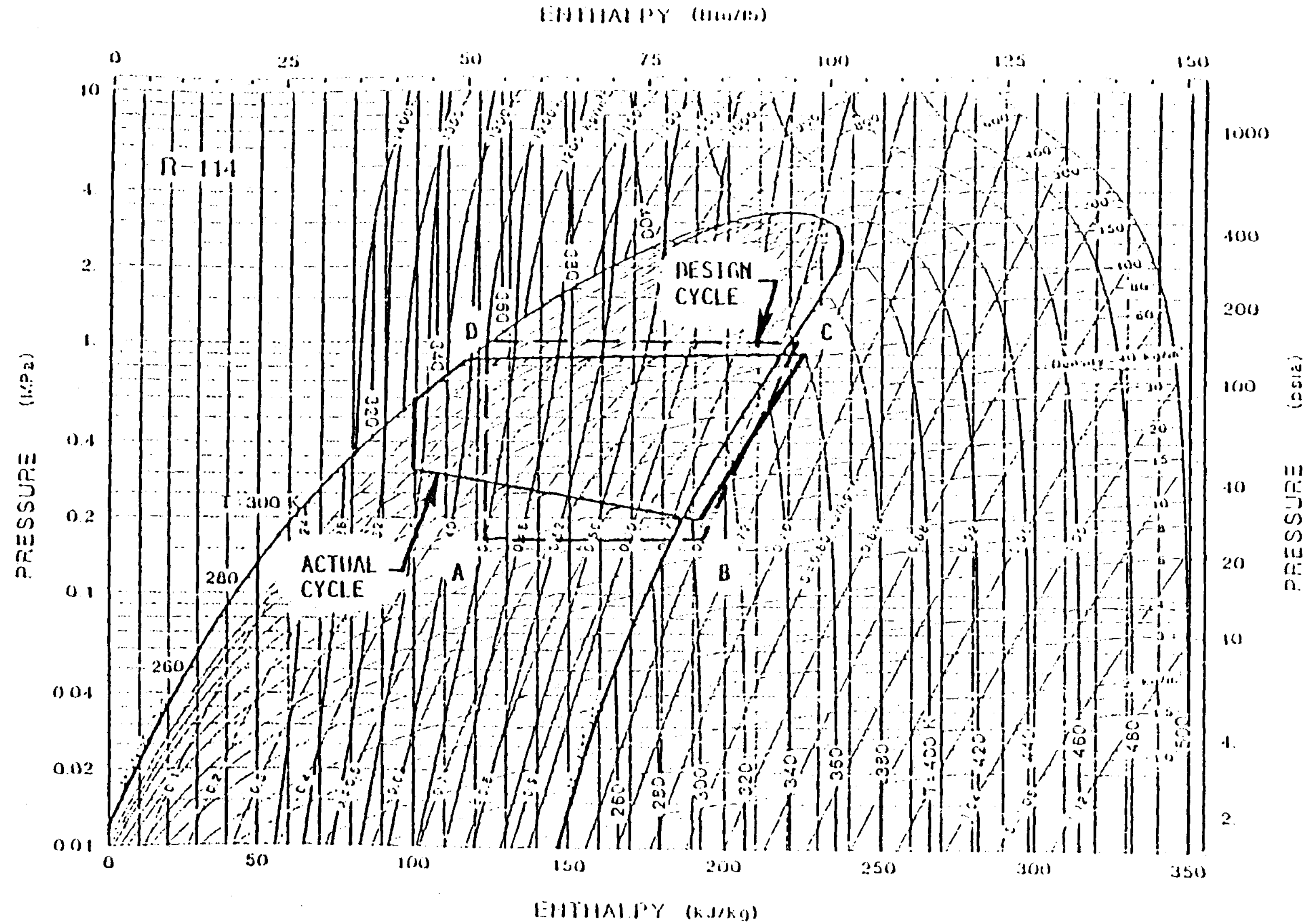

Figure 13. Design and actual heat pump cycle on pressure-enthalpy diagram for Freon 114.

in the heat pump condenser as a result of refrigerant pressure drop and additional condenser area beyond that required for condensation. Subcooling of the refrigerant (from approximately 82^{o} to $60^{o}C$) increased the energy output of the heat pump by approximately 20 kJ/kg (8.6 Btu/lb) of Freon 114 circulated, a 25 percent increase. Work of compression remained virtually unchanged while the amount of energy removed from the condenser increased. This resulted in a decrease in the amount of work input required to produce a kilowatt (ton) of refrigeration. The thermostatic expansion valve controlled the superheat condition of Freon 114 out of the evaporator at $6.7^{o}C$ as compared to the design condition of $11.1^{o}C$ superheat. A large pressure drop through the suction line reduced the suction superheat entering the compressor, thus increasing the work input by the compressor.

Actual coefficients of performance ranged from a minimum of 3.60 to a maximum of 5.55 for the entire study. The months of March and April 1982 were selected to illustrate the performance of the heat pump during a period of maintenance-free operation. During this period the average daily electrical energy consumption for the low and high temperature water heaters and the heat pump were 6.79, 3.68, and 12.60 kW-hr per day, respectively. The average coefficient of performance for the heat pump was 4.15 with a minimum of 3.93 and a maximum of 5.55.

The maximum coefficient of performance was achieved during a test period when the heat pump thermal source temperature was initially $40.5^{o}C$ and the heat sink temperature was $60^{o}C$. As the heat sink (high temperature water storage tank) temperature increased to approximately 73.9^{o}C and the heat source tank temperature decreased to 35^{o}C, the coefficient of performance dropped to a minimum of 4.1. During this test period the average coefficient of performance for the operating cycle was 4.49. Control of the heat sink temperature during the operating cycle was limited by the amount of high temperature water removed for equipment cleaning. Operation of the heat pump when the heat source (low temperature storage tank) was maintained at its maximum temperature would improve the performance by reducing the temperature lift between source and sink.

The procedure for testing heat pump performance at various water flow rates through the condenser and evaporator involved monitoring the heat pump temperatures and pressures at five-minute intervals for the entire operating cycle. Selection of water flow rates was limited by precision of the flow control valves in the water conveyance lines and maximum output of the water circulation pumps. The range of available flow rates was spanned in increments of 2.0 liters per minute (0.5 gpm). The preliminary design flow rates of 9.5 liters per minute through the condenser and 14.2 liters per minute through the evaporator were maintained as the control parameters during the tests.

Table 9 summarizes the heat pump performance at various water flow rates through the condenser and evaporator. Replication of test conditions was impossible because of variation in temperature stratification in the source and sink storage tanks from test to test. The tests served strictly as guidelines for improvement of heat pump performance under actual production conditions.

Table 9. Test results for heat pump performance at various water flow rates through the condenser and evaporator.

Test Condition	Evaporator Flow Rate (l/m)	Condenser Flow Rate (l/m)	Average COP for Heat Pump Cycle	Number of Observations	Minimum COP	Maximum COP
Variable water flow rate through evaporator	11.4	9.5	4.33	11	3.93	5.25
	13.2	"	4.20	31	3.72	5.10
	15.2	"	3.96	3	3.96	3.96
Variable water flow rate through condenser	14.2	7.6	4.01	37	3.72	4.31
	"	9.5	3.96	10	3.86	4.00
	"	11.4	3.96	22	3.38	5.55
	"	13.2	3.85	14	3.60	4.15
Average coefficient of performance for all tests			4.05	128		

Test results comparing coefficient of performance versus water flow rate through the evaporator indicated a decreased coefficient of performance for increased flow rates. A 35 percent increase in the flow rate caused an 8.5 percent decrease in the coefficient of performance. Selection of an optimum flow rate through the evaporator should reflect the purpose of the evaporator. In this research the important factor during the preliminary design was pursuit of a heat exchanger capable of transferring the maximum amount of energy from water to refrigerant. The cooling effect of the evaporator on the low temperature storage tank was of secondary importance. The greater the flow rate through the evaporator, the smaller the enthalpy change per kilogram of water and thus, the less energy transferred to the refrigerant per kilogram of water circulated. Results were considered inconclusive due to the variability of heat source temperatures during the tests. However, the results did indicate that flow rates through the evaporator should be maintained at levels corresponding to an operating situation where maximum heat transfer was achieved in the evaporator.

The same form of relationship was indicated by test results comparing coefficient of performance versus water flow rate through the condenser. The coefficient of performance decreased 4.0 percent for a flow rate increase of 74 percent. The criteria of selecting a water flow rate to provide a high water exit temperature from the condenser without producing a major reduction in recovery efficiency was established for the heat pump system. The time for recovery of water temperatures to the original levels in the high temperature storage tank was not a critical factor in system design because of long intervals between equipment wash cycles.

Low rates produced maximum exit water temperatures and reflected a greater enthalpy change per kilogram of water circulated. The lower range of flow rates (7.6 to 9.5 liters per minute) caused exit water temperatures approximately 17^{o}C above the entering water temperature while maintaining a coefficient of performance of approximately 4.00. Lower flow rates also caused a greater temperature stratification in the high temperature storage tank. Maintaining a moderate degree of stratification resulted in a shorter heat pump operating time to meet the required quantity and quality (temperature) of high grade hot water. Again, tests results were considered inconclusive because of the variability of stratification in the high temperature storage tank. Results did indicate that maintaining lower flow rates provided maximum exit water temperatures from the condenser. High flow rates resulted in lower exit water temperatures but a higher recovery efficiency.

A third test involved comparison of the heat pump operating schedule for various set point temperatures and temperature differentials controlled by the remote thermostat. The primary thrust of the test was observation of variations in heat pump operating time and high temperature storage tank stratification. Location of the thermostat in the water line between the high temperature storage tank and water heater did not accurately control the storage tank temperatures and resulted in upper tank temperatures approximately 6^{o}C above set point temperatures. Thus the operating time for the heat pump to reach and maintain desired water temperatures in the high temperature storage tank was increased.

Use of three temperature differentials of 3.3^{o}, 6.6^{o}, and 9.9^{o}C resulted in succeedingly increased heat pump operating times. Location of the thermostat also affected heat pump operating time during use of the selected temperature differentials. The heat pump was required to operate past the desired set point temperature to meet the cut out temperature of the thermostat. Smaller temperature differentials resulted in smaller temperature stratifications in the heat sink storage tank.

Electrical energy consumption by the heat pump increased slightly for increased temperature differentials, thus, the heat pump was operated for the remainder of the data collection period with a set point temperature of 71.1^{o}C and a temperature differential of 3.3^{o}C. This arrangement provided an adequate temperature and quantity of high grade hot water for the cleaning operation in addition to moderate temperature stratification in the storage tank.

Temperature Stratification in The Water Storage Tanks

Temperatures were monitored in the water storage tanks for three stratification levels. Figure 14 illustrates the average temperature stratification in the low and high temperature water storage tanks for a typical 24-hour period. The time of day scale has been adjusted to begin at 0600 hours (6:00 am) to correspond with the beginning of the milking operation.

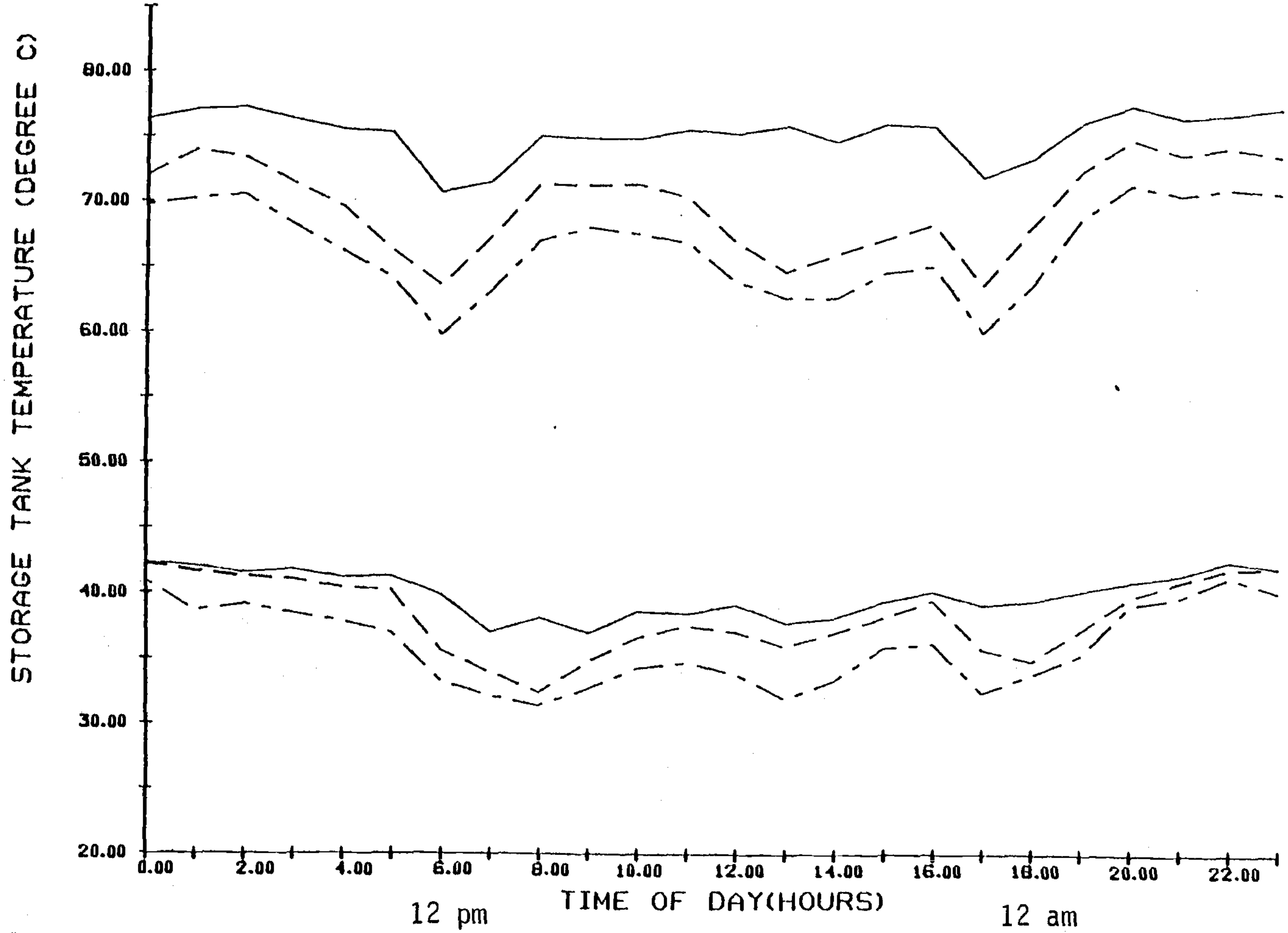

Figure 14. Average temperature stratification in low and high temperature water storage tanks for a typical 24-hour period.

Depressions in the temperature levels were indicative of water withdrawn for both low and high temperature usages. Recovery of the high temperature levels after water withdrawal illustrated the operating schedule of the heat pump. Correspondingly, the recovery of low temperature levels was a result of the energy recovery system operation. The low temperature levels continued to decrease as the high temperature levels returned to former levels. This was indicative of the removal of heat from the low temperature storage tank by the heat pump evaporator. The highest temperatures in the low temperature storage tank occurred during the early morning hours of 2100 to 2400 hours (3:00 to 6:00 am). Day time usage of the low and high grade hot water did not allow the low temperature water storage to recover to the high levels occurring in the early morning hours.

Economic Evaluation of the Energy Recycling System

Economic evaluation of an energy recycling system must consider, as a starting point, a dairy currently not recovering waste thermal energy from the milk refrigeration system. The decision to install an energy recycling system will usually be made on the basis of providing minimum or nearly minimum cost over the expected service life of the system. Several factors affecting system cost are:

(1) projected life of the system.

(2) capital cost of the system to be installed.

(3) present value of projected maintenance costs.

(4) present price of a given energy and projected escalation rate of that energy.

(5) value of money as reflected by interest rates.

The following section presents both a present worth and a simple payback period economic analysis of the energy recycling system as installed at the Oregon State University dairy research center. The present worth method provides an in-depth cost analysis of the two water heating schemes while the simple payback period serves as an indicator of the investment potential of such a water heating system for the consumer.

Present Worth Analysis

The present worth analysis determines the total equivalent dollar value required at time zero to cover all expenditures during the life of the energy recycling system. A comparison was made between the previous method of water heating with only commercial electric water heaters and the current method that included installation of an energy recovery system and water-to-water heat pump. The most economical choice of the two alternatives is the one with the minimum present worth.

Several assumptions were made for the following present worth analysis as discussed by Larsen (1978):

(1) Waste heat recovery unit and water-to-water heat pump system life assumed to be 20 years.

(2) Energy costs (electricity) were increased at an annual rate of 10 percent.

(3) Present electrical energy cost $0.040 per kW-hr.

(4) Annual average heat pump coefficient of performance assumed to be 4.05.

(5) Interest rate assumed to be 15 percent.

Present water heating costs included energy recycling system components and the cost of installation. A complete list of components and supplies for the water tempering system are included in Table 10. The cost of electrical wiring to a 220-volt power source was not included for the heat pump system as it was assumed that a dairy would have 220 power available. An installation cost for charging each system with its respective refrigerant was included as most farm operators were not equipped to perform such labor. Total cost for both systems was $4,688.00 and did not include metering devices as these were necessary for research only.

Capital cost for the previous water heating scheme involved only the cost of an additional water heater to meet the high temperature water requirements. Cost of the 379-liter commercial electric water heater was $528.

Table 10. Summary of system components and costs.

Waste Heat Recovery System		Heat Pump System	
Water-cooled condenser	$ 184.00	Condenser	$ 219.00
Storage tank	426.00	Evaporator	463.00
Insulation	45.00	Compressor	880.00
Circulation pump	100.00	Auxillary components	250.00
Differential temperature thermostat	50.00	Water storage tank	169.00
Pluming supplies	50.00	Circulation pumps	200.00
Installation	227.50	Insulation	25.00
TOTAL	$1,082.00	Plumbing supplies	50.00
		Fabrication and installation	1,250.00
		Miscellaneous supplies	100.00
		TOTAL	$3,606.00

Cost of energy to operate the two water heating schemes over their expected service life was determined by calculating the annual energy requirement in the milking parlor based on present water use. An annual cost was determined using current electrical energy prices and a factor representing the future annual electricity costs based on the rate of energy price escalation (Present Worth Factor). The present worth of energy costs were determined using the following equation:

$$\text{Present Worth of energy costs} = \frac{\text{Energy load} \times \text{Present energy price} \times \text{Present Worth Factor (PWF)} \times}{\text{Conversion efficiency}}$$

where:

$$\text{Present Worth Factor} = \sum_{m=1}^{N \text{ years}} \frac{(1 + Y/100)^m}{(1 + Z/100)^m} \tag{3}$$

and

Y = energy escalation rate, percent

Z = money interest rate, percent

m = system life, years

Examination of the previous water heating scheme indicated a low and high temperature heating load of 5,377 and 20,856 kW-hr per year, respectively. Present worth energy cost values for the low and high temperature water heating loads were $2,787 and $10,812, respectively. The heating cost for the commercial electric water heaters included an energy conversion factor of 1.0 (100 percent efficiency).

Electrical energy costs for the present water heating scheme were determined using energy consumption data recorded from November 1981 to May 1982. Daily average energy consumption by the present system was 29.8 kW-hr. Present worth energy cost was $5,639 based on a 20-year service life and electrical energy priced at $0.040 per kW-hr.

Present worth analysis for maintenance costs assumed a first year cost as a percentage of the system's initial cost. The annual escalation rate of maintenance costs was estimated to be 10 percent. Total life costs for maintenance were calculated using the following equation:

$$\begin{array}{c}\text{Present worth of}\\\text{maintenance costs}\end{array} = \begin{array}{c}\text{Capital}\\\text{costs}\end{array} \times \begin{array}{c}\text{First year costs}\\\text{of maintenance } (1\tfrac{1}{2}\%)\end{array} \times \begin{array}{c}\text{Present Worth}\\\text{Factor (PWF)}\end{array}$$

Present worth costs for maintenance of the previous and present water heating schemes were \$79 and \$700, respectively.

Total capital costs involved were assumed to have been borrowed by the consumer. The loan was paid off during the life of the system at an interest rate of 15 percent. The present worth of these interest payments was calculated for the life of the systems and represented a savings in income tax for the dairy farmer. The following equation determined the savings:

$$\text{Present worth of tax savings} = \text{Capital costs} \times \text{Assumed tax bracket (25\%)} \times \text{Present worth factor for interest}$$

These present worth savings to the dairy farmer were deducted from the total present worth for each system. An income tax bracket of 25 percent was assumed for the dairy farmer. Present worth of the tax savings for the previous and present water heating schemes were \$125 and \$1,092, respectively.

Table 11 summarizes the present worth cost analysis for the two water heating schemes. The minimum present worth cost was \$10,145, established for the energy recovery and water-to-water heat pump system. The present worth cost analysis did not include any tax credit for installation of energy conservation equipment as this varies from state to state. A tax credit for energy conservation equipment would reduce the present worth cost of only the energy recycling system.

Table 11. Summary of present worth analysis for two water heating schemes[a].

	Electric Water Heaters	Energy Recycling System
Capital cost	$ 528	$ 4,687
Present worth of energy costs	13,600	5,639
Present worth of maintenance costs	103	911
Present worth of income tax savings	-125	-1,092
TOTAL	$14,106	$10,145

[a]20 year life, 15 percent interest rate, 10 percent energy escalation rate

Payback Analysis

Another method of economic analysis is the simple payback period. Payback period is defined as the number of years required to recover the initial investment. The payback period for this analysis was the time at which initial investment and annual expenses with compound interest was equal to total energy savings with compound interest for the energy recovery and water-to-water heat pump system.

Value of the initial investment at the end of the payback period was

$$I = I_o(1 + Z/100)^n$$

where

I_o = initial investment (dollars)

n = payback period (years)

Annual expenses (taxes, maintenance and replacement costs) (O) were determined as a fixed percentage (f) of the initial investment and accumulated dependent upon an economic inflation rate (X). In equation form, this became

$$O = fI_o \frac{(1 + Z/100)^n - (1 + X/100)^n}{Z - X}$$

Annual energy savings (E) were expected to increase at an energy escalation rate (Y) equal to the economic inflation rate. Accumulated energy savings over the payback period were

$$E = E_o \frac{(1 + Z/100)^n - (1 + Y/100)^n}{Z - Y}$$

Definition of payback requires that initial investment plus annual expenses equal total savings, or

$$I + O = E$$

The actual energy recycling system produced an annual energy savings of $613 based on an average daily energy reduction of 42 kW-hr and electrical energy cost of $0.04 per kW-hr. A 15 percent interest rate, 10 percent energy escalation rate, and 10 percent inflation rate resulted in a payback period (n) of approximately 12.5 years.

Uncertainty in the economic market must be considered to realistically reflect the variability of payback period analyses. Sensitivity of the payback period was determined by modifying the baseline parameters (interest rate, energy escalation rate, and economic inflation rate) to study possible changes in the economic market. Of those three parameters, variation in interest rate produced the largest corresponding change in payback period. Figure 15 illustrates the effect of varying the interest rate from eight to 16 percent with a corresponding variation in payback period from 8.5 to 13.2 years.

Payback period serves as an indicator of risk when used in conjunction with a discounted cash flow method such as the present worth analysis. The apparent risk of investment in such a water heating system lessens as the payback period shortens. Present worth analysis encourages careful consideration of all factors influencing cost of the energy recovery and water-to-water heat pump systems and was the more accurate economic analysis. The difference between present worth costs ($3,961) represented the additional savings incurred over the expected life if any energy recycling system was installed. This becomes particularly important in view of uncertain energy costs facing today's dairy farmer.

The present worth analysis indicated the energy recovery and water-to-water heat pump system was a far superior method of tempering water. Further improvements in the water-to-water heat pump performance and decreased system cost would reduce initial cost and increase energy savings. Thus, the risk of investment in such a system for tempering water in milk parlors would be reduced.

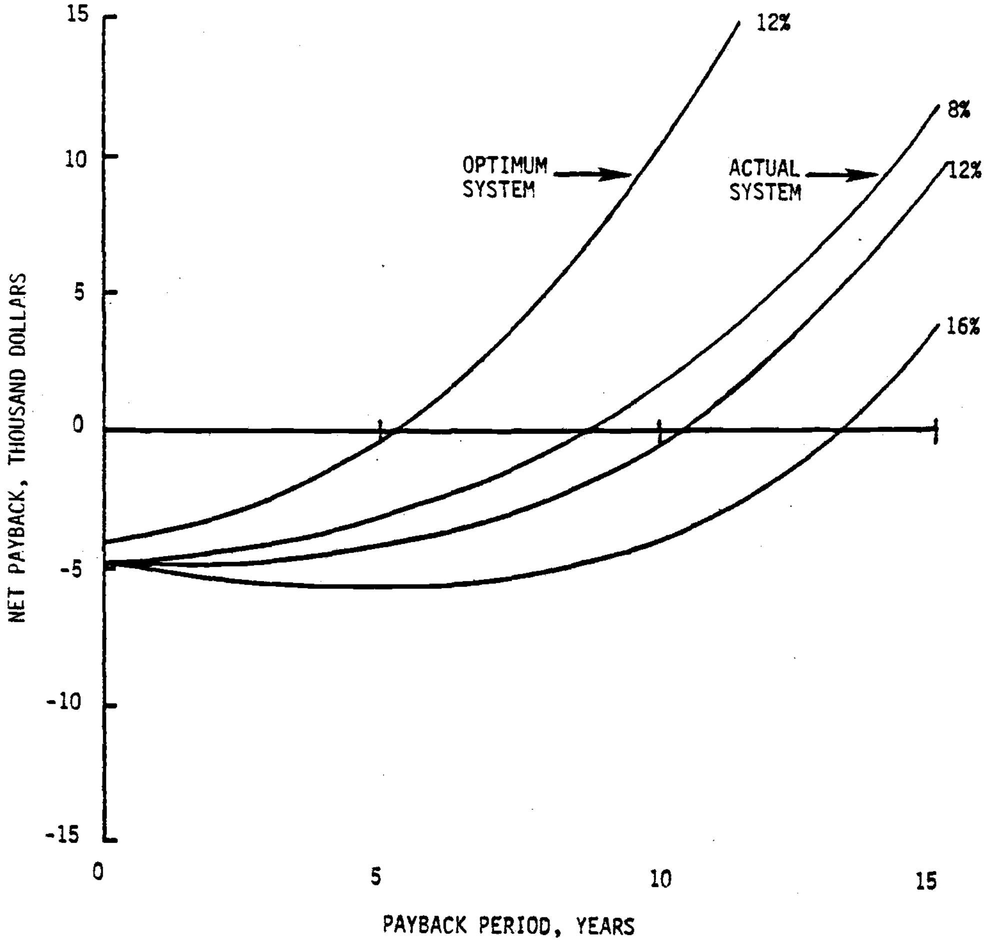

Figure 15. Payback period as a function of variable interest rates with 10 percent energy escalation and inflation rates.

V. DEVELOPMENT OF ENERGY RECYCLING SYSTEM MODEL

The development of a computer simulation model was initiated to provide insight into the operating schedule of the waste heat recovery system and water-to-water heat pump for a selected period of time. Methodology and development of the governing equations, computer simulation scheme, and comparison of simulation results with actual data are presented in this section.

The simulation model was developed with the following specific objectives:

(1) Study the operating schedule of the present waste heat recovery system and water-to-water heat pump for a representative period. This would provide a means of predicting whether the heat pump could be operated by either the present remote thermostat method, a time clock to control heat pump operating time, or a combination of the two.

(2) Study operating characteristics of the present water-to-water heat pump to attain information for further improvement in component size optimization including water storage facilities.

The model was not constructed to simulate the heat pump as one entity due to the complex thermodynamic characteristics of the refrigerant during circulation through the heat pump circuit. Rather, the heat pump condenser and evaporator, and the water-cooled condenser in the milk refrigeration system were treated as individual black-box entities.

Exit water temperatures for each entity were determined as a function of entrance water temperatures. In addition, functional equations were determined for temperature stratification levels in both the low and high temperature water storage tanks.

Methodology and Development of Governing Equations

Liquid Storage Tanks

Experimental results from previous research have shown that water storage tanks exhibit temperature stratification with height. Lavan and Thompson (1977) reported the degree of thermal stratification in a liquid storage tank was a function of storage tank configuration, incoming fluid mass flow rates, inlet and outlet port locations, and the temperature differential between the incoming fluid and liquid storage mass. An analytical study of both the low and high temperature water storage tanks for inclusion in the computer simulation model required development of a mathematical model of the storage tanks. An initial attempt involved an analysis similar to the fully stratified model as developed by Duffie and Beckman (1980) for solar liquid thermal storage tanks. This model divided the tank into N-sections or nodes of equal volume and considered the energy balance in each volume. Functions were included in the model to introduce incoming flows to the section where fluid density difference between incoming fluid and tank fluid mass was a minimum. The assumption of minimal mixing resulted in an unrealistic temperature stratification. A second approach was developed where each node was considered a fully-mixed section. Internal mixing between nodes occurred with incoming and outgoing flows. This assumed that the inlet and outlet ports made maximum use of the storage tanks' size and configuration to attain a more complete mixing.

The low temperature water storage tank was considered first. Figure 16 represents the low temperature water storage tank nodes with water inflows and outflows.

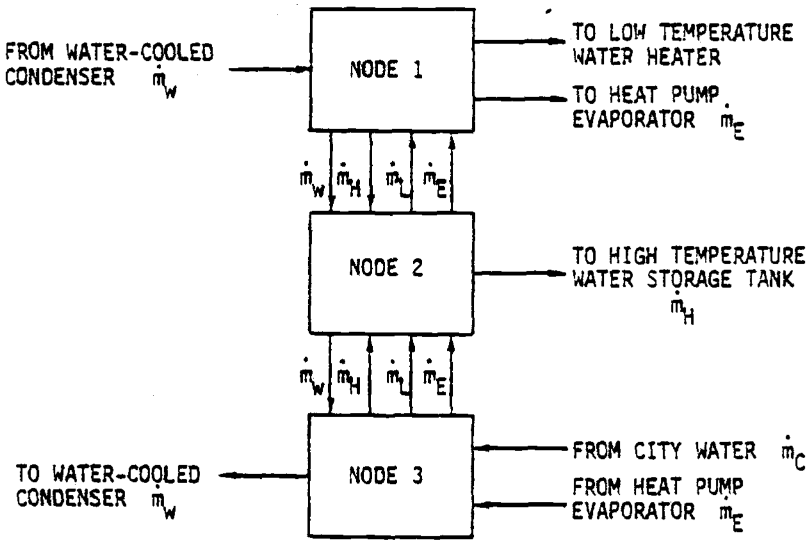

Figure 16. Nodal representation of low temperature water storage tank.

An energy balance for a fully-mixed section may be written as:

Rate of energy transfer = Rate of energy addition - Rate of energy removal - Heat loss to environment

Rewriting this relationship in heat transfer notation yields:

$$(MC_p)\frac{dT_s}{dt} = Q - L - (UA)(T_s - T_a)$$

The use of three stratification levels in each storage tank represented a reasonable compromise between a fully-mixed storage tank and a highly stratified storage tank. Applying the above energy balance equation to the three nodes of the low temperature water storage tank yields:

Node 1

$$(MC_p)\frac{dT_1}{dt} = (UA)(T_a - T_1) + \dot{m}_W(T_W - T_1) + \dot{m}_L(T_2 - T_1) + \dot{m}_E(T_2 - T_1) - \dot{m}_H(T_1 - T_2) - \dot{m}_W(T_1 - T_2)$$

Node 2

$$(MC_p)\frac{dT_2}{dt} = (UA)(T_a - T_2) + \dot{m}_W(T_1 - T_2) - \dot{m}_H(T_1 - T_2) - \dot{m}_L(T_1 - T_2) - \dot{m}_E(T_1 - T_2) - \dot{m}_W(T_2 - T_3) + \dot{m}_L(T_2 - T_3) + \dot{m}_H(T_2 - T_3) + \dot{m}_E(T_2 - T_3)$$

Node 3

$$(MC_p)\frac{dT_3}{dt} = (UA)(T_a - T_3) + \dot{m}_W(T_2 - T_3) + \dot{m}_C(T_C - T_3) + \dot{m}_E(T_E - T_3) - \dot{m}_L(T_2 - T_3) - \dot{m}_H(T_2 - T_3) - \dot{m}_E(T_2 - T_3)$$

Figure 17 represents the high temperature water storage tank nodes with the water inflows and outflows. The same type of energy

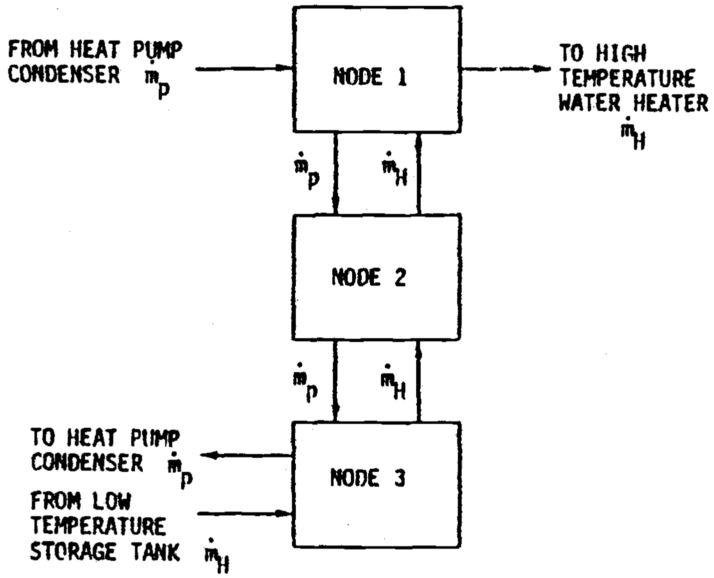

Figure 17. Nodal representation of high temperature water storage tank.

balance was applied to the high temperature water storage tank as the low temperature water storage tank. The energy balance equations for the three nodes were as follows:

Node 1

$$(MC_p)\frac{dT_1}{dt} = (UA)(T_a - T_1) + \dot{m}_P(T_P - T_1) - \dot{m}_P(T_1 - T_2) - \dot{m}_H(T_1 - T_2)$$

Node 2

$$(MC_p)\frac{dT_2}{dt} = (UA)(T_a - T_2) + \dot{m}_p(T_1 - T_2) + \dot{m}_H(T_2 - T_3) - \dot{m}_p(T_2 - T_3) - \dot{m}_H(T_1 - T_2)$$

Node 3

$$(MC_p)\frac{dT_3}{dt} = (UA)(T_a - T_3) + \dot{m}_p(T_2 - T_3) + \dot{m}_H(T_2 - T_3)$$

Heat Pump Condenser

Two methods of heat exchanger design were used to analyze the heat pump condenser. Figure 18 represents the temperature profile of the water and refrigerant flow in a condenser which includes refrigerant subcooling. Both a condensation and subcooling section are included with the composite section representing the total heat exchange through the condenser. The following assumptions were made for this

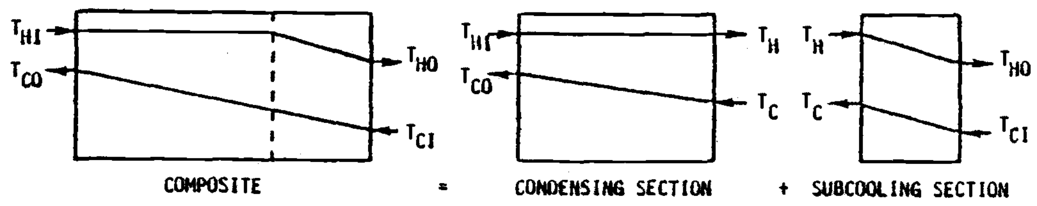

Figure 18. Temperature profile through condenser.

analysis as indicated by Welty et al. (1976):

(1) Negligible pressure drop considered through the condenser.

(2) Linear heat transfer occurred between refrigerant and water.

(3) No fouling resistance occurred.

(4) Overall heat transfer coefficient (U) was uniform through the entire condenser.

The total energy transfer rate for the condensing section is defined by the expression:

$$Q = (UA)\,(LMTD)$$

where LMTD is the log mean temperature difference and is defined as:

$$LMTD = \left(\frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)} \right)$$

Solving this expression in terms of the water exit temperature yields:

$$T_{C\ out} = T_{C\ in} + (T_H - T_{C\ in})(1 - e^{-UA/\dot{m}C_p})$$

where:

$T_{C\ out}$ = exit water temperature

$T_{C\ in}$ = entering water temperature

T_H = refrigerant condensing temperature

Evaluation of the condensing section using the LMTD method was considered a good approximation for determining water exit temperature because of the small differential between the entering and exiting temperatures of the refrigerant.

The condenser acts as a simple counter-flow heat exchanger in the subcooling section and the Number-of-Transfer Units (NTU) method was used to model this section. This method of analysis is based on the heat exchanger effectiveness, E, which when written in terms of total energy transfer, becomes:

$$Q = EW_{min} (T_{H\ in} - T_{C\ in})$$

where the heat exchanger effectiveness is defined as:

$$E = \frac{1 - \exp\left[\frac{UA}{W_{min}}\left(1 - \frac{W_{min}}{W_{max}}\right)\right]}{1 - \left(\frac{W_{min}}{W_{max}}\right)\exp\left[\frac{UA}{W_{min}}\left(1 - \frac{W_{min}}{W_{max}}\right)\right]}$$

Solving the total energy transfer rate equation in terms of water exit temperature yields:

$$T_{C\ out} = T_{C\ in} + (T_{H\ in} - T_{C\ in})\left(\frac{(1 - e^{D})}{\frac{W_{min}}{W_{max}} - e^{D}}\right)$$

where:

$T_{H\ in}$ = entering refrigerant temperature

W_{min} = $\dot{m}_H \times C_{p,H}$

W_{max} = $\dot{m}_C \times C_{p,C}$

D = $\frac{UA}{W_{min}} (1 - W_{min}/W_{max})$

Use of the NTU method in the subcooling section represented a better approximation of water exit temperature because of the greater temperature differential in the refrigerant from being subcooled. The entering water temperature, $T_{C\ in}$, in this subcooling section was the exit water temperature, $T_{C\ out}$, from the condensing section. In both sections all variables were known or determined from manufacturer's specifications with the exception of the entrance and exit water temperatures. The heat pump refrigerant condensing temperature was assumed constant over its operating interval (i.e., 82.2°C for the time the heat pump was operating). In actuality the refrigerant temperature was related to water temperature and influenced by the non-uniform opening and closing of the thermostatic expansion valve. No attempt was made in this model to account for these variations as the overall boundary conditions were sufficient in analyzing the components as black-box entities. After performing the necessary calculations, the composite governing equation for the heat pump condenser reduced to the linear form:

$$T_{C\ out} = A \cdot T_{C\ in} + B$$

where the coefficients A and B were derived from manufacturer's specifications. Complete calculations for the heat pump condenser analysis are included in Appendix B.

<u>Water-cooled Condenser in Milk Refrigeration System</u>

The same assumptions for the heat pump condenser held true for the water-cooled condenser. The major difference for the water-cooled condenser analysis was the refrigerant condensing temperature, assumed to be 48.8°C for Freon 22. No subcooling section was developed as the energy recovery unit was designed only for absorbing refrigerant superheat and heat liberated by condensation of the refrigerant. The composite governing function was the same linear form as the heat pump condenser with different coefficients for A and B.

Heat Pump Evaporator

The evaporator analysis included both an evaporation section and a superheating section. Figure 19 represents the temperature profile of the water and refrigerant flow in an evaporator. The same assumptions for the condenser analysis were applicable for the evaporator.

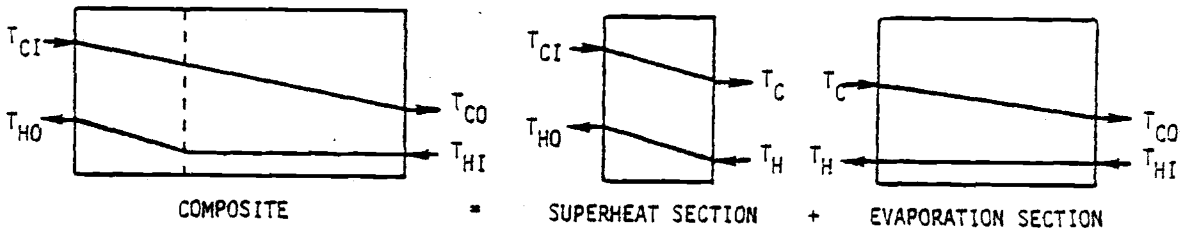

Figure 19. Temperature profile through evaporator.

The LMTD method was utilized for the energy transfer analysis in the evaporating section because of the small temperature differential between the entering and exiting temperature of refrigerant. Solving the LMTD energy transfer equation in terms of the evaporator exit water temperature resulted in:

$$T_{C\ out} = T_{C\ in} - (T_{C\ in} - T_H)\ (1 - e^{-UA/mC_p})$$

where T_H is the refrigerant evaporating temperature. The NTU method was used to analyze the energy transfer rate in the superheating section because of the greater temperature differential in the refrigerant through this section. Solving the NTU energy transfer equation for the exit water temperature yields:

$$T_{C\ out} = T_{C\ in} - (T_{C\ in} - T_{H\ in})\ \frac{(1 - e^D)}{\frac{W_{min}}{W_{max}} - e^D}$$

where $T_{H\ in}$ is the entering refrigerant temperature. The evaporating temperature for the refrigerant was assumed constant at 29.4^{o}C over the operating interval of the heat pump. Solution for the composite section yields:

$$T_{C\ out} = C \cdot T_{C\ in} + D$$

where the coefficients C and D were derived from manufacturer's specifications. Complete calculations for the evaporator analysis are included in Appendix B.

Computer Model Format Using GASP-IV Simulation Language

The FORTRAN-based computer simulation language, GASP-IV, was selected to model the energy recycling system because of its specific functional capabilities. A GASP-IV simulation program consists of both user defined and GASP-IV defined parts. The user portion includes subroutines for initialization, equations defining state variables and state events, event processing and code definitions, and special data collection and reporting instructions. GASP-IV provides subroutines for the mode controller, initialization of data and events, collection, storage, and retrival of data, computation and reporting of statistics, monitoring of programs and error reporting, and miscellaneous support. A flow chart illustrating the general outline of a complete GASP-IV simulation program as taken from Pritsker (1974) is shown in Figure 20.

The user defined subroutines written for the energy recycling system simulation model involved a simplistic logic and no flowcharts for these subroutines will be included in this section. A description of the events that occur in the milking parlor during a 24-hour period are as follows:

(A) Discrete Events

Event (1): Milking begins. Low temperature water used to prepare cows for milking.

Event (2): Milking ends. Low temperature water use ends.

Event (3): Milk refrigeration system begins operating. Water circulated through water-cooled condenser from low temperature water storage tank.

Event (4): Milk refrigeration system stops operating. Water circulated through water-cooled condenser stops.

Event (5): Milking equipment wash cycle begins. High temperature water used to clean and sanitize equipment.

Event (6): Wash cycle ends. High temperature water use stops.

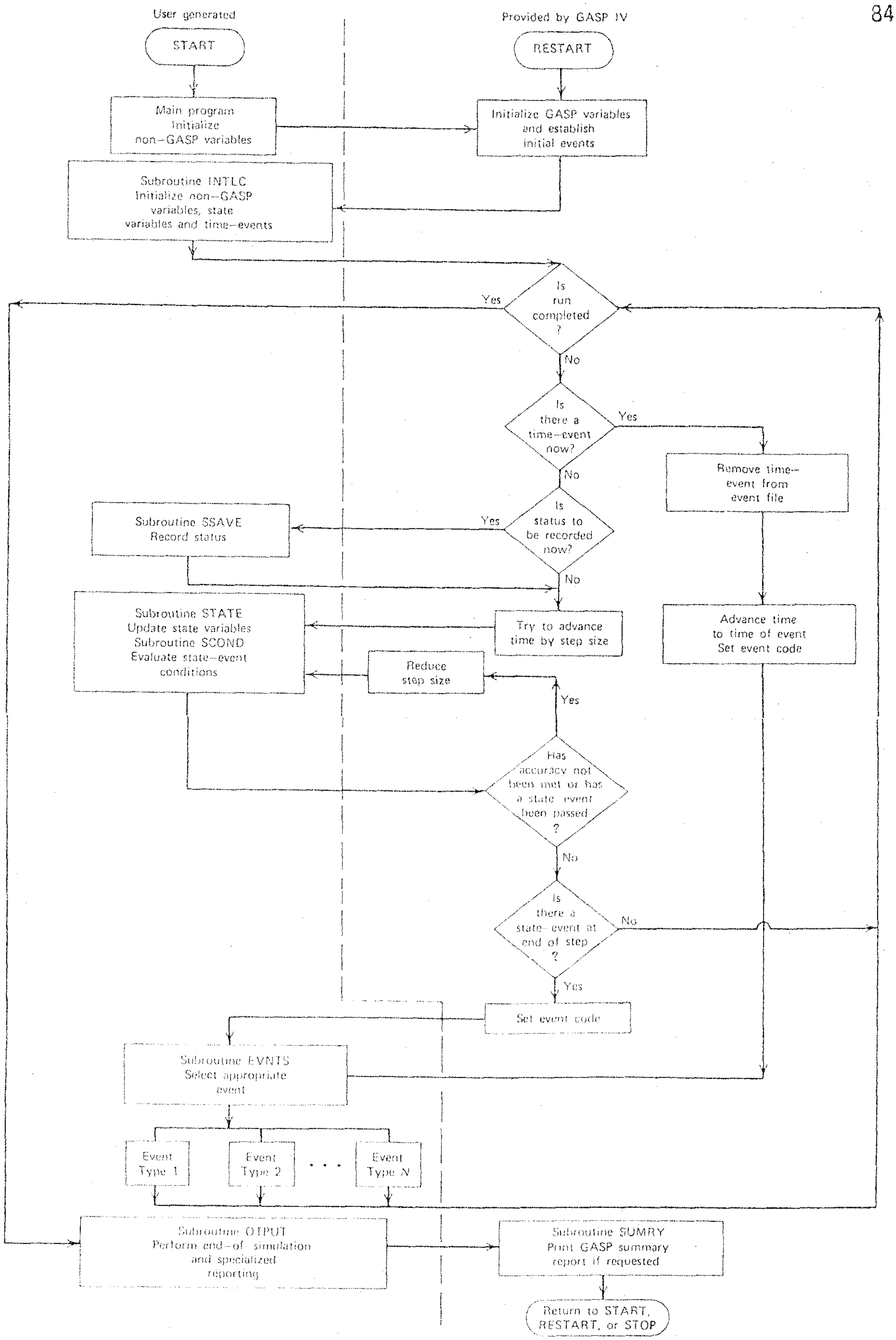

Figure 20. Flow chart of a GASP-IV simulation program.

Event (7): Milk storage tank wash cycle begins. High temperature water used for tank cleaning (wash cycle occurs every other day).

Event (8): Milk storage tank wash cycle ends. High temperature water use stops.

Event (9): Heat pump begins operation. Water circulated through the heat pump condenser from the high temperature storage tank and through the heat pump evaporator from the low temperature storage tank.

Event (10): Heat pump operation stops. Water circulation stops.

(B) State Events

- Temperature in Node 1 of high temperature water storage tank reaches 73.9^{o}C. Heat pump is shut off (go to Event 10).
- Temperature in Node 1 of high temperature water storage tank drops to 65.5^{o}C. Heat pump is started (go to Event 9).
- Temperature of water entering water-cooled condenser in milk refrigeration system reaches 40.5^{o}C. Water circulation through water-cooled condesner stops (go to Event 4).
- Temperature of water entering water-cooled condenser drops below 40.5^{o}C. Water circulation through water-cooled condenser starts (go to Event 3).

Figure 21 represents the time schedule of discrete events for the milking parlor operation in a 24-hour period.

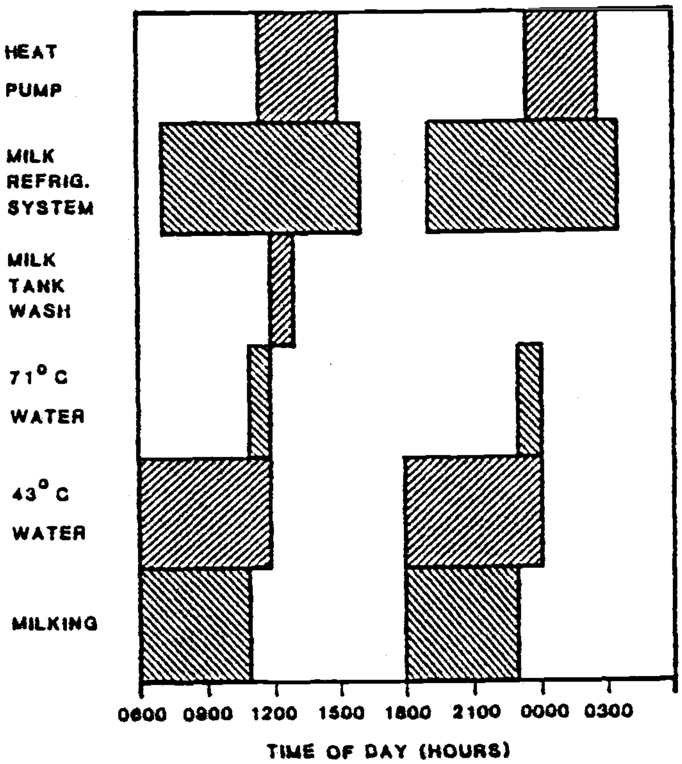

Figure 21. Time schedule of events in OSU milking parlor.

Subroutines

The user defined subroutines for this energy recycling system simulation model were as follows:

(1) INTLC

Subroutine initialized all variables and beginning events. Table 12 includes the variables, definitions, initial values, and model name.

Table 12. Variables and definitions in simulation models.

Variable	Definition	Initial Value	Model Variable
$T_{SH,1}$	Water temperature at Node 1 of high temperature tank	73.9°C	SS(1)
$T_{SH,2}$	Water temperature at Node 2 of high temperature tank	71.1°C	SS(2)
$T_{SH,3}$	Water temperature at Node 3 of high temperature tank	68.3°C	SS(3)
$T_{SL,1}$	Water temperature at Node 1 of low temperature tank	40.5°C	SS(4)
$T_{SL,2}$	Water temperature at Node 2 of low temperature tank	40.0°C	SS(5)
$T_{SL,3}$	Water temperature at Node 3 of low temperature tank	37.8°C	SS(6)
$T_{W,R}$	Water temperature returning from water-cooled condenser	40.2°C	SS(7)
$T_{E,R}$	Water temperature returning from heat pump evaporator	31.0°C	SS(8)
$T_{C,R}$	Water temperature returning from heat pump condenser	76.9°C	SS(9)
T_A	Ambient air temperature -- April	10.0°C	ATEMP
T_{City}	Temperature of city water supply -- April	11.1°C	CITY
m_C	Mass flow rate through water-cooled condenser (908 kg/hr)	0.0 kg/hr	WCCM
m_W	Mass flow rate through heat pump condenser (726 kg/hr)	0.0 kg/hr	HPCM
m_E	Mass flow rate through heat pump evaporator (863 kg/hr)	0.0 kg/hr	HPEM
m_L	Mass flow rate -- low temperature water heater (49.0 kg/hr)	0.0 kg/hr	WMILK
m_H	Mass flow rate -- high temperature water heater (568 kg/hr)	0.0 kg/hr	HMFR
M_{SH}	Mass of water in each high temperature storage tank	151.3 kg	WMASH
M_{SL}	Mass of water in each low temperature storage tank	277.9 kg	WMASL
UA_I	Heat loss coefficient -- area product		
	Low temperature storage tank		
	Node 1	0.577 W/C	UA(1)
	Node 2	0.439 W/C	UA(2)
	Node 3	0.577 W/C	UA(3)
	High temperature storage tank		
	Node 1	0.654 W/C	UA(4)
	Node 2	0.556 W/C	UA(5)
	Node 3	0.654 W/C	UA(6)

(2) EVNTS

Subroutine provided the event code definitions as previously listed.

(3) STATE

Subroutine provided the equations for the state variables. GASP-IV uses a Runge-Kutta-England (RKE) algorithm to integrate the differential equations. Step size and error allowance are user specified.

(4) SCOND

Subroutine contained conditions defining state events. Flags were used to denote the crossing of a condition and thus, a state event occurrence.

(5) SSAVE

Subroutine contained data collection and plotting instructions.

Data Collection and Format of Results

Collection of statistics involved the two variables; WCC-TIME, the water-cooled condenser operating time, and HP-TIME, the heat pump operating time. The two variables were monitored and recorded during the 24-hour period. Plots of low and high temperature water storage tank nodes versus time for the 24-hour period were made to study temperature stratification variation. Appendix C contains a complete listing of the computer simulation program and data file.

Comparison of Simulation Data

Accuracy of the simulation results was monitored for the following:

(1) Heat pump and energy recovery system operating time.

(2) Temperature stratification in low and high temperature water storage tanks

Actual data were recorded for the heat exchanger entering and exiting water temperatures for selected short duration periods of operation. Figures 22 through 24 illustrate the exit water temperatures as predicted by the heat exchanger functions and as determined during data collection for the water-cooled condenser heat pump condenser, and heat pump evaporator, respectively.

Prediction of the exit water temperature from the water-cooled condenser of the energy recovery system was accurately described by the theoretically-determined water-cooled condenser function. Actual data indicated a linear relationship between the entering and exiting water temperatures through the water-cooled condenser. A coefficient (R^2) value of 0.9828 was realized for the 50 pairs of data monitored. The small divergence of the actual exiting temperature in the upper range of entering water temperatures can be explained by the increase in the Freon 22 refrigerant superheat as the entering water temperature approached the maximum allowable entering temperature of 40.5^{o}C. The increase in the compressor discharge temperature allowed the heat exchanger to maintain a significant temperature differential between the entering refrigerant and water temperatures and thus, maintain a linear energy transfer relationship at the higher entering water temperatures. The model assumed a constant refrigerant

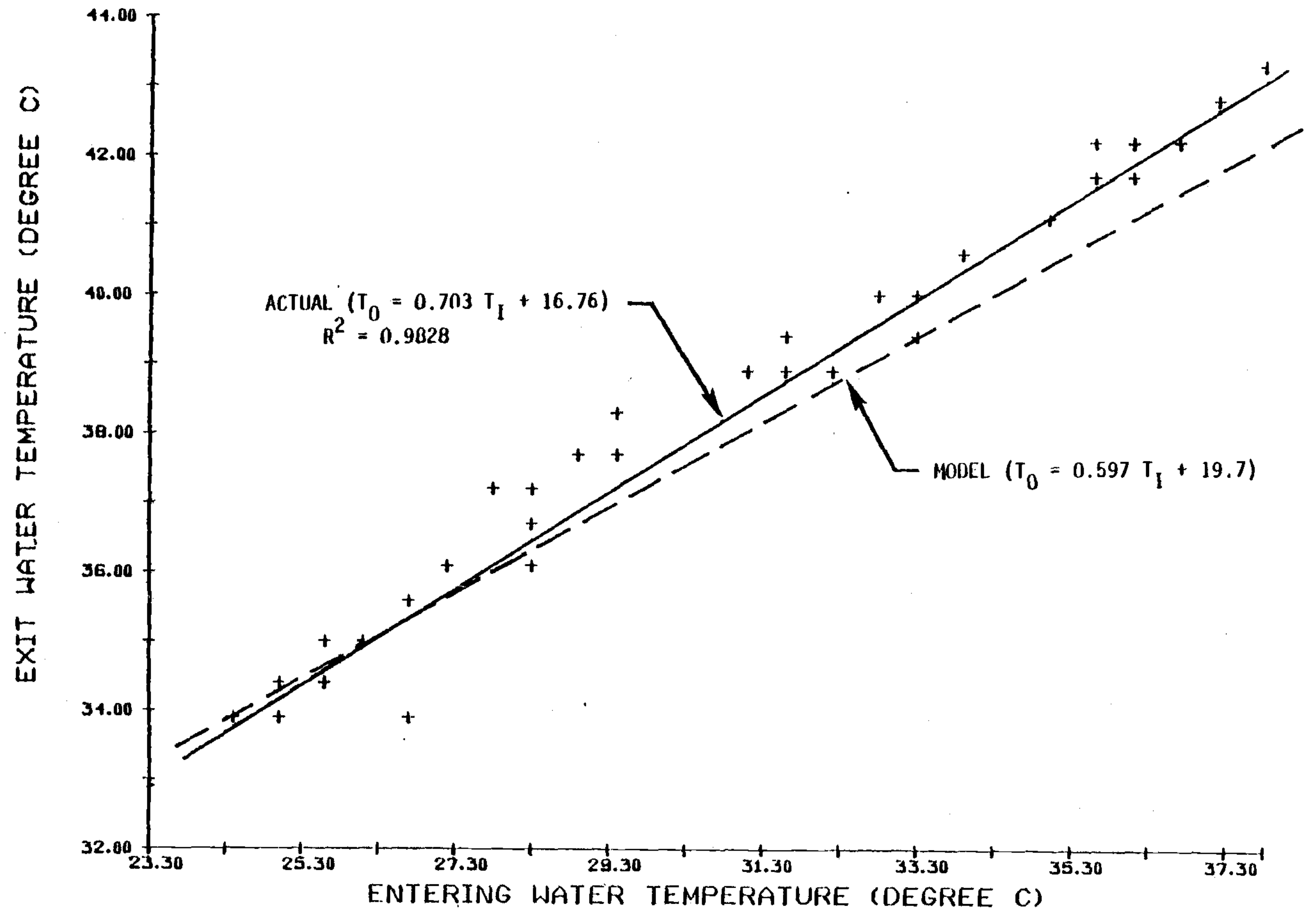

Figure 22. Exit water temperature vs. entrance water temperature for water-cooled condenser in energy recovery system.

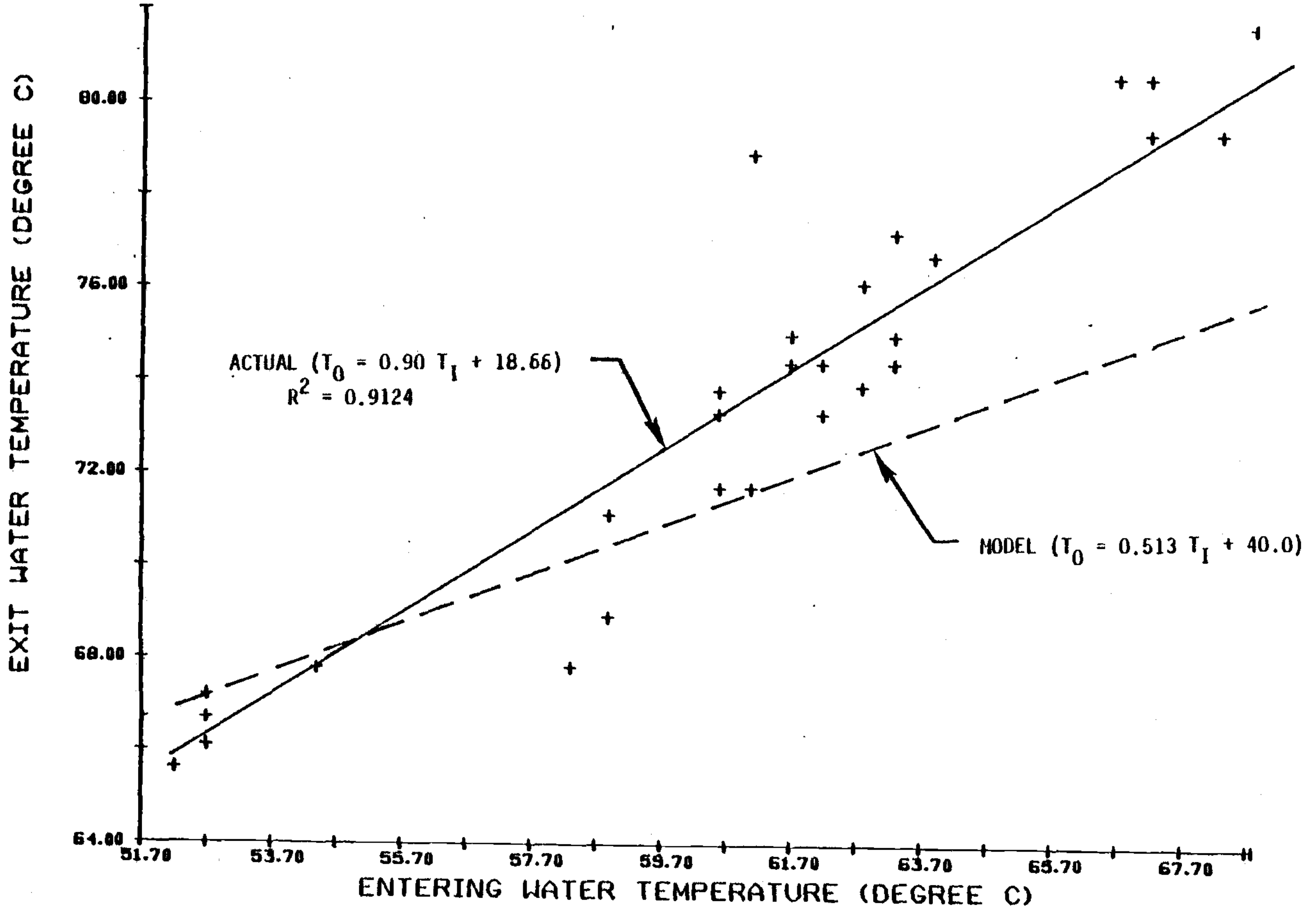

Figure 23. Exit water temperature vs. entrance water temperature for heat pump condenser, 9.5 liters/minute (2.5 gpm).

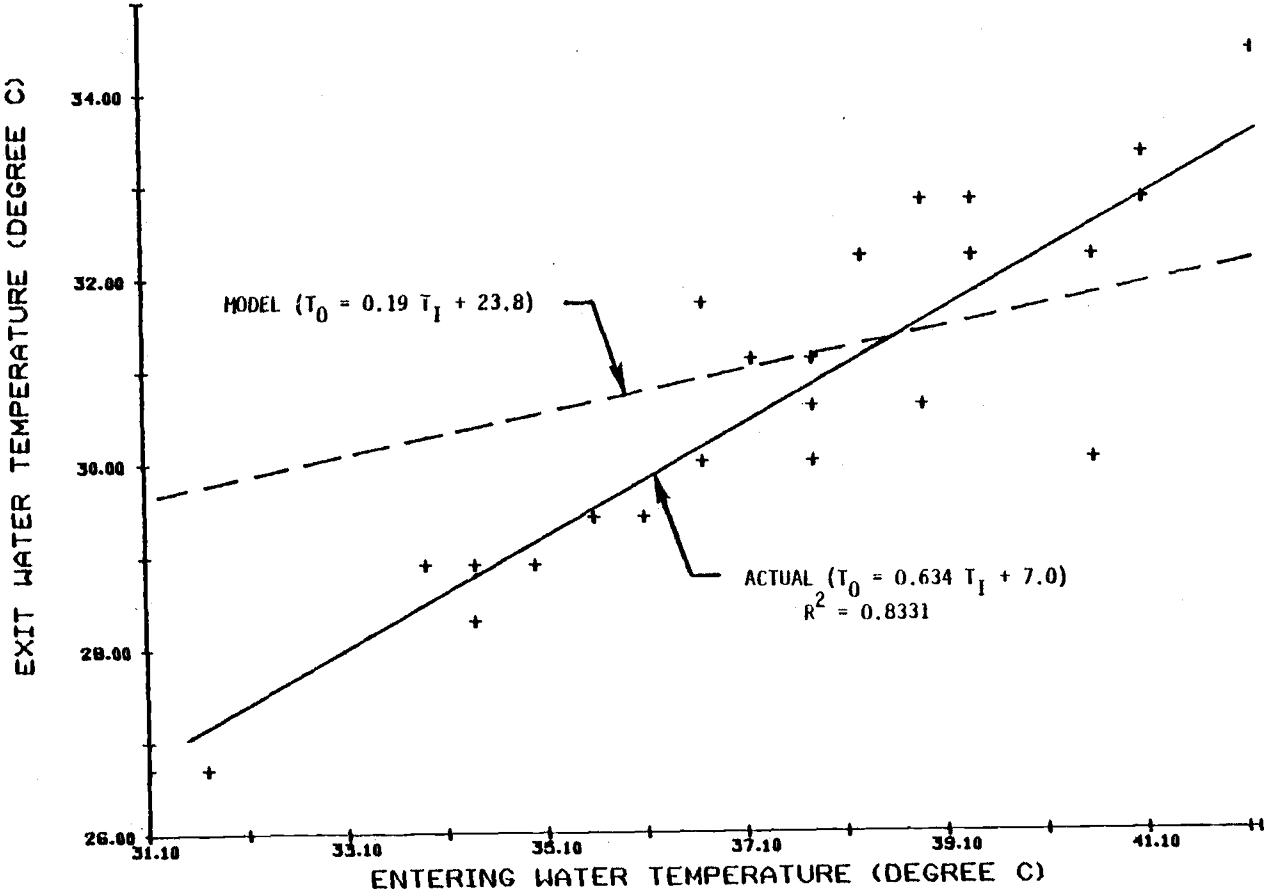

Figure 24. Exit water temperature vs. entrance water temperature for heat pump evaporator, 13.2 liters/minute (3.5 gpm).

condensing temperature of 48.9°C while in reality the refrigerant condensing temperature varied as a function of the energy removed in the water-cooled condenser.

A linear relationship also existed between the entering and exiting water temperatures through the heat pump condenser. An R^2 value of 0.9124 was realized for the 40 pairs of data collected. The heat pump condenser theoretical model significantly under-predicted the existing water temperatures in the range of entering water temperatures above 60°C. This was largely due to the assumptions in the development of the theoretical model that specified a constant condensing temperature of 82.2°C and defined the condensing section of the condenser as being the first 85 percent of its area. Actual system performance indicated an increased refrigerant condensing temperature above 82.2°C at entering water temperatures above 60.0°C. This would allow an increased energy transfer rate over that indicated by the model with a constant refrigerant condensing temperature. Definition of a specific condensing and/or subcooling region in the heat pump condenser was highly unpredictable and was only used as an initial starting point for development of this model. Further refinement would be required to accurately describe Freon 114's operating characteristics through the condenser. A revised model of the heat pump condenser should include the capacity for variable refrigerant condensing temperatures during the heat pump operating cycle, variable water flow rates through the condenser, and improvement in determining the condensing and subcooling regions.

The heat pump evaporator displayed a more scattered array of exiting water temperatures versus entering water temperatures as compared to the condenser functions. The linear regression function for the 40 pairs of entering and exiting water temperatures resulted in an R^2 value of 0.8331. The theoretical model over-predicted the exiting water temperatures in the lower range (31^o to 35^oC) of entering water temperatures and under-predicted exiting water temperatures in the upper range (39^o to 42^oC) of entering water temperatures. The small slope of 0.19 for the model indicated the exiting water temperature approached the refrigerant evaporation temperature irregardless of the entering water temperature. Assumption of a constant refrigerant evaporation temperature of 29.4^oC for the entire heat pump operating cycle limited the amount of energy transfer for the small temperature differential between the refrigerant and water entering the evaporator in the lower range of entering water temperatures. Conversely, energy transfer increased substantially in the upper range of entering water temperatures as a result of an increased temperature differential. Actual data indicated a wide variation in the evaporation temperature for Freon 114 over the range of entering water temperatures monitored. Separation of the evaporation and superheat region in the evaporator had the effect of increased divergence of the model from actual data. These regions were extremely variable as refrigerant thermal characteristics were continuously changing during the evaporation and superheat portion of the heat pump cycle. Actual performance by the heat pump evaporator proved to be inefficient due to unequal refrigerant flow distribution in its double row configuration. This resulted in poor conductive energy transfer between water and refrigerant and thus, a reduction in overall heat transfer. An improvement in the theoretical model should include the capacity for variable refrigerant evaporation temperatures and water flow rates in addition to determining evaporation and superheat regions.

The interaction between the functions for the heat pump condenser, heat pump evaporator, and water-cooled condenser and the temperature stratification levels in the high and low temperature storage tanks was the primary focus of the simulation. Figure 25 illustrates temperature stratification in the high and low temperature water storage tanks for a 24-hour period as predicted by the simulation model. The Time of Day scale began at 0.00 hours (6:00 am) to coincide with the beginning of the milking operation. Depression of the temperatures was indicative of withdrawals for the low and high temperature water uses and heat loss to the environment. The simulation maintained the highest temperature stratification levels during the time period prior to high temperature water use for equipment washing. Low grade hot water replaced the high grade hot water as the high grade hot water was removed from storage for the wash cycle. This reduced the temperature levels in the low temperature storage and thus, the heat source temperature for the heat pump. This condition produced a longer operating period for the heat pump to maintain the heat sink temperature required in the storage tank.

Comparison of the predicted temperature stratification in Figure 25 with the actual measured values in Figure 14 of the RESULTS AND DISCUSSION section indicated the model was able to predict the general pattern of temperature stratification in both the low and high temperature storage tanks. One consideration made in comparing the actual data with the predicted values was the use of average hourly data for the plot of actual temperature stratification

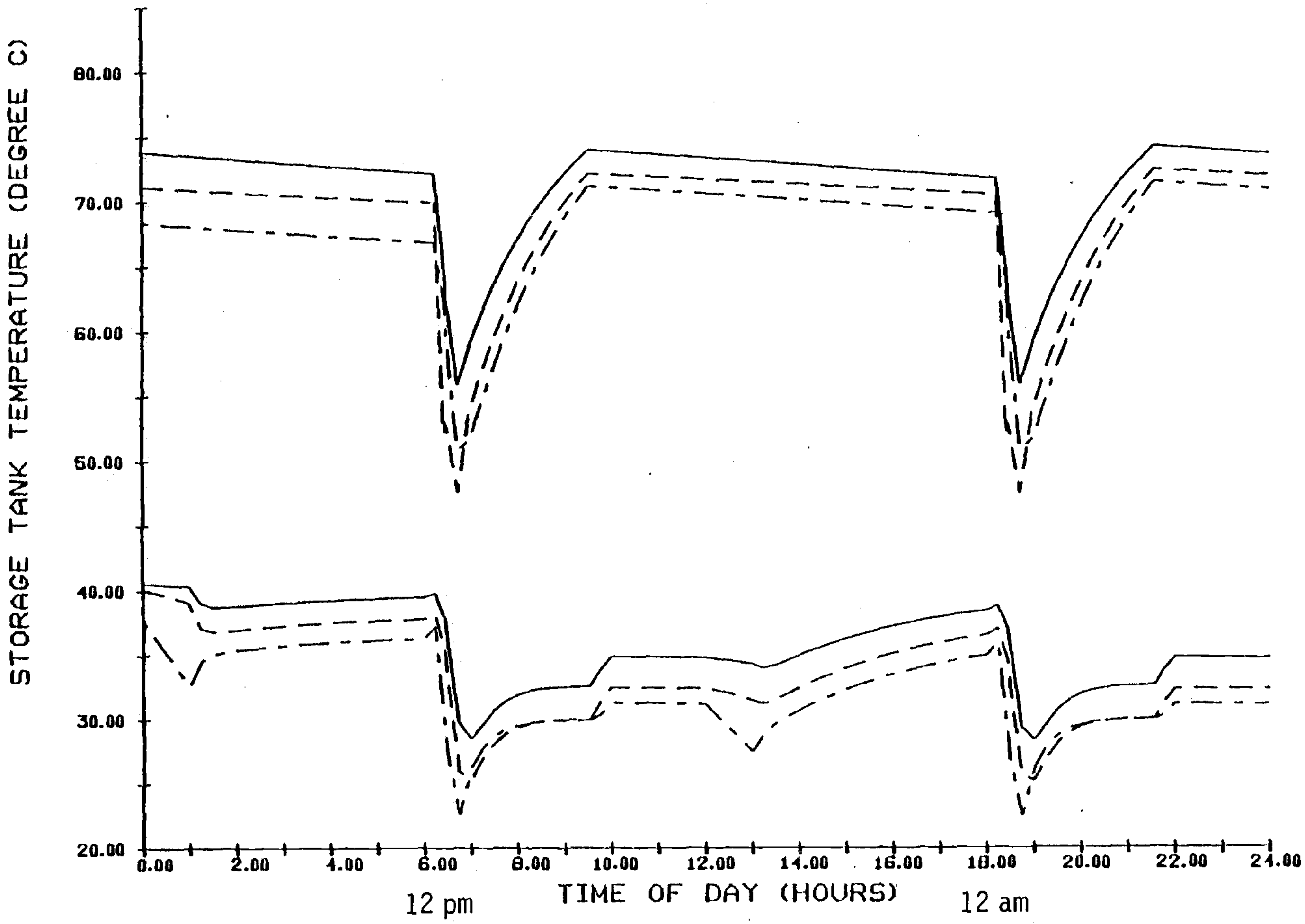

Figure 25. Predicted temperature stratification in low and high temperature water storage tank for a 24-hour period.

over a 24-hour period. Values were indicative of hourly temperatures for the seven months of data collection and included a wide variation in points. The simulation model used specified values for one 24-hour period representing an "average" condition during the month of April. The major divergence of actual from predicted occurred in the low temperature storage tank during the time period of 20:00 to 24:00 (2:00 to 6:00 am). The actual data showed the low temperature levels to rise to the original temperatures of approximately 40° to $42^{\circ}C$. The simulation model predicted a smaller rise in temperatures during this period due to low temperature water consumption and the primary milk refrigeration system shutting off after 22:00 hours (4:00 am). The divergence resulted from a smaller quantity of low temperature water being used during the night shift milking and cleaning operation. Hand washing of miscellaneous equipment and other uses were limited to the daytime operation.

Temperature depression during the water withdrawal period for equipment washing was more pronounced in the simulation. This was attributed to the predicted mixing characteristics of the tanks' configuration. Again, the comparison of an average hourly value with a time specific value contributed to the pronounced difference.

Predicted heat pump and water-cooled condenser operating time by the simulation were accurate for an operating period of 24 hours. The simulation model predicted an average heat pump operating time of 3 .1 hours twice per day. Actual heat pump operating time averaged 2.4 to 3.4 hours twice per day during the seven months of data collection. These values were based on visual observation for selected days.

The operating period for the water-cooled condenser was predicted to be 9.0 hours twice per day during the simulation. This was the original assumption for the operating hours of the primary milk refrigeration system and indicated that water entering the water-cooled condenser never attained the cut-out temperature of 40.5^{o}C to disengage the water-cooled condenser. Observation of the actual system confirmed an on-off cycling was occurring as the temperature of the water entering the water-cooled condenser approached 40.5^{o}C. The water-cooled condenser ran approximately 90 percent of the time the primary milk refrigeration system operated. Observations showed the water-cooled condenser operating time amounted to approximately eight hours twice per day.

Overall performance of the energy recycling system simulation model was in good agreement with the actual system performance for temperature stratification in the low and high temperature water storage tanks and the operating periods of the heat pump and water-cooled condenser. A larger base of simulation predictions are required to more accurately compare the model's performance with a long-term actual system performance.

Anticipated Performance of An Optimized System

Performance of the combined energy recovery and water-to-water heat pump system during periods of maintenance-free operation plus results from the system simulation model indicated several areas where additional energy savings could be achieved. Wide acceptance of this water tempering concept would also create a large market for commercially manufactured systems. Mass production of unitary-type water tempering systems could substantially reduce equipment and fabrication costs. Although time and resources did not allow actual optimization of system component sizes and operating schedules, the following management options would save energy.

(1) Reset thermostat in low temperature electric water heater to 41^{o}C. Limit water heater operation to periods of low grade hot water demand with a timer. Insulate and minimize water conveyance line length from water heater to end use.

(2) Limit operation of water-to-water heat pump to periods when desuperheater heat exchanger has built thermal energy source temperatures (low grade water storage tank) to maximum or nearly maximum following completion of the milking sequence. This would decrease the low temperature water heater operating time plus allow the water-to-water heat pump to operate at its maximum coefficient of performance.

(3) Use a timer to limit operation of the high temperature electric water heater to the period immediately preceding high grade hot water usage.

Based on the above operating scheme, electrical energy consumption for low temperature needs would be nearly eliminated. As previously stated, excess low grade thermal energy was available during the milk refrigeration process. Energy consumption by the water-to-water heat pump would be reduced as a result of more optimum heat source and sink temperatures and shortened operating times. Energy consumption in both electric water heaters would also decrease resulting from limited operating time.

As illustrated in Table 8, energy consumption by the low temperature water heater was 26.9 kW-hr/1000 liters (10.9 kW-hr/day). This may be reduced to 2.5 kW/hr/1000 liter (1 kW-hr/day) or less. High temperature energy consumption of 24.4 kW-hr/1000 liter (18.9 kW-hr/day) may be reduced as low as eight to 15 kW-hr/1000 liter (6 to 12 kW-hr/day) based on an average heat pump coefficient of performance of 4.5 and no additional heating by the high temperature water heater. Thus, total electrical energy consumption may be as low as 10 kW-hr/1000 liters (7 kW-hr/day). Resultant energy savings beyond those indicated by the present system would amount to 22.8 kW-hr/day or $0.91/day. Total energy savings for an optimally operated system would be $950 per year. Combining the energy savings with an estimated mass production system cost of $4000 would result in a payback period of 5.2 years. Assuming a 12 percent interest rate, 10 percent energy escalation rate, and 10 percent inflation rate, improved system operation and reduced cost could shorten the payback period by approximately five years as shown in Figure 15.

VI. SUMMARY AND CONCLUSIONS

An investigation was performed to determine the energy conservation potential and economic feasibility, under actual operating conditions, of a water-to-water heat pump specifically designed and operated to temper process water for cleansing and sanitizing milking parlor equipment. Commercially designed and manufactured water-to-water heat pumps of the size and specific capabilities required for use in milking parlors are presently unavailable. Information documenting the use of water-to-water heat pumps for on-farm applications is required before the consumer can consider its potential. Once sufficient evidence of the energy savings produced by the heat pump has been gathered, then acceptance by the dairy industry will increase dramatically.

The water-to-water heat pump was assembled using off-the-shelf refrigeration components selected for Freon 114's (high temperature refrigerant) thermal characteristics and sized to meet the 71.1^{o}C water heating load at the 130 cow Oregon State University dairy research center. Performance data were recorded for the 10.6 kW nominally rated heat pump specifically constructed for this application using recovered thermal energy from the milk refrigeration system as its heat source. Results indicated the heat pump was capable of providing the required quantity and quality of high temperature (71.1^{o}C) water without substantially reducing low grade hot (40.5^{o}C) water (heat pump heat source) temperatures. Actual heat pump coefficient of performance increased 38 percent over the design value, the result of increased refrigerant subcooling in the condenser.

Design, fabrication, and evaluation of the water-to-water heat pump, under actual production conditions, was the primary objective of this research effort. Optimization of heat pump component size and fabrication of the actual system to improve performance was beyond the scope of this investigation.

The following specific conclusions were drawn from this investigation:

(1) Performance data indicated that in-series operation of a desuperheater energy recovery system and water-to-water heat pump have excellent energy conservation potential on dairy farms. Electrical energy consumption for tempering low and high grade process hot water decreased from 110.0 to 51.3 kW-hr per 1,000 liters of water, an overall reduction of 53 percent.

(2) Electrical energy savings of approximately $51 per month were realized after installation of the energy recycling system based on an energy cost of $0.04 per kW-hr. The payback period to recover the $4,690 initial investment for the complete energy recycling system was 12.5 years.

(3) Coefficients of performance ranging from 3.6 to 5.6 were realized under actual production conditions with proper water management practices. Average coefficient of performance for the study was 4.05. During a period of maintenance-free operation (March - April 1982) the average coefficient of performance was 4.15.

(4) Proper water (energy) management practices at the dairy would result in coefficients of performance ranging from

4.0 to 5.5 as evidenced by several test periods where strict water use control was maintained.

(5) Operating the heat pump at maximum heat source water temperatures improved the coefficient of performance. The smaller the temperature differential between heat source and sink storage tanks, the higher the coefficient of performance. Energy transfer in the heat exchangers was improved by maintaining moderate temperature stratification in both heat source and sink storage tanks.

(6) Actual and predicted system performance indicated that optimal system operation could yield additional energy savings of $340 per year or more. Combining the energy savings with system cost reductions could produce a payback period of five years or less.

Overall performance of the complete energy recycling system indicated the heat pump components selected were accurately sized to meet the required high temperature water heating load for a 130 cow dairy. Water storage facilities were also of sufficient capacity to maintain adequate water quality, providing conservative water usage was practiced.

Sizing a water-to-water heat pump and energy recovery system for a dairy ultimately depends on that dairy's water and energy management practices. As herd size expands, low grade hot water used for cow preparation increases proportionately while tempered water for equipment cleansing does not. Therefore, sizing the energy recovery system including the low temperature water storage tank is critical to maintain low grade hot water needs. However, a water-to-water heat pump designed for one herd may meet high grade hot water needs for a range of herd sizes depending on the time of temperature recovery allowed.

A computer model designed to simulate the energy recycling system was developed. The simulation model adequately predicted heat pump and energy recovery system operating time and temperature stratification in the low and high temperature water storage tanks.

The simulation model accurately predicted the following:

(1) temperature stratification in the low and high temperature water storage tanks for a 24-hour period;

(2) daily heat pump and energy recovery system operating times;

(3) water temperature change through the water-cooled condenser in the energy recovery system.

The simulation model did not accurately predict the following:

(1) water temperature change through the heat pump condenser;

(2) water temperature change through the heat pump evaporator.

VII. RECOMMENDATIONS FOR FUTURE RESEARCH

The water-to-water heat pump specifically designed to temper water to levels required for sanitizing equipment in the 130 head OSU dairy research center proved that it was capable of creating substantial energy savings. This heat pump was designed and components were selected based on sound engineering judgments. Performance of the entire energy conservation system and especially the water-to-water heat pump was superior to values projected during the initial design and system has proven itself capable of significant energy savings through reduced expenditures for water heating. The concept of this energy recovery system that includes a high temperature water-to-water heat pump is readily adaptable to any milking parlor that has a mechanical refrigeration system to cool milk.

Before the research system can be made commercially available to the consumer, further developmental research is needed. The two major areas where additional development is required are component optimization and operation experience in a dairy where water (energy) conservation is a high priority and would be rigidly practiced. Recommendations for further research include the following:

(1) Optimization of heat pump component sizes (including heat source and sink storage tanks) to maximize performance for specific sized milking operations. An initial step would be fine tuning the present heat pump configuration to determine the maximum obtainable average coefficient of performance. This would involve a detailed analysis of water flow rates through

the evaporator and condenser; testing the heat pump under various superheat conditions using the hand expansion valve; and operating the system on its most efficient schedule. Evaluation at this level would be made to recommend further component modification without sacrificing performance. Adjustment in component size and/or configuration would be the next step to improve heat transfer in the heat exchangers and reevaluate the compressor requirements. Interrelated with this would be selection or manufacture of components specifically designed for Freon 114's thermal characteristics.

(2) Evaluation of the water storage facilities to deterine the possibility of eliminating the storage tanks and use of the commercial water heaters as both storage and back-up in case of heat pump failure.

(3) Attempt to define a water-to-water heat pump system that commercial heat pump manufacturers could develop and market. This would involve fabrication of a unitary-type heat pump using components designed for Freon 114's specific thermal characteristics and sized to meet heating loads as required for various sized dairy operations.

(4) Identify the most efficient heat pump operating schedule. This would include observation of the present operating sequence and its effect on the heat source reservoir temperature using an event recorder; identifying the time

periods when the heat source temperature is maximum; and testing the heat pump under various control schemes. Use of a time clock to control the heat pump and commercial water heaters would allow better utilization of maximum heat source temperatures between periods of low grade hot water use and boost the heat sink to desired temperatures in advance of high grade hot water use. A time clock could also be used in conjunction with the remote thermostat to meet high temperature requirements and minimize heat pump operating time.

(5) Develop an improved computer model that includes the complex thermodynamic characteristics of the heat pump cycle for Freon 114. The improved version would aid in refining component size, estimating water storage capacities, and identifying a more efficient operating schedule. The model should also be expanded to include the capacity of analyzing system requirements for various herd capacities.

(6) Install the present energy recovery system and water-to-water heat pump in a privately-owned and operated dairy of equivalent herd size to the OSU dairy and maintain rigid water and energy conservation practices. Evaluation under actual production conditions would better indicate the full potential for the energy recycling system.

BIBLIOGRAPHY

American Society of Agricultural Engineers. 1981. Agricultural Engineers Yearbook. May.

American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc. 1980. ASHRAE Systems Guide and Data Book.

American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc. 1981. ASHRAE Handbook of Fundamentals.

Ambrose, E. R. 1966. Heat Pumps and Electric Heating. New York: John Wiley and Sons Company.

Bickert, W. 1979. Recovery of milk heat on dairy farms in the United States. Ninth International Congress of Agricultural Engineers Paper No. IV-3-IIE.

Braud, H. J. 1979. Water source heat pumps for agricultural applications. American Society of Agricultural Engineers Paper No. 79-4562. December.

Canada, J. R. and J. A. White, Jr. 1980. Capital Investment Decision Analysis and Engineering. Englewood Cliffs, New Jersey: Prentice-Hall, Inc.

Cromarty, A. S. 1968. Water heating with heat energy recovered from a refrigerated farm tank. Journal of Agricultural Engineering Research 13(2):225-240.

Downing, D. Engineer at E. I. DuPont, Inc. 1981. Personal communication.

Duffie, J. A. and W. A. Beckman. 1980. Solar Engineering of Thermal Processes. New York: John Wiley and Sons Company.

Elwell, D. L., W. L. Roller, and H. M. Keener. 1980. Alternate energy for heating dairy process water. American Society of Agricultural Engineers Paper No. 80-3032. June.

Gilman, S. F. 1975. Heat pumps -- What they are and what they do. Solar Energy Heat Pump Systems for Heating and Cooling Buildings. Proceedings of workshop conducted at Pennsylvania State University. June.

Johnson, N. Halstead and Mitchell of Scottsboro, Alabama. 1981. Personal communication.

Healy, C. T. and T. I. Wetherington, Jr. 1965. Water heating by recovery of rejected heat from heat pumps. ASHRAE Journal. April. pp. 68-74.

Hellickson, M. L. and J. E. Kirby. 1979. Solar and waste heat tempered water for milking parlors. American Society of Agricultural Engineers Paper No. 79-4046. June.

Hellickson, M. L. 1980. Solar and waste heat tempered water for milking parlors. Final report for Pacific Power and Light Company and Portland General Electric Company. Contract No. 70177894. June. 27 pp.

Hustrilid, A. and H. A. Cloud. 1952. Heating water with the heat pump. Agricultural Engineering 15(June):357-359.

Kemler, E. N. and S. Oglesby. 1950. Heat Pump Applications. New York: McGraw and Hill Book Company.

Koelsch, R. K. 1979. Heat recovery in the dairy barn: How much is it worth? American Society of Agricultural Engineers Paper No. 79-3535. December.

Kowallski, T. Engineer, Copeland Corporation, Sidney, Ohio. 1981. Personal communication.

Lang, F. 1979. Widening the search for alternative energies. Power Farming Magazine 88(June):43-44.

Larsen, M. B. 1978. Economics of solar heating systems -- Willamette Valley, Oregon. Mechanical Engineering Department, Oregon State University. February.

Lavan, Z. and J. Thompson. 1979. Experimental study of thermally stratified hot water storage tanks. Solar Energy 19:519-524.

Means, P. M. 1980. Experimental evaluation and numerical modeling of a water source heat pump evaporator. M.S. thesis, Oregon State University. December.

Moore, P. B. 1976. Heat pumps: Past - present - future. Heat Pump Technology Conference. Oklahoma State University. October.

Pietsch, J. 1977. The unitary heat pump industry -- 25 years of progress. ASHRAE Journal 12(July):15-18.

Pritsker, A. A. B. 1974. The GASP-IV Simulation Language. New York: John Wiley and Sons Company, Inc.

Reay, D. A. 1979. Heat Recovery Systems. London, England: E. and F. N. Spon.

Roller, W. L., H. M. Keener, G. E. Meyer, and D. L. Elwell. 1977. Simulation of energy alternatives for dairy process water heating. American Society of Agricultural Engineers Paper No. 77-3541. December.

Skinner, S. D. 1980. Heating water in dairies with dairy refrigeration reject heat. American Society of Agricultural Engineers Paper No. 80-3031. June.

Sporn, P., E. R. Ambrose, and T. Baumeister. 1947. Heat Pumps. New York: John Wiley and Sons Company, Inc.

Stipanuk, D. M., R. K. Koelsch, and R. C. Roberg. 1979. Energy and economic analysis of heat recovery versus solar water heating on dairy farms. American Society of Agricultural Engineers Paper No. 79-3537. December.

Stoeker, W. F. 1980. Design of Thermal Systems. New York: McGraw and Hill Company, Inc.

Thompson, J. F. and W. C. Fairbanks. 1979. Refrigeration disuperheaters for energy conservation on dairies. American Society of Agricultural Engineers Paper No. 79-3534. December.

Timmons, M. B., D. C. Ludington, and L. D. Albright. 1977. Development of a simulation model for the milk house. American Society of Agricultural Engineers Paper No. 77-3538. December.

Turner, C. N. 1959. Bulk milk cooler heats water. New York Farm Electriciation Council Report. Department of Agricultural Engineering. Cornell University, New York.

Turner, C. N. 1960. Bulk milk cooler heats water. New York Farm Electrification Council Report. Department of Agricultural Engineering. Cornell University. New York.

United States Department of Agriculture. 1977. A Guide to Energy Savings For the Dairy Farmer.

Welty, J. R., C. E. Wicks, and R. E. Wilson. 1976. Fundamentals of Momentum, Heat, and Mass Transfer. New York: John Wiley and Sons Company, Inc.

Westinghouse Corporation. 1979. Templifier heat pump water heater. Commercial-Industrial Templifier Department. Staunton, Virginia. February. 25 pp.

Zastrow, O. W. 1948. Heating by refrigeration. Agricultural Engineering 22(May):203-204.

APPENDIX A
COMPONENT SPECIFICATIONS

Energy Recovery System

Water-cooled condenser

Manufacturer -- Halstead and Mitchell
Model number -- R300
Nominal rating -- 10.6 kW (3 ton)
Configuration -- Coaxial tube-in-tube
Counterflow
Water tube length -- 10.3 m (33.75 ft)

Differential temperature thermostat

Manufacturer -- Heliotrope General
Model Number -- Delta-T DTT-80
Thermistor sensor model -- SAS-3

Circulation pumps

Manufacturer -- Grundfos
Model number -- UP 26-64SF
Single phase -- 115 volts -- 1/12 HP
Single stage direct drive centrifugal pump

Water storage tank

Manufacturer -- W. L. Jackson Manufacturing Co.
Model number -- Medalist 80220-V
Galvanized hydropneumatic tank rated at 75 psi
Capacity -- 454 liters (220 gallons)

Water-to-water heat pump

Compressor

Manufacturer -- Copeland Corporation
Model number -- BRE 2/0750/TFC
Displacement -- 37.2 m^3/hr (1,315 CFH)
Nominal rating -- 26.4 kW (7.5 tons -- Freon 22)
Three-phase, 208/230 volts, 60 cycle

Condenser

Manufacturer -- Halstead and Mitchell
Model number -- R300
Nominal rating -- 10.6 kW (3 ton)
Configuration -- Coaxial tube-in-tube
Counterflow
Water tube length -- 10.3 m (33.75 ft)

Evaporator

Manufacturer -- Halstead and Mitchell
Model number -- R500
Nominal rating -- 17.6 kW (5 ton) heat exchanger operated as condenser
Configuration -- Coaxial tube-in-tube Counterflow
Water tube length -- 21.1 m (69.5 ft)

Thermostatic expansion valve

Manufacturer -- Sporlan Valve Company
Model number -- SBE-6-L
Nominal rating -- 10.6 kW (3 ton)
Superheat -- 6.7^oC (12^oF)

Water storage tank

Manufacturer -- W. L. Jackson Manufacturing Co.
Model Number -- Medalist 3120-L7
Galvanized hydropneumatic tank rated at 75 psi
Capacity -- 354 liters (120 gallons)

Remote thermostat

Manufacturer -- Johnson Control
Model number -- Series A19-SPDT
Adjustable setpoint temperature 40^o - 120^oC (100^o - 240^oF)
Adjustable temperature differential 3.3^o - 13^oC (6^o - 24^oF)

APPENDIX B
PRELIMINARY CALCULATIONS

Water Storage Tanks

Low Temperature Storage Tanks

The low temperature storage tank was sized to adequately supply low grade hot water requirements for a single milking operation and the heat pump heat source. Energy available during the milk cooling process was based on a 130 cow herd averaging 36.0 kg of milk per cow-day. The following heat transfer equation was used to determine the energy available in cooling milk from 36^{o} to $4^{o}C$:

$$Q_A = m\, C_p \Delta t$$

where:

Q_A = total energy, kJ

m = mass of milk, kg

C_p = specific heat of milk, $kJ/kg^{o}K$

Δt = temperature differential, ^{o}K

The amount of energy available for a 24-hour period assuming a plate heat exchanger efficiency of 75 percent was as follows:

$$Q_A = (130 \text{ cows})(36.0 \text{ kg/cow-day})(3.56 \text{ kJ/kg}^{o}\text{K})(32^{o}\text{K})(0.75)$$
$$= 401{,}940 \text{ kJ/day or } 112 \text{ kW-hr/day}$$

Energy required to raise the city water temperature from an average of $10^{o}C$ to the required temperature of $43.3^{o}C$ was based on pre-1981 water use records.

$$Q_{LT} = (1{,}423 \text{ kg/day})(4.187 \text{ kJ/day})(33.3^{o}\text{K})$$
$$= 198{,}400 \text{ kJ/day or } 55 \text{ kW-hr day}$$

Energy available for use as the heat pump heat source was

$$Q_{HP} = Q_A - Q_{LT}$$

$$= 401{,}940 - 198{,}400 = 203{,}530 \text{ kJ/day or } 57 \text{ kW-hr/day}$$

Energy required to raise the temperature of the low grade hot water to that of the high grade (71.1^{o}C) hot water was

$$Q_{HT} = (863 \text{ kg water/day})(4.187 \text{ kJ/kg}^{o}\text{K})(27.8^{o}\text{K})$$

$$= 100{,}460 \text{ kJ/day or } 28 \text{ kW-hr/day}$$

Energy available to the heat pump exceeded that required to meet the high temperature demand.

$$Q_{HP} > Q_{HT}$$

$$203{,}530 \text{ kJ/day} > 100{,}460 \text{ kJ/day}$$

The analysis did not account for system inefficiencies, energy losses to the environment, or changes in the dairy water and energy management practices. An 833 liter storage tank was selected to hold the low grade hot water required for a single milking operation (700 liters) assuming the time period between milking operations was long enough for the water temperatures in the tank to recover to their original levels.

<u>High Temperature Storage Tank</u>

The high temperature storage tank was sized based on high grade hot water requirements. This did not involve any calculations for sizing. A 454 liter tank was sufficient to hold the high temperature water used for a single milking operation (390 liters).

Heat Pump Components

Condenser Selection. Conditions assumed were city water entered storage tank at $10^{o}C$ and a mass flow rate of 570 kg/hr ($71.1^{o}C$ water removal rate). Resulting mixing established a water temperature of $48^{o}C$. Peak water heating load for these conditions assumed a water mass flow rate of 570 kg/hr and exit water temperature of $76^{o}C$ from condenser.

$$Q = (570 \text{ kg/hr})(4.178 \text{ kJ/kg}^{o}\text{K})(28^{o}\text{K}) = 67{,}300 \text{ kJ/hr}$$

Selection was limited to a 10.6 and 17.7 kW nominally rated unit. Refrigerant mass flow rate through units was based on stated entrance and exit condition.

10.6 kW

$$\dot{m}_R = \frac{Q}{h_C - h_D} = \frac{34{,}128 \text{ kJ/hr}}{(219.3 - 122.3 \text{ kJ/kg})} = 356 \text{ kg/hr}$$

17.7 kW

$$\dot{m}_R = \frac{56{,}800 \text{ kJ/hr}}{97 \text{ kJ/kg}} = 586 \text{ kg/hr}$$

Evaporator Selection. Energy removed in the evaporator assumed entrance and exit water temperatures of $43.3^{o}C$ and $32.2^{o}C$, respectively and a water mass flow rate of 570 kg/hr.

$$Q = 570 \text{ kg/hr})(4.178 \text{ kJ/kg}^{o}\text{K})(11.1^{o}\text{C}) = 26{,}700 \text{ kJ/hr}$$

Refrigerant mass flow rates were based on stated entrance and exit conditions.

10.6 kW

$$\dot{m} = \frac{Q}{(h_B - h_A)}$$

$$\dot{m}_R = \frac{34{,}128 \text{ kJ/hr}}{(192 - 122.3 \text{ kJ/kg})} = 490 \text{ kg/hr}$$

17.7 kW

$$\dot{m}_R = \frac{56{,}880 \text{ kJ/hr}}{69.7 \text{ kJ/kg}} = 816 \text{ kg/hr}$$

Compressor Selection. Energy requirement to increase the refrigerant from the stated evaporator exit conditions to the condenser entrance condition was

$$\dot{Q}_C = \dot{m}_R (h_C - h_B)$$

$$= (490 \text{ kg/hr})(219.3 - 192) = 13{,}500 \text{ kJ/hr}$$

Assuming a 75 percent compressor efficiency, this resulted in an energy input to the compressor of:

$$Q_C = \frac{13{,}500 \text{ kJ/hr}}{3{,}235 \text{ kW/kJ/hr}(0.75)}$$

$$= 5.8 \text{ kW } (7.5 \text{ HP})$$

Energy Recycling System Model Functions

Heat Pump Condenser

Overall heat transfer coefficient for the condenser was based on the manufacturer's specifications using the following energy transfer equation:

$$Q = UA\,\Delta t$$

where:

Q = rated energy transfer capacity, W

U = overall heat transfer coefficient, $W/m^2\,{}^oK$

A = tube surface area, m^2

Δt = temperature differential (R_{114} in - water in), ${}^oK$

solving for U:

$$U = \frac{10{,}600\ W}{(0.413\ m^2)(355^o - 327^oK)}$$

$$U = 919\ W/m^2\,{}^oK$$

Analysis of the condensing section utilized the LMTD method assuming condensation occurred in the initial 85 percent of the condenser. The LMTD method was used in the form:

$$T_{C\ out} = T_{C\ in} + (T_H - T_{C\ in})(1 - e^{-UA/\dot{m}C_p})$$

Solving for the last term yields:

$$UA/\dot{m}C_p = \frac{(919\ W/m^2\,{}^oK)(0.413\ m^2)(0.85)}{(794\ kg/hr)(4.187\ kJ/kgK)}\ (3.6\ kJ/W\ hr)$$

$$= 0.35$$

where 794 kg/hr and 4.187 kJ/kgK represented water mass flow rate and specific heat at 60^oC, respectively. The equation then reduces to

$$T_{C\ out} = T_{C\ in} + (82.2 - T_{C\ in})(1 - e^{-0.35})$$

$$= 0.705\ T_{C\ in} + 24.22$$

The subcooling section utilized the NTU method of analysis and a subcooling region of 15 percent. The NTU method was used in the form:

$$T_{C\ out} = T_{C\ in} + (T_{H\ in} - T_{C\ in})\frac{(1 - e^D)}{\frac{W_{min}}{W_{max}} - e^D}$$

Solving the variables for a refrigerant flow rate of 500 kg/hr and temperature of 82°C yields

$$W_{min} = (500\ \text{kg/hr})(1.25\ \text{kJ/kg}^{\circ}\text{K}) = 625\ \text{kJ/hr}^{\circ}\text{K}$$

$$W_{max} = (794\ \text{kg/hr})(4.191\ \text{kJ/kg}^{\circ}\text{K}) = 3{,}329\ \text{kJ/hrK}$$

$$\frac{W_{min}}{W_{max}} = 0.188$$

$$D = \frac{UA}{W_{min}}\ (1 - \frac{W_{min}}{W_{max}})$$

$$D = \frac{(919\ \text{W/m}^2)(0.413\ \text{m}^2)(0.15)(3.5\ \text{kJ/whr})(1 - 0.188)}{625\ \text{kJ/hr}^{\circ}\text{K}}$$

$$= 0.266$$

$$e^D = 1.305$$

$$\text{efficiency,}\quad E = \frac{1 - e^D}{\frac{W_{min}}{W_{max}} - e^D} = \frac{1 - 1.305}{0.188 - 1.305} = 0.273$$

$$T_{C\ out} = T_{C\ in} + (82.2 - T_{C\ in})(0.273)$$

$$T_{C\ out} = 0.727\ T_{C\ in} + 22.4$$

Combining the two sections yields the composite section:

$$T_{C\ out} = 0.727(0.705\ T_{C\ in} + 24.22) + 22.4$$

$$= 0.513\ T_{C\ in} + 40.0$$

Water-cooled Condenser. The same procedure was followed for the water-cooled condenser assuming a water mass flow rate of 910 kg/hr, no refrigerant subcooling, and a refrigerant temperature of 48.8^{o}C. Thus, only the LMTD method was used. An overall heat transfer coefficient of 1,320 w/m^{2} ^{o}K was established for a temperature differential of 20^{o}C.

$$\frac{UA}{\dot{m}C_p} = \frac{(1,316.8\ W/m^2K)(0.413\ m^2)(3.6\ kJ/Whr)}{(910\ kg/hr)\ (4.183kJ/kgK)}$$

$$= 0.515$$

$$T_{C\ out} = T_{C\ in} + (48.8 - T_{C\ in})(1 - e^{-0.515})$$

$$T_{C\ out} = 0.597T_{C\ in} + 19.7$$

Heat Pump Evaporator. Use of a water-cooled condenser as an evaporator was more difficult to analyze as the condenser is not designed to efficiently operate as an evaporator. Unequal refrigerant flow distribution into the evaporator resulted in a smaller effective surface area being utilized for heat transfer thus, a surface area half of that available was assumed in developing a function for the evaporator. This assumption was supported by the manufacturer's recommendation to oversize the condenser to meet the required heat transfer. Water and refrigerant mass flow rates were 794 and 500 kg/hr, respectively.

The overall heat transfer coefficient was established at 1,898 W/m^{2} ^{o}K for a temperature differential of 11.1^{o}K. The evaporation section was assumed at 85 percent of the surface area. Using the LMTD method in the evaporation section yields:

$$T_{C\ out} = T_{C\ in} - (T_{C\ in} - T_H)(1 - e^{-UA/\dot{m}C_p})$$

where:

$$\frac{UA}{\dot{m}C_p} = \frac{(1{,}898\ W/m^2K)(0.413\ m^2)(0.85)(3.6\ kJ/Whr)}{(794\ kg/hr)\ (4.178\ kJ/kg^oK)}$$

$$= 0.723$$

$$T_{C\ out} = T_{C\ in} - (T_{C\ in} - 29.4)(1 - e^{-0.723})$$

$$T_{C\ out} = 0.515\ T_{C\ in} + 14.27$$

The superheat section involved 15 percent of the available surface area and a refrigerant temperature of 29.4^{o}C.

$$W_{min} = (500\ kg/hr)(0.732\ kJ/kg^oK) = 366\ kJ/hr^oK$$

$$W_{max} = (794\ kg/hr)(4.178\ kJ/kg^oK) = 3{,}317\ kJ/hr^oK$$

$$\frac{W_{min}}{W_{max}} = 0.110$$

$$D = \frac{(1{,}898\ W/m^2K)(0.413\ m^2)(0.15)(3.6\ kJ/Whr)}{366\ kJ/hr^oK}(1 - 0.110)$$

$$= 1.03$$

$$e^D = 2.8$$

$$E = \frac{1 - 2.8}{0.110 - 2.8} = 0.67$$

$$T_{C\ out} = T_{C\ in} - (T_{C\ in} - 29.4)(0.67)$$

$$= 0.33\ T_{C\ in} + 19.67$$

The composite section result was

$$T_{C\ out} = 0.33(0.515\ T_{C\ in} + 14.27) + 16.67$$

$$T_{C\ out} = 0.19\ T_{C\ in} + 23.8$$

APPENDIX C

COMPUTER SIMULATION PROGRAM FOR ENERGY RECYCLING SYSTEM

```
C
C     THE FOLLOWING PROGRAM IS A SIMULATION MODEL OF A WATER-TO-WATER
C     HEAT PUMP FOR TEMPERING WATER AT THE OREGON STATE UNIVERSITY DAIRY
C     CENTER.  INCLUDED IN THE MODEL ARE THE WATER COOLED CONDENSER USED
C     FOR RECLAIMING THE REJECTED HEAT FROM THE MILK REFRIGERATION SYSTEM,
C     THE LOW TEMPERATURE AND HIGH TEMPERATURE WATER STORAGE TANKS, AND
C     THE HEAT PUMP CONDENSER AND EVAPORATOR FUNCTIONS.  THIS SIMULATION
C     ONLY LOOKS AT TEMPERATURE CHANGES THROUGHOUT THE WATER CONVEYANCE
C     SYSTEM.  THE HEAT PUMP CHARACTERISTICS ARE NOT INCLUDED AT THE
C     PRESENT TIME AND ARE CONSIDERED CONSTANT OVER ITS OPERATING INTERVAL.
C
C
      PROGRAM MAIN(INPUT,OUTPUT,TAPE5=INPUT,TAPE5=OUTPUT,TAPE7,TAPE8)
      DIMENSION NSET(200)
      COMMON QSET(200)
*CALL GCOM1
*CALL GCOM2
      COMMON/UCOM1/UA(6),ATEMP,COLD,CITY,HPCM,LAT,HPEM,
     1WMILK,WMASH,WMASL,HMFR,WCCM,HTIME,HPTIM,WTIME,TWCCT
      EQUIVALENCE(NSET(1),QSET(1))
      NCRDR=5
      NPRNT=6
      CALL GASP
      STOP
      END
C
C
C
C
C
C
      SUBROUTINE INTLC
C
*CALL GCOM1
*CALL GCOM2
      COMMON/UCOM1/UA(6),ATEMP,COLD,CITY,HPCM,LAT,HPEM,
     1WMILK,WMASH,WMASL,HMFR,WCCM,HTIME,HPTIM,WTIME,TWCCT
C
C     INTIALIZE THE STATE VARABLES
C
C     HIGH TEMPERATURE STORAGE TANK
C
      SS(1)=73.8
      SS(2)=71.1
      SS(3)=68.3
C
C     LOW TEMPERATURE STORAGE TANK
C
      SS(4)=40.5
      SS(5)=40.0
      SS(6)=37.7
C
C     WATER COOLED CONDENSER
C
      SS(7)=37.7
C
C     HEAT PUMP EVAPORATOR
C
      SS(8)=31.0
C
C     HEAT PUMP CONDENSER
C
      SS(9)=76.9
```

```
C
C*****INITIALIZE DD(*)
C
      DO 5 I=1,6
      DD(I)=0.0
    5 CONTINUE
C
C*****INITIALIZE WATER FLOW RATES
C
      HPCM=0.0
      HPEM=0.0
      WCCM=0.0
      HMFR=0.0
      WMILK=0.0
C
C*****INITIALIZE HEAT LOSS COEFFICIENT*AREA PARAMETERS
C
      UA(1)=0.6535
      UA(2)=0.5027
      UA(3)=0.6535
      UA(4)=0.5765
      UA(5)=0.4392
      UA(6)=0.5765
C
C*****INITIALIZE MISCELLANEOUS VARIABLES
C
      ATEMP=10.0
      CITY=11.1
      HTIME=0.0
      HPTIM=0.0
      WTIME=0.0
      TWCCT=0.0
C
C*****WATER MASS IN TANK SECTION
C
      WMASH=151.3
      WMASL=277.8
C
C*****INITIALIZE EVENTS
C
C     MILKING BEGINS AT TIME = 0.0
C
      ATRIB(1)=0.0
      ATRIB(2)=1.0
      CALL FILEM(1)
C
      RETURN
      END
C
C
C
C
C
C
      SUBROUTINE EVNTS(IX)
C
*CALL GCOM1
*CALL GCOM2
      COMMON/UCOM1/UA(6),ATEMP,COLD,CITY,HPCM,LAT,HPEM,
     1WMILK,WMASH,WMASL,HMFR,WCCM,HTIME,HPTIM,WTIME,TWCCT
C
      GOTO (101,102,103,104,105,106,107,108,109,110,111),IX
C
```

```
C***********************************************************************
C
C     MILKING EVENT BEGINS. LOW TEMPERATURE WATER IS USED TO PREPARE COWS
C     FOR MILKING.  WATER FLOWRATE IS 49.2 KG/HR. FLOWRATE IS ASSUMED
C     CONSTANT FOR 5 HOUR MILKING INTERVAL.
C
  101 WMILK=49.2
      COLD=WMILK
C
C     SCHEDULE END OF MILKING
C
      ATRIB(1)=TNOW+6.0
      ATRIB(2)=2.0
      CALL FILEM(1)
C
C     SCHEDULE START OF REFRIGERATION SYSTEM FOR MILK COOLING PROCESS
C
      ATRIB(1)=TNOW+1.0
      ATRIB(2)=3.0
      ATRIB(3)=1.0
      CALL FILEM(1)
C
C     SCHEDULE END OF WATER COOLED CONDENSER OPERATION. (REFRIGERATION
C     SYSTEM SHUTS OFF)
C
      ATRIB(1)=TNOW+10.0
      ATRIB(2)=4.0
      ATRIB(3)=0.0
      CALL FILEM(1)
C
      RETURN
C
C***********************************************************************
C
C     END OF MILKING EVENT. RESET LOW TEMPERATURE WATER FLOW RATE TO 0.0
C     AND SCHEDULE NEXT MILKING EVENT.
C
  102 WMILK=0.0
      COLD=WMILK
C
      ATRIB(1)=TNOW+6.0
      ATRIB(2)=1.0
      CALL FILEM(1)
C
C     SCHEDULE WASH CYCLE TO START.
C
      ATRIB(1)=TNOW+0.25
      ATRIB(2)=5.0
      CALL FILEM(1)
C
      RETURN
C
C***********************************************************************
C
C     START REFRIGERATION SYSTEM FOR MILK COOLING PROCESS.  WATER COOLED
C     CONDENSER CIRCULATES WATER TO HEAT LOW TEMP. STORAGE TANK.
C     REFRIGERATION SYSTEM STARTS UP 1 HOUR AFTER MILKING BEGINS AND
C     CONTINUES FOR 9 HOURS.(APPROXIMATE)
C
  103 WCCM=907.9
      WTIME=TNOW
C
      RETURN
```

```
C
C*********************************************************************
C
C     END OF WATER COOLED CONDENSER OPERATION.
C
  104 WCCM=0.0
      TWCCT=TNOW-WTIME
      CALL COLCT(TWCCT,2)
C
      RETURN
C
C********************************************************************C
C
C     WASH CYCLE BEGINS.  HIGH TEMPERATURE WATER IS USED FOR EQUIPMENT
C     CLEANSING AND LOW TEMPERATURE WATER REPLACES THE HIGH TEMP. WATER
C     IN THE STORAGE TANK.  CITY WATER ENTERS THE LOW TEMP. STORAGE TANK
C
  105 HMFR=567.6
      COLD=HMFR
C
C     SCHEDULE END OF WASH CYCLE.
C
      ATRIB(1)=TNOW+0.5
      ATRIB(2)=6.0
      CALL FILEM(1)
C
      RETURN
C*********************************************************************
C     WASH CYCLE ENDS.  RESET HIGH TEMP. WATER FLOW RATE TO 0.0.
C
  106 HMFR=0.0
      COLD=HMFR
C
      RETURN
C
C*********************************************************************
C
C     MILK STORAGE TANK WASH CYCLE BEGINS.  HIGH TEMPERATURE WATER USED.
C     TANK WASHED EVERY OTHER DAY.
C
  107 HMFR=378.3
      COLD=HMFR
C
C     SCHEDULE END OF TANK WASH CYCLE
C
      ATRIB(1)=TNOW+0.5
      ATRIB(2)=8.0
      CALL FILEM(1)
C
C     SCHEDULE NEXT TANK WASH EVENT
C
      ATRIB(1)=TNOW+48.0
      ATRIB(2)=7.0
      CALL FILEM(1)
C
      RETURN
C
C*********************************************************************
C
C     END OF TANK WASH
C
  108 HMFR=0.0
```

```
      COLD=HMFR
C
      RETURN
C
C***********************************************************************
C
C     START HEAT PUMP OPERATION
C
  109 HPCM=726.3
      HPEM=862.5
      HTIME=TNOW
C
      RETURN
C
C***********************************************************************
C
C     STOP HEAT PUMP OPERATION
C
  110 HPCM=0.0
      HPEM=0.0
C
C     COLLECT STATISTICS ON HEAT PUMP RUNNING TIME
C
      HPTIM=TNOW-HTIME
      CALL COLCT(HPTIM,1)
C
      RETURN
C
C***********************************************************************
C
C*****CHECK FLAGS FOR STATE EVENT OCCURING
C
C     SHUT OFF WATER COOLED CONDENSER IF LFLAG(1) > 0
C
  111 LAT=NFIND(1.0,5,1,3,0.0)
      IF(LAT.LE.0) GOTO 300
      CALL COPY(LAT)
      IF(ATRIB(3).LT.1.0) GOTO 300
      IF(LFLAG(1).GT.0) GOTO 104
C
C     START WATER COOLED CONDENSER IF LFLAG(2) < 0
C
      IF(LFLAG(2).LT.0) GOTO 103
C
C     START HEAT PUMP OPERATION IF LFLAG(3) < 0
C
  300 IF(LFLAG(3).LT.0) GOTO 109
C
C     STOP HEAT PUMP OPERATION IF LFLAG(4) > 0
C
      IF(LFLAG(4).GT.0) GOTO 110
C
      RETURN
      END
C
C
C
C
C
C
      SUBROUTINE STATE
C
*CALL GCOM1
```

```
*CALL GCOM2
      COMMON/UCOM1/UA(6),ATEMP,COLD,CITY,HPCM,LAT,HPEM,
     1WMILK,WMASH,WMASL,HMFR,WCCM,HTIME,HPTIM,WTIME,TWCCT
C
C
C     DIFFERENTIAL EQUATIONS GOVERNING TEMPERATURE CHANGES
C
C*****HIGH TEMPERATURE STORAGE TANK
C
C     TOP THIRD OF TANK
C
      DD(1)=(UA(1)*(ATEMP-SS(1))+HPCM*(SS(9)-SS(1))-HPCM*(SS(1)-SS(2))-
     1HMFR*(SS(1)-SS(2)))/WMASH
C
C     MIDDLE THIRD OF TANK
C
      DD(2)=(UA(2)*(ATEMP-SS(2))+HPCM*(SS(1)-SS(2))+HMFR*(SS(5)-SS(2))-
     1HPCM*(SS(2)-SS(3)))/WMASH
C
C     BOTTOM THIRD OF TANK
C
      DD(3)=(UA(3)*(ATEMP-SS(3))+HPCM*(SS(2)-SS(3))+HMFR*(SS(2)-SS(3)))
     1/WMASH
C
C*****LOW TEMPERATURE STORAGE TANK
C
C     TOP THIRD OF TANK
C
      DD(4)=(UA(4)*(ATEMP-SS(4))+WCCM*(SS(7)-SS(4))+WMILK*(SS(5)-SS(4))+
     1HPEM*(SS(5)-SS(4))-HMFR*(SS(4)-SS(5))-WCCM*(SS(4)-SS(5)))/WMASL
C
C     MIDDLE THIRD OF TANK
C
      DD(5)=(UA(5)*(ATEMP-SS(5))+WCCM*(SS(4)-SS(5))+HMFR*(SS(4)-SS(5))
     1-WMILK*(SS(4)-SS(5))-HPEM*(SS(4)-SS(5))-WCCM*(SS(5)-SS(6))
     2+WMILK*(SS(6)-SS(5))+HMFR*(SS(6)-SS(5))+HPEM*(SS(6)-SS(5)))/WMASL
C
C     BOTTOM THIRD OF TANK
C
      DD(6)=(UA(6)*(ATEMP-SS(6))+WCCM*(SS(5)-SS(6))+COLD*(CITY-SS(6))
     1+HPEM*(SS(8)-SS(6))-WMILK*(SS(5)-SS(6))-HMFR*(SS(5)-SS(6))
     2-HPEM*(SS(5)-SS(6)))/WMASL
C
C*****WATER COOLED CONDENSER ON MILK COOLING REFRIGERATION SYSTEM
C
      IF(WCCM.LE.0.0) GOTO 100
      SS(7)=0.597*SS(6)+19.7
C
C*****HEAT PUMP EVAPORATOR
C
      IF(HPEM.LE.0.0) GOTO 100
      SS(8)=0.19*SS(4)+23.8
C
C*****HEAT PUMP CONDENSER
C
      SS(9)=0.513*SS(3)+40.0
  100 RETURN
C
      END
C
C
C
C
C
```

```
C
      SUBROUTINE SCOND
C
*CALL GCOM1
*CALL GCOM2
      COMMON/UCOM1/UA(6),ATEMP,COLD,CITY,HPCM,LAT,HPEM,
     1WMILK,WMASH,WMASL,HMFR,WCCM,HTIME,HPTIM,WTIME,TWCCT
C
C*****CROSSING CONDITIONS FOR WATER COOLED CONDENSER OPERATION.
C
C     IF ENTERING TEMP., SS(7), > 40.5C THEN STOP WATER COOLED CONDENSER
C
      LFLAG(1)=KROSS(6,0,0.0,40.5,1,5.0)
C
C     IF ENTERING TEMP.,SS(7), < 40.5C, THEN RESTART CONDENSER
C
      LFLAG(2)=KROSS(6,0,0.0,40.5,-1,10.0)
C
C*****CROSSING CONDITIONS FOR HEAT PUMP OPERATION.
C
C     IF TOP THIRD OF HIGH TEMP TANK,SS(1), < 65.5C THEN START HEAT PUMP
C
      LFLAG(3)=KROSS(1,0,0.0,65.5,-1,5.0)
C
C     IF TOP THIRD OF TANK SS(1) > 73.9C, THEN STOP HEAT PUMP
C
      LFLAG(4)=KROSS(1,0,0.0,73.9,1,5.0)
C
      RETURN
      END
C
C
C
C
C
      SUBROUTINE SSAVE
C
*CALL GCOM1
*CALL GCOM2
      COMMON/UCOM1/UA(6),ATEMP,COLD,CITY,HPCM,LAT,HPEM,
     1WMILK,WMASH,WMASL,HMFR,WCCM,HTIME,HPTIM,WTIME,TWCCT
C
C*****SUBPROGRAM FOR COLLECTING DATA ON STATE VARIABLES AND PLOTTING RESULTS
C
C     TEMPERATURES OF STORAGE TANK SECTIONS ARE COLLECTED AND PLOTTED VS.
C     TIME.
C
      CALL GPLOT(SS(1),TNOW,1)
C
      RETURN
      END
```

APPENDIX D
MONTHLY WATER USE DATA

	Water Use in Liters Per Day		
Month	43.3°C	71.1°C	City Water Supplied
11/81	425.5	761.3	3,051.6
12/81	648.0	735.3	3,359.5
1/82	780.7	860.6	4,733.5
2/82	490.4	748.2	3,681.2
3/82	410.8	773.5	3,197.0
4/82	154.8	803.5	3,111.8
5/82	113.0	780.7	3,174.9

APPENDIX E

MONTHLY ELECTRICAL ENERGY CONSUMPTION DATA

Month	Electrical Energy Consumption in kW-hr Per Day: Low Temperature Water Heater	High Temperature Water Heater	Water-to-Water Heat Pump	Total
11/81	6.1 (2.1)*	5.2 (1.0)	11.8	23.1
12/81	14.5 (2.4)	3.8 (0.8)	13.5	31.8
1/82	22.9 (2.2)	13.5 (2.1)	11.3**	47.7
2/82	12.7 (1.8)	8.3 (1.3)	11.5**	32.5
3/82	7.4 (2.3)	3.5 (1.2)	12.2	23.1
4/82	6.2 (1.9)	3.8 (1.4)	13.0	23.0
5/82	6.1 (2.3)	5.1 (2.0)	11.9	23.1

* Peak energy demand

** Heat pump problems necessitated a shut down period for repairs.